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Module 1.1

Steam - The Energy Fluid

Steam - The Energy Fluid

It is useful to introduce the topic of steam by considering its many uses and benefits, before entering an overview of the steam plant or any technical explanations.

Steam has come a long way from its traditional associations with locomotives and the Industrial Revolution. Steam today is an integral and essential part of modern technology. Without it, our food, textile, chemical, medical, power, heating and transport industries could not exist or perform as they do.

Steam provides a means of transporting controllable amounts of energy from a central, automated boiler house, where it can be efficiently and economically generated, to the point of use. Therefore as steam moves around a plant it can equally be considered to be the transport and provision of energy.

For many reasons, steam is one of the most widely used commodities for conveying heat energy. Its use is popular throughout industry for a broad range of tasks from mechanical power production to space heating and process applications.



Fig. 1.1.1 An 18th century steam engine. Photography courtesy of Kew Bridge Steam Museum, London



Fig. 1.1.2 A modern packaged steam heat exchange system used for producing hot water

Steam is efficient and economic to generate

Water is plentiful and inexpensive. It is non-hazardous to health and environmentally sound. In its gaseous form, it is a safe and efficient energy carrier. Steam can hold five or six times as much potential energy as an equivalent mass of water.

When water is heated in a boiler, it begins to absorb energy. Depending on the pressure in the boiler, the water will evaporate at a certain temperature to form steam. The steam contains a large quantity of stored energy which will eventually be transferred to the process or the space to be heated.

It can be generated at high pressures to give high steam temperatures. The higher the pressure, the higher the temperature. More heat energy is contained within high temperature steam so its potential to do work is greater.

- Modern shell boilers are compact and efficient in their design, using multiple passes and efficient burner technology to transfer a very high proportion of the energy contained in the fuel to the water, with minimum emissions.
- The boiler fuel may be chosen from a variety of options, including combustible waste, which makes the steam boiler an environmentally sound option amongst the choices available for providing heat. Centralised boiler plant can take advantage of low interruptible gas tariffs, because any suitable standby fuel can be stored for use when the gas supply is interrupted.
- Highly effective heat recovery systems can virtually eliminate blowdown costs, return valuable condensate to the boiler house and add to the overall efficiency of the steam and condensate loop.

The increasing popularity of Combined Heat and Power (CHP) systems demonstrates the high regard for steam systems in today's environment and energy-conscious industries.



Fig. 1.1.3

Steam can easily and cost effectively be distributed to the point of use

Steam is one of the most widely used media to convey heat over distances. Because steam flows in response to the pressure drop along the line, expensive circulating pumps are not needed.

Due to the high heat content of steam, only relatively small bore pipework is required to distribute the steam at high pressure. The pressure is then reduced at the point of use, if necessary. This arrangement makes installation easier and less expensive than for some other heat transfer fluids.

Overall, the lower capital and running costs of steam generation, distribution and condensate return systems mean that many users choose to install new steam systems in preference to other energy media, such as gas fired, hot water, electric and thermal oil systems.

Steam is easy to control

Because of the direct relationship between the pressure and temperature of saturated steam, the amount of energy input to the process is easy to control, simply by controlling the saturated steam pressure. Modern steam controls are designed to respond very rapidly to process changes.

The item shown in Figure 1.1.4 is a typical two port control valve and pneumatic actuator assembly, designed for use on steam. Its accuracy is enhanced by the use of a pneumatic valve positioner.

The use of two port valves, rather than the three port valves often necessary in liquid systems, simplifies control and installation, and may reduce equipment costs.



Fig. 1.1.4 Typical two port control valve with a pneumatic actuator and positioner

Energy is easily transferred to the process

Steam provides excellent heat transfer. When the steam reaches the plant, the condensation process efficiently transfers the heat to the product being heated.

Steam can surround or be injected into the product being heated. It can fill any space at a uniform temperature and will supply heat by condensing at a constant temperature; this eliminates temperature gradients which may be found along any heat transfer surface - a problem which is so often a feature of high temperature oils or hot water heating, and may result in quality problems, such as distortion of materials being dried.

Because the heat transfer properties of steam are so high, the required heat transfer area is relatively small. This enables the use of more compact plant, which is easier to install and takes up less space in the plant. A modern packaged unit for steam heated hot water, rated to 1 200 kW and incorporating a steam plate heat exchanger and all the controls, requires only 0.7 m² floor space. In comparison, a packaged unit incorporating a shell and tube heat exchanger would typically cover an area of two to three times that size.

The modern steam plant is easy to manage

Increasingly, industrial energy users are looking to maximise energy efficiency and minimise production costs and overheads. The Kyoto Agreement for climate protection is a major external influence driving the energy efficiency trend, and has led to various measures around the globe, such as the Climate Change Levy in the UK. Also, in today's competitive markets, the organisation with the lowest costs can often achieve an important advantage over rivals. Production costs can mean the difference between survival and failure in the marketplace.

Ways of increasing energy efficiency include monitoring and charging energy consumption to relevant departments. This builds an awareness of costs and focuses management on meeting targets. Variable overhead costs can also be minimised by ensuring planned, systematic maintenance; this will maximise process efficiency, improve quality and cut downtime.

Most steam controls are able to interface with modern networked instrumentation and control systems to allow centralised control, such as in the case of a SCADA system or a Building/Energy Management System. If the user wishes, the components of the steam system can also operate independently (standalone).

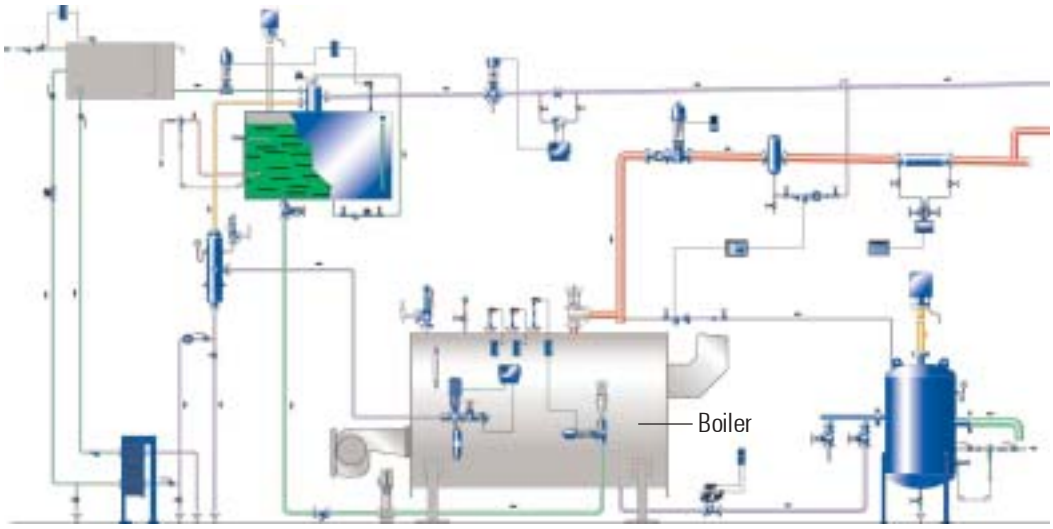


Fig. 1.1.5 A modern boiler house package

With proper maintenance a steam plant will last for many years, and the condition of many aspects of the system is easy to monitor on an automatic basis. When compared with other systems, the planned management and monitoring of steam traps is easy to achieve with a trap monitoring system, where any leaks or blockages are automatically pinpointed and immediately brought to the attention of the engineer.

This can be contrasted with the costly equipment required for gas leak monitoring, or the time-consuming manual monitoring associated with oil or water systems.



Fig. 1.1.6 Just some of the products manufactured using steam as an essential part of the process

In addition to this, when a steam system requires maintenance, the relevant part of the system is easy to isolate and can drain rapidly, meaning that repairs may be carried out quickly.

In numerous instances, it has been shown that it is far less expensive to bring a long established steam plant up to date with sophisticated control and monitoring systems, than to replace it with an alternative method of energy provision, such as a decentralised gas system. The case studies refered to in Module 1.2 provide real life examples.

Todays state-of-the-art technology is a far cry from the traditional perception of steam as the stuff of steam engines and the Industrial Revolution. Indeed, steam is the preferred choice for industry today. Name any well known consumer brand, and in nine cases out of ten, steam will have played an important part in production.

Steam is flexible

Not only is steam an excellent carrier of heat, it is also sterile, and thus popular for process use in the food, pharmaceutical and health industries. It is also widely used in hospitals for sterilisation purposes.

The industries within which steam is used range from huge oil and petrochemical plants to small local laundries. Further uses include the production of paper, textiles, brewing, food production, curing rubber, and heating and humidification of buildings.

Many users find it convenient to use steam as the same working fluid for both space heating and for process applications. For example, in the brewing industry, steam is used in a variety of ways during different stages of the process, from direct injection to coil heating.



Fig. 1.1.7 Clean steam pipeline equipment used in pharmaceutical process plant



Fig. 1.1.8 These brewing processes all use steam

Steam is also intrinsically safe - it cannot cause sparks and presents no fire risk. Many petrochemical plants utilise steam fire-extinguishing systems. It is therefore ideal for use in hazardous areas or explosive atmospheres.

Other methods of distributing energy

The alternatives to steam include water and thermal fluids such as high temperature oil. Each method has its advantages and disadvantages, and will be best suited to certain applications or temperature bands.

Compared to steam, water has a lower potential to carry heat, consequently large amounts of water must be pumped around the system to satisfy process or space heating requirements. However, water is popular for general space heating applications and for low temperature processes (up to 120°C) where some temperature variation can be tolerated.

Thermal fluids, such as mineral oils, may be used where high temperatures (up to 400°C) are required, but where steam cannot be used. An example would include the heating of certain chemicals in batch processes. However thermal fluids are expensive, and need replacing every few years - they are not suited to large systems. They are also very 'searching' and high quality connections and joints are essential to avoid leakage.

Different media are compared in Table 1.1.1, which follows. The final choice of heating medium depends on achieving a balance between technical, practical and financial factors, which will be different for each user.

Broadly speaking, for commercial heating and ventilation, and industrial systems, steam remains the most practical and economic choice.

Table 1.1.1 Comparison of heating media with steam

Steam	Hot water	High temperature oils
High heat content Latent heat approximately 2 100 kJ/kg	Moderate heat content Specific heat 4.19 kJ/kg°C	Poor heat content Specific heat often 1.69-2.93 kJ/kg°C
Inexpensive Some water treatment costs	Inexpensive Only occasional dosing	Expensive
Good heat transfer coefficients	Moderate coefficients	Relatively poor coefficients
High pressure required for high temperatures	High pressure needed for high temperatures	Low pressures only to get high temperatures
No circulating pumps required Small pipes	Circulating pumps required Large pipes	Circulating pumps required Even larger pipes
Easy to control with two way valves	More complex to control - three way valves or differential pressure valves may be required	More complex to control - three way valves or differential pressure valves may be required.
Temperature breakdown is easy through a reducing valve	Temperature breakdown more difficult	Temperature breakdown more difficult
Steam traps required	No steam traps required	No steam traps required
Condensate to be handled	No condensate handling	No condensate handling
Flash steam available	No flash steam	No flash steam
Boiler blowdown necessary	No blowdown necessary	No blowdown necessary
Water treatment required to prevent corrosion	Less corrosion	Negligible corrosion
Reasonable pipework required	Searching medium, welded or flanged joints usual	Very searching medium, welded or flanged joints usual
No fire risk	No fire risk	Fire risk
System very flexible	System less flexible	System inflexible

The benefits of steam - a summary:

Table 1.1.2 Steam benefits

Inherent benefits	System benefits
Water is readily available Water is inexpensive Steam is clean and pure Steam is inherently safe Steam has a high heat content Steam is easy to control due to the pressure/temperature relationship Steam gives up its heat at a constant temperature	Small bore pipework, compact size and less weight No pumps, no balancing Two port valves - cheaper Maintenance costs lower than for dispersed plant Capital cost is lower than for dispersed plant SCADA compatible products Automation; fully automated boiler houses fulfil requirements such as PM5 and PM60 in the UK Low noise Reduced plant size (as opposed to water) Longevity of equipment Boilers enjoy flexible fuel choice and tariff Systems are flexible and easy to add to
Environmental factors	**Uses**
Fuel efficiency of boilers Condensate management and heat recovery Steam can be metered and managed Links with CHP/waste heat Steam makes environmental and economic sense	Steam has many uses - chillers, pumps, fans, humidification Sterilisation Space heating Range of industries

Questions

1. How does the heat carrying capacity of steam compare with water ?

a| It is about the same ☐

b| It is less than water ☐

c| More than water ☐

d| It depends on the temperature ☐

2. Which of the following is true of steam ?

a| It carries much more heat than water ☐

b| Its heat transfer coefficient is more than thermal oil and water ☐

c| Pumps are not required for distribution ☐

d| All of the above ☐

3. The amount of energy carried by steam is adjusted by

a| Controlling steam pressure ☐

b| Controlling steam flow ☐

c| Controlling condensation ☐

d| Controlling boiler feeedwater temperature ☐

4. Approximately how much potential energy will steam hold compared to an equivalent mass of water?

a| Approximately the same ☐

b| Half as much ☐

c| 5 to 6 times as much ☐

d| Twice as much ☐

5. How does steam give up its heat ?

a| By cooling ☐

b| By radiation ☐

c| By conduction ☐

d| By condensation ☐

6. Which of the following statements is not true ?

a| Steam is less searching than high temperature oil or water ☐

b| Steam pipes will be smaller than water or high temperature oil pipes ☐

c| Temperature breakdown of water and oil is easier than steam ☐

d| Steam plant is smaller than water plant. ☐

Answers

1: c, 2: d, 3: a, 4: c, 5: d, 6: c

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Module 1.2

Steam and the Organisation

Steam and the Organisation

The benefits described are not of interest to all steam users. The benefits of steam, as a problem solver, can be subdivided according to different viewpoints within a business. They are perceived differently depending on whether you are a chief executive, a manager or at operating level.

The questions these people ask about steam are markedly different.

Chief executive

The highest level executive is concerned with the best energy transfer solution to meet the strategic and financial objectives of the organisation.

If a company installs a steam system or chooses to upgrade an existing system, a significant capital investment is required, and the relationship with the system, and the system provider, will be long and involved.

Chief executives and senior management want answers to the following questions:

Q. What kind of capital investment does a steam system represent ?

A steam system requires only small bore pipes to satisfy a high heat requirement. It does not require costly pumps or balancing, and only two port valves are required.

This means the system is simpler and less expensive than, for example, a high temperature hot water system. The high efficiency of steam plant means it is compact and makes maximum use of space, something which is often at a premium within plant.

Furthermore, upgrading an existing steam system with the latest boilers and controls typically represents 50% of the cost of removing it and replacing it with a decentralised gas fired system.



Fig. 1.2.1

Q. How will the operating and maintenance costs of a steam system affect overhead costs ?

Centralised boiler plant is highly efficient and can use low interruptible tariff fuel rates. The boiler can even be fuelled by waste, or form part of a state-of-the-art Combined Heat and Power plant.

Steam equipment typically enjoys a long life - figures of thirty years or more of low maintenance life are quite usual.

Modern steam plant, from the boiler house to the steam using plant and back again, can be fully automated. This dramatically cuts the cost of manning the plant.

Sophisticated energy monitoring equipment will ensure that the plant remains energy efficient and has a low manning requirement.

All these factors in combination mean that a steam system enjoys a low lifetime cost.

Q. If a steam system is installed, how can the most use be made of it ?

Steam has a range of uses. It can be used for space heating of large areas, for complex processes and for sterilisation purposes.

Using a hospital as an example, steam is ideal because it can be generated centrally at high pressure, distributed over long distances and then reduced in pressure at the point of use. This means that a single high pressure boiler can suit the needs of all applications around the hospital, for example, heating of wards, air humidification, cooking of food in large quantities and sterilisation of equipment.

It is not as easy to cater for all these needs with a water system.

Q. What if needs change in the future ?

Steam systems are flexible and easy to add to. They can grow with the company and be altered to meet changing business objectives.

Q. What does using steam say about the company ?

The use of steam is environmentally responsible. Companies continue to choose steam because it is generated with high levels of fuel efficiency. Environmental controls are increasingly stringent, even to the extent that organisations have to consider the costs and methods of disposing of plant before it is installed. All these issues are considered during the design and manufacture of steam plant.

Management level

A manager will consider steam as something that will provide a solution to a management problem, as something that will benefit and add value to the business. The manager's responsibility is to implement initiatives ordered by senior executives. A manager would ask "How will steam enable successful implementation of this task ?"

Managers tend to be practical and focused on completing a task within a budget. They will choose to use steam if they believe it will provide the greatest amount of practicality and expediency, at a reasonable cost.

They are less concerned with the mechanics of the steam system itself. A useful perspective would be that the manager is the person who wants the finished product, without necessarily wanting to know how the machinery that produces it is put together.

Managers need answers to the following questions:

Q. Will steam be right for the process ?

Steam serves many applications and uses. It has a high heat content and gives up its heat at a constant temperature. It does not create a temperature gradient along the heat transfer surface, unlike water and thermal oils, which means that it may provide more consistent product quality. As steam is a pure fluid, it can be injected directly into the product or made to surround the product being heated. The energy given to the process is easy to control using two port valves, due to the direct relationship between temperature and pressure.



Fig. 1.2.2

Q. If a steam system is installed, how can the most use be made of it ?

Steam has a wide variety of uses. It can be used for space heating over large areas, and for many complex manufacturing processes.

On an operational level, condensate produced by a manufacturing process can be returned to the boiler feedtank. This can significantly reduce the boiler fuel and water treatment costs, because the water is already treated and at a high temperature.

Lower pressure steam can also be produced from the condensate in a flash vessel, and used in low pressure applications such as space heating.

Q. What does steam cost to produce ?

Water is plentiful and inexpensive, and steam boilers are highly efficient because they extract a large proportion of the energy contained within the fuel. As mentioned previously, central boiler plant can take advantage of low interruptible fuel tariffs, something which is not possible for decentralised gas systems which use a constant supply of premium rate fuel.

Flash steam and condensate can be recovered and returned to the boiler or used on low pressure applications with minimal losses.

Steam use is easy to monitor using steam flowmeters and SCADA compatible products.

For real figures, see 'The cost of raising steam', later in this Module.

In terms of capital and operating costs, it was seen when answering the concerns of the chief executive that steam plant can represent value for money in both areas.

Q. Is there enough installation space ?

The high rates of heat transfer enjoyed by steam means that the plant is smaller and more compact than water or thermal oil plant. A typical modern steam to hot water heat exchanger package rated to 1 200 kW occupies only 0.7 m² floor space. Compare this to a hot water calorifier which may take up a large part of a plant room.

Q. Not wishing to think too much about this part of the process, can a total solution be provided ?

Steam plant can be provided in the form of compact ready-to-install packages which are installed, commissioned and ready to operate within a very short period of time. They offer many years of trouble-free operation and have a low lifetime cost.

Technical personnel/operators

At the operating level, the day-to-day efficiency and working life of individuals can be directly affected by the steam plant and the way in which it operates. These individuals want to know that the plant is going to work, how well it will work, and the effect this will have on their time and resources.

Technical personal/operators need answers to the following questions:

Q. Will it break down ?

A well designed and maintained steam plant should have no cause to break down. The mechanics of the system are simple to understand and designed to minimise maintenance. It is not unusual for items of steam plant to enjoy 30 or 40 years of trouble-free life.

Q. When maintenance is required, how easy is it ?

Modern steam plant is designed to facilitate rapid easy maintenance with minimum downtime. The modern design of components is a benefit in this respect. For example, swivel connector steam traps can be replaced by undoing two bolts and slotting a new trap unit into place. Modern forged steam and condensate manifolds incorporate piston valves which can be maintained in-line with a simple handheld tool.

Sophisticated monitoring systems target the components that really need maintenance, rather than allowing preventative maintenance to be carried out unnecessarily on working items of plant. Control valve internals can simply be lifted out and changed in-line, and actuators can be reversed in the field. Mechanical pumps can be serviced, simply by removing a cover, which has all the internals attached to it. Universal pipeline connectors allow steam traps to be replaced in minutes.

An important point to note is that when maintenance of the system is required, a steam system is easy to isolate and will drain rapidly, meaning that repairs can be quickly actioned. Any minor leaks that do occur are non-toxic. This is not always the case with liquid systems, which are slower and more costly to drain, and may include toxic or difficult to handle thermal fluids.

Q. Will it look after itself ?

A steam system requires maintenance just like any other important part of the plant, but thanks to today's modern steam plant design, manning and maintenance requirements and the lifetime costs of the system are low. For example, modern boiler houses are fully automated. Feedwater treatment and heating burner control, boiler water level, blowdown and alarm systems are all carried out by automatic systems. The boiler can be left unmanned and only requires testing in accordance with local regulations.

Similarly, the steam plant can be managed centrally using automatic controls, flowmetering and monitoring systems. These can be integrated with a SCADA system.

Manning requirements are thus minimised.

Industries and processes which use steam:

Table 1.2.1 Steam users

Heavy users	Medium users	Light users
Food and drinks	Heating and ventilating	Electronics
Pharmaceuticals	Cooking	Horticulture
Oil refining	Curing	Air conditioning
Chemicals	Chilling	Humidifying
Plastics	Fermenting	
Pulp and paper	Treating	
Sugar refining	Cleaning	
Textiles	Melting	
Metal processing	Baking	
Rubber and tyres	Drying	
Shipbuilding		
Power generation		

Interesting uses for steam:

- Shrink-wrapping meat.
- Depressing the caps on food jars.
- Exploding corn to make cornflakes.
- Dyeing tennis balls.
- Repairing underground pipes (steam is used to expand and seal a foam which has been pumped into the pipe. This forms a new lining for the pipe and seals any cracks).
- Keeping chocolate soft, so it can be pumped and moulded.
- Making drinks bottles look attractive but safe, for example tamper-proof, by heat shrinking a film wrapper.
- Drying glue (heating both glue and materials to dry on a roll).
- Making condoms.
- Making bubble wrap.
- Peeling potatoes by the tonne (high pressure steam is injected into a vessel full of potatoes. Then it is quickly depressurised, drawing the skins off).
- Heating swimming pools.
- Making instant coffee, milk or cocoa powder.
- Moulding tyres.
- Ironing clothes.
- Making carpets.
- Corrugating cardboard.
- Ensuring a high quality paint finish on cars.
- Washing milk bottles.
- Washing beer kegs.
- Drying paper.
- Ensuring medicines and medical equipment are sterile.
- Cooking potato chips.
- Sterilising wheelchairs.
- Cooking pieces of food, for example seafood, evenly in a basket using injected steam for heat, moisture and turbulence at the same time.
- Cooking large vats of food by direct injection or jacket heating.

……and hundreds more.

The cost of raising steam

In today's industry, the cost of supplying energy is of enormous interest. Table 1.2.2 shows provisional industrial fuel prices for the United Kingdom, obtained from a recent Digest of UK Energy Statistics, which were available in 2001.

Table 1.2.2 UK fuel prices - 2001 (provisional)

Fuel	Size of consumer	2001
Coal (£ per tonne)	Small Medium Large	55.49 46.04 33.85
Heavy fuel oil (£ per tonne)	Small Medium Large	142.73 136.15 119.54
Gas oil (£ per tonne)	Small Medium Large	230.48 224.61 204.30
Electricity (pence per kWh)	Small Medium Large	4.89 3.61 2.76
Gas (pence per kWh)	Small Medium Large	1.10 0.98 0.78

The cost of raising steam based on the above costs

All figures exclude the Climate Change Levy (which came into force in April 2001) although the oil prices do include hydrocarbon oil duty.

The cost of raising steam is based on the cost of raising one tonne (1 000 kg) of steam using the fuel types listed and average fuel cost figures.

Table 1.2.3 UK steam costs - 2001 (provisional)

Fuel			Average unit cost (£)	Unit of supply	Cost of raising 1 000 kg of steam (£)
Oil	**Heavy**	**(3 500 s)**	0.074 0	Per litre	9.12
	Medium oil	**(950 s)**	0.091 8	Per litre	11.31
	Light oil	**(210 s)**	0.100 0	Per litre	12.32
	Gas oil	**(35 s)**	0.105 4	Per litre	12.99
Natural gas	**Firm**		0.006 3	Per kWh	6.99
	Interruptible		0.005 0	Per kWh	5.55
Coal			35.160 0	Per Tonne	3.72
Electricity			0.036 7	Per kWh	25.26

Boiler efficiency

A modern steam boiler will generally operate at an efficiency of between 80 and 85%. Some distribution losses will be incurred in the pipework between the boiler and the process plant equipment, but for a system insulated to current standards, this loss should not exceed 5% of the total heat content of the steam. Heat can be recovered from blowdown, flash steam can be used for low pressure applications, and condensate is returned to the boiler feedtank. If an economiser is fitted in the boiler flue, the overall efficiency of a centralised steam plant will be around 87%.

This is lower than the 100% efficiency realised with an electric heating system at the point of use, but the typical running costs for the two systems should be compared. It is clear that the cheapest option is the centralised boiler plant, which can use a lower, interruptible gas tariff rather than the full tariff gas or electricity, essential for a point of use heating system. The overall efficiency of electricity generation at a power station is approximately 30 to 35%, and this is reflected in the unit charges.

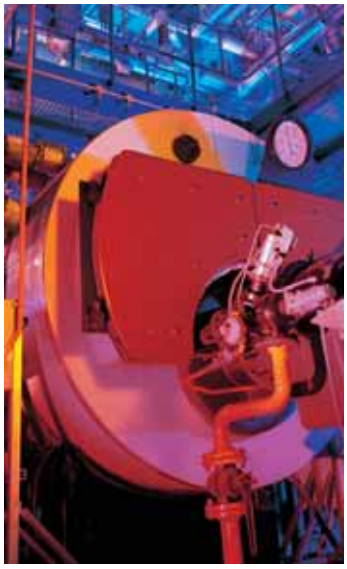


Fig. 1.2.3

Components within the steam plant are also highly efficient. For example, steam traps only allow condensate to drain from the plant, retaining valuable steam for the process. Flash steam from the condensate can be utilised for lower pressure processes with the assistance of a flash vessel.

The following pages introduce some real life examples of situations in which a steam user had, initially, been poorly advised and/or had access to only poor quality or incomplete information relating to steam plant. In both cases, they almost made decisions which would have been costly and certainly not in the best interests of their organisation.

Some identification details have been altered.

Case study: UK West Country hospital considers replacing their steam system

In one real life situation in the mid 1990's, a hospital in the West of England considered replacing their aged steam system with a high temperature hot water system, using additional gas fired boilers to handle some loads. Although new steam systems are extremely modern and efficient in their design, older, neglected systems are sometimes encountered and this user needed to take a decision either to update or replace the system.

The financial allocation to the project was £2.57 million over three years, covering professional fees plus VAT.

It was shown, in consultation with the hospital, that only £1.2 million spent over ten years would provide renewal of the steam boilers, pipework and a large number of calorifiers. It was also clear that renewal of the steam system would require a much reduced professional input. In fact, moving to high temperature hot water (HTHW) would cost over £1.2 million **more** than renewing the steam system.

The reasons the hospital initially gave for replacing the steam system were:

- With a HTHW system, it was thought that maintenance and operating costs would be lower.
- The existing steam plant, boilers and pipework needed replacing anyway.

Maintenance costs for the steam system were said to include insurance of calorifiers, steam trap maintenance, reducing valves and water treatment plant, also replacement of condensate pipework.

Operating costs were said to include water treatment, make-up water, manning of the boiler house, and heat losses from calorifiers, blowdown and traps.

The approximate annual operating costs the hospital was using for HTHW versus steam, are given in the Table 1.2.4.

Table 1.2.4 Operating costs

Utility	Steam (£)	HTHW (£)
Fuel	245 000 0	180 000 37 500
Attendance	57 000	0
Maintenance	77 000	40 000
Water treatment	8 000	0
Water	400	100
Electricity	9 000	12 000
Spares	10 000	5 000
Total	**£406 400**	**£274 600**

Additional claims in favour of individual gas fired boilers were given as:

- No primary mains losses.
- Smaller replacement boilers.
- No stand-by fuel requirement.

The costings set out above made the HTHW system look like the more favourable option in terms of operating costs.

The new HTHW system would cost £1 953 000 plus £274 600 per annum in operating and maintenance costs. This, in effect, meant decommissioning a plant and replacing it at a cost in excess of £2 million, to save just over £130 000 a year.

The following factors needed to be taken into account:

- The £130 000 saving using HTHW is derived from £406 400 - £274 600. The steam fuel cost can be reduced to the same level as for HTHW by using condensate return and flash steam recovery. This would reduce the total by £65 000 to £341 400.
- The largest savings claimed were due to the elimination of manned boilers. However, modern boiler houses are fully automated and there is no manning requirement.
- The £37 000 reduction in maintenance costs looked very optimistic considering that the HTHW solution included the introduction of 16 new gas fired boilers, 4 new steam generators and 9 new humidifiers. This would have brought a significant maintenance requirement.
- The steam generators and humidifiers had unaccounted for fuel requirements and water treatment costs. The fuel would have been supplied at a premium rate to satisfy the claim that stand-by fuel was not needed. In contrast, centralised steam boilers can utilise low cost alternatives at interruptible tariff.
- The savings from lower mains heat losses (eliminated from mains-free gas fired boilers) were minimal against the total costs involved, and actually offset by the need for fuel at premium tariff.
- The proposal to change appeared entirely motivated by weariness with the supposed low efficiency calorifiers – however on closer inspection it can be demonstrated that steam to water calorifiers are 84% efficient, and the remaining 16% of heat contained in the condensate can almost all be returned to the boiler house. Gas fired hot water boilers struggle to reach the 84% efficiency level even at full-load. Unused heat is just sent up the stack. Hot water calorifiers are also much larger and more complicated, and the existing plant rooms were unlikely to have much spare room.
- A fact given in favour of replacing the steam system was the high cost of condensate pipe replacement. This statement tells us that corrosion was taking place, of which the commonest cause is dissolved gases, which can be removed physically or by chemical treatment. Removing the system because of this is like replacing a car because the ashtrays are full !

- A disadvantage given for steam systems was the need for insurance inspection of steam/water calorifiers. However, HTHW calorifiers also require inspection !
- A further disadvantage given was the need to maintain steam pressure reducing valves. But water systems contain three port valves with a significant maintenance requirement.
- The cost of make-up water and water treatment for steam systems was criticised. However, when a steam system requires maintenance, the relevant part can be easily isolated and quickly drained with few losses (this minimises downtime). In contrast, a water system requires whole sections to be cooled and then drained off. It must then be refilled and purged of air after maintenance. HTHW systems also require chemical treatment, just like steam systems.

Presented with these explanations, the hospital realised that much of the evidence they had been basing their decision on was biased and incomplete. The hospital engineering team reassessed the case, and decided to retain their steam plant and bring it up to date with modern controls and equipment, saving a considerable amount of money.

Trace heating

Trace heating is a vital element in the reliable operation of pipelines and storage/process vessels, across a broad range of industries.

A steam tracer is a small steam pipe which runs along the outer surface of a (usually) larger process pipe. Heat conductive paste is often used between the tracer and the process pipe. The two pipes are then insulated together. The heat provided from the tracer (by conduction) prevents the contents of the larger process pipe from freezing (anti-frost protection for water lines) or maintains the temperature of the process fluid so that it remains easy to pump.

Tracing is commonly found in the oil and petrochemical industries, but also in the food and pharmaceutical sectors, for oils, fats and glucose. Many of these fluids can only be pumped at temperatures well above ambient. In chemical processing, a range of products from acetic acid through to asphalt, sulphur and zinc compounds may only be moved through pipes if maintained at a suitable temperature.

For the extensive pipe runs found in much of process industry, steam tracing remains the most popular choice. For very short runs or where no steam supply is available, electrical tracing is often chosen, although hot water is also used for low temperature requirements. The relative benefits of steam and electric tracing are summarised in Table 1.2.5.

Table 1.2.5 The relative merits of steam and electric trace heating

	Steam trace heating	Electric trace heating
Robustness - ability to resist adverse weather and physical abuse	Good	Poor
Flexibility - ability to meet demands of different products	Excellent	Poor
Safety - suitability for use in hazardous areas	Excellent	Cannot be used in all zones
Energy costs per GJ	0 to £2.14	£8.64
System life	Long	Limited
Reliability	High	High
Ease by which the system can be extended	Easy	Difficult
Temperature control - accuracy of maintaining temperature	Very good/high	Excellent
Suitability for large plant	Excellent	Moderate
Suitability for small plant	Moderate	Good
Ease of tracer installation	Moderate	Requires specialist skills
Cost of maintenance	Low	Moderate
Specialised maintenance staff requirement	No	Yes
Availability as turnkey project	Yes	Yes

Case study: UK oil refinery uses steam tracing for 4 km pipeline

In 1998, a steam trace heating system was installed at one of the UK's largest oil refineries.

Background

The oil company in question is involved in the export of a type of wax product. The wax has many uses, such as insulation in electric cabling, as a resin in corrugated paper and as a coating used to protect fresh fruit.

The wax has similar properties to candle wax. To enable it to be transported any distance in the form of a liquid, it needs to be maintained at a certain temperature. The refinery therefore required a pipeline with critical tracing.

The project required the installation of a 200 mm diameter product pipeline, which would run from a tank farm to a marine terminal out at sea – a pipeline of some 4 km in length.

The project began in April 1997, installation was completed in August 1998, and the first successful export of wax took place a month later.

Although the refinery management team was originally committed to an electric trace solution, they were persuaded to look at comparative design proposals and costings for both electric and steam trace options.

The wax application

The key parameter for this critical tracing application was to provide tight temperature control of the product at 80°C, but to have the ability to raise the temperature to 90°C for start-up or re-flow conditions. Other critical factors included the fact that the product would solidify at temperatures below 60°C, and spoil if subjected to temperatures above 120°C.

Steam was available on site at 9 bar g and 180°C, which immediately presented problems of excessive surface temperatures if conventional schedule 80 carbon steel trace pipework were to be used. This had been proposed by the contractor as a traditional steam trace solution for the oil company.

The total tracer tube length required was 11.5 km, meaning that the installation of carbon steel pipework would be very labour intensive, expensive and impractical. With all the joints involved it was not an attractive option.

However, today's steam tracing systems are highly advanced technologically. Spirax Sarco and their partner on the project, a specialist tracing firm, were able to propose two parallel runs of insulated copper tracer tube, which effectively put a layer of insulation between the product pipe and the steam tracer. This enabled the use of steam supply at 9 bar g, without the potential for hot spots which could exceed the critical 120°C product limitation.

The installation benefit was that as the annealed ductile steam tracer tubing used was available in continuous drum lengths, the proposed 50 m runs would have a limited number of joints, reducing the potential for future leaks from connectors.

This provided a reliable, low maintenance solution.

After comprehensive energy audit calculations, and the production of schematic installation drawings for costing purposes, together with some careful engineering, the proposal was to use the existing 9 bar g distribution system with 15 mm carbon steel pipework to feed the tracing system, together with strainers and temperature controls. Carbon steel condensate pipework was used together with lightweight tracing traps which minimised the need for substantial fabricated supports.

The typical tracer runs would be 50 m of twin isolated copper tracer tubing, installed at the 4 and 8 o'clock positions around the product pipe, held to the product pipeline with stainless steel strap banding at 300 mm intervals.

The material and installation costs for steam trace heating were about 30% less than the electric tracing option. In addition, ongoing running costs for the steam system would be a fraction of those for the electrical option.

Before the oil company management would commit themselves to a steam tracing system, they not only required an extended product warranty and a plant performance guarantee, but also insisted that a test rig should be built to prove the suitability of the self-acting controlled tracer for such an arduous application.

Spirax Sarco were able to assure them of the suitability of the design by referral to an existing installation elsewhere on their plant, where ten self-acting controllers were already installed and successfully working on the trace heating of pump transfer lines.

The oil company was then convinced of the benefits of steam tracing the wax product line and went on to install a steam tracing system.

Further in-depth surveys of the 4 km pipeline route were undertaken to enable full installation drawings to be produced. The company was also provided with on-site training for personnel on correct practices and installation procedures.

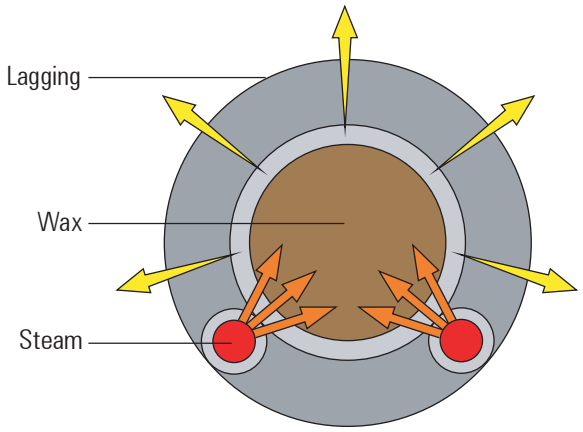


Fig. 1.2.4

After installation the heat load design was confirmed and the product was maintained at the required 80°C.

The oil company executives were impressed with the success of the project and chose to install steam tracing for another 300 m long wax product line in preference to electric tracing, even though they were initially convinced that electric tracing was the only solution for critical applications.

Questions

1. How does the cost of upgrading a steam system compare with installing a decentralised gas fired system?

a| It costs the same to upgrade the steam system. ☐

b| It costs twice as much to upgrade the steam system. ☐

c| It costs 75% as much to upgrade the steam system. ☐

d| It costs half as much to upgrade the steam system. ☐

2. Which of the following uses for steam could be found in a hospital?

a| Space heating. ☐

b| Sterilisation. ☐

c| Cooking. ☐

d| All of the above. ☐

3. Which of the following statements is true?

a| Steam creates a temperature gradient along the heat transfer surface, ensuring consistent product quality. ☐

b| Steam gives up its heat at a constant temperature without a gradient along the heat transfer surface, ensuring consistent product quality. ☐

c| High temperature oils offer a constant temperature along the heat transfer surface, which leads to poor product quality. ☐

d| High temperature oils can be directly injected into the product to be heated. ☐

4. A hot water calorifier can occupy much of a plant room. How much floor space does a modern steam to hot water packaged unit need if it is rated at 1200 kW?

a| 0.7 m² ☐

b| 7.0 m² ☐

c| 1.2 m² ☐

d| 12 m² ☐

5. Why is steam inexpensive to produce?

a| Steam boilers can use a variety of fuels. ☐

b| Steam boilers can utilise the heat from returned condensate. ☐

c| Steam boilers can be automated. ☐

d| All of the above. ☐

6. Which of the following statements best describes steam tracing?

a| Steam is injected into the process pipe to keep the contents moving. ☐

b| An electric jacket is used to heat the process piping. ☐

c| A steam tracer is a small steam pipe which runs along the outside of a process pipe. ☐

d| A tracer is a small water filled pipe which runs along the outside of a process pipe. ☐

Answers

1: d, 2: d, 3: b, 4: a, 5: d, 6: c

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Module 1.3

The Steam and Condensate Loop

The Steam and Condensate Loop

This Module of *The Steam and Condensate Loop* is intended to give a brief, non-technical overview of the steam plant. It offers an overall explanation of how the different parts of the steam plant relate to each other - and represents useful reading for anyone who is unfamiliar with the topic, prior to progressing to the next Block, or, indeed, before undertaking any form of detailed study of steam theory or steam plant equipment.

The boiler house

The boiler

The boiler is the heart of the steam system. The typical modern packaged boiler is powered by a burner which sends heat into the boiler tubes.

The hot gases from the burner pass backwards and forwards up to 3 times through a series of tubes to gain the maximum transfer of heat through the tube surfaces to the surrounding boiler water. Once the water reaches saturation temperature (the temperature at which it will boil at that pressure) bubbles of steam are produced, which rise to the water surface and burst. The steam is released into the space above, ready to enter the steam system. The stop or crown valve isolates the boiler and its steam pressure from the process or plant.

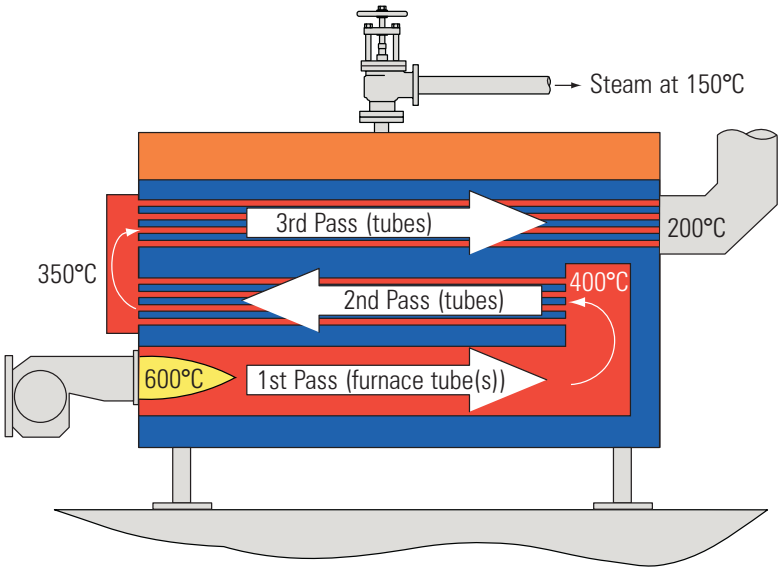


Fig. 1.3.1 Typical heat path through a smoke tube shell boiler

If steam is pressurised, it will occupy less space. Steam boilers are usually operated under pressure, so that more steam can be produced by a smaller boiler and transferred to the point of use using small bore pipework. When required, the steam pressure is reduced at the point of use. As long as the amount of steam being produced in the boiler is as great as that leaving the boiler, the boiler will remain pressurised. The burner will operate to maintain the correct pressure. This also maintains the correct steam temperature, because the pressure and temperature of saturated steam are directly related.

The boiler has a number of fittings and controls to ensure that it operates safely, economically, efficiently and at a consistent pressure.

Feedwater

The quality of water which is supplied into the boiler is important. It must be at the correct temperature, usually around 80°C, to avoid thermal shock to the boiler, and to keep it operating efficiently. It must also be of the correct quality to avoid damage to the boiler.



Fig. 1.3.2 A sophisticated feedtank system where the water is being heated by steam injection

Ordinary untreated potable water is not entirely suitable for boilers and can quickly cause them to foam and scale up. The boiler would become less efficient and the steam would become dirty and wet. The life of the boiler would also be reduced.

The water must therefore be treated with chemicals to reduce the impurities it contains.

Both feedwater treatment and heating take place in the feedtank, which is usually situated high above the boiler. The feedpump will add water to the boiler when required. Heating the water in the feedtank also reduces the amount of dissolved oxygen in it. This is important, as oxygenated water is corrosive.

Blowdown

Chemical dosing of the boiler feedwater will lead to the presence of suspended solids in the boiler. These will inevitably collect in the bottom of the boiler in the form of sludge, and are removed by a process known as bottom blowdown. This can be done manually - the boiler attendant will use a key to open a blowdown valve for a set period of time, usually twice a day.

Other impurities remain in the boiler water after treatment in the form of dissolved solids. Their concentration will increase as the boiler produces steam and consequently the boiler needs to be regularly purged of some of its contents to reduce the concentration. This is called control of total dissolved solids (TDS control). This process can be carried out by an automatic system which uses either a probe inside the boiler, or a small sensor chamber containing a sample of boiler water, to measure the TDS level in the boiler. Once the TDS level reaches a set point, a controller signals the blowdown valve to open for a set period of time. The lost water is replaced by feedwater with a lower TDS concentration, consequently the overall boiler TDS is reduced.

Level control

If the water level inside the boiler were not carefully controlled, the consequences could be catastrophic. If the water level drops too low and the boiler tubes are exposed, the boiler tubes could overheat and fail, causing an explosion. If the water level becomes too high, water could enter the steam system and upset the process.

For this reason, automatic level controls are used. To comply with legislation, level control systems also incorporate alarm functions which will operate to shut down the boiler and alert attention if there is a problem with the water level. A common method of level control is to use probes which sense the level of water in the boiler. At a certain level, a controller will send a signal to the feedpump which will operate to restore the water level, switching off when a predetermined level is reached. The probe will incorporate levels at which the pump is switched on and off, and at which low or high level alarms are activated. Alternative systems use floats.

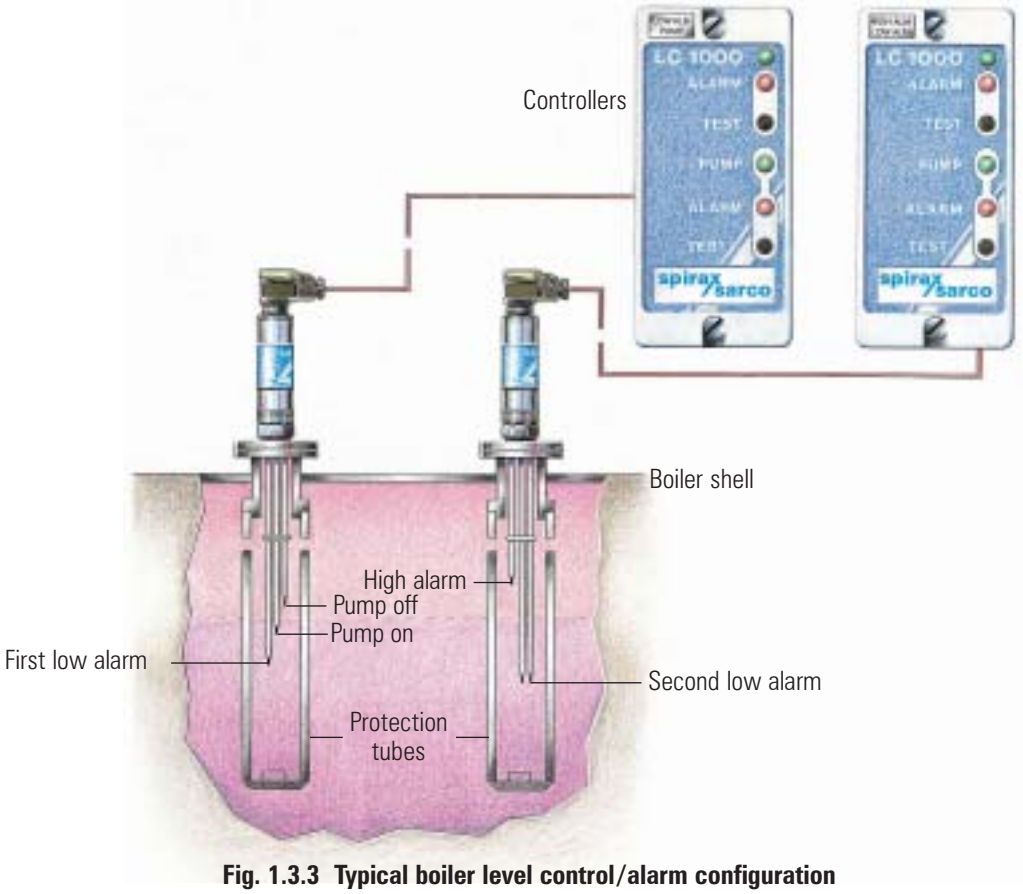


Fig. 1.3.3 Typical boiler level control/alarm configuration

It is a legal requirement in most countries to have two independent low level alarm systems.

The flow of steam to the plant

When steam condenses, its volume is dramatically reduced, which results in a localised reduction in pressure. This pressure drop through the system creates the flow of steam through the pipes.

The steam generated in the boiler must be conveyed through the pipework to the point where its heat energy is required. Initially there will be one or more main pipes or steam mains which carry steam from the boiler in the general direction of the steam using plant. Smaller branch pipes can then distribute the steam to the individual pieces of equipment.

Steam at high pressure occupies a lower volume than at atmospheric pressure. The higher the pressure, the smaller the bore of pipework required for distribution of a given mass of steam.

Steam quality

It is important to ensure that the steam leaving the boiler is delivered to the process in the right condition. To achieve this the pipework which carries the steam around the plant normally incorporates strainers, separators and steam traps.

A strainer is a form of sieve in the pipeline. It contains a mesh through which the steam must pass. Any passing debris will be retained by the mesh. A strainer should regularly be cleaned to avoid blockage. Debris should be removed from the steam flow because it can be very damaging to plant, and may also contaminate the final product.

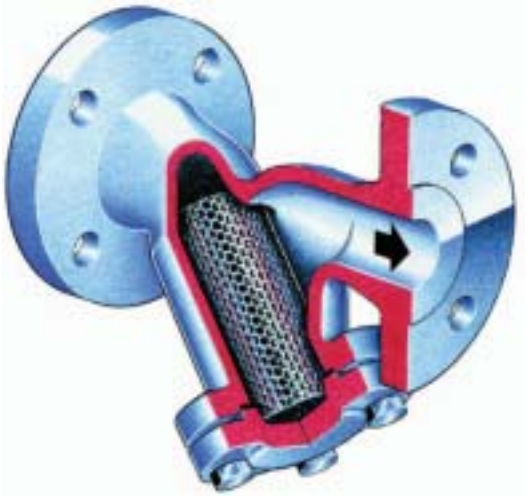


Fig. 1.3.4 Cut section of a strainer

The steam should be as dry as possible to ensure it is carrying heat effectively. A separator is a body in the pipeline which contains a series of plates or baffles which interrupt the path of the steam. The steam hits the plates, and any drops of moisture in the steam collect on them, before draining from the bottom of the separator.

Steam passes from the boiler into the steam mains. Initially the pipework is cold and heat is transferred to it from the steam. The air surrounding the pipes is also cooler than the steam, so the pipework will begin to lose heat to the air. Insulation fitted around the pipe will reduce this heat loss considerably.

When steam from the distribution system enters the steam using equipment the steam will again give up energy by: **a)** warming up the equipment and **b)** continuing to transfer heat to the process. As steam loses heat, it turns back into water. Inevitably the steam begins to do this as soon as it leaves the boiler. The water which forms is known as condensate, which tends to run to the bottom of the pipe and is carried along with the steam flow. This must be removed from the lowest points in the distribution pipework for several reasons:

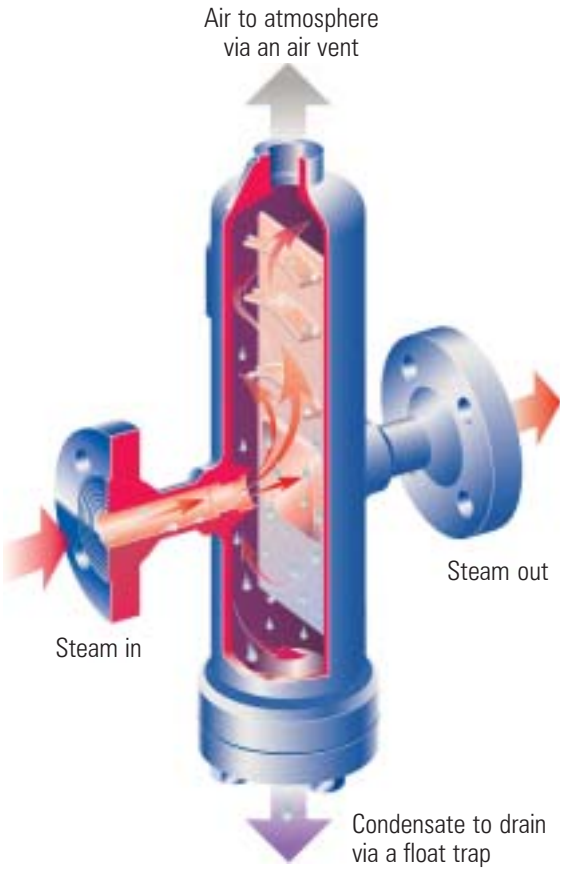


Fig. 1.3.5 Cut section of a separator showing operation

- Condensate does not transmit heat effectively. A film of condensate inside plant will reduce the efficiency with which heat is transferred.
- When air dissolves into condensate, it becomes corrosive.
- Accumulated condensate can cause noisy and damaging waterhammer.
- Inadequate drainage leads to leaking joints.

A device known as a steam trap is used to release condensate from the pipework whilst preventing the steam from escaping from the system. It can do this in several ways:

- A float trap uses the difference in density between steam and condensate to operate a valve. As condensate enters the trap, a float is raised and the float lever mechanism opens the main valve to allow condensate to drain. When the condensate flow reduces the float falls and closes the main valve, thus preventing the escape of steam.
- Thermodynamic traps contain a disc which opens to condensate and closes to steam.
- In bimetallic thermostatic traps, a bimetallic element uses the difference in temperature between steam and condensate to operate the main valve.
- In balanced pressure thermostatic traps, a small liquid filled capsule which is sensitive to heat operates the valve.

Once the steam has been employed in the process, the resulting condensate needs to be drained from the plant and returned to the boiler house. This process will be considered later in this Module.

Pressure reduction

As mentioned before, steam is usually generated at high pressure, and the pressure may have to be reduced at the point of use, either because of the pressure limitations of the plant, or the temperature limitations of the process.

This is achieved using a pressure reducing valve.

Steam at the point of use

A large variety of steam using plant exists. A few examples are described below:

- **Jacketed pan -** Large steel or copper pans used in the food and other industries to boil substances - anything from prawns to jam. These large pans are surrounded by a jacket filled with steam, which acts to heat up the contents.
- **Autoclave -** A steam-filled chamber used for sterilisation purposes, for example medical equipment, or to carry out chemical reactions at high temperatures and pressures, for example the curing of rubber.
- **Heater battery -** For space heating, steam is supplied to the coils in a heater battery. The air to be heated passes over the coils.
- **Process tank heating -** A steam filled coil in a tank of liquid used to heat the contents to the desired temperature.
- **Vulcaniser -** A large receptacle filled with steam and used to cure rubber.
- **Corrugator -** A series of steam heated rollers used in the corrugation process in the production of cardboard.
- **Heat exchanger -** For heating liquids for domestic/industrial use.

Control of the process

Any steam using plant will require some method to control the flow of steam. A constant flow of steam at the same pressure and temperature is often not what is required – a gradually increasing flow will be needed at start-up to gently warm the plant, and once the process reaches the desired temperature, the flow must be reduced.

Control valves are used to control the flow of steam. The actuator, see Figure 1.3.6, is the device that applies the force to open or close the valve. A sensor monitors conditions in the process, and transmits information to the controller. The controller compares the process condition with the set value and sends a corrective signal to the actuator, which adjusts the valve setting.

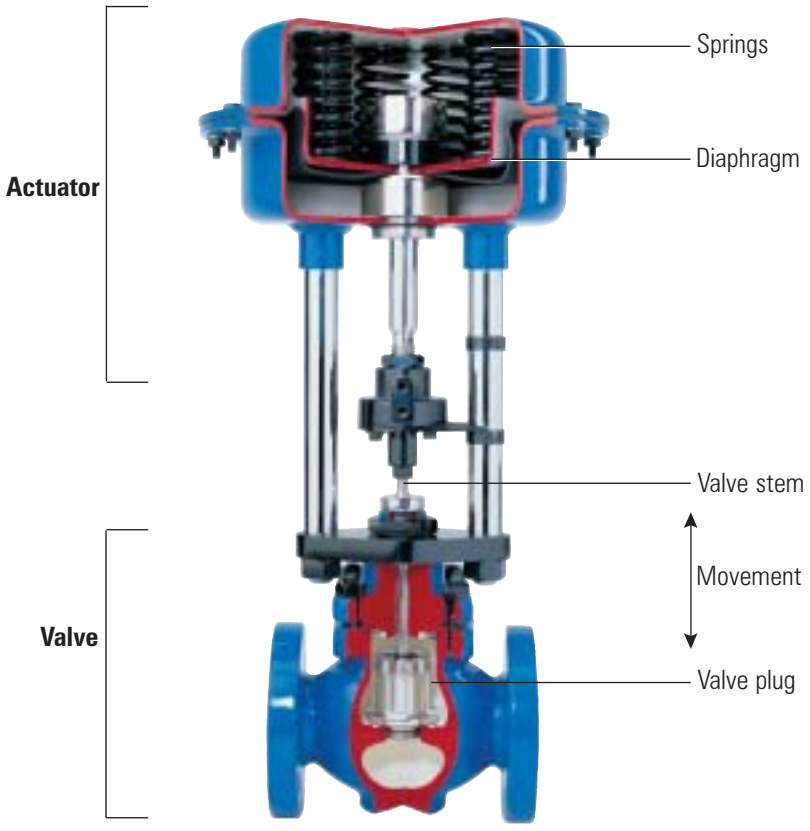


Fig. 1.3.6 A pneumatically operated two port control valve

A variety of control types exist:

- **Pneumatically actuated valves -** Compressed air is applied to a diaphragm in the actuator to open or close the valve.
- **Electrically actuated valves -** An electric motor actuates the valve.
- **Self-acting -** There is no controller as such - the sensor has a liquid fill which expands and contracts in response to a change in process temperature. This action applies force to open or close the valve.

Condensate removal from plant

Often, the condensate which forms will drain easily out of the plant through a steam trap. The condensate enters the condensate drainage system. If it is contaminated, it will probably be sent to drain. If not, the valuable heat energy it contains can be retained by returning it to the boiler feedtank. This also saves on water and water treatment costs.

Sometimes a vacuum may form inside the steam using plant. This hinders condensate drainage, but proper drainage from the steam space maintains the effectiveness of the plant. The condensate may then have to be pumped out.

Mechanical (steam powered) pumps are used for this purpose. These, or electric powered pumps, are used to lift the condensate back to the boiler feedtank.

A mechanical pump, see Figure 1.3.7, is shown draining an item of plant. As can be seen, the steam and condensate system represents a continuous loop.

Once the condensate reaches the feedtank, it becomes available to the boiler for recycling.

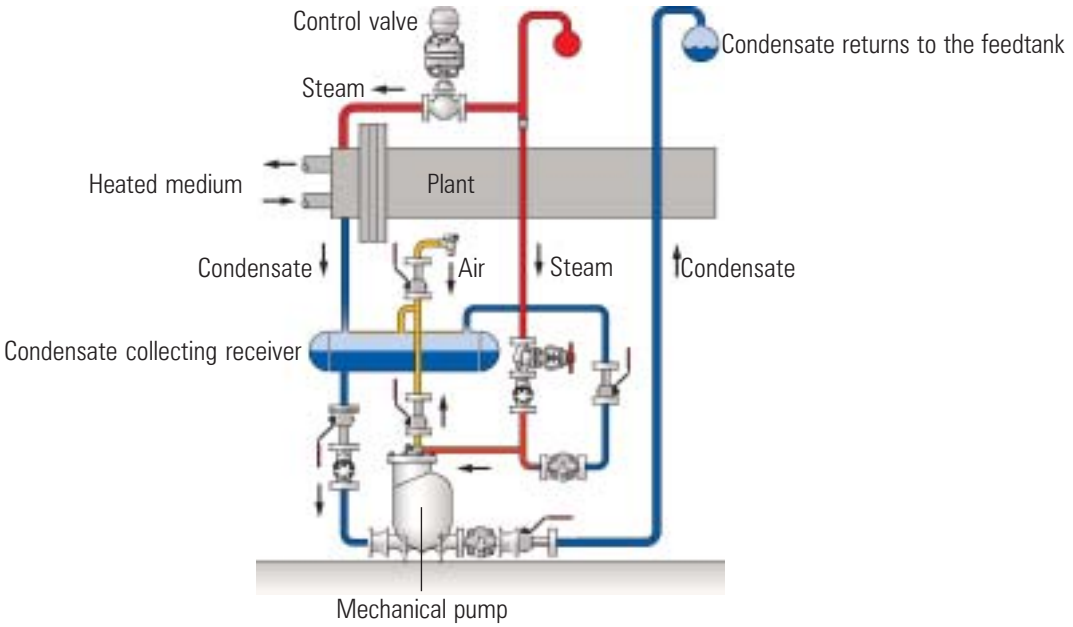


Fig. 1.3.7 Condensate recovery and return

Energy monitoring

In today's energy conscious environment, it is common for customers to monitor the energy consumption of their plant.

Steam flowmeters are used to monitor the consumption of steam, and used to allocate costs to individual departments or items of plant.

Questions

1. What is the purpose of the multi-flue passes in a boiler ?

a| To reduce the amount of flue gases exhausted ☐

b| To help produce drier steam ☐

c| To provide more even generation of steam bubbles ☐

d| To give a greater heat transfer area to the water ☐

2. What is the purpose of the boiler feedtank ?

a| To store chemically treated water for the boiler ☐

b| To provide a reservoir of hot water for the boiler ☐

c| To collect condensate returning from the plant ☐

d| All of the above ☐

3. The boiler feedtank is heated to approximately what temperature ?

a| 80°C ☐

b| 20°C ☐

c| Steam temperature ☐

d| It isn't heated, all heating takes place in the boiler ☐

4. What is the purpose of boiler bottom blowdown ?

a| To remove total dissolved solids in the boiler water ☐

b| To remove separated out oxygen ☐

c| To dilute the boiler water to reduce TDS ☐

d| To remove solids which collect in the bottom of the boiler ☐

5. What is used to remove suspended water particles in a steam main ?

a| A separator and steam trap ☐

b| A strainer and steam trap ☐

c| A strainer ☐

d| A reducing valve ☐

6. Which of the following is the purpose of a boiler automatic level control ?

a| To provide TDS control ☐

b| To maintain a specified level of water ☐

c| To comply with legislation ☐

d| To take corrective action if the boiler alarms sound ☐

Answers

1: d, 2: d, 3: a, 4: d, 5: a, 6: b

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Module 2.1

Engineering Units

Engineering Units

Throughout the engineering industries, many different definitions and units have been proposed and used for mechanical and thermal properties.

The problems this caused led to the development of an agreed international system of units (or SI units: Système International d'Unités). In the SI system there are seven well-defined base units from which the units of other properties can be derived, and these will be used throughout this publication.

The SI base units include length (in metres), mass (in kilograms), time (in seconds) and temperature (in kelvin). The first three will hopefully need no further explanation, while the latter will be discussed in more detail later.

The other SI base units are electric current (in amperes), amount of substance (in moles) and luminous intensity (in candela). These may be familiar to readers with a background in electronics, chemistry and physics respectively, but have little relevance to steam engineering nor the contents of *The Steam and Condensate Loop.*

Table 2.1.1 shows the derived units that are relevant to this subject, all of which should be familiar to those with any general engineering background. These quantities have all been assigned special names after famous pioneers in the development of science and engineering.

Table 2.1.1 Named quantities in derived SI units

Quantity	Name	Symbol	SI base unit	Derived unit
Area	square metre	A	m^2	-
Volume	cubic metre	V	m^3	-
Velocity	metre per second	u	m/s	-
Acceleration	metre per second squared	a	m/s^2	-
Force	newton	N	$kg\ m/s^2$	J/m
Energy	joule	J	$kg\ m^2/s^2$	N m
Pressure or stress	pascal	Pa	$kg\ m/s^2$	N/m^2
Power	watt	W	$kg\ m^2/s^3$	J/s

There are many other quantities that have been derived from SI base units, which will also be of significance to anyone involved in steam engineering. These are provided in Table 2.1.2.

Table 2.1.2 Other quantities in derived SI units

Quantity	SI base unit	Derived unit
Mass density	kg/m^3	kg/m^3
Specific volume (v_g)	m^3/kg	m^3/kg
Specific enthalpy (h)	m^2/s^2	J/kg
Specific heat capacity (c_p)	m^2/s^2K	J/kg K
Specific entropy	m^2/s^2K	J/kg K
Heat flowrate	$m^2\ kg/s^3$	J/s or W
Dynamic viscosity	kg/m s	N s/m^2

Dot notation

This convention is used to identify a compound unit incorporating rate, for example:

m = Mass (e.g. kg)
$\dot{m}$ = Mass flow per time unit (e.g. kg/h) = Mass flowrate

Multiples and submultiples

Table 2.1.3 gives the SI prefixes that are used to form decimal multiples and submultiples of SI units. They allow very large or very small numerical values to be avoided. A prefix attaches directly to the name of a unit, and a prefix symbol attaches directly to the symbol for a unit.

In summary: one thousand metres may be shown as 1 km, 1000 m or 10^3 m.

Table 2.1.3 Multiples and submultiples used with SI units

Multiples			Submultiples		
Factor	**Prefix**	**Symbol**	**Factor**	**Prefix**	**Symbol**
10^{12}	tera	T	10^{-3}	milli	m
10^{9}	giga	G	10^{-6}	micro	µ
10^{6}	mega	M	10^{-9}	nano	n
10^{3}	kilo	k	10^{-12}	pico	P

Special abbreviations used in steam flowmetering applications

For historical reasons, International Standard ISO 5167 (supersedes BS 1042) which refers to flowmetering, use the following abbreviations in Table 2.1.4.

Table 2.1.4 Symbols used in flowmetering applications

Symbol	Definition	Unit
q_m	Mass flowrate	kg/s or kg/h
q_v	Volume flowrate	m^3/s
Q_L	Liquid flowrate	l/min
Q_S	Gas flowrate at STP	l/min
Q_F	Gas flowrate actual	l/min
Q_E	Equivalent water flowrate	l/min
D_S	Density of gas at STP	kg/m^3
D_F	Density of gas actual	kg/m^3
P_S	Standard pressure (1.013 bar a)	bar a
P_F	Actual flow pressure	bar a
T_S	Standard temperature	°C
T_F	Actual flow temperature	°C

STP - Standard temperature and pressure

These are the standard conditions for measurement of the properties of matter. The standard temperature is the freezing point of pure water, 0°C or 273.15°K. The standard pressure is the pressure exerted by a column of mercury (symbol Hg) 760 mm high, often designated 760 mm Hg. This pressure is also called one atmosphere and is equal to 1.01325 x 10^6 dynes per square centimetre, or approximately 14.7 lb per square inch. The density (mass per volume) of a gas is usually reported as its value at STP. Properties that cannot be measured at STP are measured under other conditions; usually the values obtained are then mathematically extrapolated to their values at STP.

Symbols

Table 2.1.5 shows the symbols and typical units used in *The Steam and Condensate Loop.*

Table 2.1.5 Symbols and units of measure used in *The Steam and Condensate Loop*

Symbol	Definition	Unit
A	Cross sectional area of a conduit, for the operating condition	m^2 or mm^2
c_P	Specific heat capacity at constant pressure	kJ/kg °C or kJ/kg K
C_V	Specific heat capacity at constant volume	kJ/m^3 °C or kJ/m^3 K
D	Diameter of the circular cross section of a conduit	m or mm
d	Orifice diameter	m or mm
g	Acceleration due to gravity	9.81 m/s^2
Hz	The unit of frequency (number of cycles per second)	Hz or kHz
J	Joule, the unit of energy	J or kJ
L	Length	m
M	Molar mass of a fluid	kg/mol
N	Newton, the unit of force	N or kN
Pa	Unit of pressure (Pascal)	Pa or kPa
p	Static pressure of a fluid	bar or kPa
Δp	Differential pressure	bar or kPa
m	Fundamental unit of length (metre)	m
m	Mass	kg
$\dot{m}$	Mass flowrate	kg/s or kg/h
$\dot{m}_s$	Steam mass flowrate	kg/s or kg/h
Q	Quantity of heat	kJ
$\dot{Q}$	Heat transfer rate	kJ/s (kW)
R	Radius	m or mm
Re_D	Reynolds number referred to diameter D	Dimensionless
s	Fundamental unit of time (second)	
Sr	Strouhal number	Dimensionless
σ	Stress	N/m^2
T_S	Steam temperature	K or °C
T_L	Liquid (or product) temperature	K or °C
ΔT	Temperature difference or change	K or °C
t	Time	s or h
u	Velocity of a fluid	m/s
μ	Dynamic viscosity of a fluid	Pa s or cP
ν	Kinematic viscosity	cSt
ρ	Density of a fluid	kg/m^3
$\dot{V}$	Volume flowrate	m^3/s or m^3/h
W	Unit of energy flow (Watt)	W (J/s)
V (vg)	Volume (Specific volume)	m^3 (m^3/kg)
H (hg)	Enthalpy (Specific enthalpy)	kJ (kJ/kg)
S (sg)	Entropy (Specific entropy)	kJ K (kJ/kg K)
U (ug)	Internal energy (specific internal energy)	kJ (kJ/kg)

Subscripts used with properties

When using enthalpy, entropy and internal energy, subscripts as shown below are used to identify the phase, for example:

Subscript **f** = Fluid or liquid state, for example h_f: liquid enthalpy

Subscript **fg** = Change of state liquid to gas, for example h_{fg}: enthalpy of evaporation

Subscript **g** = Total, for example h_g: total enthalpy

Note that, by convention, the total heat in superheated steam is signified by **h**.

It is also usual, by convention, to signify sample quantities in capital letters, whilst unit quantities are signified in lower case letters.

For example:
Total enthalpy in a sample of superheated steam H kJ
Specific enthalpy of superheated steam h kJ/kg

Temperature

The temperature scale is used as an indicator of thermal equilibrium, in the sense that any two systems in contact with each other with the same value are in thermal equilibrium.

The Celsius (°C) scale

This is the scale most commonly used by the engineer, as it has a convenient (but arbitrary) zero temperature, corresponding to the temperature at which water will freeze.

The absolute or K (kelvin) scale

This scale has the same increments as the Celsius scale, but has a zero corresponding to the minimum possible temperature when all molecular and atomic motion has ceased. This temperature is often referred to as absolute zero (0 K) and is equivalent to -273.15°C.

The two scales of temperature are interchangeable, as shown in Figure 2.1.1 and expressed in Equation 2.1.1.

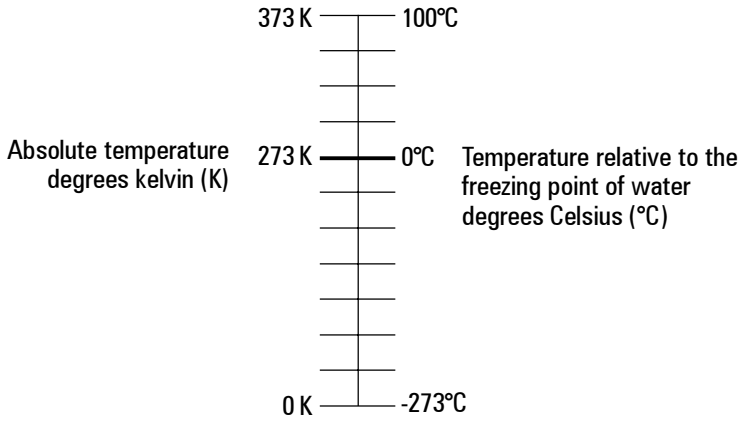


Fig. 2.1.1 Comparison of absolute and gauge temperatures

$$T\,(K) = \text{Temperature}\,(°C) + 273.15 \qquad \textbf{Equation 2.1.1}$$

The SI unit of temperature is the kelvin, which is defined as $1 \div 273.15$ of the thermodynamic temperature of pure water at its triple point (0.01°C). An explanation of triple point is given in Module 2.2.

Most thermodynamic equations require the temperature to be expressed in kelvin. However, temperature difference, as used in many heat transfer calculations, may be expressed in either °C or K. Since both scales have the same increments, a temperature difference of 1°C has the same value as a temperature difference of 1 K.

Pressure

The SI unit of pressure is the pascal (Pa), defined as 1 newton of force per square metre (1 N/m²). As Pa is such a small unit the kPa (1 kilonewton/m²) or MPa (1 Meganewton/m²) tend to be more appropriate to steam engineering.

However, probably the most commonly used metric unit for pressure measurement in steam engineering is the bar. This is equal to 10^5 N/m², and approximates to 1 atmosphere. This unit is used throughout this publication.

Other units often used include lb/in² (psi), kg/cm², atm, in H_2O and mm Hg. Conversion factors are readily available from many sources.

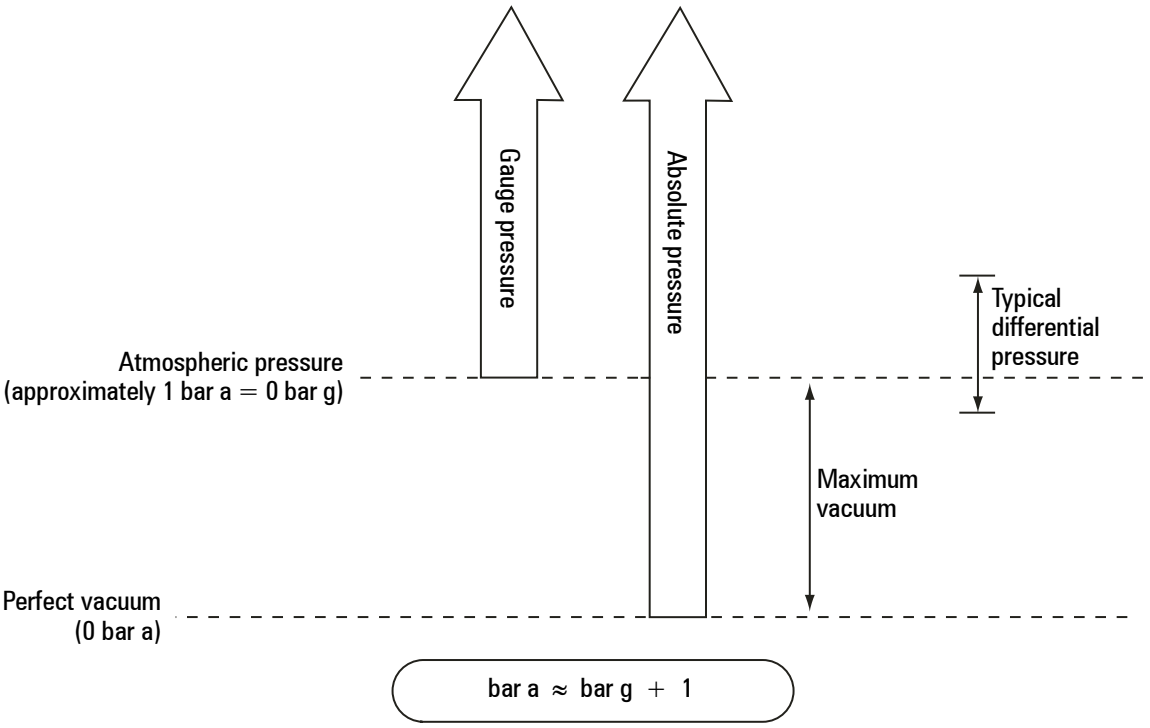


Fig. 2.1.2 Comparison of absolute and gauge pressures

Absolute pressure (bar a)

This is the pressure measured from the datum of a perfect vacuum i.e. a perfect vacuum has a pressure of 0 bar a.

Gauge pressure (bar g)

This is the pressure measured from the datum of the atmospheric pressure. Although in reality the atmospheric pressure will depend upon the climate and the height above sea level, a generally accepted value of 1.013 25 bar a (1 atm) is often used. This is the average pressure exerted by the air of the earth's atmosphere at sea level.

Gauge pressure = Absolute pressure - Atmospheric pressure

Pressures above atmospheric will always yield a positive gauge pressure. Conversely a vacuum or negative pressure is the pressure below that of the atmosphere. A pressure of -1 bar g corresponds closely to a perfect vacuum.

Differential pressure

This is simply the difference between two pressures. When specifying a differential pressure, it is not necessary to use the suffixes 'g' or 'a' to denote either gauge pressure or absolute pressure respectively, as the pressure datum point becomes irrelevant.

Therefore, the difference between two pressures will have the same value whether these pressures are measured in gauge pressure or absolute pressure, as long as the two pressures are measured from the same datum.

Density and specific volume

The density (ρ) of a substance can be defined as its mass (m) per unit volume (V). The specific volume (v_g) is the volume per unit mass and is therefore the inverse of density. In fact, the term 'specific' is generally used to denote a property of a **unit mass** of a substance (see Equation 2.1.2).

$$\rho = \frac{m}{V} = \frac{1}{v_g} \qquad \textbf{Equation 2.1.2}$$

Where:
ρ = Density (kg/m³)
m = Mass (kg)
V = Volume (m³)
v_g = Specific volume (m³/kg)

The SI units of density (ρ) are kg/m³, conversely, the units of specific volume (v_g) are m³/kg.

Another term used as a measure of density is specific gravity. It is a ratio of the density of a substance (ρ_s) and the density of pure water (ρ_w) at standard temperature and pressure (STP). This reference condition is usually defined as being at atmospheric pressure and 0°C. Sometimes it is said to be at 20°C or 25°C and is referred to as normal temperature and pressure (NTP).

$$\text{Specific gravity} = \frac{\text{Density of substance } \rho_s}{\text{Density of water } \rho_w} \qquad \textbf{Equation 2.1.3}$$

The density of water at these conditions is approximately 1 000 kg/m³. Therefore substances with a density greater than this value will have a specific gravity greater than 1, whereas substances with a density less than this will have a specific gravity of less than 1.

Since specific gravity is a ratio of two densities, it is a dimensionless variable and has no units. Therefore in this case the term specific does not indicate it is a property of a unit mass of a substance. Specific gravity is also sometimes known as the relative density of a substance.

Heat, work and energy

Energy is sometimes described as the ability to do work. The transfer of energy by means of mechanical motion is called work. The SI unit for work and energy is the joule, defined as 1 N m.

The amount of mechanical work carried out can be determined by an equation derived from Newtonian mechanics:

Work = Force x Displacement

It can also be described as the product of the applied pressure and the displaced volume:

Work = Applied pressure x Displaced volume

Example 2.1.1
An applied pressure of 1 Pa (or 1 N/m²) displaces a volume of 1 m³. How much work has been done?

Work done = 1 N/m² x 1 m³ = 1 N m (or 1 J)

The benefits of using SI units, as in the above example, is that the units in the equation actually cancel out to give the units of the product.

The experimental observations of J. P. Joule established that there is an equivalence between mechanical energy (or work) and heat. He found that the same amount of energy was required to produce the same temperature rise in a specific mass of water, regardless of whether the energy was supplied as heat or work.

The total energy of a system is composed of the internal, potential and kinetic energy. The temperature of a substance is directly related to its internal energy (u_g). The internal energy is associated with the motion, interaction and bonding of the molecules within a substance. The external energy of a substance is associated with its velocity and location, and is the sum of its potential and kinetic energy.

The transfer of energy as a result of the difference in temperature alone is referred to as heat flow. The watt, which is the SI unit of power, can be defined as 1 J/s of heat flow.

Other units used to quantify heat energy are the British Thermal Unit (Btu: the amount of heat to raise 1 lb of water by 1°F) and the kilocalorie (the amount of heat to raise 1 kg of water by 1°C). Conversion factors are readily available from numerous sources.

Specific enthalpy

This is the term given to the total energy, due to both pressure and temperature, of a fluid (such as water or steam) at any given time and condition. More specifically it is the sum of the internal energy and the work done by an applied pressure (as in Example 2.1.1).

The basic unit of measurement is the joule (J). Since one joule represents a very small amount of energy, it is usual to use kilojoules (kJ = 1 000 joules).

The specific enthalpy is a measure of the total energy of a unit mass, and its units are usually kJ/kg.

Specific heat capacity

The enthalpy of a fluid is a function of its temperature and pressure. The temperature dependence of the enthalpy can be found by measuring the rise in temperature caused by the flow of heat at constant pressure. The constant-pressure heat capacity c_p, is a measure of the change in enthalpy at a particular temperature.

Similarly, the internal energy is a function of temperature and specific volume. The constant-volume heat capacity c_v, is a measure of the change in internal energy at a particular temperature and constant volume.

Because the specific volumes of solids and liquids are generally smaller, then unless the pressure is extremely high, the work done by an applied pressure can be neglected. Therefore, if the enthalpy can be represented by the internal energy component alone, the constant-volume and constant-pressure heat capacities can be said to be equal.

Therefore, for solids and liquids: $c_p \approx c_v$

Another simplification for solids and liquids assumes that they are incompressible, so that their volume is only a function of temperature. This implies that for incompressible fluids the enthalpy and the heat capacity are also only functions of temperature.

The specific heat capacity represents the amount of energy required to raise 1 kg by 1°C, and can be thought of as the ability of a substance to absorb heat. Therefore the SI units of specific heat capacity are kJ/kg K (kJ/kg °C). Water has a large specific heat capacity (4.19 kJ/kg °C) compared with many fluids, which is why both water and steam are considered to be good carriers of heat.

The amount of heat energy required to raise the temperature of a substance can be determined from Equation 2.1.4.

$$Q = m\, c_p\, \Delta T \qquad \textbf{Equation 2.1.4}$$

Where:
Q = Quantity of energy (kJ)
m = Mass of the substance (kg)
c_p = Specific heat capacity of the substance (kJ/kg °C)
ΔT = Temperature rise of the substance (°C)

This equation shows that for a given mass of substance, the temperature rise is linearly related to the amount of heat provided, assuming that the specific heat capacity is constant over that temperature range.

Example 2.1.2
Consider a quantity of water with a volume of 2 litres, raised from a temperature of 20°C to 70°C.

At atmospheric pressure, the density of water is approximately 1 000 kg/m³. As there are 1 000 litres in 1 m³, then the density can be expressed as 1 kg per litre (1 kg/l). Therefore the mass of the water is 2 kg.

The specific heat capacity for water can be taken as 4.19 kJ/kg °C over low ranges of temperature.

Therefore: $Q = 2 \text{ kg} \times 4.19 \text{ kJ/kg°C} \times (70 - 20)\text{°C} = 419 \text{ kJ}$

If the water was then cooled to its original temperature of 20°C, it would also release this amount of energy in the cooling application.

Entropy (S)

Entropy is a measure of the degree of disorder within a system. The greater the degree of disorder, the higher the entropy. The SI units of entropy are kJ/kg K (kJ/kg °C).

In a solid, the molecules of a substance arrange themselves in an orderly structure. As the substance changes from a solid to a liquid, or from a liquid to a gas, the arrangement of the molecules becomes more disordered as they begin to move more freely. For any given substance the entropy in the gas phase is greater than that of the liquid phase, and the entropy in the liquid phase is more than in the solid phase.

One characteristic of all natural or spontaneous processes is that they proceed towards a state of equilibrium. This can be seen in the second law of thermodynamics, which states that heat cannot pass from a colder to a warmer body.

A change in the entropy of a system is caused by a change in its heat content, where the change of entropy is equal to the heat change divided by the average absolute temperature, Equation 2.1.5.

$$\text{Change in entropy } (\Delta S) = \frac{\text{Change in enthalpy } (\Delta H)}{\text{Average absolute temperature } (\Delta T)} \qquad \textbf{Equation 2.1.5}$$

When unit mass calculations are made, the symbols for entropy and enthalpy are written in lower case, Equation 2.1.6.

$$\text{Change in specific entropy } (\Delta s) = \frac{\text{Change in specific enthalpy } (\Delta h)}{\text{Average absolute temperature } (\Delta T)} \qquad \textbf{Equation 2.1.6}$$

To look at this in further detail, consider the following examples:

Example 2.1.3
A process raises 1 kg of water from 0 to 100°C (273 to 373 K) under atmospheric conditions.

Specific enthalpy at 0°C (h_f) = 0 kJ/kg (from steam tables)

Specific enthalpy of water at 100°C (h_f) = 419 kJ/kg (from steam tables)

Calculate the change in specific entropy

Since this is a change in specific entropy of water, the symbol 's' in Equation 2.1.6 takes the suffix 'f' to become s_f.

Calculate: $\text{Change in specific entropy } (\Delta s_f) = \dfrac{\text{Change in specific enthalpy } (\Delta h)}{\text{Average absolute temperature } (\Delta T)}$

Therefore:

$$\Delta s_f = \frac{419 - 0}{\left(\frac{273 + 373}{2}\right)}$$

$$\Delta s_f = \frac{419}{323}$$

$$\Delta s_f = \mathbf{1.297 \text{ kJ/kg K}}$$

Example 2.1.4

A process changes 1 kg of water at 100°C (373 K) to saturated steam at 100°C (373 K) under atmospheric conditions.

Calculate the change in specific entropy of evaporation

Since this is the entropy involved in the change of state, the symbol 's' in Equation 2.1.6 takes the suffix '$_{fg}$' to become s_{fg}.

Specific enthalpy of evaporation of steam at 100°C (373 K) (h_{fg}) = 2 258 kJ/kg (from steam tables)

Specific enthalpy of evaporation of water at 100°C (373 K) (h_{fg}) = 0 kJ/ks (from steam tables)

Calculate:

$$\text{Change in specific entropy } (\Delta s_{fg}) = \frac{\text{Change in specific enthalpy } (\Delta h)}{\text{Average absolute temperature } (\Delta T)}$$

Therefore:

$$\text{Change in specific entropy } \Delta s_{fg} = \frac{2\,258 - 0}{\left(\frac{373 + 373}{2}\right)}$$

$$\Delta s_{fg} = \frac{2\,258}{373}$$

$$\mathbf{\Delta s_{fg} = 6.054\ kJ/kg\ K}$$

The total change in specific entropy from water at 0°C to saturated steam at 100°C is the sum of the change in specific entropy for the water, plus the change of specific entropy for the steam, and takes the suffix '$_g$' to become the total change in specific entropy s_g.

Therefore:

$$\text{Change in specific entropy } (\Delta s_g) = \Delta s_f + \Delta s_{fg}$$

$$\Delta s_g = 1.297 \text{ (from Example 2.1.3)} + 6.054 \text{ (from above)}$$

$$\mathbf{\Delta s_g = 7.351\ kJ/kg\ K}$$

Example 2.1.5
A process superheats 1 kg of saturated steam at atmospheric pressure to 150°C (423 K). Determine the change in entropy.

Specific total enthalpy of steam at atmospheric pressure and at 100°C (373 K) (h_g) = 2 675 kJ/kg (from steam tables)

Specific total enthalpy of steam at atmospheric pressure and at 150°C (423 K) (h) = 2 777 kJ/kg (from steam tables)

Change in specific enthalpy (Δh) = 102 kJ/kg

$$\text{Average absolute temperature} = \frac{373 + 423}{2}$$

Average absolute temperature = 398 K

$$\text{Change in specific entropy } (\Delta s) = \frac{\text{Change in specific enthalpy } (\Delta h)}{\text{Average absolute temperature } (\Delta T)} \qquad \textbf{Equation 2.1.6}$$

$$\text{Change in specific entropy } (\Delta s) = \frac{102}{398}$$

Change in specific entropy (Δs) = 0.256 kJ/kg K

Total change in specific entropy (Δs) = Δs_g + additional entropy due to superheating (Δs)

The change in total specific entropy = 7.351 kJ/kg K (from Example 2.1.4) + 0.256 kJ/kg K

The total change in specific entropy = 7.607 kJ/kg K

As the entropy of saturated water is measured from a datum of 0.01°C, the entropy of water at 0°C can, for practical purposes, be taken as zero. The total change in specific entropy in this example is based on an initial water temperature of 0°C, and therefore the final result happens to be very much the same as the specific entropy of steam that would be observed in steam tables at the final condition of steam at atmospheric pressure and 150°C.

Entropy is discussed in greater detail in Module 2.15, Entropy - A Basic Understanding, and in Module 2.16, Entropy - Its Practical Use.

Questions

1. Given water has a specific heat capacity of 4.19 kJ/kg °C, what quantity of heat is required to raise the temperature of 2 500 l of water from 10°C to 80°C?

a| 733 250 kJ ☐

b| 175 000 kJ ☐

c| 175 kJ ☐

d| 41 766 kJ ☐

2. A pressure of 10 bar absolute is specified. What is the equivalent pressure in gauge units?

a| 8 bar g ☐

b| 11 bar g ☐

c| 9 bar g ☐

d| 12 bar g ☐

3. A valve has an upstream pressure of 8 bar absolute and a downstream pressure of 5 bar g. What is the pressure differential across the valve?

a| 3 bar ☐

b| 4 bar ☐

c| 7 bar ☐

d| 2 bar ☐

4. What quantity of heat is given up when 1 000 l of water is cooled from 50°C to 20°C?

a| 125 700 kJ ☐

b| 30 000 KJ ☐

c| 30 000 kJ/kg ☐

d| 125 700 kJ/kg ☐

5. 500 l of fuel oil is to be heated from 25°C to 65°C. The oil has a relative density of 0.86 and a specific heat capacity of 1.88 kJ/kg°C. How much heat will be required?

a| 17 200 kJ ☐

b| 37 600 kJ ☐

c| 32 336 kJ ☐

d| 72 068 kJ ☐

6. A thermometer reads 160°C. What is the equivalent temperature in K?

a| 433 K ☐

b| 192 K ☐

c| 113 K ☐

d| 260 K ☐

Answers

1: a, 2: c, 3: d, 4: a, 5: c, 6: a

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Module 2.2

What is Steam?

What is Steam?

A better understanding of the properties of steam may be achieved by understanding the general molecular and atomic structure of matter, and applying this knowledge to ice, water and steam.

A molecule is the smallest amount of any element or compound substance still possessing all the chemical properties of that substance which can exist. Molecules themselves are made up of even smaller particles called atoms, which define the basic elements such as hydrogen and oxygen.

The specific combinations of these atomic elements provide compound substances. One such compound is represented by the chemical formula H_2O, having molecules made up of two atoms of hydrogen and one atom of oxygen.

The reason water is so plentiful on the earth is because hydrogen and oxygen are amongst the most abundant elements in the universe. Carbon is another element of significant abundance, and is a key component in all organic matter.

Most mineral substances can exist in the three physical states (solid, liquid and vapour) which are referred to as phases. In the case of H_2O, the terms ice, water and steam are used to denote the three phases respectively.

The molecular structure of ice, water, and steam is still not fully understood, but it is convenient to consider the molecules as bonded together by electrical charges (referred to as the hydrogen bond). The degree of excitation of the molecules determines the physical state (or phase) of the substance.

Triple point

All the three phases of a particular substance can only coexist in equilibrium at a certain temperature and pressure, and this is known as its triple point.

The triple point of H_2O, where the three phases of ice, water and steam are in equilibrium, occurs at a temperature of 273.16 K and an absolute pressure of 0.006 112 bar. This pressure is very close to a perfect vacuum. If the pressure is reduced further at this temperature, the ice, instead of melting, sublimates directly into steam.

Ice

In ice, the molecules are locked together in an orderly lattice type structure and can only vibrate. In the solid phase, the movement of molecules in the lattice is a vibration about a mean bonded position where the molecules are less than one molecular diameter apart.

The continued addition of heat causes the vibration to increase to such an extent that some molecules will eventually break away from their neighbours, and the solid starts to melt to a liquid state. At atmospheric pressure, melting occurs at 0°C. Changes in pressure have very little effect on the melting temperature, and for most practical purposes, 0°C can be taken as the melting point. However, it has been shown that the melting point of ice falls by 0.0072°C for each additional atmosphere of pressure. For example, a pressure of 13.9 bar g would be needed to reduce the melting temperature by 0.1°C.

Heat that breaks the lattice bonds to produce the phase change while not increasing the temperature of the ice, is referred to as enthalpy of melting or heat of fusion. This phase change phenomenon is reversible when freezing occurs with the same amount of heat being released back to the surroundings.

For most substances, the density decreases as it changes from the solid to the liquid phase. However, H_2O is an exception to this rule as its density increases upon melting, which is why ice floats on water.

Water

In the liquid phase, the molecules are free to move, but are still less than one molecular diameter apart due to mutual attraction, and collisions occur frequently. More heat increases molecular agitation and collision, raising the temperature of the liquid up to its boiling temperature.

Enthalpy of water, liquid enthalpy or sensible heat (h_f) of water

This is the heat energy required to raise the temperature of water from a datum point of 0°C to its current temperature.

At this reference state of 0°C, the enthalpy of water has been arbitrarily set to zero. The enthalpy of all other states can then be identified, relative to this easily accessible reference state.

Sensible heat was the term once used, because the heat added to the water produced a change in temperature. However, the accepted terms these days are liquid enthalpy or enthalpy of water.

At atmospheric pressure (0 bar g), water boils at 100°C, and 419 kJ of energy are required to heat 1 kg of water from 0°C to its boiling temperature of 100°C. It is from these figures that the value for the specific heat capacity of water (C_p) of 4.19 kJ/kg °C is derived for most calculations between 0°C and 100°C.

Steam

As the temperature increases and the water approaches its boiling condition, some molecules attain enough kinetic energy to reach velocities that allow them to momentarily escape from the liquid into the space above the surface, before falling back into the liquid.

Further heating causes greater excitation and the number of molecules with enough energy to leave the liquid increases. As the water is heated to its boiling point, bubbles of steam form within it and rise to break through the surface.

Considering the molecular structure of liquids and vapours, it is logical that the density of steam is much less than that of water, because the steam molecules are further apart from one another. The space immediately above the water surface thus becomes filled with less dense steam molecules.

When the number of molecules leaving the liquid surface is more than those re-entering, the water freely evaporates. At this point it has reached boiling point or its saturation temperature, as it is saturated with heat energy.

If the pressure remains constant, adding more heat does not cause the temperature to rise any further but causes the water to form saturated steam. The temperature of the boiling water and saturated steam within the same system is the same, but the heat energy per unit mass is much greater in the steam.

At atmospheric pressure the saturation temperature is 100°C. However, if the pressure is increased, this will allow the addition of more heat and an increase in temperature without a change of phase.

Therefore, increasing the pressure effectively increases both the enthalpy of water, and the saturation temperature. The relationship between the saturation temperature and the pressure is known as the steam saturation curve (see Figure 2.2.1).

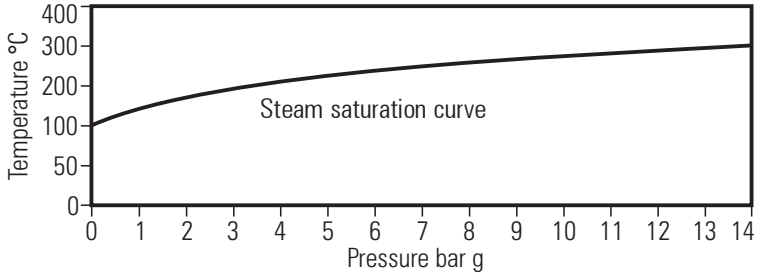


Fig. 2.2.1 Steam saturation curve

Water and steam can coexist at any pressure on this curve, both being at the saturation temperature. Steam at a condition above the saturation curve is known as superheated steam:

- Temperature above saturation temperature is called the degree of superheat of the steam.
- Water at a condition below the curve is called sub-saturated water.

If the steam is able to flow from the boiler at the same rate that it is produced, the addition of further heat simply increases the rate of production. If the steam is restrained from leaving the boiler, and the heat input rate is maintained, the energy flowing into the boiler will be greater than the energy flowing out. This excess energy raises the pressure, in turn allowing the saturation temperature to rise, as the temperature of saturated steam correlates to its pressure.

Enthalpy of evaporation or latent heat (h_{fg})

This is the amount of heat required to change the state of water at its boiling temperature, into steam. It involves no change in the temperature of the steam/water mixture, and all the energy is used to change the state from liquid (water) to vapour (saturated steam).

The old term latent heat is based on the fact that although heat was added, there was no change in temperature. However, the accepted term is now enthalpy of evaporation.

Like the phase change from ice to water, the process of evaporation is also reversible. The same amount of heat that produced the steam is released back to its surroundings during condensation, when steam meets any surface at a lower temperature.

This may be considered as the useful portion of heat in the steam for heating purposes, as it is that portion of the total heat in the steam that is extracted when the steam condenses back to water.

Enthalpy of saturated steam, or total heat of saturated steam

This is the total energy in saturated steam, and is simply the sum of the enthalpy of water and the enthalpy of evaporation.

$$h_g = h_f + h_{fg} \qquad \textbf{Equation 2.2.1}$$

Where:
h_g = Total enthalpy of saturated steam (Total heat) (kJ/kg)
h_f = Liquid enthalpy (Sensible heat) (kJ/kg)
h_{fg} = Enthalpy of evaporation (Latent heat) (kJ/kg)

The enthalpy (and other properties) of saturated steam can easily be referenced using the tabulated results of previous experiments, known as steam tables.

The saturated steam tables

The steam tables list the properties of steam at varying pressures. They are the results of actual tests carried out on steam. Table 2.2.1 shows the properties of dry saturated steam at atmospheric pressure - 0 bar g.

Table 2.2.1 Properties of saturated steam at atmospheric pressure

Pressure bar g	Saturation temperature °C	Enthalpy (energy) in kJ/kg			Volume of dry saturated steam m^3/kg
		Water h_f	Evaporation h_{fg}	Steam h_g	
0	100	419	2 257	2 676	1.673

Example 2.2.1

At atmospheric pressure (0 bar g), water boils at 100°C, and 419 kJ of energy are required to heat 1 kg of water from 0°C to its saturation temperature of 100°C. Therefore the specific enthalpy of water at 0 bar g and 100°C is 419 kJ/kg, as shown in the steam tables (see Table 2.2.2).

Another 2 257 kJ of energy are required to evaporate 1 kg of water at 100°C into 1 kg of steam at 100°C. Therefore at 0 bar g the specific enthalpy of evaporation is 2 257 kJ/kg, as shown in the steam tables (see Table 2.2.2).

Therefore: Specific enthalpy of steam $h_g = 419 + 2\,257$

$$h_g = 2\,676 \text{ kJ/kg at 0 bar g}$$

However, steam at atmospheric pressure is of a limited practical use. This is because it cannot be conveyed under its own pressure along a steam pipe to the point of use.

Note: Because of the pressure/volume relationship of steam, (volume is reduced as pressure is increased) it is usually generated in the boiler at a pressure of at least 7 bar g. The generation of steam at higher pressures enables the steam distribution pipes to be kept to a reasonable size.

As the steam pressure increases, the density of the steam will also increase. As the specific volume is inversely related to the density, the specific volume will decrease with increasing pressure.

Figure 2.2.2 shows the relationship of specific volume to pressure. This highlights that the greatest change in specific volume occurs at lower pressures, whereas at the higher end of the pressure scale there is much less change in specific volume.

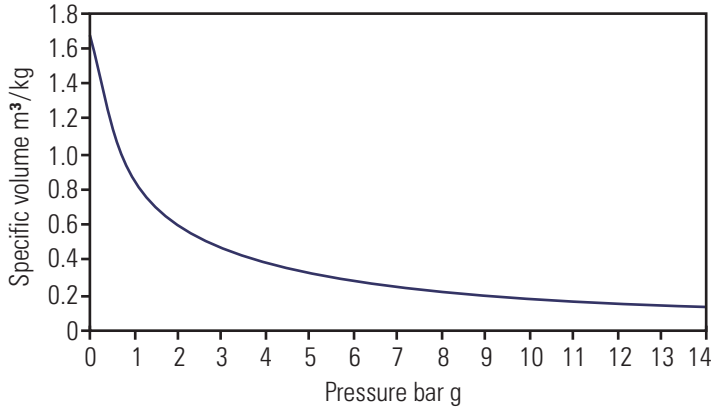


Fig. 2.2.2 Steam pressure/specific volume relationship

The extract from the steam tables shown in Table 2.2.2 shows specific volume, and other data related to saturated steam.

At 7 bar g, the saturation temperature of water is 170°C. More heat energy is required to raise its temperature to saturation point at 7 bar g than would be needed if the water were at atmospheric pressure. The table gives a value of 721 kJ to raise 1 kg of water from 0°C to its saturation temperature of 170°C.

The heat energy (enthalpy of evaporation) needed by the water at 7 bar g to change it into steam is actually less than the heat energy required at atmospheric pressure. This is because the specific enthalpy of evaporation decreases as the steam pressure increases.

However, as the specific volume also decreases with increasing pressure, the amount of heat energy transferred in the same volume actually increases with steam pressure.

Table 2.2.2 Extract from the saturated steam tables

Pressure bar g	Saturation temperature °C	Enthalpy kJ/kg			Volume of dry saturated steam m³/kg
		Water h_f	Evaporation h_{fg}	Steam h_g	
0	100	419	2 257	2 676	1.673
1	120	506	2 201	2 707	0.881
2	134	562	2 163	2 725	0.603
3	144	605	2 133	2 738	0.461
4	152	641	2 108	2 749	0.374
5	159	671	2 086	2 757	0.315
6	165	697	2 066	2 763	0.272
7	170	721	2 048	2 769	0.240

Dryness fraction

Steam with a temperature equal to the boiling point at that pressure is known as dry saturated steam. However, to produce 100% dry steam in an industrial boiler designed to produce saturated steam is rarely possible, and the steam will usually contain droplets of water.

In practice, because of turbulence and splashing, as bubbles of steam break through the water surface, the steam space contains a mixture of water droplets and steam.

Steam produced in any shell-type boiler (see Block 3), where the heat is supplied only to the water and where the steam remains in contact with the water surface, may typically contain around 5% water by mass.

If the water content of the steam is 5% by mass, then the steam is said to be 95% dry and has a dryness fraction of 0.95.

The actual enthalpy of evaporation of wet steam is the product of the dryness fraction (χ) and the specific enthalpy (h_{fg}) from the steam tables. Wet steam will have lower usable heat energy than dry saturated steam.

$$\text{Actual enthalpy of evaporation} = h_{fg}\chi \qquad \textbf{Equation 2.2.2}$$

Therefore:

$$\text{Actual total enthalpy} = h_f + h_{fg}\chi \qquad \textbf{Equation 2.2.3}$$

Because the specfic volume of water is several orders of magnitude lower than that of steam, the droplets of water in wet steam will occupy negligible space. Therefore the specific volume of wet steam will be less than dry steam:

$$\text{Actual specific volume} = v_g\chi \qquad \textbf{Equation 2.2.4}$$

Where v_g is the specific volume of dry saturated steam.

Example 2.2.2
Steam at a pressure of 6 bar g having a dryness fraction of 0.94 will only contain 94% of the enthalpy of evaporation of dry saturated steam at 6 bar g. The following calculations use figures from steam tables:

$$\text{Actual total enthalpy} = 697.5 \text{ kJ/kg} + (2066.0 \text{ kJ/kg} \times 0.94)$$
$$= \mathbf{2\,639.5\ kJ/kg}$$
$$\text{Actual specific volume} = 0.272 \text{ m}^3\text{/kg} \times 0.94$$
$$= \mathbf{0.256\ m^3/kg}$$

The steam phase diagram

The data provided in the steam tables can also be expressed in a graphical form. Figure 2.2.3 illustrates the relationship between the enthalpy and temperature of the various states of water and steam; this is known as a phase diagram.

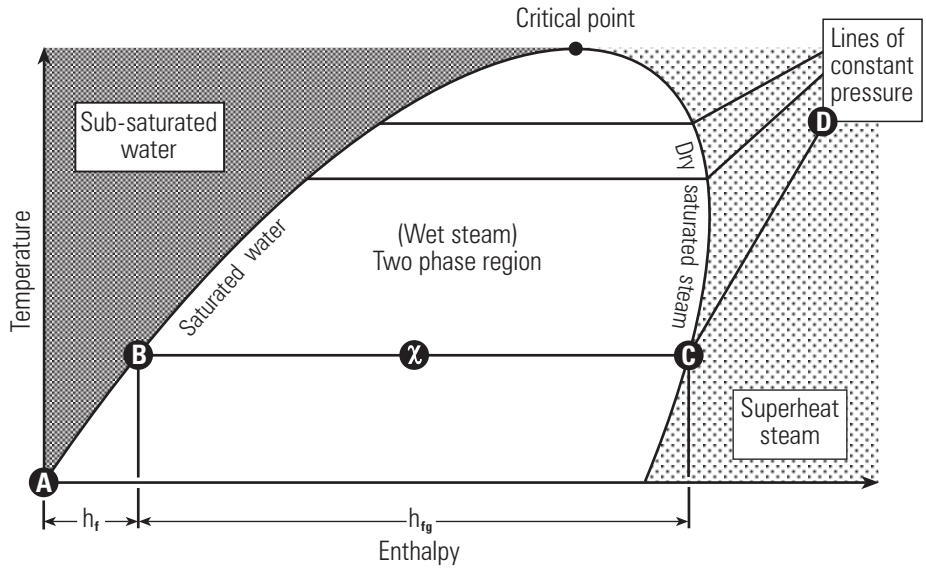


Fig. 2.2.3 Temperature enthalpy phase diagram

As water is heated from 0°C to its saturation temperature, its condition follows the saturated water line until it has received all of its liquid enthalpy, h_f, (**A** - **B**).

If further heat continues to be added, the water changes phase to a water/vapour mixture and continues to increase in enthalpy while remaining at saturation temperature ,h_{fg}, (**B** - **C**).

As the water/vapour mixture increases in dryness, its condition moves from the saturated liquid line to the saturated vapour line. Therefore at a point exactly halfway between these two states, the dryness fraction (χ) is 0.5. Similarly, on the saturated steam line, the steam is 100% dry.

Once it has received all of its enthalpy of evaporation, it reaches the saturated steam line. If it continues to be heated after this point the pressure remains constant but the temperature of the steam will begin to rise as superheat is imparted (**C** - **D**).

The saturated water and saturated steam lines enclose a region in which a water/vapour mixture exists - wet steam. In the region to the left of the saturated water line only water exists, and in the region to the right of the saturated steam line only superheated steam exists.

The point at which the saturated water and saturated steam lines meet is known as the critical point. As the pressure increases towards the critical point the enthalpy of evaporation decreases, until it becomes zero at the critical point. This suggests that water changes directly into saturated steam at the critical point.

Above the critical point the steam may be considered as a gas. The gaseous state is the most diffuse state in which the molecules have an almost unrestricted motion, and the volume increases without limit as the pressure is reduced.

The critical point is the highest temperature at which water can exist. Any compression at constant temperature above the critical point will not produce a phase change.

Compression at constant temperature below the critical point however, will result in liquefaction of the vapour as it passes from the superheated region into the wet steam region.

The critical point occurs at 374.15°C and 221.2 bar a for steam. Above this pressure the steam is termed supercritical and no well-defined boiling point applies.

Flash steam

The term 'flash steam' is traditionally used to describe steam issuing from condensate receiver vents and open-ended condensate discharge lines from steam traps. How can steam be formed from water without adding heat?

Flash steam occurs whenever water at high pressure (and a temperature higher than the saturation temperature of the low-pressure liquid) is allowed to drop to a lower pressure. Conversely, if the temperature of the high-pressure water is lower than the saturation temperature at the lower pressure, flash steam cannot be formed. In the case of condensate passing through a steam trap, it is usually the case that the upstream temperature is high enough to form flash steam. See Figure 2.2.4.

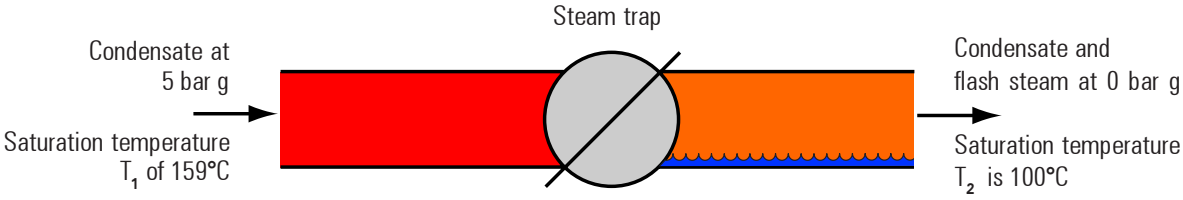


Fig. 2.2.4 Flash steam formed because $T_1 > T_2$

Consider a kilogram of condensate at 5 bar g and a saturation temperature of 159°C passing through a steam trap to a lower pressure of 0 bar g. The amount of energy in one kilogram of condensate at saturation temperature at 5 bar g is 671 kJ. In accordance with the first law of thermodynamics, the amount of energy contained in the fluid on the low-pressure side of the steam trap must equal that on the high-pressure side, and constitutes the principle of conservation of energy.

Consequently, the heat contained in one kilogram of low-pressure fluid is also 671 kJ. However, water at 0 bar g is only able to contain 419 kJ of heat, subsequently there appears to be an imbalance of heat on the low-pressure side of 671 – 419 = 252 kJ, which, in terms of the water, could be considered as excess heat.

This excess heat boils some of the condensate into what is known as flash steam and the boiling process is called flashing. Therefore, the one kilogram of condensate which existed as one kilogram of liquid water on the high pressure side of the steam trap now partly exists as both water and steam on the low-pressure side.

The amount of flash steam produced at the final pressure (P_2) can be determined using Equation 2.2.5:

$$\text{Proportion of flash steam} = \frac{(h_f \text{ at } P_1) - (h_f \text{ at } P_2)}{h_{fg} \text{ at } P_2} \qquad \textbf{Equation 2.2.5}$$

Where:
P_1 = Initial pressure
P_2 = Final pressure
h_f = Liquid enthalpy (kJ/kg)
h_{fg} = Enthalpy of evaporation (kJ/kg)

Example 2.2.3 The case where the high pressure condensate temperature is higher than the low pressure saturation temperature.

Consider a quantity of water at a pressure of 5 bar g, containing 671 kJ/kg of heat energy at its saturation temperature of 159°C. If the pressure was then reduced down to atmospheric pressure (0 bar g), the water could only exist at 100°C and contain 419 kJ/kg of heat energy. This difference of 671 - 419 = 252 kJ/kg of heat energy, would then produce flash steam at atmospheric pressure.

Therefore : $$\text{Flash steam produced} = \frac{671 - 419}{2\,257}$$

Total flash steam = 0.112 kg steam/kg water (or 11.2%)

The proportion of flash steam produced can be thought of as the ratio of the excess energy to the enthalpy of evaporation at the final pressure.

Example 2.2.4 The case where the high pressure condensate temperature is lower than the low pressure saturation temperature.

Consider the same conditions as in Example 2.2.3, with the exception that the high-pressure condensate temperature is at 90°C, that is, sub-cooled below the atmospheric saturation temperature of 100°C. Note: It is not usually practical for such a large drop in condensate temperature from its saturation temperature (in this case 159°C to 90°C); it is simply being used to illustrate the point about flash steam not being produced under such circumstances.

In this case, the sub-saturated water table will show that the liquid enthalpy of one kilogram of condensate at 5 bar g and 90°C is 377 kJ. As this enthalpy is less than the enthalpy of one kilogram of saturated water at atmospheric pressure (419 kJ), there is no excess heat available to produce flash steam. The condensate simply passes through the trap and remains in a liquid state at the same temperature but lower pressure, atmospheric pressure in this case. See Figure 2.2.5.

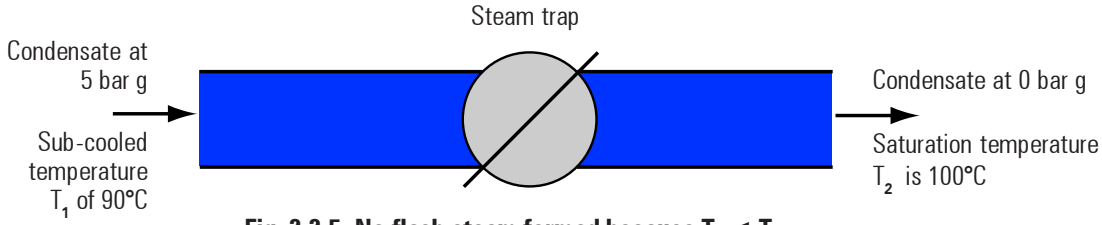


Fig. 2.2.5 No flash steam formed because $T_1 < T_2$

The vapour pressure of water at 90°C is 0.7 bar absolute. Should the lower condensate pressure have been less than this, flash steam would have been produced.

The principles of conservation of energy and mass between two process states

The principles of the conservation of energy and mass allow the flash steam phenomenon to be thought of from a different direction.

Consider the conditions in Example 2.2.3.

1 kg of condensate at 5 bar g and 159°C produces 0.112 kg of flash steam at atmospheric pressure. This can be illustrated schematically in Figure 2.2.5. The total mass of flash and condensate remains at 1 kg.

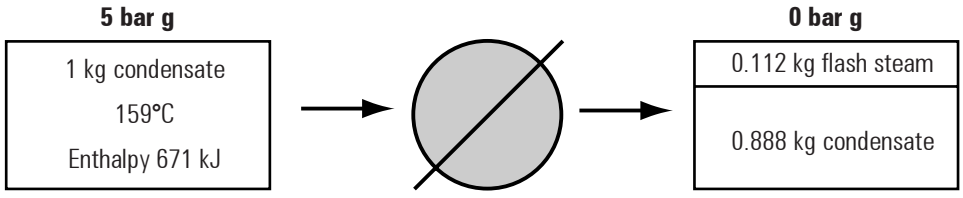


Fig. 2.2.6 The principle of energy conservation between two process states

The principle of energy conservation states that the total energy in the lower-pressure state must equal the total energy in the higher-pressure state. Therefore, the amount of heat in the flash steam and condensate must equal that in the initial condensate of 671 kJ.

Steam tables give the following information:

Total enthalpy of saturated water at atmospheric pressure (h_f) = 419 kJ/kg

Total enthalpy in saturated steam at atmospheric pressure (h_g) = 2 675 kJ/kg

Therefore, at the lower pressure state of 0 bar g,

Total enthalpy in the water = 0.888 kg x 419 kJ/kg = 372 kJ (A)

Total enthalpy in the steam = 0.112 kg x 2 675 kJ/kg = 299 kJ (B)

Total enthalpy in condensate and steam at the lower pressure = A + B = 671 kJ

Therefore, according to the steam tables, the enthalpy expected in the lower-pressure state is the same as that in the higher-pressure state, thus proving the principle of conservation of energy.

Questions

1. If steam at 5 bar absolute has a dryness fraction of 0.96 what will be its specific enthalpy of evaporation?

a| 2 002 kJ/kg ☐
b| 2 108 kJ/kg ☐
c| 2 195 kJ/kg ☐
d| 2 023 kJ/kg ☐

2. What is the volume of steam at 7 bar g having a dryness fraction of 0.95?

a| 0.252 m^3/kg ☐
b| 0.228 m^3/kg ☐
c| 0.240 m^3/kg ☐
d| 0.272 m^3/kg ☐

3. 500 kg/h of condensate at 7 bar g passes through a steam trap to atmospheric pressure. How much flash steam will be released?

a| 252.54 kg/h ☐
b| 56.42 kg/h ☐
c| 73.73 kg/h ☐
d| 66.9 kg/h ☐

4. Referring to Question 3, how much condensate will be available to return to the boiler feedtank?

a| 433 kg/h ☐
b| 500 kg/h ☐
c| 426.27 kg/h ☐
d| 443.58 kg/h ☐

5. Referring to Question 3 what will be the temperature of the condensate and flash steam?

a| 170°C ☐
b| 165°C ☐
c| 100°C ☐
d| 175°C ☐

6. As steam pressure increases the enthalpy/m^3:-

a| Remains the same ☐
b| Increases ☐
c| Reduces ☐

Answers

1: d, 2: b, 3: d, 4: a, 5: c, 6: b

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Module 2.3

Superheated Steam

Superheated Steam

If the saturated steam produced in a boiler is exposed to a surface with a higher temperature, its temperature will increase above the evaporating temperature.

The steam is then described as superheated by the number of temperature degrees through which it has been heated above saturation temperature.

Superheat cannot be imparted to the steam whilst it is still in the presence of water, as any additional heat simply evaporates more water. The saturated steam must be passed through an additional heat exchanger. This may be a second heat exchange stage in the boiler, or a separate superheater unit. The primary heating medium may be either the hot flue gas from the boiler, or may be separately fired.

Superheated steam has its applications in, for example, turbines where the steam is directed by nozzles onto a rotor. This causes the rotor to turn. The energy to make this happen can only have come from the steam, so logically the steam has less energy after it has gone through the turbine rotor. If the steam was at saturation temperature, this loss of energy would cause some of the steam to condense.

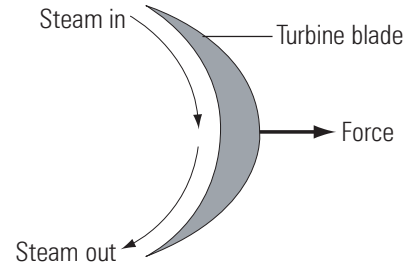


Fig. 2.3.1 Steam and force on a turbine blade

Turbines have a number of stages; the exhaust steam from the first rotor will be directed to a second rotor on the same shaft. This means that saturated steam would get wetter and wetter as it went through the successive stages. Not only would this promote waterhammer, but the water particles would cause severe erosion within the turbine. The solution is to supply the turbine with superheated steam at the inlet, and use the energy in the superheated portion to drive the rotor until the temperature/pressure conditions are close to saturation; and then exhaust the steam.

Another very important reason for using superheated steam in turbines is to improve thermal efficiency.

The thermodynamic efficiency of a heat engine such as a turbine, may be determined using one of two theories:

- The Carnot cycle, where the change in temperature of the steam between the inlet and outlet is compared to the inlet temperature.
- The Rankine cycle, where the change in heat energy of the steam between the inlet and outlet is compared to the total energy taken from the steam.

Example 2.3.1
A turbine is supplied with superheated steam at 90 bar a/450°C.
The exhaust is at 0.06 bar a (partial vacuum) and 10% wet.
Saturated temperature = 36.2°C.

Note: The values used for the temperature and energy content in the following examples are from steam tables.

$$\text{Carnot efficiency } \eta_C = \frac{T_i - T_e}{T_i} \qquad \textbf{Equation 2.3.1}$$

2.3.1.1 Determine the Carnot efficiency (η_C)

Where:
T_i = Temperature at turbine inlet (450°C) = 723.0 K
T_e = Temperature at turbine exhaust (36.2°C) = 309.2 K

$$\eta_C = \frac{723.0 - 309.2}{723.0}$$

Carnot efficiency η_C = 0.572 (57.2%)

$$\text{Rankine efficiency } \eta_R = \frac{H_i - H_e}{H_i - h_e} \qquad \textbf{Equation 2.3.2}$$

2.3.1.2 Determine the Rankine efficiency (η_R)

Where:

H_i = Heat at turbine inlet — H_i = 3 256 kJ/kg (from superheated steam tables)

H_e = Heat at turbine exhaust — H_e = heat in steam + heat in water:

heat in steam at 0.06 bar a (h_{fg}) = 2 415 kJ/kg

heat in water at 0.06 bar a (h_f) = 152 kJ/kg

As this steam is 10% wet the actual heat in the steam is 90% of h_{fg} = (0.9 x 2 415)

and the actual heat in the water is 10% of h_f = (0.1 x 152)

$$H_e = (0.9 \times 2415) + (0.1 \times 152)$$

$$H_e = 2188.7 \text{ kJ/kg}$$

h_e = Sensible heat in condensate — h_e = 152 kJ/kg (from steam tables)

$$\eta_R = \frac{3256 - 2188.7}{3256 - 152}$$

Rankine efficiency η_R = 0.344 (34.4%)

Examination of the figures for either of the cycles indicates that to achieve high efficiency:

- The temperature or energy at the turbine inlet should be as high as possible. This means as high a pressure and temperature as is practically possible.

  Superheated steam is the simplest way of providing this.

- The temperature or energy in the exhaust must be as low as possible. This means as low a pressure and temperature as is practically possible, and is usually achieved by a condenser on the turbine exhaust.

Notes:

- The figures calculated in Examples 2.3.1.1 and 2.3.1.2 are for thermodynamic efficiency, and must not be confused with mechanical efficiency.
- Although the efficiency figures appear to be very low, they must not be viewed in isolation, but rather used to compare one type of heat engine with another. For example, gas turbines, steam engines and diesel engines.

Superheated steam tables

The superheated steam tables display the properties of steam at various pressures in much the same way as the saturated steam tables. However, with superheated steam there is no direct relationship between temperature and pressure. Therefore at a particular pressure it may be possible for superheated steam to exist at a wide range of temperatures.

In general, saturated steam tables give gauge pressure, superheated steam tables give absolute pressure.

Table 2.3.1 Extract from superheated steam tables

Absolute pressure bar a	Units	Temperature (°C)					
		150	200	250	300	400	500
1.013	V_g (m³/kg)	1.912	2.145	2.375	2.604	3.062	3.519
	u_g (kJ/kg)	2 583	2 659	2 734	2 811	2 968	3 131
	h_g (kJ/kg)	2 777	2 876	2 975	3 075	3 278	3 488
	s_g (kJ/kg)	7.608	7.828	8.027	8.209	8.537	8.828

Example 2.3.2
How much more heat does superheated steam with a temperature of 400°C and a pressure of 1.013 bar a (0 bar g) have than saturated steam at the same pressure?

h_g for saturated steam at 1.013 bar a = 2 676 kJ/kg (from saturated steam tables)

h_g for steam at 1.013 bar a and 400°C = 3 278 kJ/kg (from superheated steam tables)

Enthalpy in the superheat = 3 278 kJ/kg - 2 676 kJ/kg

Enthalpy in the superheat = 602 kJ/kg

This may sound a useful increase in energy, but in fact it will actually make life more difficult for the engineer who wants to use steam for heating purposes.

From the energy in the superheat shown, the specific heat capacity can be determined by dividing this value by the temperature difference between saturation temperature (100°C) and the superheated steam temperature (400°C):

$$\text{Specific heat capacity} = \frac{602 \text{ kJ/kg}}{400°C - 100°C}$$

Specific heat capacity = 2.0 kJ/kg °C

However, unlike the specific heat capacity of water, the specific heat capacity for superheated steam varies considerably with pressure and temperature and cannot be taken as a constant.

The value of 2.0 kJ/kg °C given above is therefore only the mean specific heat capacity over the specified temperature range for that pressure.

There is no direct relationship between temperature, pressure and the specific heat capacity of superheated steam. There is, however, a general trend towards an increase in specific heat capacity with increasing pressure at low degrees of superheat, but this is not always the case.

Typical value range: 2.0 kJ/kg °C at 125°C and 1.013 bar a (0 bar g)

3.5 kJ/kg °C at 400°C and 120 bar a.

Can superheated steam be used in process heat exchangers and other heating processes?
Although not the ideal medium for transferring heat, superheated steam is sometimes used for process heating in many steam plants around the world, especially in the HPIs (Hydrocarbon Processing Industries) which produce oils and petrochemicals. This is more likely to be because superheated steam is already available on site for power generation, being the preferred energy source for turbines, rather than because it has any advantage over saturated steam for heating purposes. To be clear on this point, in most cases, saturated steam should be used for heat transfer processes, even if it means desuperheating the steam to do so. HPIs often desuperheat steam to within about ten degrees of superheat. This small degree of superheat is removed readily in the first part of the heating surface. Greater amounts of superheat are more difficult, and often uneconomic to deal with and (for heating purposes) are best avoided.

There are quite a few reasons why superheated steam is not as suitable for process heating as saturated steam:

Superheated steam has to cool to saturation temperature before it can condense to release its latent heat (enthalpy of evaporation). The amount of heat given up by the superheated steam as it cools to saturation temperature is relatively small in comparison to its enthalpy of evaporation.

If the steam has only a few degrees of superheat, this small amount of heat is quickly given up before it condenses. However, if the steam has a large degree of superheat, it may take a relatively long time to cool, during which time the steam is releasing very little energy.

Unlike saturated steam, the temperature of superheated steam is not uniform. Superheated steam has to cool to give up heat, whilst saturated steam changes phase. This means that temperature gradients over the heat transfer surface may occur with superheated steam.

In a heat exchanger, use of superheated steam can lead to the formation of a dry wall boiling zone, close to the tube sheet. This dry wall area can quickly become scaled or fouled, and the resulting high temperature of the tube wall may cause tube failure.

This clearly shows that in heat transfer applications, steam with a large degree of superheat is of little use because it:

- Gives up little heat until it has cooled to saturation temperature.
- Creates temperature gradients over the heat transfer surface as it cools to saturation temperature.
- Provides lower rates of heat transfer whilst the steam is superheated.
- Requires larger heat transfer areas.

So, superheated steam is not as effective as saturated steam for heat transfer applications. This may seem strange, considering that the rate of heat transfer across a heating surface is directly proportional to the temperature difference across it. If superheated steam has a higher temperature than saturated steam at the same pressure, surely superheated steam should be able to impart more heat? The answer to this is 'no'. This will now be looked at in more detail.

It is true that the temperature difference will have an effect on the rate of heat transfer across the heat transfer surface, as clearly shown by Equation 2.5.3.

$$\dot{Q} = U\,A\,\Delta T \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
U = Overall thermal transmittance (heat transfer coefficient) (W/m²°C)
A = Heat transfer area (m²)
ΔT = Temperature difference between primary and secondary fluid (°C)

Equation 2.5.3 also shows that heat transfer will depend on the overall heat transfer coefficient 'U', and the heat transfer area 'A'.

For any single application, the heat transfer area might be fixed. However, the same cannot be said of the 'U' value; and this is the major difference between saturated and superheated steam. The overall 'U' value for superheated steam will vary throughout the process, but will always be much lower than that for saturated steam. It is difficult to predict 'U' values for superheated steam, as these will depend upon many factors, but generally, the higher the degree of superheat, the lower the 'U' value.

Typically, for a horizontal steam coil surrounded with water, 'U' values might be as low as 50 to 100 W/m²°C for superheated steam but 1 200 W/m²°C for saturated steam, as depicted in Figure 2.3.2.

For steam to oil applications, the 'U' values might be considerably less, perhaps as low as 20 W/m²°C for superheated steam and 150 W/m²°C for saturated steam.

In a shell and tube heat exchanger, 100 W/m²°C for superheated steam and 500 W/m²°C for saturated steam can be expected. These figures are typical; actual figures will vary due to other design and operational considerations.

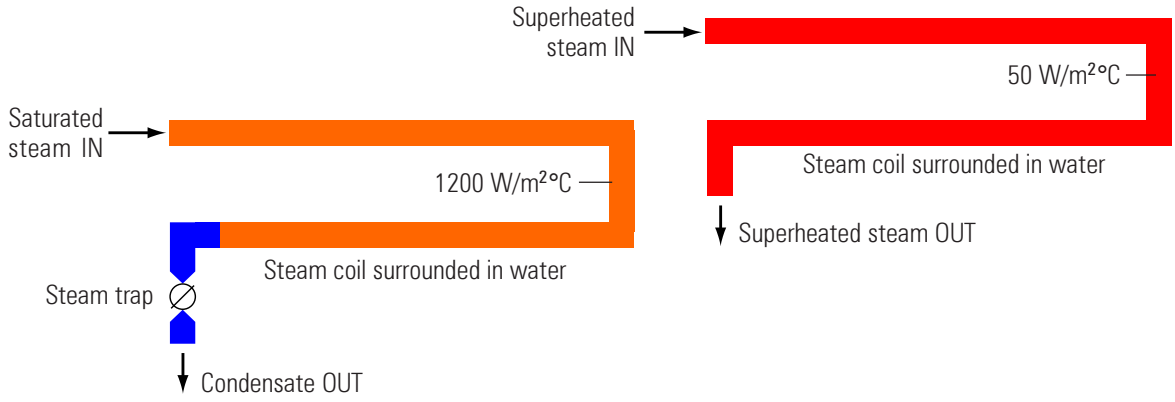


Figure 2.3.2 Typical 'U' values for superheated and saturated steam coils in water

Although the temperature of superheated steam is always higher than saturated steam at the same pressure, its ability to transfer heat is therefore much lower. The overall effect is that superheated steam is much less effective at transferring heat than saturated steam at the same pressure. The next Section 'Fouling' gives more detail.

Not only is superheated steam less effective at transferring heat, it is very difficult to quantify using Equation 2.5.3, $\dot{Q} = U\ A\ \Delta T$, as the temperature of the steam will fall as it gives up its heat while passing along the heating surface.

Predicting the size of heat transfer surfaces utilising superheated steam is difficult and complex. In practice, the basic data needed to perform such calculations is either not known or empirically obtained, putting their reliability and accuracy in doubt.

Clearly, as superheated steam is less effective at transferring heat than saturated steam, then any heating area using superheated steam would have to be larger than a saturated steam coil operating at the same pressure to deliver the same heat flowrate.

If there is no choice but to use superheated steam, it is not possible to maintain steam in its superheated state throughout the heating coil or heat exchanger, since as it gives up some of its heat content to the secondary fluid, it cools towards saturation temperature. The amount of heat above saturation is quite small compared with the large amount available as condensation occurs.

The steam should reach saturation relatively soon in the process; this allows the steam to condense to produce higher heat transfer rates and result in a higher overall 'U' value for the whole coil, see Figure 2.3.3.

To help to enable this, superheated steam used for heat transfer purposes should not hold more than about 10°C of superheat.

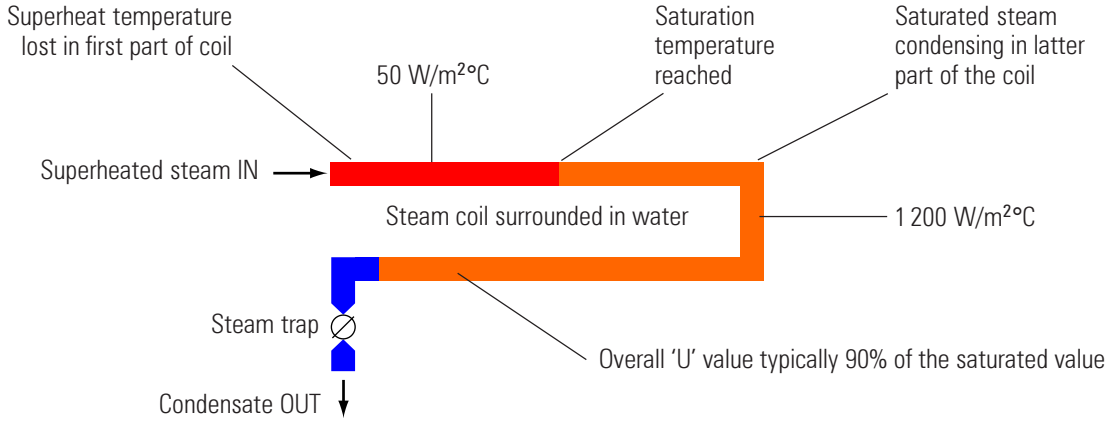


Figure 2.3.3 Less superheat allows the steam to condense in the major part of the coil thus increasing the overall 'U' value approaching that of saturated steam.

If this is so, it is relatively easy and practical to design a heat exchanger or a coil with a heating surface area based upon saturated steam at the same pressure, by adding on a certain amount of surface area to allow for the superheat. Using this guideline, the first part of a coil will be used purely to reduce the temperature of superheated steam to its saturation point. The rest of the coil will then be able to take advantage of the higher heat transfer ability of the saturated steam. The effect is that the overall 'U' value may not be much less than if saturated steam were supplied to the coil.

From practical experience, if the extra heating area needed for superheated steam is 1% per 2°C of superheat, the coil (or heat exchanger) will be large enough. This seems to work up to 10°C of superheat. It is not recommended that superheated steam above 10°C of superheat be used for heating purposes due to the probable disproportionate and uneconomic size of the heating surface, the propensity for fouling by dirt, and the possibility of product spoilage by the high and uneven superheat temperatures.

Fouling

Fouling is caused by deposits building up on the heat transfer surface adding a resistance to heat flow. Many process liquids can deposit sludge or scale on heating surfaces, and will do so at a faster rate at higher temperatures. Further, superheated steam is a dry gas. Heat flowing from the steam to the metal wall must pass through the static films adhering to the wall, which resist heat flow.

By contrast, the condensation of saturated steam causes the movement of steam towards the wall, and the release of large quantities of latent heat right at the condensing surface. The combination of these factors means that the overall heat transfer rates are much lower where superheated steam is present, even though the temperature difference between the steam and the secondary fluid is higher.

Example 2.3.3 Sizing a tube bundle for superheated steam

Superheated steam at 3 bar g with 10°C of superheat (154°C) is to be used as the primary heat source for a shell and tube process heat exchanger with a heating load of 250 kW, heating an oil based fluid from 80°C to 120°C (making the arithmetic mean secondary temperature (ΔT_{AM}) 100°C). Estimate the area of primary steam coil required.

(Arithmetic mean temperature differences are used to keep this calculation simple; in practice, logarithmic mean temperatures would be used for greater accuracy. Please refer to Module 2.5 'Heat Transfer' for details on arithmetic and logarithmic mean temperature differences).

First, consider the coil if it were heated by saturated steam at 3 bar g (144°C).

The 'U' value for saturated steam heating oil via a new carbon steel coil is taken to be 500 W/m² °C.

ΔT_{AM} = Saturated steam temperature (144°C) – Mean secondary temp. (100°C)
= 44°C

Using Equation 2.5.3: $\dot{Q} = U A \Delta T$

$250\,000 = 500 \text{ W/m}^2\text{°C} \times A \times 44\text{°C}$

$$A = \frac{250\,000}{500 \times 44}$$

$A = 11.4 \text{ m}^2$

Therefore, if saturated steam were used, the heating coil area = 11.4 m²

The degree of superheat is 10°C. Allowing 1% extra heating area per 2°C of superheat, the extra amount of coil $= \frac{10\%}{2}$

= 5% extra heating area

Heating area = 11.4 m² + 5%

= 11.4 + 0.6

= 12 m²

Adding on another 5% for future fouling:

= 12 + 5%

= 12.54 + 0.6

= 12.6 m²

Other applications using superheated steam

All the above applies when steam is flowing through a relatively narrow passage, such as the tubes in a shell and tube heat exchanger or the plates in a plate heat exchanger.

In some applications, perhaps a drying cylinder in a paper machine, superheated steam is admitted to a greater volume, when its velocity plummets to very small values. Here, the steam near the wall of the cylinder quickly drops in temperature to near saturation and condensation begins. The heat flow through the wall is then the same as if the cylinder were supplied with saturated steam. Superheat is present only within the 'core' in the steam space and has no discernible effect on heat transfer rates.

There are instances where the presence of superheat can actually reduce the performance of a process, where steam is being used as a process material.

One such process might involve moisture being imparted to the product from the steam as it condenses, such as, the conditioning of animal feedstuff (meal) prior to pelletising. Here the moisture provided by the steam is an essential part of the process; superheated steam would over-dry the meal and make pelletising difficult.

The effects of reducing steam pressure

In addition to the use of an additional heat exchanger (generally called a 'superheater'), superheat can also be imparted to steam by allowing it to expand to a lower pressure as it passes through the orifice of a pressure reducing valve. This is termed a throttling process with the lower pressure steam having the same enthalpy (apart from a small amount lost to friction in passing through the valve) as the upstream high pressure steam. However, the temperature of the throttled steam will always be lower than that of the supply steam.

The state of the throttled steam will depend upon:

- The pressure of the supply steam.
- The state of the supply steam.
- The pressure drop across the valve orifice.

For supply steam below 30 bar g in the dry saturated state, any drop in pressure will produce superheated steam after throttling. The degree of superheat will depend on the amount of pressure reduction.

For supply steam above 30 bar g in the dry saturated state, the throttled steam might be superheated, dry saturated, or even wet, depending on the amount of pressure drop. For example, dry saturated steam at 60 bar g would have to be reduced to approximately 10.5 bar g to produce dry saturated steam. Any less of a pressure drop will produce wet steam, while any greater pressure drop would produce superheated steam.

Equally, the state of the supply steam at any pressure will influence the state of the throttled steam. For example, wet steam at a pressure of 10 bar g and 0.95 dryness fraction would need to be reduced to 0.135 bar g to produce dry saturated steam. Any less of a pressure drop would produce wet steam while any greater pressure drop would superheat the throttled steam.

Example 2.3.4 Increasing the dryness of wet steam with a control valve

Steam with a dryness fraction (χ) of 0.95 is reduced from 6 bar g to 1 bar g, using a pressure reducing valve.

Determine the steam conditions after the pressure reducing valve.

From steam tables; At 6 bar g $h_f = 697.5$ kJ/kg

$h_{fg} = 2\,066.0$ kJ/kg

At 1 bar g $h_f = 505.6$ kJ/kg

$h_{fg} = 2\,201.1$ kJ/kg

Therefore actual total enthalpy at 6 bar g (valve inlet):

Actual total enthalpy at 6 bar g $= h_f + (h_{fg}\chi)$

$= 697.5 + (2\,066 \times 0.95)$

Actual total enthalpy at 6 bar g = 2 660.2 kJ/kg

This quantity of heat energy is retained by the steam as the pressure is reduced to 1 bar g.

Total enthalpy of dry saturated steam, at 1 bar g = 505.6 + 2 201.1

Total enthalpy of dry saturated steam, at 1 bar g = 2 706.7 kJ/kg

As the actual enthalpy of the steam at 1 bar g is less than the enthalpy of dry saturated steam at 1 bar g, then the steam is not superheated and still retains a proportion of moisture in its content.

However as $h_f + (h_{fg}\chi)$ = (total enthalpy in the steam)

At 1 bar g: $505.6 + (2\,201.1 \times \chi) = 2\,660.2$ kJ/kg

Therefore:

$$\chi = \frac{2\,660.2 \text{ kJ/kg} - 505.6 \text{ kJ/kg}}{2\,201.1 \text{ kJ/kg}}$$

χ = 0.979 or 97.9% dry steam at 1 bar g

Since the total enthalpy after the pressure reducing valve is less than the total enthalpy of steam at 1 bar g, the steam is still wet.

Example 2.3.5 Superheat created by a control valve

Steam with a dryness fraction of 0.98 is reduced from 10 bar g down to 1 bar g using a pressure reducing valve (as shown in Figure 2.3.4).

From steam tables: At 10 bar g h_f = 781.6 kJ/kg

h_{fg} = 2 000.1 kJ/kg

Therefore: Actual total enthalpy = $h_f + (h_{fg}\chi)$

At 10 bar g = 781.6 + (2 000.1 x 0.98)

Actual total enthalpy at 10 bar g = 2 741.7 kJ/kg

Determine the degree of superheat after the valve.

As in the previous example (2.3.4), the specific enthalpy of dry saturated steam (h_g) at 1 bar g is 2 706.7 kJ/kg.

The actual total enthalpy of the steam is greater than the total enthalpy (h_g) of dry saturated steam at 1 bar g. The steam is therefore not only 100% dry, but also has some degree of superheat.

The excess energy = 2 741.7 - 2 706.7 = 35 kJ/kg, and this is used to raise the temperature of the steam from the saturation temperature of 120°C to 136°C.

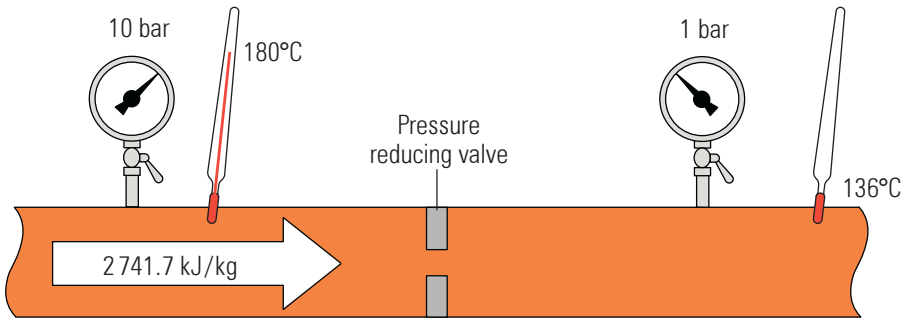


Fig. 2.3.4 The creation of superheat by pressure reduction

The degree of superheat can be determined either by using superheated steam tables, or by using a Mollier chart.

The Mollier chart

The Mollier chart is a plot of the specific enthalpy of steam against its specific entropy (s_g).

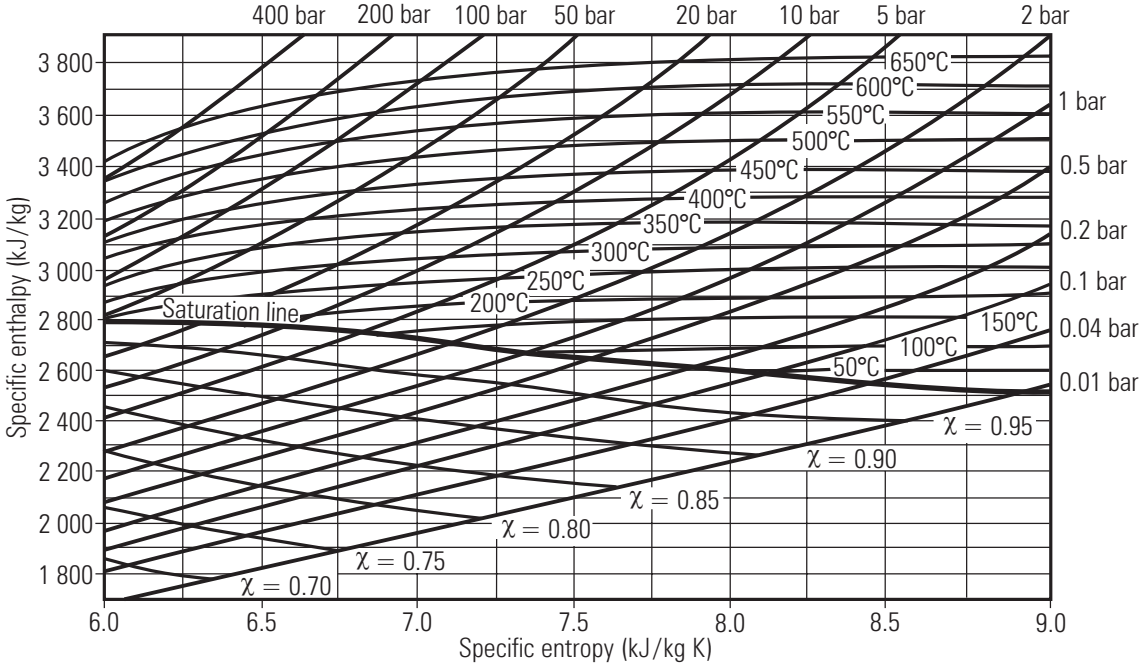


Fig. 2.3.5 Enthalpy - entropy or Mollier chart for steam

Figure 2.3.5 shows a simplified, small scale version of the Mollier chart. The Mollier chart displays many different relationships between enthalpy, entropy, temperature, pressure and dryness fraction. It may appear to be quite complicated, due to the number of lines:

- Constant enthalpy lines (horizontal).
- Constant entropy lines (vertical).
- The steam saturation curve across the centre of the chart divides it into a superheated steam region, and a wet steam region. At any point above the saturation curve the steam is superheated, and at any point below the saturation curve the steam is wet. The saturation curve itself represents the condition of dry saturated steam at various pressures.
- Constant pressure lines in both regions.
- Constant temperature lines in the superheat region.
- Constant dryness fraction (χ) lines in the wet region.

A perfect expansion, for example within a steam turbine or a steam engine, is a constant entropy process, and can be represented on the chart by moving vertically downwards from a point representing the initial condition to a point representing the final condition.

A perfect throttling process, for example across a pressure reducing valve, is a constant enthalpy process. It can be represented on the chart by moving horizontally from left to right, from a point representing the initial condition to a point representing the final condition.

Both these processes involve a reduction in pressure, but the difference lies in the way in which this is achieved.

The two examples shown in Figure 2.3.6 illustrate the advantage of using the chart to analyse steam processes; they provide a pictorial representation of such processes. However, steam processes can also be numerically represented by the values provided in the superheated steam tables.

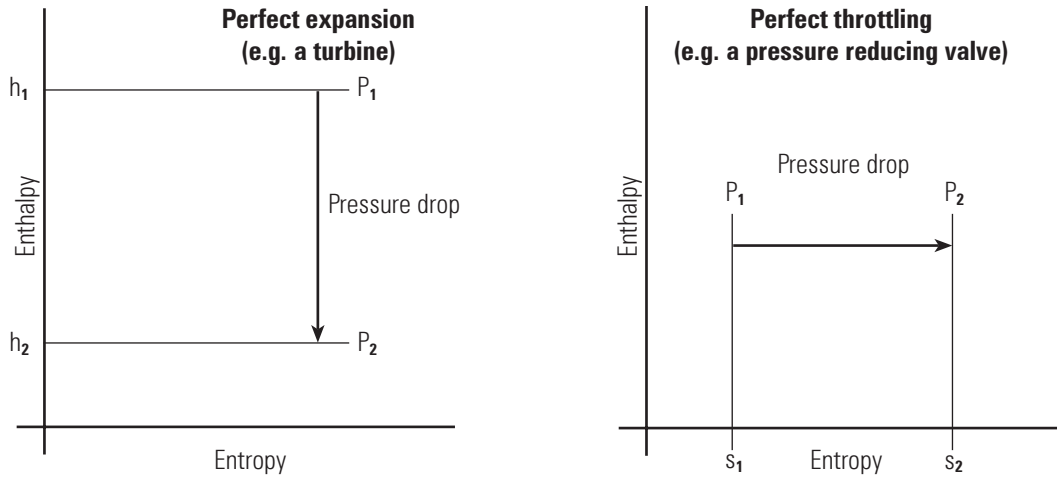


Fig. 2.3.6 Examples of expansion and throttling

Example 2.3.6 Perfect isentropic expansion resulting in work

Consider the perfect expansion of steam through a turbine. Initially the pressure is 50 bar a, the temperature is 300°C, and the final pressure is 0.04 bar a.

As the process is a perfect expansion, the entropy remains constant. The final condition can then be found by dropping vertically downwards from the initial condition to the 0.04 bar a constant pressure line (see Figure 2.3.7).

At the initial condition, the entropy is approximately 6.25 kJ/kg °C. If this line is followed vertically downwards until 0.04 bar a is reached, the final condition of the steam can be evaluated. At this point the specific enthalpy is 1 890 kJ/kg, and the dryness fraction is 0.72 (see Figure 2.3.7).

The final condition can also be determined by using the superheated steam tables.

At the initial condition (50 bar a/300°C): $h_g = 2\ 927$ kJ/kg

and $s_g = 6.212$ kJ/kg °C

For dry saturated steam at 0.04 bar a: $s_f = 0.422$ kJ/kg °C

$s_{fg} = 8.051$ kJ/kg °C

and $s_g = 8.473$ kJ/kg °C

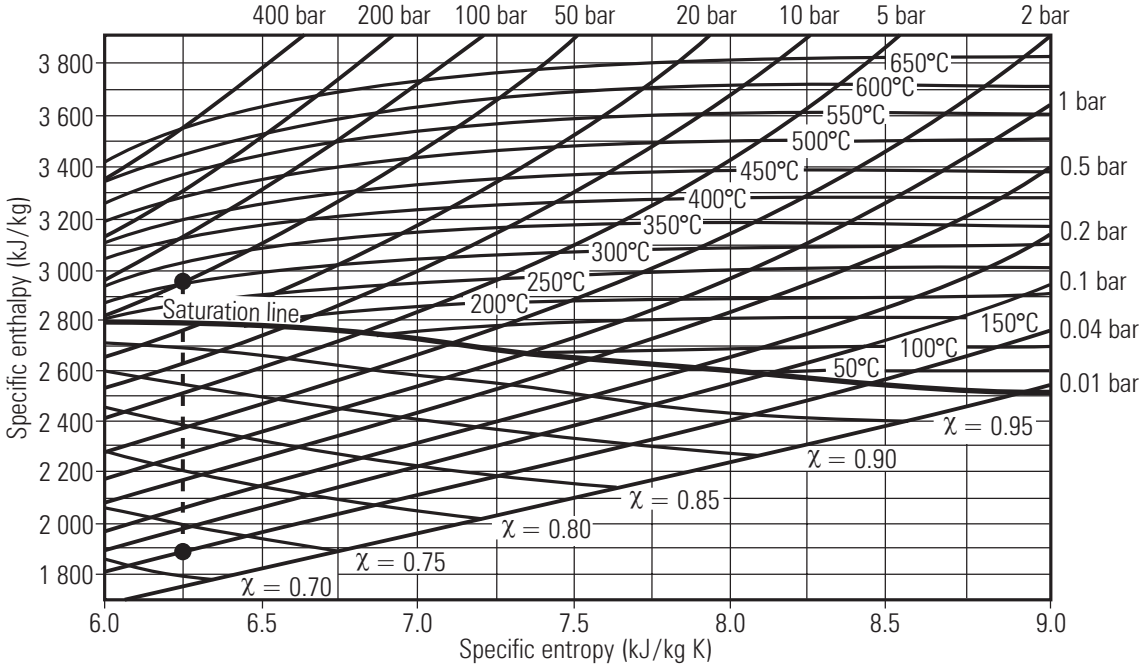


Fig. 2.3.7 Enthalpy - entropy or Mollier chart for steam - Example

Since the entropy of dry saturated steam at 0.04 bar a (8.473 kJ/kg°C) is greater than the entropy of the superheated steam at 50 bar a/300°C (6.212 kJ/kg°C), it follows that some of the dry saturated steam must have condensed to maintain the constant entropy.

As the entropy remains constant, at the final condition:

$$s_f + (s_{fg}c) = 6.212 \text{ kJ/kg °C}$$

$$0.422 + (8.051 \times c) = 6.212 \text{ kJ/kg °C}$$

$$\text{Therefore: } c = \frac{6.212 \text{ kJ/kg °C} - 0.422 \text{ kJ/kg °C}}{8.051 \text{ kJ/kg °C}}$$

$$\textbf{Dryness fraction } c = \textbf{0.72 (72\%)}$$

$$\text{Also } h_g = h_f + (h_{fg}c)$$

$$\text{At 0.04 bar g: } h_f = 121 \text{ kJ/kg}$$

$$\text{And } h_{fg} = 2433 \text{ kJ/kg}$$

$$\text{Therefore: } h_g = 121 + (2433 \times 0.72)$$

$$\textbf{Specific enthalpy } h_g = \textbf{1873 kJ/kg}$$

These answers correspond closely with the results obtained using the Mollier chart. The small difference in value between the two sets of results is to be expected, considering the inaccuracies involved in reading off a chart such as this.

Questions

1. Compared with saturated steam at the same pressure, superheated steam:

a| Contains more heat energy ☐
b| Has a greater enthalpy of evaporation ☐
c| Has a smaller specific volume ☐
d| Condenses at a higher temperature ☐

2. Which is NOT a characteristic of superheated steam:

a| It contains no water droplets ☐
b| It causes severe erosion in pipes ☐
c| It may cause uneven heating of a product ☐
d| It has a temperature above saturation ☐

3. Superheated steam at a pressure of 6 bar g:

a| Has a larger specific heat capacity than water ☐
b| Has a dryness fraction of 0.99 ☐
c| Must not be used as a heat transfer medium ☐
d| Has a temperature greater than 165°C ☐

4. If steam with a dryness fraction of 0.97 is reduced from 7 bar g to 2 bar g using a pressure reducing valve, at the final condition it has:

a| A temperature of 170.5°C and a dryness fraction of 0.97 ☐
b| A temperature of 164°C and a dryness fraction of 1 ☐
c| A temperature of 133.7°C and a dryness fraction of 0.99 ☐
d| A temperature of 149.9°C and a dryness fraction of 0.98 ☐

5. If superheated steam at 250°C and 4 bar a is reduced to 2 bar a in a steam engine, what is its final temperature?

a| 120°C ☐
b| 172°C ☐
c| 247°C ☐
d| 250°C ☐

6. Steam at 7 bar g and at 425°C:

a| Has a volume less than that at saturated temperature ☐
b| Is superheated by 254°C ☐
c| Has a specific enthalpy of 2 951 kJ/kg ☐
d| Has a specific entropy of 7.040 kJ/kg K ☐

Answers

1: a, 2: b, 3: d, 4: c, 5: b, 6: b

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Module 2.4

Steam Quality

Steam Quality

Steam should be available at the point of use:

- In the correct quantity.
- At the correct temperature and pressure.
- Free from air and incondensable gases.
- Clean.
- Dry.

Correct quantity of steam

The correct quantity of steam must be made available for any heating process to ensure that a sufficient heat flow is provided for heat transfer.

Similarly, the correct flowrate must also be supplied so that there is no product spoilage or drop in the rate of production. Steam loads must be properly calculated and pipes must be correctly sized to achieve the flowrates required.

Correct pressure and temperature of steam

Steam should reach the point of use at the required pressure and provide the desired temperature for each application, or performance will be affected. The correct sizing of pipework and pipeline ancillaries will ensure this is achieved.

However, even if the pressure gauge is correctly displaying the desired pressure, the corresponding saturation temperature may not be available if the steam contains air and/or incondensable gases.

Air and other incondensable gases

Air is present within the steam supply pipes and equipment at start-up. Even if the system were filled with pure steam the last time it was used, the steam would condense at shutdown, and air would be drawn in by the resultant vacuum.

When steam enters the system it will force the air towards either the drain point, or to the point furthest from the steam inlet, known as the remote point. Therefore steam traps with sufficient air venting capacities should be fitted to these drain points, and automatic air vents should be fitted to all remote points.

However, if there is any turbulence the steam and air will mix and the air will be carried to the heat transfer surface. As the steam condenses, an insulating layer of air is left behind on the surface, acting as a barrier to heat transfer.

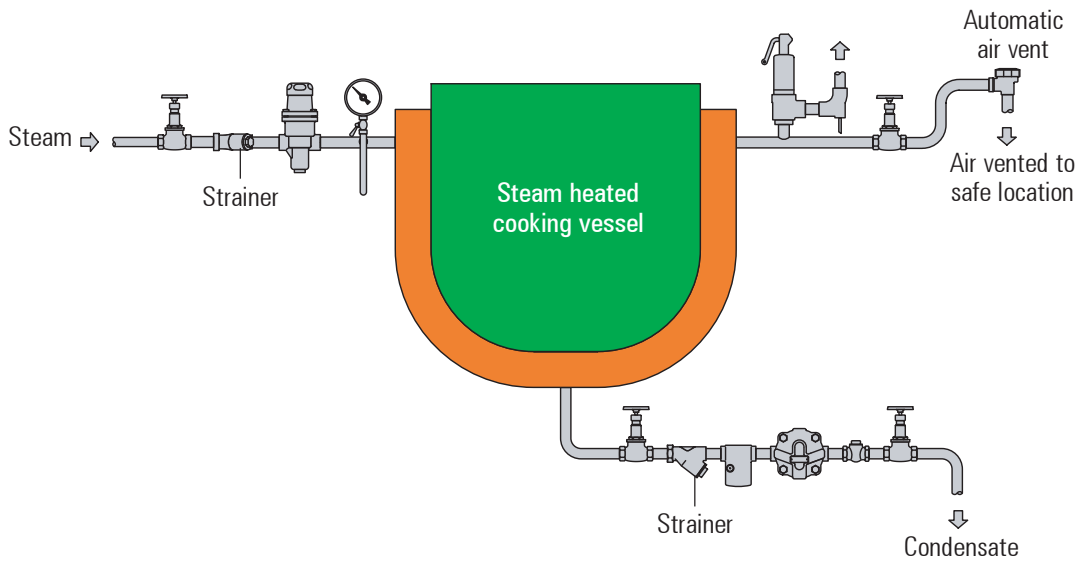


Fig. 2.4.1 Steam process equipment with an automatic air vent and strainers

Steam and air mixtures

In a mixture of air and steam, the presence of air will cause the temperature to be lower than expected. The total pressure of a mixture of gases is made up of the sum of the partial pressures of the components in the mixture.

This is known as Dalton's Law of Partial Pressures. The partial pressure is the pressure exerted by each component if it occupied the same volume as the mixture:

$$\begin{pmatrix}\text{Effective steam}\\ \text{pressure (bar a)}\end{pmatrix} = \begin{pmatrix}\text{Amount of steam as a proportion}\\ \text{of total (by volume)}\end{pmatrix} \times \begin{pmatrix}\text{Indicated pressure}\\ \text{(bar a)}\end{pmatrix}$$ **Equation 2.4.1**

Note: This is a thermodynamic relationship, so all pressures must be expressed in bar a.

Example 2.4.1

Consider a steam/air mixture made up of ¾ steam and ¼ air by volume. The total pressure is 4 bar a.

Determine the temperature of the mixture:

$$\frac{3}{4} \times 4 \text{ bar a} = 3 \text{ bar a}$$

Therefore the steam only has an effective pressure of 3 bar a as opposed to its apparent pressure of 4 bar a. The mixture would only have a temperature of 134°C rather than the expected saturation temperature of 144°C.

This phenomena is not only of importance in heat exchange applications (where the heat transfer rate increases with an increase in temperature difference), but also in process applications where a minimum temperature may be required to achieve a chemical or physical change in a product. For instance, a minimum temperature is essential in a steriliser in order to kill bacteria.

Other sources of air in the steam and condensate loop

Air can also enter the system in solution with the boiler feedwater. Make-up water and condensate, exposed to the atmosphere, will readily absorb nitrogen, oxygen and carbon dioxide: the main components of atmospheric air. When the water is heated in the boiler, these gases are released with the steam and carried into the distribution system.

Atmospheric air consists of 78% nitrogen, 21% oxygen and 0.03% carbon dioxide, by volume analysis. However, the solubility of oxygen is roughly twice that of nitrogen, whilst carbon dioxide has a solubility roughly 30 times greater than oxygen!

This means that 'air' dissolved in the boiler feedwater will contain much larger proportions of carbon dioxide and oxygen: both of which cause corrosion in the boiler and the pipework.

The temperature of the feedtank is maintained at a temperature typically no less than 80°C so that oxygen and carbon dioxide can be liberated back to the atmosphere, as the solubility of these dissolved gases decreases with increasing temperature.

The concentration of dissolved carbon dioxide is also kept to a minimum by demineralising and degassing the make-up water at the external water treatment stage.

The concentration of dissolved gas in the water can be determined using Henry's Law. This states that the mass of gas that can be dissolved by a given volume of liquid is directly proportional to the partial pressure of the gas.

This is only true however if the temperature is constant, and there is no chemical reaction between the liquid and the gas.

Cleanliness of steam

Layers of scale found on pipe walls may be either due to the formation of rust in older steam systems, or to a carbonate deposit in hard water areas. Other types of dirt which may be found in a steam supply line include welding slag and badly applied or excess jointing material, which may have been left in the system when the pipework was initially installed. These fragments will have the effect of increasing the rate of erosion in pipe bends and the small orifices of steam traps and valves.

For this reason it is good engineering practice to fit a pipeline strainer (as shown in Figure 2.4.2). This should be installed upstream of every steam trap, flowmeter, pressure reducing valve and control valve.

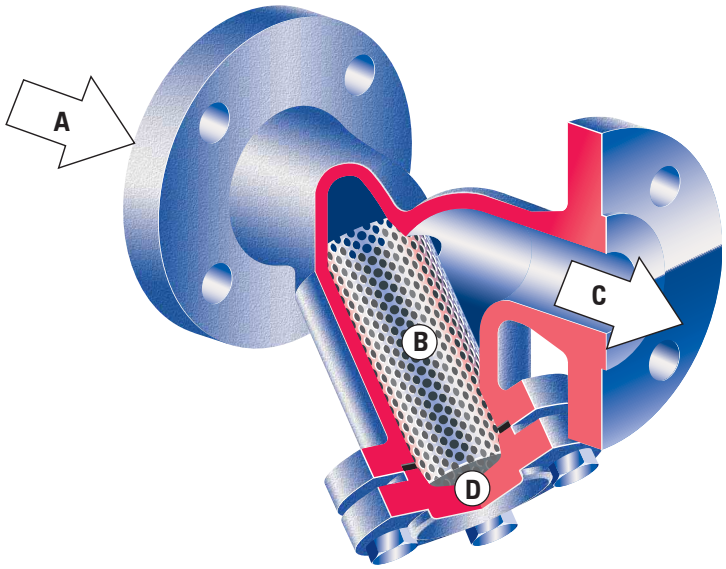


Fig. 2.4.2 A pipeline strainer

Steam flows from the inlet **A** through the perforated screen **B** to the outlet **C**. While steam and water will pass readily through the screen, dirt will be arrested. The cap **D** can be removed, allowing the screen to be withdrawn and cleaned at regular intervals.

When strainers are fitted in steam lines, they should be installed on their sides so that the accumulation of condensate and the problem of waterhammer can be avoided. This orientation will also expose the maximum strainer screen area to the flow.

A layer of scale may also be present on the heat transfer surface, acting as an additional barrier to heat transfer. Layers of scale are often a result of either:

- Incorrect boiler operation, causing impurities to be carried over from the boiler in water droplets.
- Incorrect water treatment in the boiler house.

The rate at which this layer builds up can be reduced by careful attention to the boiler operation and by the removal of any droplets of moisture.

Dryness of steam

Incorrect chemical feedwater treatment and periods of peak load can cause priming and carryover of boiler feedwater into the steam mains, leading to chemical and other material being deposited on to heat transfer surfaces. These deposits will accumulate over time, gradually reducing the efficiency of the plant.

In addition to this, as the steam leaves the boiler, some of it must condense due to heat loss through the pipe walls. Although these pipes may be well insulated, this process cannot be completely eliminated.

The overall result is that steam arriving at the plant is relatively wet.

It has already been shown that the presence of water droplets in steam reduces the actual enthalpy of evaporation, and also leads to the formation of scale on the pipe walls and heat transfer surface.

The droplets of water entrained within the steam can also add to the resistant film of water produced as the steam condenses, creating yet another barrier to the heat transfer process.

A separator in the steam line will remove moisture droplets entrained in the steam flow, and also any condensate that has gravitated to the bottom of the pipe.

In the separator shown in Figure 2.4.3 the steam is forced to change direction several times as it flows through the body. The baffles create an obstacle for the heavier water droplets, while the lighter dry steam is allowed to flow freely through the separator.

The moisture droplets run down the baffles and drain through the bottom connection of the separator to a steam trap. This will allow condensate to drain from the system, but will not allow the passage of any steam.

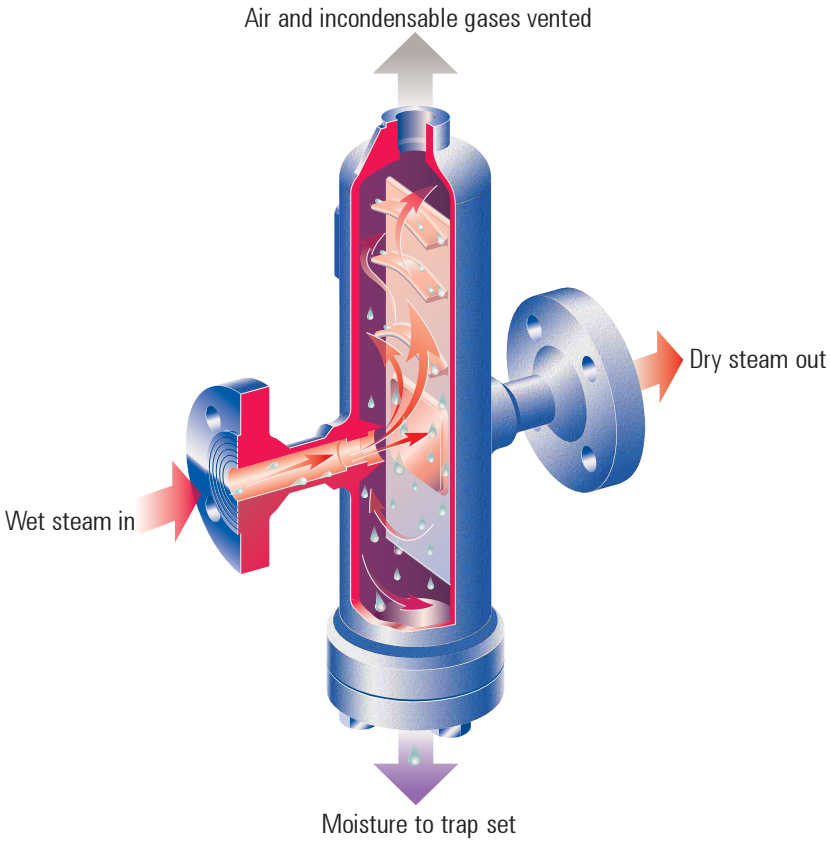


Fig. 2.4.3 A steam separator

Waterhammer

As steam begins to condense due to heat losses in the pipe, the condensate forms droplets on the inside of the walls. As they are swept along in the steam flow, they then merge into a film. The condensate then gravitates towards the bottom of the pipe, where the film begins to increase in thickness.

The build up of droplets of condensate along a length of steam pipework can eventually form a slug of water (as shown in Figure 2.4.4), which will be carried at steam velocity along the pipework (25 - 30 m/s).

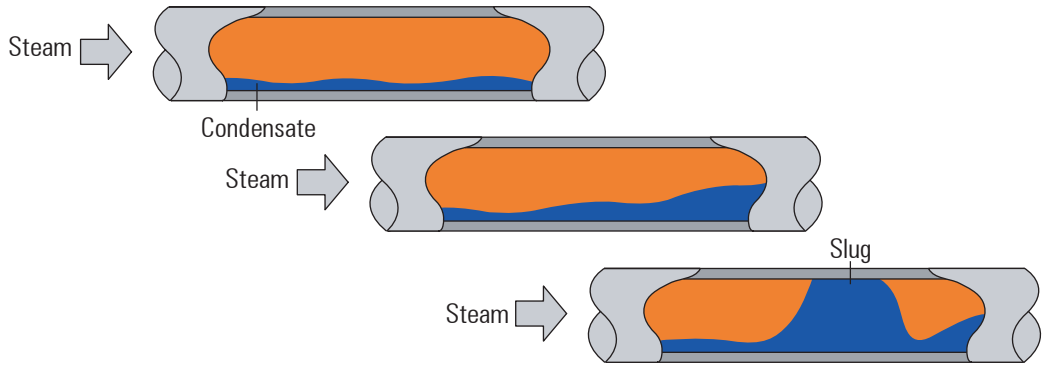


Fig. 2.4.4 Formation of a solid slug of water

This slug of water is dense and incompressible, and when travelling at high velocity, has a considerable amount of kinetic energy.

The laws of thermodynamics state that energy cannot be created or destroyed, but simply converted into a different form.

When obstructed, perhaps by a bend or tee in the pipe, the kinetic energy of the water is converted into pressure energy and a pressure shock is applied to the obstruction.

Condensate will also collect at low points, and slugs of condensate may be picked up by the flow of steam and hurled downstream at valves and pipe fittings.

These low points might include a sagging main, which may be due to inadequate pipe support or a broken pipe hanger. Other potential sources of waterhammer include the incorrect use of concentric reducers and strainers, or inadequate drainage before a rise in the steam main. Some of these are shown in Figure 2.4.5.

The noise and vibration caused by the impact between the slug of water and the obstruction, is known as waterhammer.

Waterhammer can significantly reduce the life of pipeline ancillaries. In severe cases the fitting may fracture with an almost explosive effect. The consequence may be the loss of live steam at the fracture, creating a hazardous situation.

The installation of steam pipework is discussed in detail in Block 9, Steam Distribution.

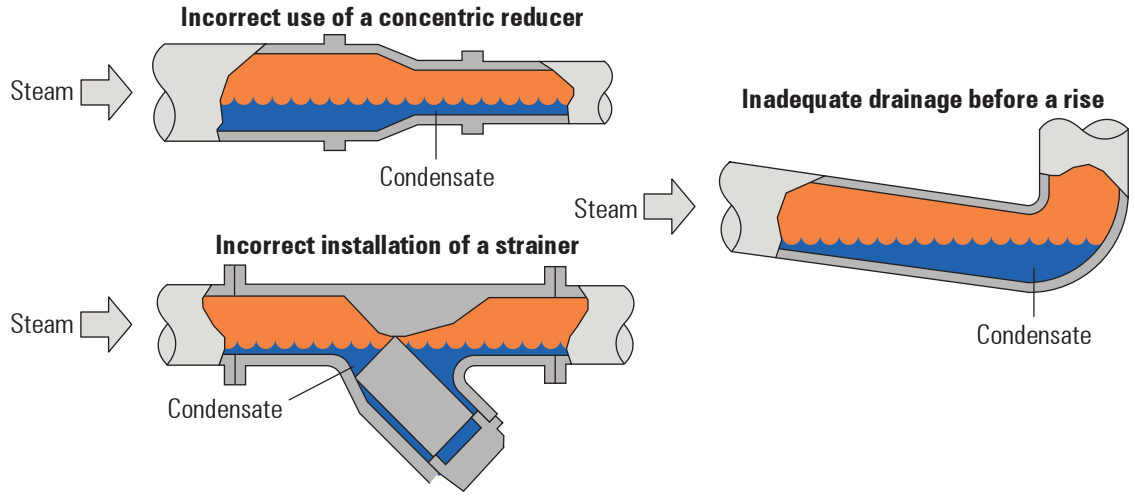


Fig. 2.4.5 Potential sources of waterhammer

Questions

1. Steam supplied at 6.5 bar g contains 20% air by volume. What is the temperature of the mixture?

a| 165°C ☐
b| 127°C ☐
c| 167°C ☐
d| 159°C ☐

2. Why is a boiler feedtank heated to approximately 85°C?

a| To reduce the energy required to raise steam ☐
b| To reduce the content of total dissolved solids in the water supplied to the boiler ☐
c| To reduce the gas content of the water ☐
d| To reduce the content of suspended solids in the water ☐

3. What is used to dry steam?

a| A separator ☐
b| A strainer ☐
c| A steam trap ☐
d| A tee piece ☐

4. What causes waterhammer?

a| Suspended water droplets ☐
b| An air/water mixture ☐
c| Strainers fitted on their sides ☐
d| Slugs of water in the steam ☐

5. How does air enter a steam system?

a| Through joints, on shut down of the steam system ☐
b| With make-up water to the boiler feedtank ☐
c| With condensate entering the boiler feedtank ☐
d| All of the above ☐

6. Why should strainers installed on steam lines be fitted on their sides?

a| To prevent the build-up of water in the strainer body ☐
b| To trap more dirt ☐
c| To reduce the frequency of cleaning ☐
d| To provide maximum screening area for the steam ☐

Answers

1: d, 2: c, 3: a, 4: d, 5: d, 6: a

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Module 2.5

Heat Transfer

Heat Transfer

In a steam heating system, the sole purpose of the generation and distribution of steam is to provide heat at the process heat transfer surface. If the required heat input rate and steam pressure are known, then the necessary steam consumption rate may be determined. This will allow the size of the boiler and the steam distribution system to be established.

Modes of heat transfer

Whenever a temperature gradient exists, either within a medium or between media, the transfer of heat will occur. This may take the form of either conduction, convection or radiation.

Conduction

When a temperature gradient exists in either a solid or stationary fluid medium, the heat transfer which takes place is known as conduction. When neighbouring molecules in a fluid collide, energy is transferred from the more energetic to the less energetic molecules. Because higher temperatures are associated with higher molecular energies, conduction must occur in the direction of decreasing temperature.

This phenomenon can be seen in both liquids and gases. However, in liquids the molecular interactions are stronger and more frequent, as the molecules are closer together. In solids, conduction is caused by the atomic activity of lattice vibrations as explained in Module 2.2.

The equation used to express heat transfer by conduction is known as Fourier's Law. Where there is a linear temperature distribution under steady-state conditions, for a one-dimensional plane wall it may be written as:

$$\dot{Q} = k\,A\,\frac{\Delta T}{\varkappa} \qquad \textbf{Equation 2.5.1}$$

Where:

$\dot{Q}$ = Heat transferred per unit time (W)
k = Thermal conductivity of the material (W/m K or W/m°C)
A = Heat transfer area (m²)
ΔT = Temperature difference across the material (K or °C)
$\varkappa$ = Material thickness (m)

Example 2.5.1

Consider a plane wall constructed of solid iron with a thermal conductivity of 70 W/m°C, and a thickness of 25 mm. It has a surface area of 0.3 m by 0.5 m, with a temperature of 150°C on one side and 80°C on the other.

Determine the rate of heat transfer:

$$\text{Heat transfer rate} = 70\ \text{W/m°C} \times (0.3 \times 0.5)\ \text{m}^2 \times \frac{(150 - 80)\text{°C}}{0.025\ \text{m}}$$

Heat transfer rate = 29 400 W (29.4 kW)

The thermal conductivity is a characteristic of the wall material and is dependent on temperature. Table 2.5.1 shows the variation of thermal conductivity with temperature for various common metals.

Table 2.5.1 Thermal conductivity (W/m °C)

Material	Thermal conductivity (W/m°C)		
	At 25°C	At 125°C	At 225°C
Iron	80	68	60
Low carbon steel	54	51	47
Stainless steel	16	17.5	19
Tungsten	180	160	150
Platinum	70	71	72
Aluminium	250	255	250
Gold	310	312	310
Silver	420	418	415
Copper	401	400	398

Considering the mechanism of heat transfer in conduction, in general the thermal conductivity of a solid will be much greater than of a liquid, and the thermal conductivity of a liquid will be greater than of a gas. Air has a particularly low thermal conductivity and this is why insulating materials often have lots of air spaces.

Convection

The transfer of heat energy between a surface and a moving fluid at different temperatures is known as convection. It is actually a combination of the mechanisms of diffusion and the bulk motion of molecules.

Near the surface where the fluid velocity is low, diffusion (or random molecular motion) dominates. However, moving away from the surface, bulk motion holds an increasing influence. Convective heat transfer may take the form of either forced convection or natural convection. Forced convection occurs when fluid flow is induced by an external force, such as a pump or an agitator. Conversely, natural convection is caused by buoyancy forces, due to the density differences arising from the temperature variations in the fluid.

The transfer of heat energy caused by a phase change, such as boiling or condensing, is also referred to as a convective heat transfer process.

The equation for convection is expressed by Equation 2.5.2 which is a derivation of Newton's Law of Cooling:

$$\dot{Q} = h\,A\,\Delta T \qquad \textbf{Equation 2.5.2}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
h = Convective heat transfer coefficient of the process (W/m² K or W/m² °C)
A = Heat transfer area of the surface (m²)
ΔT = Temperature difference between the surface and the bulk fluid (K or °C)

Example 2.5.2

Consider a plane surface 0.4 m by 0.9 m at a temperature of 20°C.
A fluid flows over the surface with a bulk temperature of 50°C.
The convective heat transfer coefficient (h) is 1600 W/m²°C.

Determine the rate of heat transfer:

$$\text{Heat transfer rate} = 1600\ \text{W/m}^2\text{°C} \times (0.4 \times 0.9)\ \text{m}^2 \times (50 - 20)\text{°C}$$

Heat transfer rate = 17 280 W (17.28 kW)

Radiation

The heat transfer due to the emission of energy from surfaces in the form of electromagnetic waves is known as thermal radiation. In the absence of an intervening medium, there is a net heat transfer between two surfaces of different temperatures. This form of heat transfer does not rely on a material medium, and is actually most efficient in a vacuum.

The general heat transfer equation

In most practical situations, it is very unusual for all energy to be transferred by one mode of heat transfer alone. The overall heat transfer process will usually be a combination of two or more different mechanisms.

The general equation used to calculate heat transfer across a surface used in the design procedure and forming a part of heat exchange theory is:

$$\dot{Q} = U\ A\ \Delta T \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W (J/s))
U = Overall heat transfer coefficient (W/m² K or W/m² °C)
A = Heat transfer area (m²)
ΔT = Temperature difference between the primary and secondary fluid (K or °C)
Note:
$\dot{Q}$ will be a mean heat transfer rate ($\dot{Q}_M$) if ΔT is a mean temperature difference (ΔT_{LM} or ΔT_{AM}).

The overall heat transfer coefficient (U)

This takes into account both conductive and convective resistance between two fluids separated by a solid wall. The overall heat transfer coefficient is the reciprocal of the overall resistance to heat transfer, which is the sum of the individual resistances.

The overall heat transfer coefficient may also take into account the degree of fouling in the heat transfer process. The deposition of a film or scale on the heat transfer surface will greatly reduce the rate of heat transfer. The fouling factor represents the additional thermal resistance caused by fluid impurities, rust formation or other reactions between the fluid and the wall.

The magnitude of the individual coefficients will depend on the nature of the heat transfer process, the physical properties of the fluids, the fluid flowrates and the physical layout of the heat transfer surface.

As the physical layout cannot be established until the heat transfer area has been determined, the design of a heat exchanger is by necessity, an iterative procedure. A starting point for this procedure usually involves selecting typical values for the overall heat transfer coefficient of various types of heat exchanger.

An accurate calculation for the individual heat transfer coefficients is a complicated procedure, and in many cases it is not possible due to some of the parameters being unknown. Therefore, the use of established typical values of overall heat transfer coefficient will be suitable for practical purposes.

Temperature difference (ΔT)

Newton's law of cooling states that the heat transfer rate is related to the instantaneous temperature difference between the hot and the cold media. In a heat transfer process, this temperature difference will vary either with position or with time. The general heat transfer equation was thus developed as an extension to Newton's law of cooling, where the mean temperature difference is used to establish the heat transfer area required for a given heat duty.

Mean temperature difference (ΔT_M)

The determination of the mean temperature difference in a flow type process like a heat exchanger will be dependent upon the direction of flow. The primary and secondary fluids may flow in the same direction (parallel flow/co-current flow), in the opposite direction (countercurrent flow), or perpendicular to each other (crossflow). When saturated steam is used the primary fluid temperature can be taken as a constant, because heat is transferred as a result of a change of phase only. The result is that the temperature profile is no longer dependent on the direction of flow.

However, as the secondary fluid passes over the heat transfer surface, the highest rate of heat transfer occurs at the inlet and progressively decays along its travel to the outlet. This is simply because the temperature difference between the steam and secondary fluid reduces with the rise in secondary temperature.

The resulting temperature profile of the steam and secondary fluid is typically as shown in Figure 2.5.1.

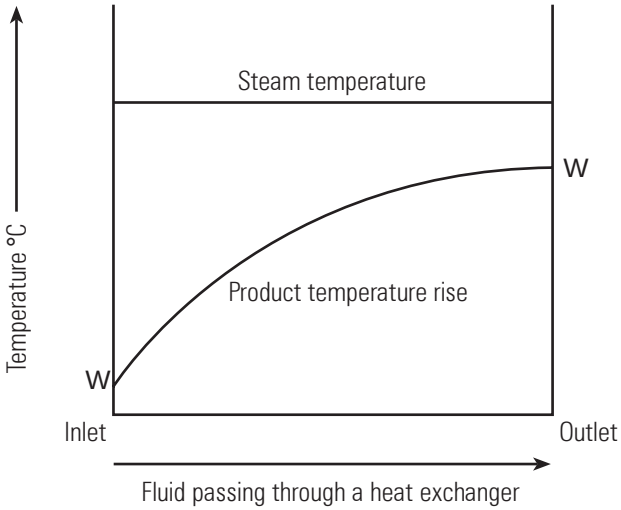


Fig. 2.5.1 Product temperature rise (LMTD)

The rise in secondary temperature is non-linear and is best represented by a logarithmic calculation. For this purpose the mean temperature difference chosen is termed the Logarithmic Mean Temperature Difference or LMTD or ΔT_{LM}.

An easier (but less accurate) way to calculate the mean temperature difference is to consider the Arithmetic Mean Temperature Difference or AMTD or ΔT_{AM}. This considers a linear increase in the secondary fluid temperature and for quick manual calculations, will usually give a satisfactory approximation of the mean temperature difference to be used in Equation 2.5.3. The AMTD temperature profile is shown in Figure 2.5.2.

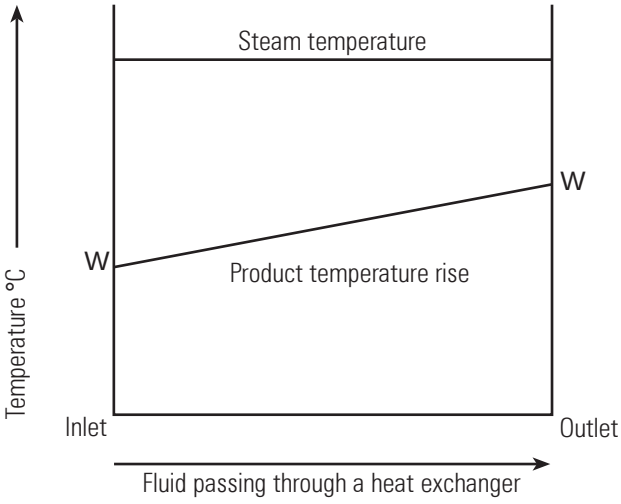


Fig. 2.5.2 Product temperature rise (AMTD)

The arithmetic mean temperature difference (AMTD):

$$\Delta T_{AM} = \left(\frac{T_{p1} + T_{p2}}{2} \right) - \left(\frac{T_{s1} + T_{s2}}{2} \right)$$

Where:
T_{p1} = Primary fluid in temperature
T_{p2} = Primary fluid out temperature
T_{s1} = Secondary fluid in temperature
T_{s2} = Secondary fluid out temperature

For steam, where the temperature of the primary fluid (steam) remains constant, this equation may be simplified to:

$$\Delta T_{AM} = T_s - \left(\frac{T_1 + T_2}{2} \right) \qquad \textbf{Equation 2.5.4}$$

Where:
T_s = Steam temperature (°C)
T_1 = Secondary fluid in temperature (°C)
T_2 = Secondary fluid out temperature (°C)

Because there is no temperature change on the steam side, the AMTD normally provides a satisfactory analysis of the heat transfer process, which is easy to manipulate in manual calculations.

However, a log mean temperature difference can also be used, which accounts for the non-linear change in temperature of the secondary fluid.

The log mean temperature difference (LMTD):

$$\Delta T_{LM} = \frac{(T_s - T_1) - (T_s - T_2)}{\ln \left(\frac{T_s - T_1}{T_s - T_2} \right)}$$

For steam, where the temperature of the primary fluid (steam) remains constant, this equation may be simplified to:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln \left(\frac{T_s - T_1}{T_s - T_2} \right)} \qquad \textbf{Equation 2.5.5}$$

Where:
T_s = Steam temperature (°C)
T_1 = Secondary fluid in temperature (°C)
T_2 = Secondary fluid out temperature (°C)
ln = A mathematical function known as 'natural logarithm'

Both Equations 2.5.4 and 2.5.5 assume that there is no change in the specific heat capacity or the overall heat transfer coefficient, and that there are no heat losses.

In reality the specific heat capacity may change as a result of temperature variations. The overall heat transfer coefficient may also change because of variations in fluid properties and flow conditions. However, in most applications the deviations will be almost negligible and the use of mean values will be perfectly acceptable.

In many cases the heat exchange equipment will be insulated from its surroundings, but the insulation will not be 100% efficient. Therefore, the energy transferred between the steam and the secondary fluid may not represent all of the heat lost from the primary fluid.

Example 2.5.3
Steam at 2 bar g is used to heat water from 20°C to 50°C.
The saturation temperature of steam at 2 bar g is 134°C.

Determine the arithmetic and the log mean temperature differences:

$$\Delta T_{AM} = 134 - \frac{20 + 50}{2}$$

$$\mathbf{\Delta T_{AM} = 99°C}$$

$$\Delta T_{LM} = \frac{50 - 20}{\ln\left(\frac{134 - 20}{134 - 50}\right)} = \frac{30}{\ln\left(\frac{114}{84}\right)} = \frac{30}{\ln(1.3571)} = \frac{30}{0.3053}$$

$$\mathbf{\Delta T_{LM} = 98.24°C}$$

In this example the AMTD and the LMTD have a similar value. This is because the secondary fluid temperature rise is small in comparison with the temperature difference between the two fluids.

Example 2.5.4
Consider a pressurised process fluid tank, which is heated from 10°C to 120°C using steam at 4.0 bar g. The saturation temperature of steam at 4.0 bar g is 152°C.

Determine the arithmetic and log mean temperature differences:

$$\Delta T_{AM} = 152 - \frac{10 + 120}{2}$$

$$\mathbf{\Delta T_{AM} = 87°C}$$

$$\Delta T_{LM} = \frac{120 - 10}{\ln\left(\frac{152 - 10}{152 - 120}\right)} = \frac{110}{\ln\left(\frac{142}{32}\right)} = \frac{110}{\ln(4.4375)} = \frac{110}{1.49}$$

$$\mathbf{\Delta T_{LM} = 73.8°C}$$

Because the secondary fluid temperature rise is large in comparison with the temperature difference between the two fluids, the discrepancy between the two results is more significant.

By using the AMTD rather than the LMTD, the calculated heat transfer area would be almost 15% smaller than that required.

Barriers to heat transfer

The metal wall may not be the only barrier in a heat transfer process. There is likely to be a film of air, condensate and scale on the steam side. On the product side there may also be baked-on product or scale, and a stagnant film of product.

Agitation of the product may eliminate the effect of the stagnant film, whilst regular cleaning on the product side should reduce the scale.

Regular cleaning of the surface on the steam side may also increase the rate of heat transfer by reducing the thickness of any layer of scale, however, this may not always be possible. This layer may also be reduced by careful attention to the correct operation of the boiler, and the removal of water droplets carrying impurities from the boiler.

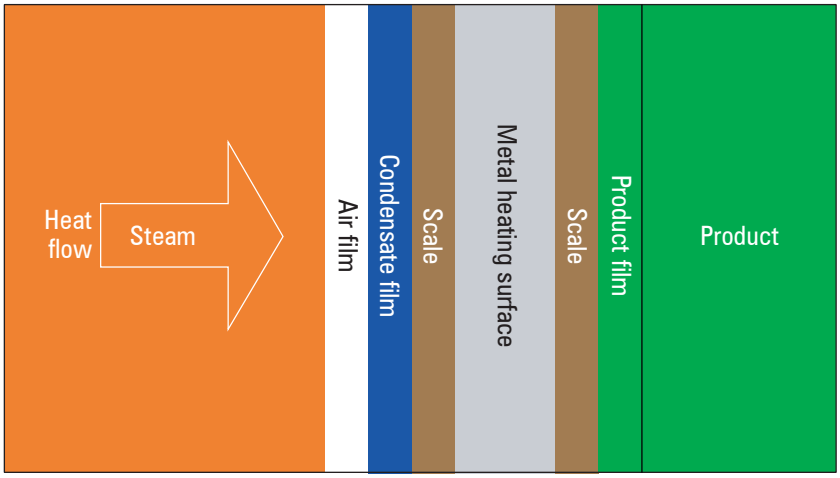


Fig. 2.5.3 Heat transfer layers

Filmwise condensation

The elimination of the condensate film, is not quite as simple. As the steam condenses to give up its enthalpy of evaporation, droplets of water may form on the heat transfer surface. These may then merge together to form a continuous film of condensate. The condensate film may be between 100 and 150 times more resistant to heat transfer than a steel heating surface, and 500 to 600 times more resistant than copper.

Dropwise condensation

If the droplets of water on the heat transfer surface do not merge immediately and no continuous condensate film is formed, 'dropwise' condensation occurs. The heat transfer rates which can be achieved during dropwise condensation, are generally much higher than those achieved during filmwise condensation.

As a larger proportion of the heat transfer surface is exposed during dropwise condensation, heat transfer coefficients may be up to ten times greater than those for filmwise condensation.

In the design of heat exchangers where dropwise condensation is promoted, the thermal resistance it produces is often negligible in comparison to other heat transfer barriers. However, maintaining the appropriate conditions for dropwise condensation have proved to be very difficult to achieve.

If the surface is coated with a substance that inhibits wetting, it may be possible to maintain dropwise condensation for a period of time. For this purpose, a range of surface coatings such as Silicones, PTFE and an assortment of waxes and fatty acids are sometimes applied to surfaces in a heat exchanger on which condensation is to be promoted. However, these coatings will gradually lose their effectiveness due to processes such as oxidation or fouling, and film condensation will eventually predominate.

As air is such a good insulator, it provides even more resistance to heat transfer. Air may be between 1 500 and 3 000 times more resistant to heat flow than steel, and 8 000 to 16 000 more resistant than copper. This means that a film of air only 0.025 mm thick may resist as much heat transfer as a wall of copper 400 mm thick! Of course all of these comparative relationships depend on the temperature profiles across each layer.

Figure 2.5.4 illustrates the effect this combination of layers has on the heat transfer process. These barriers to heat transfer not only increase the thickness of the entire conductive layer, but also greatly reduce the mean thermal conductivity of the layer.

The more resistant the layer to heat flow, the larger the temperature gradient is likely to be. This means that to achieve the same desired product temperature, the steam pressure may need to be significantly higher.

The presence of air and water films on the heat transfer surfaces of either process or space heating applications is not unusual. It occurs in all steam heated process units to some degree.

To achieve the desired product output and minimise the cost of process steam operations, a high heating performance may be maintained by reducing the thickness of the films on the condensing surface. In practice, air will usually have the most significant effect on heat transfer efficiency, and its removal from the supply steam will increase heating performance.

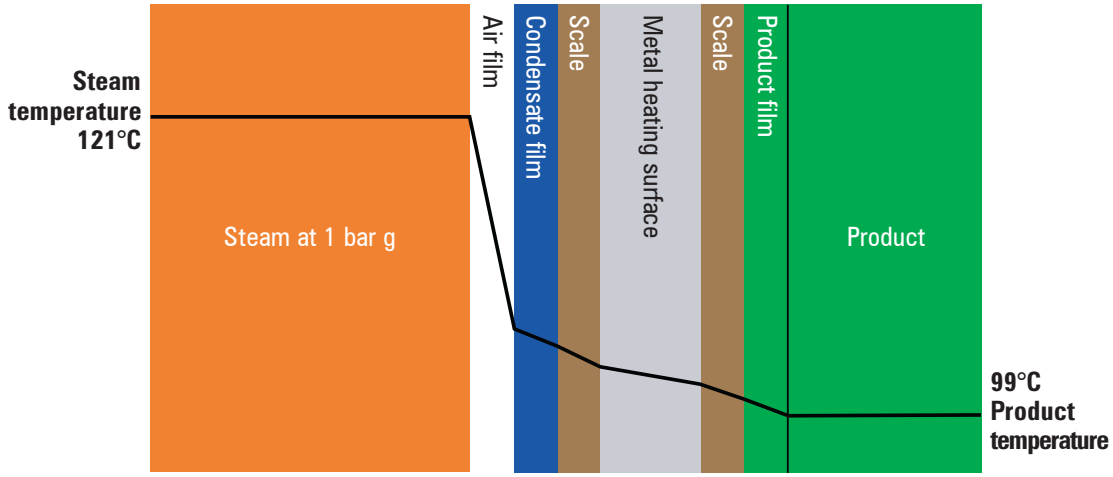


Fig. 2.5.4 Temperature gradients across heat transfer layers

Defining the overall heat transfer coefficient (U value)

The five main most commonly related terms associated with the subject of heat transfer are:

1.	Heat flowrate	$\dot{Q}$	(W)
2.	Thermal conductivity	k	(W/m°C)
3.	Thermal resistivity	r	(m°C/W)
4.	Thermal resistance	R	(m² °C/W)
5.	Thermal transmittance	U	(W/m² °C)

The following text in this Module describes them and how they are related to each other.

The traditional method for calculating heat transfer across a plane wall considers the use of an overall heat transfer coefficient 'U', or more correctly, the overall thermal transmittance between one side of the wall and the other.

U values are quoted for a wide range and combination of materials and fluids and are usually influenced by empirical data and operating experience. The previously mentioned films of condensate, air, scale, and product either side of the metal wall can have a significant effect on the overall thermal transmittance and because of this, it is worth considering the whole issue of heat transfer across a simple plane wall and then a multi-layer barrier.

Heat transfer by conduction through a simple plane wall

A good way to start is by looking at the simplest possible case, a metal wall with uniform thermal properties and specified surface temperatures.

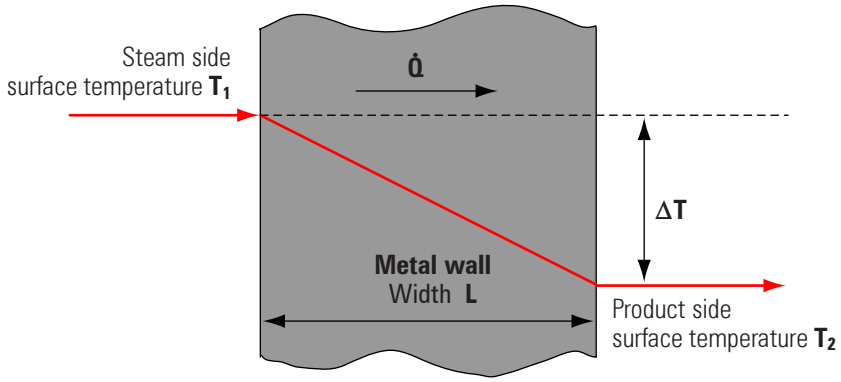


Fig. 2.5.5 Conductive heat transfer through a plane wall

T_1 and T_2 are the surface temperatures either side of the metal wall, of thickness L; and the temperature difference between the two surfaces is ΔT.

Ignoring the possible resistance to heat flow at the two surfaces, the process of heat flow through the wall can be derived from Fourier's law of conduction as shown in Equation 2.5.1. The term 'barrier' refers to a heat resistive film or the metal wall of a heat exchanger.

$$\dot{Q} = k\,A\,\frac{\Delta T}{\varkappa} \qquad \textbf{Equation 2.5.1}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
k = Thermal conductivity of the barrier (W/m K or W/m°C)
A = Heat transfer area (m²)
ΔT = Temperature difference across the barrier (K or °C)
$\varkappa$ = Barrier thickness (m)

It is possible to rearrange Equation 2.5.1 into Equation 2.5.6.

$$\dot{Q} = A\,\frac{\Delta T}{\varkappa/k} \qquad \textbf{Equation 2.5.6}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
A = Heat transfer area (m²)
ΔT = Temperature difference across the barrier (°C)
$\varkappa/k$ = Barrier thickness/material thermal conductivity $\left(\frac{m^2\,°C}{W}\right)$

It can be seen from their definitions in Equation 2.5.6 that $\varkappa/k$ is the thickness of the barrier divided by its inherent property of thermal conductivity. Simple arithmetic dictates that if the length ($\varkappa$) of the barrier increases, the value $\varkappa/k$ will increase, and if the value of the barrier conductivity (k) increases, then the value of $\varkappa/k$ will decrease. A characteristic that would behave in this fashion is that of thermal resistance. If the length of the barrier increases, the resistance to heat flow increases; and if the conductivity of the barrier material increases the resistance to heat flow decreases. It can be concluded that the term $\varkappa/k$ in Equation 2.5.6 relates to the thermal resistance of a barrier of known length.

The results of simple electrical theory parallel the equations appertaining to heat flow. In particular, the concept of adding resistances in series is possible, and is a useful tool when analysing heat transfer through a multi-layer barrier, as will be seen in a later section of this module.

Equation 2.5.6 can now be restated in terms of thermal resistance, where:

$$\text{Resistance } R = \frac{\text{Thickness}}{\text{Conductivity}}$$

$$R = \frac{x}{k} \left[\frac{m}{W/m^\circ C} \right]$$

$$R = \frac{x}{k} \left[\frac{m^2 °C}{W} \right]$$

as shown in Equation 2.5.7

$$\dot{Q} = A \frac{\Delta T}{R} \qquad \textbf{Equation 2.5.7}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
A = Heat transfer area (m^2)
ΔT = Temperature difference across the barrier (°C)
R = Thermal resistance of the barrier (m^2 °C/W)

Thermal resistance denotes a characteristic of a particular barrier, and will change in accordance to its thickness and conductivity.

In contrast, the barrier's ability to resist heat flow does not change, as this is a physical property of the barrier material. This property is called 'thermal resistivity'; it is the inverse of thermal conductivity and is shown in Equation 2.5.8.

$$r = \frac{1}{k} \qquad \textbf{Equation 2.5.8}$$

Where:
r = Thermal resistivity (m°C/W)
k = Thermal conductivity (W/m°C)

Equally, thermal conductivity $k = \frac{1}{r}$

If the thermal resistance is $\frac{x}{k}$ (from Equation 2.5.6), and $k = \frac{1}{r}$

Then thermal resistance is $\frac{x}{1/r} = x\,r$ = thickness x thermal resistivity

Relating the overall resistance to the overall U value

The usual problem that has to be solved in heat transfer applications is the rate of heat transfer, and this can be seen from the general heat transfer formula, Equation 2.5.3.

$$\dot{Q} = U\,A\,\Delta T \qquad \textbf{Equation 2.5.3}$$

Where:
U = The overall thermal transmittance (W/m²°C)

By comparing Equations 2.5.3 and 2.5.7, it must be true that:

$$Q = U\,A\,\Delta T = A\,\frac{\Delta T}{R}$$

and therefore,

$$U = \frac{1}{R} \qquad \textbf{Equation 2.5.9}$$

Therefore, U value (thermal transmittance) is the inverse of resistance.

Heat flow through a multi-layer barrier

As seen in Figure 2.5.4, a practical application would be the metal wall of a heat exchanger tube or plate which uses steam on one side to heat water on its other. It can also be seen that various other barriers are present slowing down the heat flow, such as an air film, a condensate film, a scale film, and a stationary film of secondary water immediately adjacent to the heating surface.

These films can be thought of as 'fouling' the flow of heat through the barrier, and consequently these resistances are considered by heat exchanger designers as 'fouling factors'.

All of these films, in addition to the resistance of the metal wall, constitute a resistance to heat flow and, as in an electrical circuit, these resistances can be added to form an overall resistance.

Therefore:

As : $U = \frac{1}{R}$ the overall U is the inverse of the sum of the resistance as shown in Equation 2.5.10.

$$U = \frac{1}{R_1 + R_2 + R_3 + R_4 + R_5 + R_6} \qquad \textbf{Equation 2.5.10}$$

Where:
R_1 = Resistance of the air film
R_2 = Resistance of the condensate film
R_3 = Resistance of the scale film on the steam side
R_4 = Resistance of the of the metal wall
R_5 = Resistance of the scale film on the water side
R_6 = Resistance of the product film

As resistance is x/k as shown in Equation 2.5.6, then Equation 2.5.10 can be rewritten as Equation 2.5.11:

$$U = \frac{1}{x_1/k_1 + x_2/k_2 + x_3/k_3 + x_4/k_4 + x_5/k_5 + x_6/k_6}$$ **Equation 2.5.11**

Where:

$$x_1/k_1 = \frac{\text{Thickness of air film}}{\text{Thermal conductivity of air}}$$

$$x_2/k_2 = \frac{\text{Thickness of condensate film}}{\text{Thermal conductivity of condensate}}$$

$$x_3/k_3 = \frac{\text{Thickness of scale film on steam side}}{\text{Thermal conductivity of scale}}$$

$$x_4/k_4 = \frac{\text{Thickness of metal wall}}{\text{Thermal conductivity of metal}}$$

$$x_5/k_5 = \frac{\text{Thickness of scale film on water side}}{\text{Thermal conductivity of scale}}$$

$$x_6/k_6 = \frac{\text{Thickness of water film}}{\text{Thermal conductivity of water}}$$

Table 2.5.2 Typical thermal conductivities of various materials

Material	Thermal conductivity W/m°C
Air	0.025
Condensate	0.4
Scale	0.1 to 1
Water	0.6
Steel	50
Copper	400

The thermal conductivities will alter depending on the film material (and temperature). For instance, air roughly has thirty times greater resistance to heat flow than water. For this reason, it is relatively more important to remove air from the steam supply before it reaches the heat exchanger, than to remove water in the form of wet steam. Of course, it is still sensible to remove wet steam at the same time.

The resistance of air to steel is roughly two thousand times more, and the resistance of air to copper is roughly twenty thousand times more. Because of the high resistances of air and water to that of steel and copper, the effect of small thicknesses of air and water on the overall resistance to heat flow can be relatively large.

There is no point in changing a steel heat transfer system to copper if air and water films are still present; there will be little improvement in performance, as will be proven in Example 2.5.5.

Air and water films on the steam side can be eradicated by good engineering practice simply by installing a separator and float trap set in the steam supply prior the control valve. Scale films on the steam side can also be reduced by fitting strainers in the same line.

Scale on the product side is a little more difficult to treat, but regular cleaning of heat exchangers is sometimes one solution to this problem. Another way to reduce scaling is to run heat exchangers at lower steam pressures; this reduces the steam temperature and the tendency for scale to form from the product, especially if the product is a solution like milk.

Example 2.5.5

Consider a steam to water heat exchanger where the air film, condensate film and scale on the steam side is 0.2 mm thick; on the water side, the water and scale films are 0.05 mm and 0.1 mm thick respectively.

The thickness of the steel walled heating surface is 6 mm.

Table 2.5.3 The resistance of the barriers including steel tube

Material	Thickness 'x' mm	Conductivity 'k' (W/m°C)	Resistance $R = x/k$ (m²°C/W)
Air	0.2	0.025	0.008
Condensate	0.2	0.4	0.000 5
Scale steam side	0.2	0.5	0.000 4
Steel tube	6.0	50.0	0.000 12
Water	0.05	0.6	0.000 08
Scale water side	0.1	0.5	0.000 2

From Equation 2.5.6:

1. Calculate the overall U value (U_1) from the conditions shown in Table 2.5.3

$$U = \frac{1}{x_1/k_1 + x_2/k_2 + x_3/k_3 + x_4/k_4 + x_5/k_5 + x_6/k_6} \qquad \textbf{Equation 2.5.11}$$

Where :

$x_1/k_1 = 0.008$

$x_2/k_2 = 0.000\,5$

$x_3/k_3 = 0.000\,4$

$x_4/k_4 = 0.000\,12$

$x_5/k_5 = 0.000\,08$

$x_6/k_6 = 0.000\,2$

$$U_1 = \frac{1}{0.008 + 0.000\,5 + 0.000\,4 + 0.000\,12 + 0.000\,08 + 0.000\,2}$$

$$U_1 = \frac{1}{0.009\,3}$$

$$U_1 = \mathbf{108\ W/m^2\,°C}$$

2. Remove the air and the condensate from the steam supply

Now consider the same heat exchanger where the air and condensate have been removed by a separator in the steam supply.

Calculate U_2

$$U_2 = \frac{1}{0.000\,4 + 0.000\,12 + 0.000\,08 + 0.000\,2}$$

$$U_2 = \frac{1}{0.000\,8}$$

$$\mathbf{U_2 = 1250\ W/m^2\,°C}$$

It can be seen from U_2 that by fitting a separator in the steam supply to this heat exchanger, and assuming that all air and condensate has been removed from the steam, the thermal transmittance is more than 11 times greater than the original value.

3. Remove the scale on the steam and water sides

Now consider reducing the scale on the steam side by fitting a strainer in the steam line, and reducing the scale on the water side by operating at a lower steam pressure.

Calculate U_3

$$U_3 = \frac{1}{0.000\,12 + 0.000\,08}$$

$$U_3 = \frac{1}{0.000\,2}$$

$$\mathbf{U_3 = 5\,000\ W/m^2\,°C}$$

The thermal transmittance has increased another fourfold by eradicating the scale.

4. Revert to the original conditions but change from steel tube to copper tube of the same thickness.

Table 2.5.4 The resistance of the barriers including copper tube

Material	Thickness 'x' mm	Conductivity 'k' (W/m°C)	Resistance $R = x/k$ (m² °C/W)
Air	0.2	0.025	0.008
Condensate	0.2	0.4	0.000 5
Scale steam side	0.2	0.5	0.000 4
Copper tube	6.0	400.0	0.000 015
Water	0.05	0.6	0.000 08
Scale water side	0.1	0.5	0.000 2

Calculate U_4

$$U_4 = \frac{1}{0.008 + 0.000\,5 + 0.000\,4 + 0.000\,12 + 0.000\,08 + 0.000\,2}$$

$$U_4 = \frac{1}{0.009\,2}$$

$$\mathbf{U_4 = 108.7\ W/m^2\,°C}$$

It can be seen that the greater conductivity offered by the copper over the steel has made very little difference to the overall thermal transmittance of the heat exchanger, due to the dominating effect of the air and other fouling factors.

Please note that, in practice, other factors will influence the overall U value, such as the velocities of the steam and water passing through the heat exchanger tubes or plates, and the combination of heat transfer by convection and radiation.

Also, it is unlikely that the fitting of a separator and strainer will completely eradicate the presence of air, wet steam, and scale from inside a heat exchanger. The above calculations are only being shown to highlight the effects of these on heat transfer. However, any attempt to remove such barriers from the system will generally prove successful, and is virtually guaranteed to increase heat transfer in steam heating plant and equipment as soon as this is done.

Rather than having to calculate individual resistances of film barriers, Tables exist showing overall U values for different types of heat exchange application such as steam coil heating of water or oil. These are documented in Module 2.10, 'Heating with coils and jackets'.

U values for heat exchangers vary considerably due to factors such as design ('shell and tube' or 'plate and frame' construction), material of construction, and the type of fluids involved in the heat transfer function.

Questions

1. What is the conductive heat transfer rate per unit area across a copper wall 5 mm thick, if the temperature on one side is 100°C and the temperature on the other is 40°C?

a| 21 000 W/m² ☐
b| 120 kW ☐
c| 4 800 kW/m² ☐
d| 33.3 W/mm² ☐

2. The rate of convective heat transfer from a plane surface with an area of 1.5 m² to a fluid in motion is 40 kW. If the surface temperature is 15°C and the fluid temperature is 40°C, what is the convective heat transfer coefficient?

a| 1 067 W/m² °C ☐
b| 667 kW °C/m² ☐
c| 1 500 kW m² °C ☐
d| 2 400 kW/m² °C ☐

3. According to the heat transfer equation, the heat transfer rate varies with:

a| The flowrate of the secondary fluid ☐
b| The mass flowrate of steam ☐
c| The temperature rise of the secondary fluid ☐
d| The mean temperature difference between the two fluids ☐

4. Steam at 3 bar g is used to heat water from 10°C to 80°C. What is the difference between the AMTD and the LMTD in this case?

a| 70°C ☐
b| 4.3°C ☐
c| 99°C ☐
d| 10°C ☐

5. The temperature gradient across a heat transfer layer is an indication of:

a| The thickness of the heat transfer layer ☐
b| The steam pressure ☐
c| The thermal conductivity of the heat transfer layer ☐
d| The mean temperature difference between the two fluids ☐

6. One side of a plane surface is at 25°C. A fluid at 70°C flows across the other surface. The convective heat transfer coefficient is 1 600 W/m² °C. What surface area is required to transfer 68 kW?

a| 0.944 m² ☐
b| 0.447 m² ☐
c| 0.894 m² ☐
d| 1.888 m² ☐

Answers

1: c, 2: a, 3: d, 4: b, 5: c, 6: a

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Module 2.6

Methods of Estimating Steam Consumption

Methods of Estimating Steam Consumption

The optimum design for a steam system will largely depend on whether the steam consumption rate has been accurately established. This will enable pipe sizes to be calculated, while ancillaries such as control valves and steam traps can be sized to give the best possible results. The steam demand of the plant can be determined using a number of different methods:

- **Calculation** - By analysing the heat output on an item of plant using heat transfer equations, it may be possible to obtain an estimate for the steam consumption. Although heat transfer is not an exact science and there may be many unknown variables, it is possible to utilise previous experimental data from similar applications. The results acquired using this method are usually accurate enough for most purposes.
- **Measurement** - Steam consumption may be determined by direct measurement, using flowmetering equipment. This will provide relatively accurate data on the steam consumption for an existing plant. However, for a plant which is still at the design stage, or is not up and running, this method is of little use.
- **Thermal rating** - The thermal rating (or design rating) is often displayed on the name-plate of an individual item of plant, as provided by the manufacturers. These ratings usually express the anticipated heat output in kW, but the steam consumption required in kg/h will depend on the recommended steam pressure.

  A change in any parameter which may alter the anticipated heat output, means that the thermal (design) rating and the connected load (actual steam consumption) will not be the same. The manufacturer's rating is an indication of the ideal capacity of an item and does not necessarily equate to the connected load.

Calculation

In most cases, the heat in steam is required to do two things:

- To produce a **change** in temperature in the product, that is providing a 'heating up' component.
- To **maintain** the product temperature as heat is lost by natural causes or by design, that is providing a 'heat loss' component.

In any heating process, the 'heating up' component will decrease as the product temperature rises, and the differential temperature between the heating coil and the product reduces. However, the heat loss component will increase as the product temperature rises and more heat is lost to the environment from the vessel or pipework.

The total heat demand at any time is the sum of these two components.

The equation used to establish the amount of heat required to raise the temperature of a substance (Equation 2.1.4, from module 1), can be developed to apply to a range of heat transfer processes.

$$Q = m\, c_p\, \Delta T \qquad \textbf{Equation 2.1.4}$$

Where:

Q = Quantity of energy (kJ)
m = Mass of the substance (kg)
c_p = Specific heat capacity of the substance (kJ/kg °C)
ΔT = Temperature rise of the substance (°C)

In its original form this equation can be used to determine a total amount of heat energy over the whole process. However, in its current form, it does not take into account the **rate** of heat transfer. To establish the rates of heat transfer, the various types of heat exchange application can be divided into two broad categories:

- Non-flow type applications - where the product being heated is a fixed mass and a single batch within the confines of a vessel.
- Flow type applications - where a heated fluid constantly flows over the heat transfer surface.

Non-flow type applications

In non-flow type applications the process fluid is held as a single batch within the confines of a vessel. A steam coil situated in the vessel, or a steam jacket around the vessel, may constitute the heating surface. Typical examples include hot water storage calorifiers as shown in Figure 2.6.1 and oil storage tanks where a large circular steel tank is filled with a viscous oil requiring heat before it can be pumped.

Some processes are concerned with heating solids; typical examples are tyre presses, laundry ironers, vulcanisers and autoclaves.

In some non-flow type applications, the process heat up time is unimportant and ignored. However, in others, like tanks and vulcanisers, it may not only be important but crucial to the overall process.

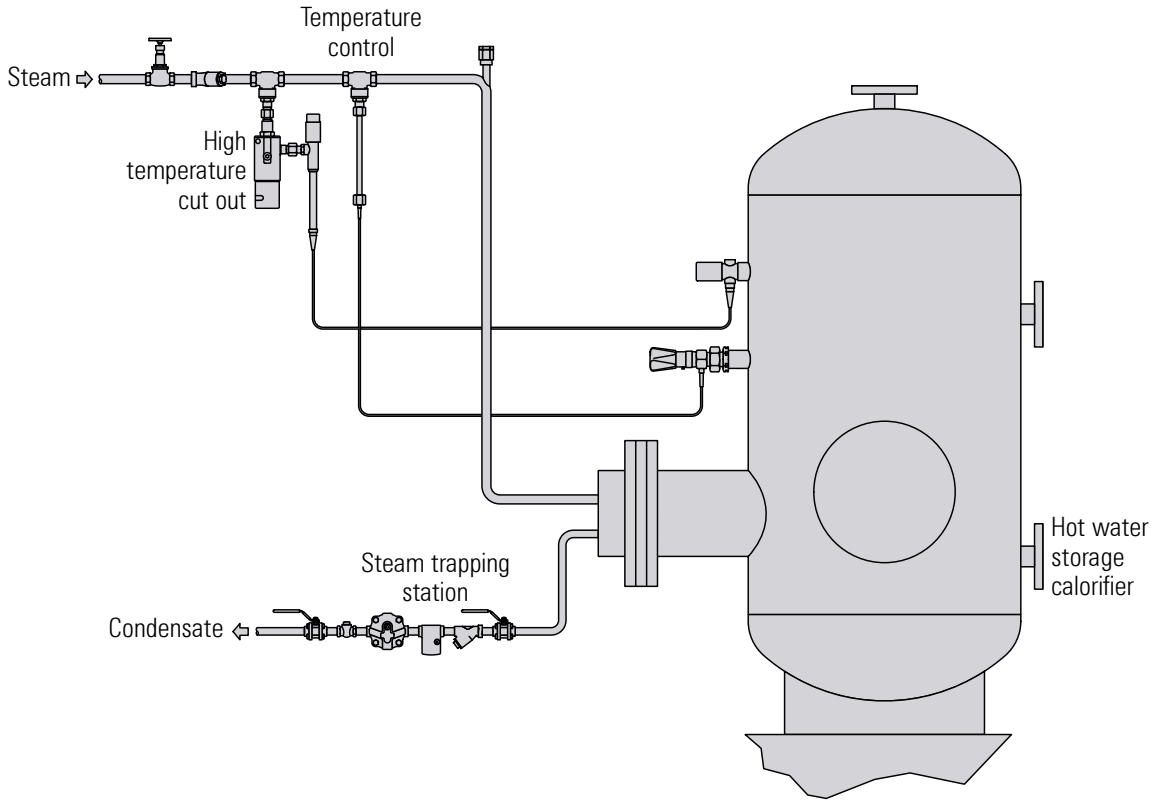


Fig. 2.6.1 Hot water storage - a non-flow application

Consider two non-flow heating processes requiring the same amount of heat energy but different lengths of time to heat up. The heat transfer rates would differ while the amounts of total heat transferred would be the same.

The mean rate of heat transfer for such applications can be obtained by modifying Equation 2.1.4 to Equation 2.6.1:

$$\dot{Q} = \frac{m\ c_p\ \Delta T}{t} \qquad \textbf{Equation 2.6.1}$$

Where:
$\dot{Q}$ = Mean heat transfer rate (kW (kJ/s))
m = Mass of the fluid (kg)
c_p = Specific heat capacity of the fluid (kJ/kg °C)
ΔT = Increase in fluid temperature (°C)
t = Time for the heating process (seconds)

Example 2.6.1
Calculating the mean heat transfer rate in a non-flow application.

A quantity of oil is heated from a temperature of 35°C to 120°C over a period of 10 minutes (600 seconds). The volume of the oil is 35 litres, its specific gravity is 0.9 and its specific heat capacity is 1.9 kJ/kg°C over that temperature range.

Determine the rate of heat transfer required:

As the density of water at Standard Temperature and Pressure (STP) is 1 000 kg/m³

The density of the oil $\rho_o = 0.9 \times 1000$

$\rho_o = 900$ kg/m³

As 1 000 litres = 1 m³, $\rho_o = 900$ kg/m³

$\rho_o = \mathbf{0.9\ kg/l}$

Therefore the mass of the oil = 0.9×35

$\dot{Q} = 31.5$ kg

$$\dot{Q} = \frac{31.5\ \text{kg} \times 1.9\ \text{kJ/kg°C} \times (120 - 35)\text{°C}}{600\ \text{seconds}}$$

$\mathbf{\dot{Q} = 8.48\ kJ/s\ (8.48\ kW)}$

Equation 2.6.1 can be applied whether the substance being heated is a solid, a liquid or a gas. However, it does not take into account the transfer of heat involved when there is a change of phase.

The quantity of heat provided by the condensing of steam can be determined by Equation 2.6.2:

$$Q = m_s\ h_{fg} \qquad \textbf{Equation 2.6.2}$$

Where:
Q = Quantity of heat (kJ)
m_s = Mass of steam (kg)
h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)

It therefore follows that the steam consumption can be determined from the heat transfer rate and vice-versa, from Equation 2.6.3:

$$\dot{Q} = \dot{m}_s \, h_{fg} \qquad \textbf{Equation 2.6.3}$$

Where:
$\dot{Q}$ = Mean heat transfer rate (kW or kJ/s)
$\dot{m}_s$ = Mean steam consumption (kg/s)
h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)

If it is assumed at this stage that the heat transfer is 100% efficient, then the heat provided by the steam must be equal to the heat requirement of the fluid to be heated. This can then be used to construct a heat balance, in which the heat energy supplied and required are equated:

$$\text{Primary side} = \dot{Q} = \text{Secondary side}$$

$$\dot{m}_s \, h_{fg} = \dot{Q} = \frac{m \, c_p \, \Delta T}{t} \qquad \textbf{Equation 2.6.4}$$

Where:
$\dot{m}_s$ = Mean steam consumption rate (kg/s)
h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)
$\dot{Q}$ = Mean heat transfer rate (kW (kJ/s))
m = Mass of the secondary fluid (kg)
c_p = Specific heat capacity of the secondary fluid (kJ/kg °C)
ΔT = Temperature rise of the secondary fluid (°C)
t = Time for the heating process (seconds)

Example 2.6.2
A tank containing 400 kg of kerosene is to be heated from 10°C to 40°C in 20 minutes (1 200 seconds), using 4 bar g steam. The kerosene has a specific heat capacity of 2.0 kJ/kg °C over that temperature range. h_{fg} at 4.0 bar g is 2 108.1 kJ/kg. The tank is well insulated and heat losses are negligible.

Determine the steam flowrate

$$\dot{Q} = \frac{400 \text{ kg} \times 2.0 \text{ kJ/kg °C} \times (40 - 10)\text{°C}}{1\,200 \text{ seconds}}$$

$$\dot{\mathbf{Q}} = \mathbf{20\ kJ/s}$$

Therefore:

$$\dot{m}_s = \frac{20 \text{ kJ/s}}{2108.1 \text{ kJ/kg}}$$

$$\dot{m}_s = 0.0095 \text{ kg/s}$$

$$\dot{\mathbf{m}}_s = \mathbf{34.2\ kg/h}$$

In some non-flow type applications, the length of time of the batch process may not be critical, and a longer heat up time may be acceptable. This will reduce the instantaneous steam consumption and the size of the required plant equipment.

Flow type applications

Typical examples include shell and tube heat exchangers, see Figure 2.6.2 (also referred to as non-storage calorifiers) and plate heat exchangers, providing hot water to heating systems or industrial processes. Another example would be an air heater battery where steam gives up its heat to the air that is constantly passing through.

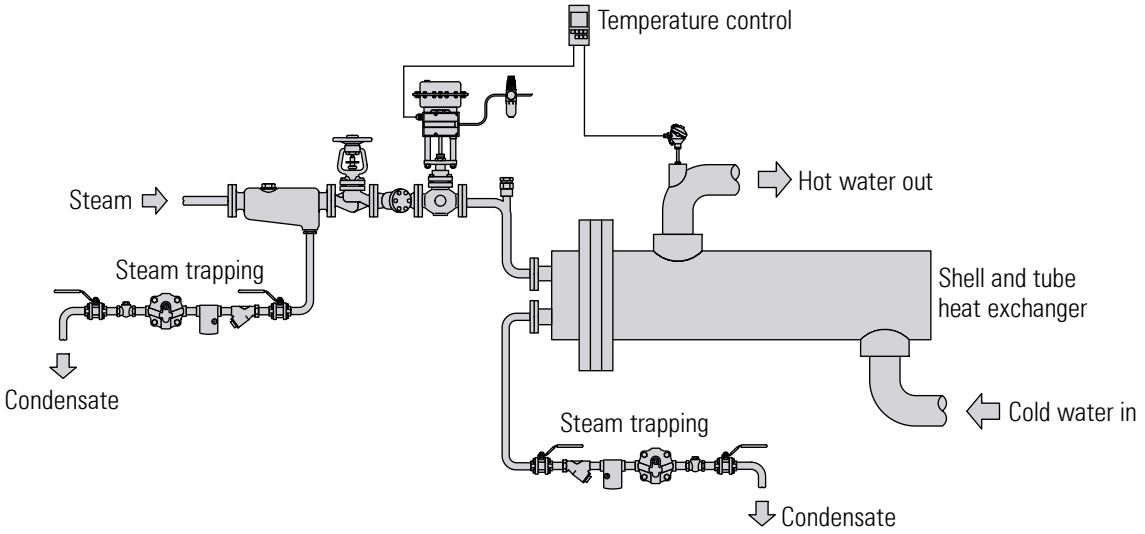


Fig 2.6.2 Non-storage calorifier

Figure 2.6.3 provides a typical temperature profile in a heat exchanger with a constant secondary fluid flowrate. The condensing temperature (T_s) remains constant throughout the heat exchanger. The fluid is heated from T_1 at the inlet valve to T_2 at the outlet of the heat exchanger.

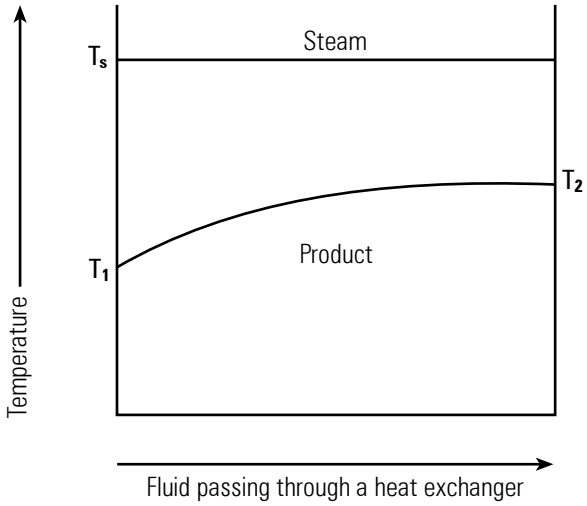


Fig. 2.6.3 Typical temperature profile in a heat exchanger

For a fixed secondary flowrate, the required heat load ($\dot{Q}$) is proportional to the product temperature rise (ΔT). Using Equation 2.6.1:

$$\dot{Q} = \frac{m\ c_p\ \Delta T}{t} \qquad \textbf{Equation 2.6.1}$$

$\frac{m}{t}$ = Product flowrate = constant

c_p = Specific heat = constant

Therefore: $\dot{Q} \propto \Delta T$

As flowrate is mass flow per unit time, the secondary flowrate is depicted in equation 2.6.1 as:

$$\frac{m}{t}$$

This can be represented by $\dot{m}$, where $\dot{m}$ is the secondary fluid flowrate in kg/s, and is shown in equation 2.6.5.

$$\dot{Q} = \dot{m}\, c_p\, \Delta T \qquad \textbf{Equation 2.6.5}$$

$\dot{Q}$ = Mean heat transfer rate (kW)
$\dot{m}$ = Mean secondary fluid flowrate (kg/s)
c_p = Specific heat capacity of the secondary fluid (kJ/kg K) or (kJ/kg°C)
ΔT = Temperature rise of the secondary fluid (K or °C)

A heat balance equation can be constructed for flow type applications where there is a continuous flow of fluid:

Primary side = $\dot{Q}$ = Secondary side

$$\dot{m}_s\, h_{fg} = \dot{Q} = \dot{m}\, c_p\, \Delta T \qquad \textbf{Equation 2.6.6}$$

Where:
$\dot{m}_s$ = Mean steam consumption rate (kg/s)
h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)
$\dot{Q}$ = Mean heat transfer rate (kW (kJ/s))
$\dot{m}$ = Mass flowrate of the secondary fluid (kg/s)
c_p = Specific heat capacity of the secondary fluid (kJ/kg °C)
ΔT = Temperature rise of the secondary fluid (°C)

Mean steam consumption

The mean steam consumption of a flow type application like a process heat exchanger or heating calorifier can be determined from Equation 2.6.6, as shown in Equation 2.6.7.

$$\dot{m}_s = \frac{\dot{m}\, c_p\, \Delta T}{h_{fg}} \qquad \textbf{Equation 2.6.7}$$

Where:
$\dot{m}_s$ = Mean steam consumption rate (kg/s)
$\dot{m}$ = Mass flowrate of the secondary fluid (kg/s)
c_p = Specific heat capacity of the secondary fluid (kJ/kg °C)
ΔT = Temperature rise of the secondary fluid (°C)
h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)

Equally, the mean steam consumption can be determined from Equation 2.6.6 as shown in Equation 2.6.8.

$$\dot{m}_s = \frac{\dot{Q}}{h_{fg}} \qquad \textbf{Equation 2.6.8}$$

But as the mean heat transfer is, itself, calculated from the mass flow, the specific heat, and the temperature rise, it is easier to use Equation 2.6.7.

Example 2.6.3

Dry saturated steam at 3 bar g is used to heat water flowing at a constant rate of 1.5 l/s from 10°C to 60°C.

h_{fg} at 3 bar g is 2 133.4 kJ/kg, and the specific heat of water is 4.19 kJ/kg °C

Determine the steam flowrate from Equation 2.6.7:

As 1 litre of water has a mass of 1 kg, the mass flowrate = 1.5 kg/s

$$\dot{m}_s = \frac{\dot{m}\ c_p\ \Delta T}{h_{fg}} \qquad \textbf{Equation 2.6.7}$$

$$\dot{m}_s = \frac{1.5 \times 4.19 \times (60 - 10)}{2\,133.4}$$

$$\dot{m}_s = 0.147\,3 \text{ kg/s}$$

$$\dot{m}_s = \mathbf{530\ kg/h}$$

At start-up, the inlet temperature, T_1 may be lower than the inlet temperature expected at the full running load, causing a higher heat demand. If the warm-up time is important to the process, the heat exchanger needs to be sized to provide this increased heat demand. However, warm-up loads are usually ignored in flow type design calculations, as start-ups are usually infrequent, and the time it takes to reach design conditions is not too important. The heat exchanger heating surface is therefore usually sized on the running load conditions.

In flow type applications, heat losses from the system tend to be considerably less than the heating requirement, and are usually ignored. However, if heat losses are large, the mean heat loss (mainly from distribution pipework) should be included when calculating the heating surface area.

Warm-up and heat loss components

In any heating process, the warm-up component will decrease as the product temperature rises, and the differential temperature across the heating coil reduces. However, the heat loss component will increase as the product and vessel temperatures rise, and more heat is lost to the environment from the vessel or pipework. The total heat demand at any time is the sum of these two components.

If the heating surface is sized only with consideration of the warm-up component, it is possible that not enough heat will be available for the process to reach its expected temperature. The heating element, when sized on the sum of the mean values of both these components, should normally be able to satisfy the overall heat demand of the application.

Sometimes, with very large bulk oil storage tanks for example, it can make sense to maintain the holding temperature lower than the required pumping temperature, as this will reduce the heat losses from the tank surface area. Another method of heating can be employed, such as an outflow heater, as shown in Figure 2.6.4.

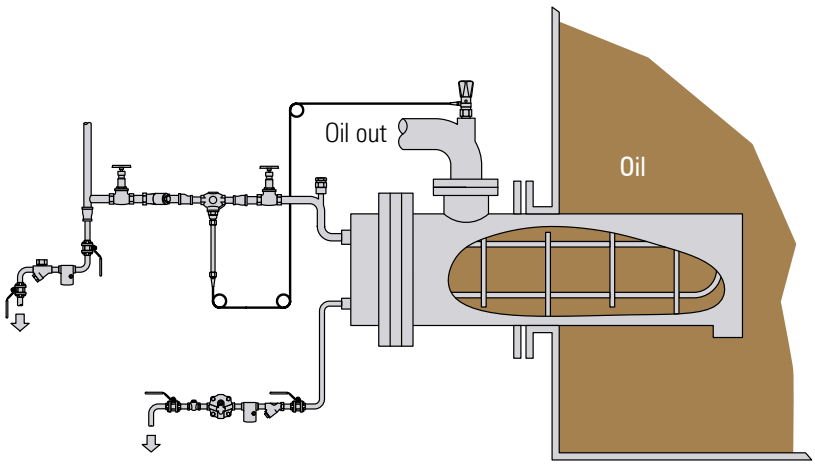


Fig. 2.6.4 An outflow heater

Heating elements are encased in a metal shroud protruding into the tank and designed such that only the oil in the immediate vicinity is drawn in and heated to the pumping temperature. Heat is therefore only demanded when oil is drawn off, and since the tank temperature is lowered, lagging can often be dispensed with. The size of outflow heater will depend on the temperature of the bulk oil, the pumping temperature and the pumping rate.

Adding materials to open topped process tanks can also be regarded as a heat loss component which will increase thermal demand. These materials will act as a heat sink when immersed, and they need to be considered when sizing the heating surface area.

Whatever the application, when the heat transfer surface needs calculating, it is first necessary to evaluate the total mean heat transfer rate. From this, the heat demand and steam load may be determined for full load and start-up. This will allow the size of the control valve to be based on either of these two conditions, subject to choice.

Questions

1. A tank of water is to be heated by a steam coil from 15°C to 65°C in 30 minutes.
The tank measures 0.7 m x 0.7 m x 1 m high.
The water is 0.8 m deep.
The specific heat capacity of the water is 4.19 kJ/kg°C.
Steam is supplied to the coil at 4 bar g.
From the given information what will be the nearest to the steam flowrate required?
(For this question ignore heat losses from the liquid surface and tank sides)

a| 78 kg/h ☐
b| 54 kg/h ☐
c| 91 kg/h ☐
d| 45 kg/h ☐

2. Referring to Question 1, what will be the effect on the required steam flowrate if the tank is heated in 1 h?

a| The steam flowrate will be halved ☐
b| The steam flowrate will be doubled ☐
c| The steam flowrate will remain the same ☐
d| The heat required to raise the water will be doubled ☐

3. In Question 1 other energy requirements should be taken into account for a more accurate final steam demand. Which of the following would account for the greatest heat requirement?

a| Losses from the tank sides ☐
b| Losses in heating the tank material ☐
c| Losses from the bottom of the tank ☐
d| Losses from the liquid surface ☐

4. An air heater battery has a rating of 50 kW when supplied with steam at 7 bar g. What will be its steam consumption?

a| 88 kg/h ☐
b| 96 kg/h ☐
c| 43 kg/h ☐
d| 72 kg/h ☐

5. If the air heater battery in Question 4 is actually supplied with steam at 5 bar g what will be the effect on its heat output?

a| The rating will be increased ☐
b| There will be no effect ☐
c| The rating will be reduced ☐
d| Condensate removal will be difficult ☐

6. Oil passing through a heater is heated from 38°C to 121°C and flows at the rate of 550 l/h.
Steam is supplied to the heater at 5 bar g.
The oil has a specific heat capacity of 1.9 kJ/kg °C, and a density of 850 kg/m³.
What will be the steam flowrate?

a| 25 kg/h ☐

b| 40 kg/h ☐

c| 35 kg/h ☐

d| 97 kg/h ☐

Answers

1: a, 2: a, 3: d, 4: a, 5: c, 6: c

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Module 2.7

Measurement of Steam Consumption

Measurement of Steam Consumption

By a steam flowmeter

The use of a steam flowmeter may be used to directly measure the steam usage of an operational item of plant. This may be used to monitor the results of energy saving schemes and to compare the efficiency of one item of plant with another. The steam can then be costed as a raw material at any stage of the production process, so that the cost of individual product lines may be determined.

It is only in comparatively rare cases that a meter cannot measure steam flow. Care should be taken, however, to ensure that the prevailing steam pressure is considered and that no other calibration factor has been overlooked.

Steam flowmetering is discussed in detail in Block 4.

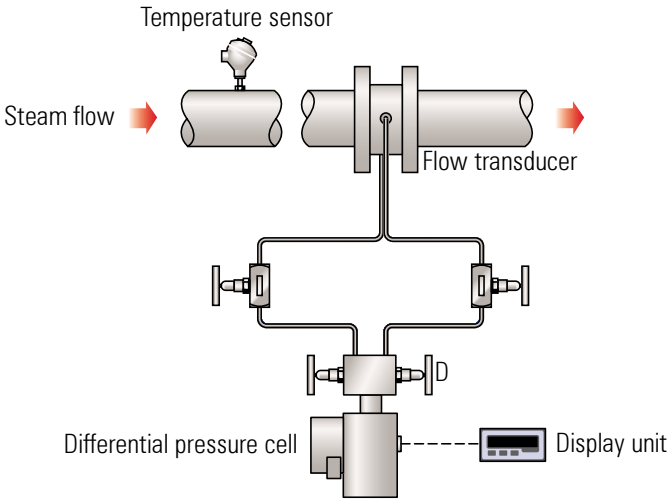


Fig. 2.7.1 Typical steam flowmeter installation

By a condensate pump

A less accurate method of estimating the steam consumption is by incorporating a counter into the body of a positive displacement pump used to pump condensate from the process. Each discharge stroke is registered, and an estimate of the capacity of each stroke is used to calculate the amount of steam condensed over a given time period.

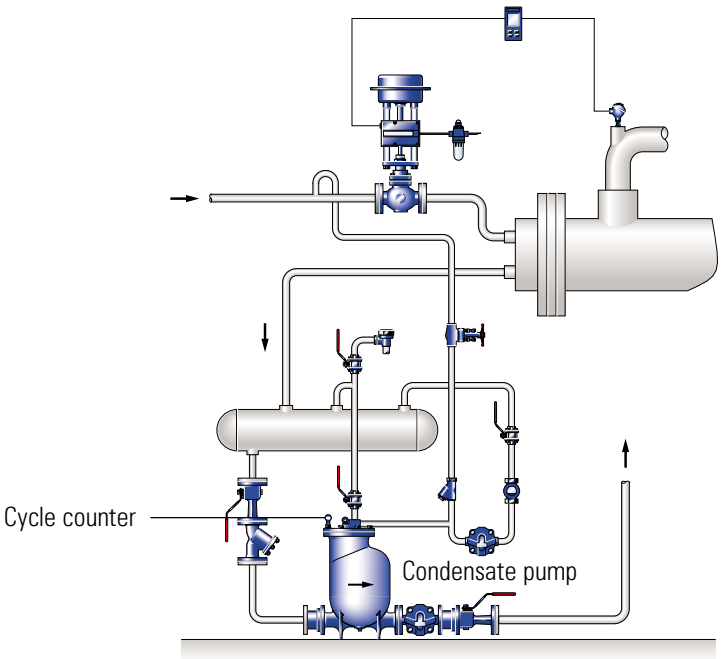


Fig. 2.7.2 Positive displacement pump with cycle counter

A purpose built electronic pump monitor can be used which enables this to be carried out automatically, converting the pump into a condensate meter. The electronic pump monitor can be read locally or can return digital data to a central monitoring system. If the pump is draining a vented receiver, a small allowance has to be made for flash steam losses.

By collecting the condensate

Steam consumption can also be established directly, by measuring the mass of condensate collected in a drum over a period of time. This may provide a more accurate method than using theoretical calculations if the flash steam losses (which are not taken into account) are small, and can work for both non-flow and flow type applications. However, this method cannot be used in direct steam injection applications, humidification or sterilisation processes, where it is not possible to collect the condensate.

Figure 2.7.3 shows a test being carried out on a jacketed pan. In this case an empty oil drum and platform scales are shown, but smaller plant can be tested just as accurately using a bucket and spring balance. This method is quite easy to set up and can be relied upon to give accurate results.

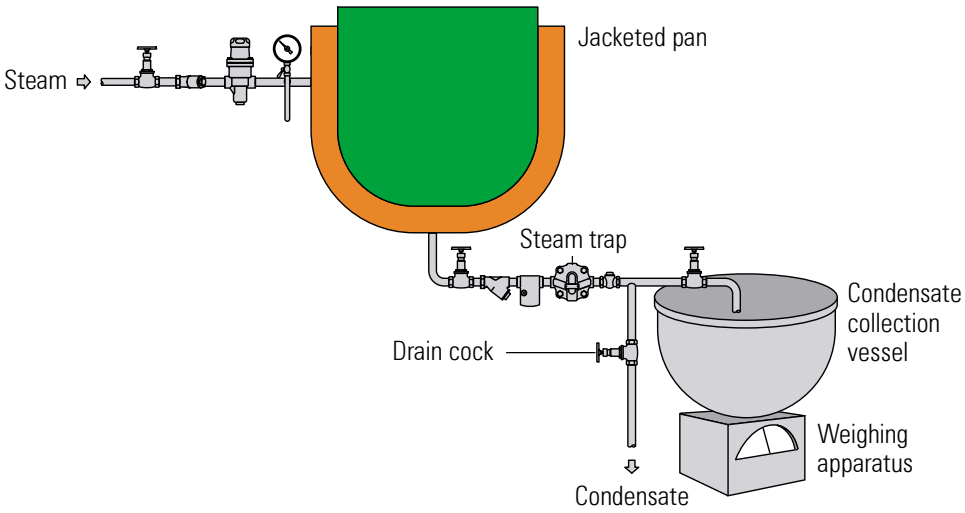


Fig. 2.7.3 Equipment for measurement of steam consumption

The drum is first weighed with a sufficient quantity of cold water. Steam is then supplied to the plant, and any condensate is discharged below the water level in the container to condense any flash steam. By noting the increase in weight over time, the mean steam consumption can be determined.

Although this method gives the mean rate of steam consumption, if the weight of condensate is noted at regular intervals during the test, the corresponding steam consumption rates can be calculated. Any obvious peaks will become apparent and can be taken into account when deciding on the capacity of associated equipment. It is important to note that the test is conducted with the condensate discharging into an atmospheric system. If the test is being used to quantify steam consumption on plant that would otherwise have a condensate back pressure, the steam trap capacity must relate to the expected differential pressure.

Care must also be taken to ensure that only condensate produced during the test run is measured. In the case of the boiling pan shown, it would be wise to drain the jacket completely through the drain cock before starting the test. At the end, drain the jacket again and add this condensate to that in the container before weighing.

The test should run for as long as possible in order to reduce the effect of errors of measurement. It is always advisable to run three tests under similar conditions and average the results in order to get a reliable answer. Discard any results that are widely different from the others and, if necessary, run further tests.

If the return system includes a collecting tank and pump, it may be possible to stop the pump for a period and measure condensate volume by carefully dipping the tank before and after a test period. Care must be taken here, particularly if the level change is small or if losses occur due to flash steam.

Questions

Relative questions on this subject will be asked in Block 4, 'Steam Flowmetering'.

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Module 2.8

Thermal Rating

Thermal Rating

Some items of manufactured plant are supplied with information on thermal output. These design ratings can be both helpful and misleading. Ratings will usually involve raising a stated amount of air, water or other fluid through a given temperature rise, using steam at a specified pressure. They are generally published in good faith with a reasonable allowance for fouling of the heat transfer surface.

It must be clear that changing any factor at all will alter the predicted heat output and thereby the steam consumption. A secondary fluid which is colder than specified will increase the demand, while steam at less than the specified pressure will reduce the ability to transfer heat.

Temperature and pressure can often be measured easily so that corrections can be applied. However, flowrates of air, water and other fluids may be far more difficult to measure. Undetected fanbelt slip or pump impeller wear can also lead to discrepancies, while lower than expected resistances applied to pumps and fans can cause flowrates to be higher than the design values.

A more common source of error arises from the assumption that the manufacturer's rating equates to actual load. A heat exchanger may be capable of meeting or exceeding a given duty, but the connected load may often only be a fraction of this. Clearly it is useful to have information on the thermal rating of plant, but care must be taken when relating this to an actual heat load.

If the load is quoted in kW, and the steam pressure is given, then steam flowrate may be determined as shown in Equation 2.8.1:

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW} \times 3\,600}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

XYZ Heat Exchanger Company

Serial Number		HX12345	
Pressures		SHELL	TUBE
	NWP	-	4.0 bar g
	MAWP	10 bar g	12.0 bar g
	MAAP	11 bar g	13.2 bar g
	TEST	15 bar g	24.0 bar g
Main shell thickness		5 mm	
Date of hydraulic test		1985	
Design code - shell		BS 853	
Design code - tubes		BS 853	
Design rating		250 kW	

Fig. 2.8.1 Typical heat exchanger manufacturer's name-plate

Questions

1. What is the result of using a heat exchanger rating to calculate its steam consumption?

a | The true connected heat load may be different from the rated figure ☐

b | The rating does not take account of the temperature of the secondary medium ☐

c | The rating is based on a steam pressure of 1.0 bar ☐

d | The rating does not allow for condensate forming in the heat exchanger ☐

2. A heat exchanger has a design rating based on a working pressure of 7 bar g. What would be the effect of supplying the exchanger with steam at 3 bar g?

a | The heat output would be greater because the enthalpy of evaporation at 3 bar g is higher than at 7 bar g ☐

b | The heat output would be greater because steam at 3 bar g has a greater volume than steam at 7 bar g ☐

c | Less weight of steam would be required because steam at 3 bar g has a higher enthalpy of evaporation than at 7 bar g ☐

d | The output would be reduced because the difference in temperature between the steam and product is reduced ☐

Answers

1: a, 2: d

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Module 2.9

Energy Consumption of Tanks and Vats

Energy Consumption of Tanks and Vats

The heating of liquids in tanks is an important requirement in process industries such as the dairy, metal treatment and textile industries. Water may need to be heated to provide a hot water utility; alternatively, a liquid may need to be heated as part of the production process itself, whether or not a chemical reaction is involved. Such processes may include boiler feedtanks, wash tanks, evaporators, boiling pans, coppers, calandrias and reboilers.

Tanks are often used for heating processes, of which there are two major categories:

- Totally enclosed tanks, such as those used for storing fuel oil, and where heat load calculations are generally straightforward.
- Open topped tanks, where heat load calculations may be complicated by the introduction of articles and materials, or by evaporative losses.

Open and closed tanks are used for a large number of process applications:

- **Boiler feedtanks -** The boiler feedtank is at the heart of any steam generation system. It provides a reservoir of returned condensate and treated make-up water, for feeding the boiler. One reason for heating the water is to reduce oxygen entering the boiler, with (theoretically) 0 ppm oxygen at 100°C.

  Boiler feedtanks are normally operated at between 80°C and 90°C.

- **Hot water tanks -** Hot water is required for a number of processes in industry. It is often heated in simple, open or closed tanks which use steam as the heating medium.

  The operating temperature can be anywhere between 40°C and 85°C depending on the application.

- **Degreasing tanks -** Degreasing is the process where deposits of grease and cooling oil are removed from metal surfaces, after machining and prior to the final assembly of the product.

  In a degreasing tank, the material is dipped into a solution, which is heated by coils to a temperature of between 90°C and 95°C.

- **Metal treatment tanks -** Metal treatment tanks, which are sometimes called vats, are used in a number of different processes:

  - To remove scale or rust.
  - To apply a metallic coating to surfaces.

  The treatment temperatures typically range from 70°C to 85°C.

- **Oil storage tanks -** Storage tanks are required to hold oils which cannot be pumped at ambient temperatures, such as heavy fuel oil for boilers. At ambient temperatures, heavy oil is very thick and must be heated to 30°C - 40°C in order to reduce its viscosity and allow it to be pumped. This means that all heavy oil storage tanks need to be provided with heating to facilitate pumping.
- **Heating tanks used in process industries -** Heating tanks are used by a number of process industries, see Table 2.9.1.

Table 2.9.1 Process industries which use heating tanks

Industry	Process	Typical temperatures
Sugar	Raw juice heating	80 to 85°C
Dairy	Hot water generation	80°C
Plating	Metal deposition	70 to 85°C
Metal / steel	Removal of rust / scale	90 to 95°C
Pharmaceutical	Wash tanks	70°C
Rubber	Heating caustic oil	140°C

In some applications the process fluid may have achieved its working temperature, and the only heat requirement may be due to losses from the solid surface of the walls and/or the losses from the liquid surface.

This Module will deal with the calculations which determine the energy requirements of tanks: the following two Modules (2.10 and 2.11) will deal with how this energy may be provided.

When determining the heat requirement of a tank or vat of process fluid, the total heat requirement may consist of some or all of a number of key components:

1. The heat required to raise the process fluid temperature from cold to its operating temperature.

2. The heat required to raise the vessel material from cold to its operating temperature.

3. The heat lost from the solid surface of the vessel to the atmosphere.

4. The heat lost from the liquid surface exposed to the atmosphere.

5. The heat absorbed by any cold articles dipped into the process fluid.

However, in many applications only some of the above components will be significant. For example, in the case of a totally enclosed well-insulated bulk oil storage tank, the total heat requirement may be made up almost entirely of the heat required to raise the temperature of the fluid.

Items 1 and 2, the energy required to raise the temperature of the liquid and the vessel material, and item 5, the heat absorbed by any cold articles dipped into the process fluid, can be found by using the Equation 2.6.1. Generally, data can be accurately defined, and hence the calculation of the heat requirement is straightforward and precise.

$$\dot{Q} = \frac{m\, c_p\, \Delta T}{t} \qquad \textbf{Equation 2.6.1}$$

Items 3 and 4, the heat losses from the vessel and liquid surfaces can be determined by using Equation 2.5.3.

However, heat loss calculations are much more complex, and usually empirical data, or tables based on several assumptions have to be relied upon. It follows that heat loss calculations are less accurate.

$$\dot{Q} = U\, A\, \Delta T \qquad \textbf{Equation 2.5.3}$$

Heat loss from the solid surface of the vessel to the atmosphere

Heat will only be transferred provided there is a difference in temperature between the surface and the ambient air.

Figure 2.9.1 provides some typical overall heat transfer coefficients for heat transfer from bare steel flat surfaces to ambient air. If the bottom of the tank is not exposed to ambient air, but is positioned flat on the ground, it is usual to consider this component of the heat loss to be negligible, and it may safely be ignored.

- For 25 mm of insulation, the U value should be multiplied by a factor of 0.2
- For 50 mm of insulation, the U value should be multiplied by a factor of 0.1

The overall heat transfer coefficients provided in Figure 2.9.1 are for 'still air' conditions only.

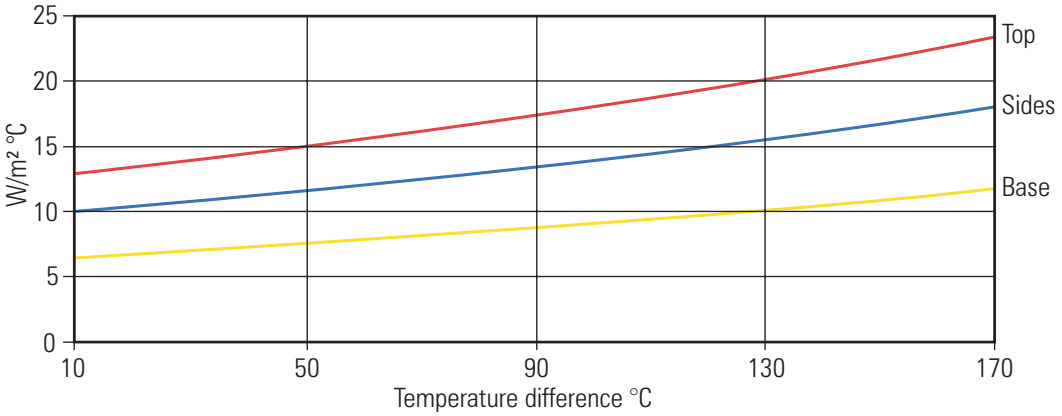


Fig. 2.9.1 Typical overall heat transfer coefficients from flat steel surfaces

Table 2.9.2 shows multiplication factors which need to be applied to these values if an air velocity is being taken into account. However, if the surface is well insulated, the air velocity is not likely to increase the heat loss by more than 10% even in exposed conditions.

Table 2.9.2 Effect on heat transfer with air movement

Velocity (m/s)	0	1	2	4	6	8	10	12	14	16
Velocity (km/h)	0	3.6	7.2	14.4	21.6	28.8	36	43.2	50.4	57.6
Factor X	1	1.4	1.7	2.4	3.0	3.6	4.1	4.5	4.9	5.2

Velocities of less than 1 m/s can be considered as sheltered conditions, whilst 5 m/s may be thought of as a gentle breeze (about 3 on the Beaufort scale), 10 m/s a fresh breeze (Beaufort 5), and 16 m/s a moderate gale (Beaufort 7).

For bulk oil storage tanks, the overall heat transfer coefficients quoted in Table 2.9.3 may be used.

Table 2.9.3 Overall heat transfer coefficients for oil tanks

Tank position	ΔT between oil and air	Overall heat transfer coefficient (W/m²°C)	
		Unlagged	Lagged
Sheltered	Up to 10°C	6.8	1.7
	Up to 27°C	7.4	1.8
	Up to 38°C	8.0	2.0
Exposed	Up to 10°C	8.0	2.0
	Up to 27°C	8.5	2.1
	Up to 38°C	9.1	2.3
Underground	Any temperature	6.8	-

Water tanks: heat loss from the water surface to the atmosphere

Figure 2.9.2 relates heat loss from a water surface to air velocity and surface temperature. In this chart 'still' air is considered to have a velocity of 1 m/s, tanks in sheltered positions outdoors consider velocities at about 4 m/s, whilst tanks in exposed positions outdoors are considered with velocities at about 8 m/s.

This chart provides the heat loss in W/m² rather than the units of the overall heat transfer coefficient of W/m²°C. This means that this value must be multiplied by the surface area to provide a rate of heat transfer, as the water to air temperature difference has already been taken into account.

Heat losses from the water surface, as shown in Figure 2.9.2 are not significantly affected by the humidity of the air. The full range of humidities likely to be encountered in practice is covered by the thickness of the curve. However, the graph considers heat losses with an air temperature of 15.6°C and 55% air humidity. Different conditions to these can be calculated from the Engineering Support Centre on the Spirax Sarco website.

To determine the heat loss from the chart, the water surface temperature must be selected from the top scale. A line should then be projected vertically downwards to the (bold) heat loss curve. For indoor tanks a line should be projected horizontally from the intersection to the left-hand scale.

For outdoor tanks a horizontal line should be projected either left or right until it intersects the required location, either sheltered or exposed. A projection vertically downwards will then reveal the heat loss on the bottom scale.

In most cases, the heat loss from the liquid surface is likely to be the most significant heat loss element. Where practical, heat loss can be limited by covering the liquid surface with a layer of polystyrene spheres which provide an insulating 'blanket'. Any solution to reduce heat losses becomes even more important when tanks are located outside in exposed positions as portrayed by the graph in Figure 2.9.2

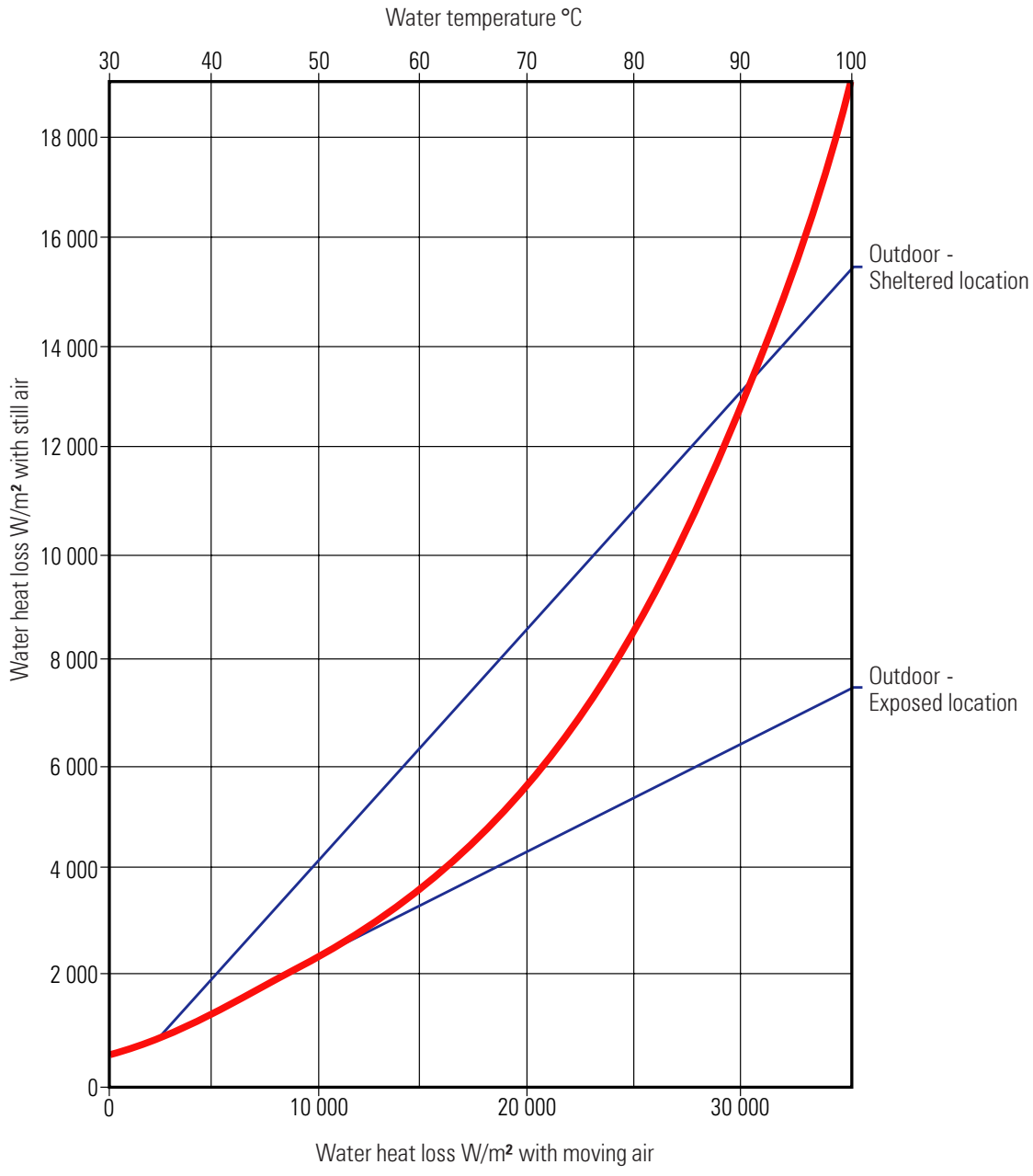


Fig. 2.9.2 Heat loss from water surfaces

Example 2.9.1

For the tank shown in Figure 2.9.3, determine:

Part 1. The mean heat transfer rate required during start-up.

Part 2. The maximum heat transfer rate required during operation.

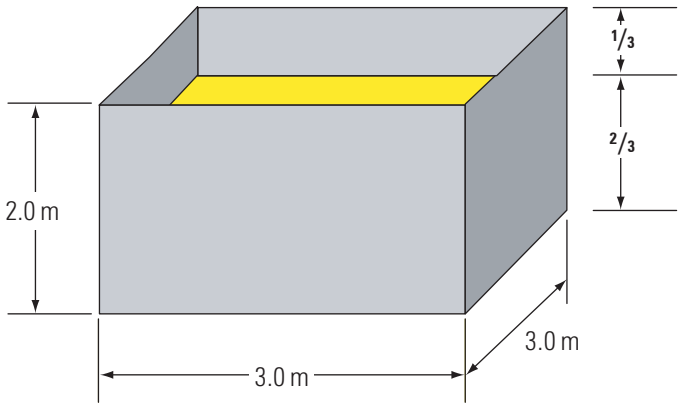


Fig. 2.9.3

- The tank is unlagged and open topped and is situated on a concrete floor inside a factory. It is 3 m long by 3 m wide by 2 m high.
  Tank total surface area = 24 m² (excluding base).
  Heat transfer coefficient from tank/air, U_1 = 11 W/m²°C.
  The tank is ⅔ full of a weak acid solution (c_p = 3.9 kJ/kg°C) which has the same density as water (1 000 kg/m³)
- The tank is fabricated from 15 mm mild steel plate.
  (Density = 7 850 kg/m³, c_p = 0.5 kJ/kg°C)
- The tank is used on alternate days, when the solution needs to be raised from the lowest considered ambient temperature of 8°C to 60°C in 2 hours, and remain at that temperature during the day.
- When the tank is up to temperature, a 500 kg steel article is to be dipped every 20 minutes without the tank overflowing. (c_p = 0.5 kJ/kg°C)

Part 1 Determine the mean heat transfer rate required during start-up $\dot{Q}_{M\ (start\text{-}up)}$

This is the sum of:

A1. Heating the liquid $\dot{Q}_{M\ (liquid)}$

A2. Heating the tank material $\dot{Q}_{M\ (tank)}$

A3. Heat losses from the sides of the tank $\dot{Q}_{M\ (sides)}$

A4. Heat losses from the liquid surface $\dot{Q}_{M\ (surface)}$

Part 1.1 Heating the liquid $\dot{Q}_{M\ (liquid)}$

Initial temperature T_1 = 8°C

Final temperature T_2 = 60°C

Temperature rise ΔT = 60 - 8

Temperature rise ΔT = 52°C

Volume of liquid = 2 x 3 x 3 x ⅔

Volume of liquid = 12 m³

Mass of liquid m = 12 000 kg

Specific heat c_p = 3.9 kJ/kg°C

Heating time t = 2 hours

t = 7 200 seconds

$$\dot{Q} = \frac{m\ c_p\ \Delta T}{t} \qquad \textbf{Equation 2.6.1}$$

$$\dot{Q}_{M\,(liquid)} = \frac{m\ c_p\ \Delta T}{7\,200}$$

$$\dot{Q}_{M\,(liquid)} = \frac{12\,000\ x\ 3.9\ x\ 52}{7\,200}$$

$$\dot{Q}_{M\,(liquid))} = \mathbf{338\ kW}$$

Part 1.2 Heating the tank material $\dot{Q}_{M\,(tank)}$

Tank plate thickness = 0.015 m

Volume of mild steel = ((2 m x 3 m x 4) + (3 m x 3 m)) x 0.015 m

Volume of mild steel = 0.495 m³

Mass of mild steel = 0.495 m³ x 7 850 kg/m³

Mass of mild steel = 3 886 kg

Using Equation 2.6.1:

$$\dot{Q}_{M\,(tank)} = \frac{m\ c_p\ \Delta T}{7\,200}$$

$$\dot{Q}_{M\,(tank)} = \frac{3\,886\ x\ 0.5\ x\ 52}{7\,200}$$

$$\dot{Q}_{M\,(tank)} = \mathbf{14\ kW}$$

Part 1.3 Heat losses from tank sides $\dot{Q}_{M\,(sides)}$

$$\dot{Q} = U\ A\ \Delta T \qquad \textbf{Equation 2.5.3}$$

Where:
ΔT is the mean temperature difference ΔT_M
$\Delta T_M = T_m - T_{amb}$
T_m = Mean liquid temperature
T_{amb} = Design ambient temperature

$$\dot{Q}_{M\,(sides)} = \frac{U_1\ A\ \Delta T_M}{1\,000}$$

$$T_m = \frac{8 + 60}{2}$$

$$\mathbf{T_m = 34°C}$$

$$T_{amb} = 8°C$$

$$\Delta T_M = T_m - T_{amb}$$

$$\Delta T_M = 34°C - 8°C$$

$$\mathbf{\Delta T_M = 26°C}$$

$$\dot{Q}_{M\,(sides)} = \frac{11\ x\ 24\ x\ 26}{1\,000}$$

$$\dot{Q}_{M\,(sides)} = \mathbf{7\ kW}$$

Part 1.4 Heat losses from liquid surface $\dot{Q}_{M\,(surface)}$

Surface area $A = 9\ m^2$

Initial water temperature $T_1 = 8°C$

Final water temperature $T_2 = 60°C$

Mean liquid temperature $T_m = \frac{8 + 60}{2}$

$$\mathbf{T_m = 34°C}$$

Heat loss estimated from Figure 2.9.2 using the mean temperature of 34°C $= 880\ W/m^2$

Area of water surface $= 9\ m^2$

Therefore: $\text{Heat loss} = 9\ m^2 \times \frac{880\ W}{m^2} \times \frac{1\ kW}{1000}$

$$\mathbf{\dot{Q}_{M\,(surface)} = 8\ kW}$$

Part 1.5 Total mean heat transfer requirement $Q_{M\,(start\text{-}up)}$

$$\dot{Q}_{M\,(start\text{-}up)} = \dot{Q}_{M\,(liquid)} + \dot{Q}_{M\,(tank)} + \dot{Q}_{M\,(sides)} + \dot{Q}_{M\,(surface)}$$

$$\dot{Q}_{M\,(start\text{-}up)} = 338\ kW + 14\ kW + 7\ kW + 8\ kW$$

$$\mathbf{\dot{Q}_{M\,(start\text{-}up)} = 367\ kW}$$

Part 2 Determine the running load, that is the maximum heat transfer rate required during operation $\dot{Q}_{(operation)}$

- In operating conditions, the liquid and tank (A1 and A2, page 2.9.6) are already up to operating temperature, so the heating components = 0.
- In operating conditions, the heat losses from the liquid and tank (A3 and A4, page 2.9.6) will be greater. This is because of the greater difference between the liquid and tank temperatures and the surroundings.
- Immersing the article in the liquid is clearly the objective of the process, so this heat load must be calculated and added to the running load heat losses.

Part 2.1 Heat losses from tank sides

$$\dot{Q} = U\,A\,\Delta T \qquad \textbf{Equation 2.5.3}$$

Where:
$\Delta T = T_f - T_{amb}$
T_f = Final liquid temperature
T_{amb} = Design ambient temperature

$$\dot{Q}_{(sides)} = \frac{U_1 \times A \times \Delta T}{1000}$$

$$\Delta T = T_f - T_{amb}$$

$$\Delta T = 60°C - 8°C$$

$$\mathbf{\Delta T = 52°C}$$

$$\dot{Q}_{(sides)} = \frac{11 \times 24 \times 52}{1000}$$

$$\mathbf{\dot{Q}_{(sides)} = 14\ kW}$$

Part 2.2 Heat losses from liquid surface $\dot{Q}_{(surface)}$

Surface area $A = 9\ m^2$

Final water temperature $T_f = 60°C$

'U' estimated from Figure 2.9.2 $= 3\ 700\ W/m^2$

$$\dot{Q}_{(surface)} = 9\ m^2 \times 3\ 700\ \frac{W}{m^2} \times \frac{1kW}{1000\ W}$$

$$\dot{Q}_{(surface)} = \mathbf{34\ kW}$$

Part 2.3 Heating the steel articles immersed in the tank $\dot{Q}_{(article)}$

Mass of article $m = 500\ kg$

Specific heat of article $c_p = 0.5\ kJ/kg\,°C$

Temperature of article when placed in tank $T_1 = 8°C$

Temperature of article when removed from tank $T_2 = 60°C$

Temperature change $\Delta T = 60 - 8$

$\Delta T = 52°C$

Time in tank (minutes) $t = 20$ minutes

$\mathbf{t = 1200\ seconds}$

$$\dot{Q} = \frac{m\ c_p\ \Delta T}{t} \qquad \textbf{Equation 2.6.1}$$

$$\dot{Q}_{(article)} = \frac{m \times c_p \times \Delta T}{1\ 200}\ kW$$

$$\dot{Q}_{(article)} = \frac{500 \times 0.5 \times 52}{1\ 200}\ kW$$

$$\dot{Q}_{(article)} = \mathbf{11\ kW}$$

Part 2.4 Total mean heat transfer requirement $\dot{Q}_{(operation)}$ (The running load)

$$\dot{Q}_{(operation)} = \dot{Q}_{(sides)} + \dot{Q}_{(surface)} + \dot{Q}_{(article)}$$

$$\dot{Q}_{(operation)} = 14 + 34 + 11\ kW$$

$$\dot{Q}_{(operation)} = \mathbf{59\ kW}$$

Note that the operational energy requirement (59 kW) is significantly less than the start-up energy requirement (367 kW). This is typical, and, where possible, the start-up period may be extended. This will have the effect of reducing the maximum energy flowrate and has the benefits of levelling demand on the boiler, and making less demand on the temperature control system.

For tanks that are to operate continuously, it is often only necessary to calculate the operating requirements i.e. the Part 2 calculations.

Questions

1. An indoors open-topped tank of water, 1.5 m wide x 2.0 m long x 1.5 m high is maintained at 85°C. The water is 1.4 m deep. The ambient temperature is 20°C and the tank is lagged with 50 mm thick insulation. There is negligible air movement over the tank. Approximately how much heat is lost from the sides of the tank?

a | 6 960 W ☐

b | 8 190 W ☐

c | 819 W ☐

d | 2 071 W ☐

2. Referring to Question 1, what will be the approximate heat loss from the liquid surface if the air velocity across the liquid surface was about 4 m/s to 5 m/s?

a | 82 kW ☐

b | 57 kW ☐

c | 75 kW ☐

d | 18 kW ☐

3. Referring to Question 2, roughly how much steam at 4 bar g is required to offset the heat lost from the liquid surface?

a | 13 kg/s ☐

b | 28 kg/h ☐

c | 46 kg/h ☐

d | 128 kg/h ☐

4. 200 kg of copper at 25°C is immersed into a tank of water based solution at 70°C. It is held there for 15 minutes. Approximately how much extra heat load is put onto the tank (c_p copper = 0.4 kJ/kg°C)?

a | 10 kW ☐

b | 15 kW ☐

c | 18 kW ☐

d | 4 kW ☐

5. Water at the rate of 1 l/s is drawn off a coil heated tank operating at 60°C and replaced with cold water at 10°C. Steam is supplied to the coil at 7 bar g.
How much steam is required to maintain the tank temperature?

a | 316 kg/h ☐

b | 387 kg/h ☐

c | 352 kg/h ☐

d | 368 kg/h ☐

6. For any particular tank temperature how does the heat loss from the lid of a closed tank compare with that of the bottom?

a | They are approximately the same ☐

b | Losses from the top are approximately double those from the bottom ☐

c | Losses from the bottom are approximately double those from the top ☐

d | Losses from the top are approximately 4 times those from the bottom ☐

Answers

1: c, 2: c, 3: d, 4: d, 5: d, 6: b

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Module 2.10

Heating with Coils and Jackets

Heating with Coils and Jackets

Vessels can be heated in a number of different ways. This module will deal with indirect heating. In these systems, the heat is transferred across a heat transfer surface. Options include:

- **Submerged steam coils** - A widely used form of heat transfer involves the installation inside a tank of a steam coil immersed in a process fluid.
- **Steam jackets** - Steam circulates in the annular space between a jacket and the vessel walls, and heat is transferred through the wall of the vessel.

Submerged steam coils

The use of tank coils is particularly common in marine applications where cargoes of crude oil, edible oils, tallow and molasses are heated in deep tanks. Many of these liquids are difficult to handle at ambient temperatures due to their viscosity. Steam heated coils are used to raise the temperature of these liquids, lowering their viscosity so that they become easier to pump.

Tank coils are also extensively used in electroplating and metal treatment. Electroplating involves passing articles through several process tanks so that metallic coatings can be deposited on to their surfaces. One of the first stages in this process is known as pickling, where materials such as steel and copper are treated by dipping them in tanks of acid or caustic solution to remove any scale or oxide (e.g. rust) which may have formed.

Steam coil sizing

Having determined the energy required (previous Module), and with knowledge of the steam pressure/temperature in the coil, the heat transfer surface may be determined using Equation 2.5.3:

$$\dot{Q} = U\,A\,\Delta T \qquad \textbf{Equation 2.5.3}$$

The heat transfer area calculated is equivalent to the surface area of the coil, and will enable an appropriate size and layout to be specified.

Determining the 'U' value

To calculate the heat transfer area, a value for the overall heat transfer coefficient, U, must be chosen. This will vary considerably with the thermal and transport properties of both fluids and a range of other conditions.

On the product side of the coil a thermal boundary layer will exist in which there is a temperature gradient between the surface and the bulk fluid. If this temperature difference is relatively large, then the natural convective currents will be significant and the heat transfer coefficient will be high.

Assisted circulation (such as stirring) that will induce forced convection, will also result in higher coefficients. As convection is partially dependent on the bulk motion of the fluid, the viscosity (which varies with temperature) also has an important bearing on the thermal boundary layer.

Additional variations can also occur on the steam side of the coil, especially with long lengths of pipe. The coil inlet may have a high steam velocity and may be relatively free from water. However, further along the length of the coil the steam velocity may be lower, and the coil may be running partially full of water. In very long coils, such as those sometimes found in seagoing tankers or in large bulk storage tanks, a significant pressure drop occurs along the length of the coil. To achieve the mean coil temperature, an average steam pressure of approximately 75% of the inlet pressure may be used. In extreme cases the average pressure used may be as low as 40% of the inlet pressure.

Another variable is the coil material itself. The thermal conductivity of the coil material may vary considerably. However, overall heat transfer is governed to a large extent by the heat resistant films, and the thermal conductivity of the coil material is not as significant as their combined effect. Table 2.10.1 provides typical overall heat transfer coefficients for various conditions of submerged steam coil application. 'U' values for steam pressures between 2 bar g and 6 bar g should be found by interpolation of the data in the table.

Table 2.10.1 Heat emission rates for steam coils submerged in water

Customary overall heat transfer coefficients		**U (W/m² °C)**
Mean steam/water temperature difference around 30°C		550 - 1300
Mean steam/water temperature difference around 60°C		1000 - 1700
Mean steam/water temperature difference around 110°C		1300 - 2700
Recommended rates		**U (W/m² °C)**
Lower pressure coils	(<2 bar g) with natural circulation of water	550
Higher pressure coils	(>6 bar g) with natural circulation of water	1100
Lower pressure coils	(<2 bar g) with assisted circulation of water	1100
Higher pressure coils	(>6 bar g) with assisted circulation of water	1700

The range of figures shown in Table 2.10.1 demonstrates the difficulty in providing definitive 'U' values. Customary figures at the higher end of the scale will apply to installations that are supplied with clean dry steam, small coils and good condensate drainage. The lower end is more applicable to poor quality steam, long coils and poor condensate drainage.

The recommended overall heat transfer coefficients will apply to typical conditions and installations. These recommended rates are empirically derived, and will generally ensure that a generous safety margin applies to the coil sizing.

In the case of fluids other than water, the heat transfer coefficient will vary even more widely due to the way in which viscosity varies with temperature. However, the values shown in Table 2.10.2 will serve as a guide for some commonly encountered substances, while Table 2.10.3 gives typical surface areas of pipes per metre length.

Table 2.10.2 Heat emission rates for steam coils submerged in miscellaneous liquids

	Medium pressure steam	**(2 - 6 bar g) with natural liquid convection**	**U (W/m² °C)**
	Light oils		170
	Heavy oils		80 - 110
*	**Fats**		30 - 60
	Medium pressure steam	**(2 - 6 bar g) with forced liquid convection**	**U (W/m² °C)**
	Light oils	**(200 sec Redwood at 38°C)**	550
	Medium oils	**(1 000 sec Redwood at 38°C)**	340
	Heavy oils	**(3 500 sec Redwood at 38°C)**	170
**	**Molasses**	**(10 000 sec Redwood at 38°C)**	85
*	**Fats**	**(50 000 sec Redwood at 38°C)**	55

* Certain materials such as tallow and margarine are solid at normal temperatures but have quite low viscosities in the molten state.

** Commercial molasses frequently contains water and the viscosity is much lower.

Table 2.10.3 Nominal surface areas of steel pipes per meter length

Nominal bore (mm)	**15**	**20**	**25**	**32**	**40**	**50**	**65**	**80**	**100**
Surface area (m²/m)	0.067	0.085	0.106	0.134	0.152	0.189	0.239	0.279	0.358

Example 2.10.1
Continuing from Example 2.9.1 determine:

Part 1. The average steam mass flowrate during start-up. (Mean heat load = 367 kW)

Part 2. The heat transfer area required.

Part 3. A recommended coil surface area.

Part 4. The maximum steam mass flowrate with the recommended heat transfer area.

Part 5. A recommendation for installation, including coil diameter and layout.

The following additional information has been provided:

- Steam pressure onto the control valve = 2.6 bar g (3.6 bar a).
- A stainless steel steam coil provides heat.
- Heat transfer coefficient from steam/coil/liquid, U = 650 W/m² °C

Part 1 Calculate the average steam mass flowrate during start-up

Steam pressure onto the control valve = 2.6 bar g (3.6 bar a)

Critical pressure drop (CPD) will occur across the control valve during start-up, therefore the minimum steam pressure in the heating coil should be taken as 58% of upstream absolute pressure. An explanation of this is given in Block 5.

$$\text{Minimum steam pressure} = 3.6 \text{ bar a} \times 58\%$$

$$\textbf{Minimum steam pressure} = \textbf{2.1bar a (1.1bar g)}$$

$$\text{Enthalpy of evaporation } (h_{fg}) \text{ at 1.1 bar g} = 2197 \text{ kJ/kg (from steam tables)}$$

$$\text{Average steam flowrate} = \frac{367 \text{ kW}}{2197 \text{ kJ/kg}}$$

$$\textbf{Average steam flowrate} = \textbf{0.167 kg/s or 602 kg/h}$$

Part 2 Calculate the heat transfer area required

Using Equation 2.5.3 : $\dot{Q} = U A \Delta T$

Where:

$\dot{Q}$ = 367 kW (Example 2.9.1)

U = 650 W/m °C (given)

ΔT = Mean temperature difference ΔT_M

Steam pressure = 2.1 bar a (see above)

Therefore: Steam temperature in the coil, T_s = **122°C** (from steam tables)

$$\text{Mean liquid temperature, } T_m = \frac{8 + 60}{2}$$

$$T_m = 34°C$$

$$\Delta T_M = T_s - T_m$$

$$\Delta T_M = 122 - 34$$

$$\Delta T_M = 88°C$$

$$\text{Therefore } 367 \text{ kW} = \frac{650 \text{ W/m}^2\text{°C} \times A \text{ m}^2 \times 88°C}{1000}$$

$$A = \frac{367 \times 1000}{650 \times 88}$$

$$A = 6.416 \text{ m}^2$$

Part 3 A recommendation for coil surface area

Because of the difficulties in providing accurate 'U' values, and to allow for future fouling of the heat exchange surface, it is usual to add 10% to the calculated heat transfer area.

Recommended heat transfer area (A) = 6.416 m² + 10%

$$\mathbf{A \approx 7\ m^2}$$

Part 4 The maximum steam mass flowrate with the recommended heat transfer area

Maximum heat transfer (and hence steam demand) will occur when the temperature difference between the steam and the process fluid is at its maximum, and should take into consideration the extra pipe area allowed for fouling.

(a) Consider the maximum heating capacity of the coil $\dot{Q}_{(coil)}$

Using Equation 2.5.3: $\dot{Q} = \mathbf{U\,A\,\Delta T}$

Where:

$U = 650$ W/m°C (given)

$A = 7$ m²

ΔT = Initial temperature difference

Initial liquid temperature $T_1 = 8$°C

Steam pressure = 1.1 bar g

Steam temperature, $T_s = 122$°C

$\Delta T = T_s - T_1 = 122 - 8$°C

$\Delta T = 114$°C

$$\dot{Q}_{(coil)} = \frac{650 \times 7 \times 114}{1\,000}$$

$$\mathbf{\dot{Q}_{(coil)} = 519\ kW}$$

(b) Steam flowrate to deliver 519 kW

$$\text{Maximum steam flowrate } (\dot{m}_s) = \frac{519\ \text{kW} \times 3\,600}{h_{fg} \text{ at } 1.1 \text{ bar g}}\ \text{kg/h}$$

$$\dot{m}_s = \frac{519\ \text{kW} \times 3\,600}{2\,197\ \text{kJ/kg}}\ \text{kg/h}$$

$$\mathbf{\dot{m}_s = 850\ kg/h}$$

Part 5 A recommendation for installation, including coil diameter and layout

(a) Determine coil diameter and length

Steam pressure = 1.1 bar g (v_g = 0.841 m³/kg - from steam tables)

Maximum steam flowrate = 850 kg/h

Maximum recommended steam velocity = 25 m/s

$$\text{Steam velocity} = \frac{\text{Volume flow/second}}{\text{Cross sectional area of pipe}} \text{ m/s}$$

$$\text{Volume flow per second} = \frac{\text{Mass flow (kg/h) x Specific volume (m}^3\text{/kg)}}{3\,600 \text{ (s/h)}}$$

$$\text{Cross sectional area of pipe} = \frac{\pi D^2}{4}$$

$$\text{Therefore: } 25 \text{ m/s} = \frac{850 \text{ kg/h} \times 0.841 \text{ m}^3\text{/kg} \times 4}{3600 \times \pi \times D^2}$$

$$D^2 = \frac{850 \times 0.841 \times 4}{3600 \pi \times 25}$$

$$\text{Diameter} = \sqrt{\frac{850 \times 0.841 \times 4}{3600 \times \pi \times 25}} \text{ metres}$$

Diameter = 0.1 m or 100 mm

From Table 2.10.3, a 100 mm pipe has a surface area of 0.358 m²/m run. This application will require:

$$\frac{7 \text{ m}^2}{0.358 \text{ m}^2\text{/m}} = 19.6 \text{ metres of 100 mm pipe}$$

It may be difficult to accommodate this length of large bore heating pipe to install in a 3 m × 3 m tank.

One solution would be to run a bank of parallel pipes between steam and condensate manifolds, set at different heights to encourage condensate to run to the lower (condensate) manifold. The drain line must fall from the bottom of the condensate manifold down to the steam trap (or pump-trap). See Figure 2.10.1 for a suggested layout.

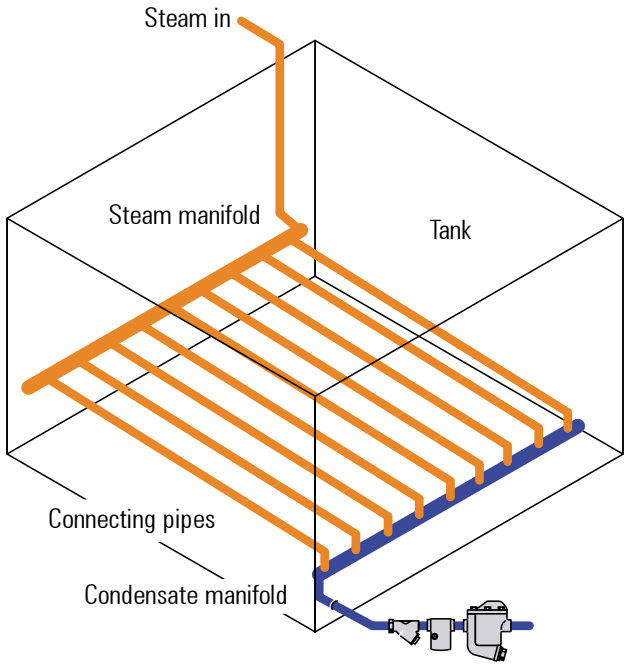


Fig. 2.10.1 Possible layout of coils in a rectangular tank

Note the steam supply is situated at one end of its manifold, whilst the trap set is at the other end. This will help steam to flow and push condensate through the coils.

In the application, the steam and condensate headers would each be 2.8 m long. As the condensate manifold is holding condensate, the heat from it will be small compared to the steam manifold and this can be ignored in the calculation.

The steam manifold should be 100 mm diameter as determined by the previous velocity calculation. This will provide a heating area of:

$$2.8 \text{ m} \times 0.358 \text{ m}^2/\text{m} = 1.0 \text{ m}^2$$

Consequently 7 m² - 1 m² = 6 m² of heat transfer area is still required, and must be provided by the connecting pipes.

Arbitrarily selecting 32 mm pipe as a good compromise between robustness and workability:

$$\frac{6 \text{ m}^2}{0.134 \text{ m}^2/\text{m}} \approx 45 \text{ metres of 32 mm pipe is required}$$

The lengths of the connecting pipes are 2.5 m.

$$\text{Therefore, the number of connecting pipes} = \frac{45 \text{ m}}{2.5 \text{ m}} \approx 18$$

CHECK

It is necessary to confirm the steam velocity through the connecting tubes:

On the basis of proportionality of heat transfer area, the steam header will condense:

$$\frac{1 \text{ m}^2}{7 \text{ m}^2} \times \frac{100}{1} = 14\% \text{ of the steam condensed in the whole coil assembly.}$$

This leaves 86% of the 850 kg/h = 731 kg/h of steam which must pass through the 18 connecting pipes and also into the lower (condensate) manifold.

$$\text{Steam velocity} = \frac{\text{Volume flow/second}}{\text{Cross sectional area of pipe}} \text{ m/s}$$

$$\text{Steam velocity} = \frac{731 \text{ kg/h} \times 0.841 \text{ m}^3/\text{kg} \times 4}{3\,600 \times 18 \text{ pipes} \times \pi \times 0.032^2} \text{ m/s}$$

Steam velocity ≈ 12 m/s per connecting pipe

Other steam coil layouts

The design and layout of the steam coil will depend on the process fluid being heated. When the process fluid to be heated is a corrosive solution, it is normally recommended that the coil inlet and outlet connections are taken over the lip of the tank, as it is not normally advisable to drill through the corrosion resistant linings of the tank side. This will ensure that there are no weak points in the tank lining, where there is a risk of leakage of corrosive liquids. In these cases the coil itself may also be made of corrosion resistant material such as lead covered steel or copper, or alloys such as titanium.

However, where there is no danger of corrosion, lifts over the tank structure should be avoided, and the steam inlet and outlet connections may be taken through the tank side. The presence of any lift will result in waterlogging of a proportion of the coil length, and possibly waterhammer, noise and leaking pipework.

Steam heating coils should generally have a gradual fall from the inlet to the outlet to ensure that condensate runs toward the outlet and does not collect in the bottom of the coil.

Where a lift is unavoidable, it should be designed to include a seal arrangement at the bottom of the lift and a small bore dip pipe, as shown in Figure 2.10.2.

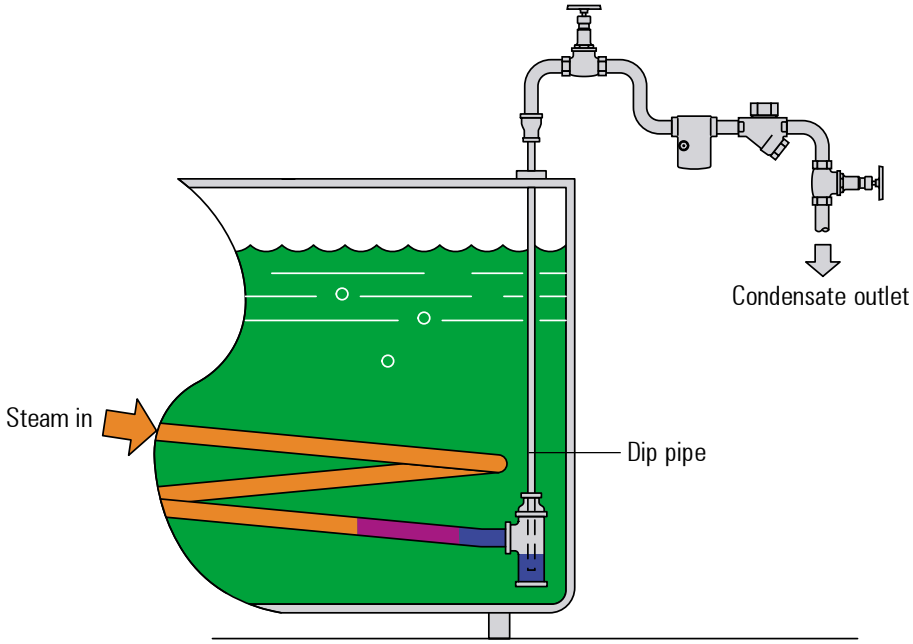


Fig. 2.10.2 Tank with a rising discharge pipe

The seal arrangement allows a small amount of condensate to collect to act as a water seal, and prevents the occurrence of steam locking. Without this seal, steam can pass over any condensate collecting in the bottom of the pipe, and close the steam trap at the top of the riser.

The condensate level would then rise and form a temporary water seal, locking the steam between the bottom of the riser and the steam trap. The steam trap remains closed until the locked steam condenses, during which time the coil continues to waterlog.

When the locked steam condenses and the steam trap opens, a slug of water is discharged up the riser. As soon as the water seal is broken, steam will enter the rising pipe and close the trap, while the broken column of water falls back to lie at the bottom of the heating coil.

The small bore dip pipe will only allow a very small volume of steam to become locked in the riser. It enables the water column to be easily maintained without steam bubbling through it, ensuring there is a steady and continuous condensate flow to the outlet.

When the seal is ultimately broken, a smaller volume of water will return to the heating coil than with an unrestricted large bore riser, but as the water seal arrangement requires a smaller volume of condensate to form a water seal, it will immediately re-form.

If the process involves articles being dipped into the liquid, it may not be convenient to install the coil at the bottom of the tank - it may be damaged by the objects being immersed in the solution. Also, during certain processes, heavy deposits will settle at the bottom of the tank and can quickly cover the heating surface, inhibiting heat transfer.

For these reasons side hung coils are often used in the electroplating industry. In such cases serpentine or plate-type coils are arranged down the side of a tank, as shown in Figure 2.10.3. These coils should also have a fall to the bottom with a water seal and a small bore dip-pipe. This arrangement has the advantage that it is often easier to install, and also easier to remove for periodic cleaning if required.

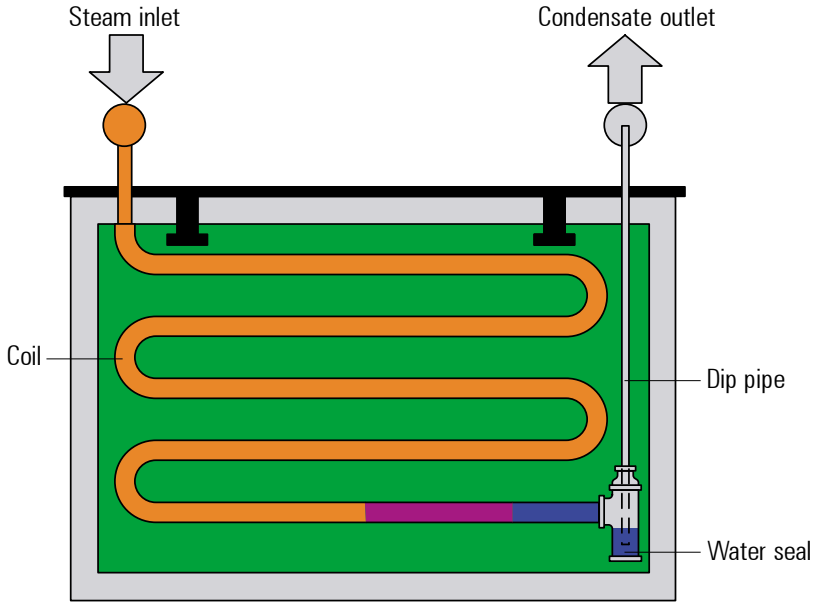


Fig. 2.10.3 Side hung coils

If articles are to be dipped into the tank, it may not be possible to use any sort of agitator to induce forced convection and prevent temperature gradients occurring throughout the tank. Whether bottom or side coils are used, it is essential that they are arranged with adequate coverage so that the heat is distributed evenly throughout the bulk of the liquid.

The diameter of the coil should provide sufficient length of coil for good distribution. A short length of coil with a large diameter may not provide adequate temperature distribution. However a very long continuous length of coil may experience a temperature gradient due to the pressure drop from end to end, resulting in uneven heating of the liquid.

Whilst the next two headings, 'Sizing the control valve' and 'The condensate removal device' are included in this Module, the new reader should refer to later Blocks and Modules in *The Learning Centre* for full and comprehensive information, before attempting sizing and selection of equipment.

Control valve arrangement

The control valve set may be either one or two valves in parallel. A single control valve, large enough to cope with the maximum flowrate encountered at start-up, may be unable to control flow accurately at the minimum expected flowrate. This could cause erratic temperature control. An alternative is to fit two temperature control valves in parallel:

- One valve (running valve) sized to control at the lower flowrate.
- A second valve (starting valve) to pass the difference between the capacity of the first valve, and the maximum flowrate.

The starting valve would have a set-point slightly lower than the running valve, so it would close first, leaving the running valve to control at low loads.

Sizing the control valve

The control valve set (either one valve or two valves in parallel).

The coil has been sized on mean heat transfer values. However, it may be better to size the control valve to supply the maximum (start-up) load. With large coils in tanks, this will help to maintain a degree of steam pressure throughout the length of the coil when the steam is turned on, helping to push condensate through the coil to the steam trapping device. If the control valve were sized on mean values, steam pressure in the coil at start-up will tend to be lower and the coil may flood.

Using one valve

Continuing with Example 2.10.1 the maximum steam load is 850 kg/h and the coil is designed to deliver this at a pressure of 1.1 bar g. A steam valve sizing chart would show that a K_V of about 20 is required to pass 850 kg/h of steam with a pressure of 2.6 bar g at the inlet of the control valve, and Critical Pressure Drop (CPD) across the valve. (Module 6.4 will show how the valve size can be determined by calculation).

A DN40 control valve with a larger K_{VS} of 25 would therefore need to be selected for the application.

If one valve is to be used, this valve must ensure the maximum heat load is catered for, while maintaining the required steam pressure in the coil to assist the drainage of condensate from it at start-up. However, for reasons previously explained, two valves may be better.

The running load is 52 kW and with the coil running at 1.1 bar g, the running steam load:

$$\dot{m}_{s\,(run)} = \frac{\dot{Q}}{h_{fg}} = \frac{52\ kW}{2\,197\ kJ/kg} = 0.0237\ kg/s$$

$$= 85\ kg/h$$

The steam valve sizing chart shows a K_V of 2 is required to pass 85 kg/h with 3.6 bar upstream, operating at critical pressure drop.

A DN15 KE type valve (K_{VS} = 4) and a DN25 piston actuated valve (K_{VS} = 18.6) operating together will cater for the start-up load. When approaching the control temperature, the larger valve would be set to shut down, allowing the smaller valve to give good control.

The condensate removal device

The selection and sizing of the condensate removal device will be very much influenced by the condensate backpressure. For the purpose of this example, it is assumed the backpressure is atmospheric pressure. The device should be sized so it is able to satisfy both of the following conditions:

1. Pass 850 kg/h of condensate with 1.1 bar g in the coil, i.e. the full-load condition.
2. Pass the condensate load when steam pressure in the coil equals the condensate backpressure, i.e. the stall load condition.

If the steam trap is only sized on the first condition, it is possible that it may not pass the stall load (the condition where the product approaches its required temperature and the control valve modulates to reduce steam pressure). The stall load may be considerable. With respect to non-flow type applications such as tanks, this may not be too serious from a thermal viewpoint because the contents of the tank will almost be at the required temperature, and have a huge reservoir of heat.

Any reduction in heat transfer at this part of the heating process may therefore have little immediate effect on the tank contents.

However, condensate will back up into the coil and waterhammer will occur, along with its associated symptoms and mechanical stresses. Tank coils in large circular tanks tend to be of robust construction, and are often able to withstand such stresses. Problems can however occur in rectangular tanks (which tend to be smaller), where vibration in the coil will have more of an effect on the tank structure. Here, the energy dissipated by the waterhammer causes vibration, which can be detrimental to the life of the coil, the tank, and the steam trap, as well as creating unpleasant noise.

With respect to flow-type applications such as plate heat exchangers, a failure to consider the stall condition will usually have serious implications. This is mainly due to the small volume in the heat exchanger.

For heat exchangers, any unwanted reduction in the heating surface area, such as that caused by condensate backing up into the steam space, can affect the flow of heat through the heating surface. This can cause the control system to become erratic and unstable, and processes requiring stable or accurate control can suffer with poor performance.

If heat exchangers are oversized, sufficient heating surface may remain when condensate backs up into the steam space, and reduction of thermal performance may not always occur. However, with heat exchangers not designed to cope with the effects of waterlogging, this can lead to corrosion of the heating surface, inevitably reducing the service life of the exchanger. Waterlogging can, in some applications, be costly. Consider a waterlogging air heater frost coil. Cold air at 4°C flowing at 3 m/s can soon freeze condensate locked in the coils, resulting in premature and unwarranted failure. Proper drainage of condensate is essential to maintain the service life of any heat exchanger and air heater.

Steam traps are devices which modulate to allow varying amounts of condensate to drain from applications under varying conditions. Float traps are steam traps designed to modulate and release condensate close to steam temperature, offering maximum plant performance, maximum plant life, and maximum return on plant investment.

When stall conditions occur, and a steam trap cannot be used, an automatic pump-trap or pump and trap in combination will ensure correct condensate drainage at all times, thus maximising the thermal capability and lifetime costs of the plant.

Steam jackets

The most commonly used type of steam jacket consists simply of an outer cylinder surrounding the vessel, as shown in Figure 2.10.4. Steam circulates in the outer jacket, and condenses on the wall of the vessel. Jacketed vessels may also be lagged, or may contain an internal air space surrounding the jacket. This is to ensure that as little steam as possible condenses on the outer jacket wall, and that the heat is transferred inwards to the vessel.

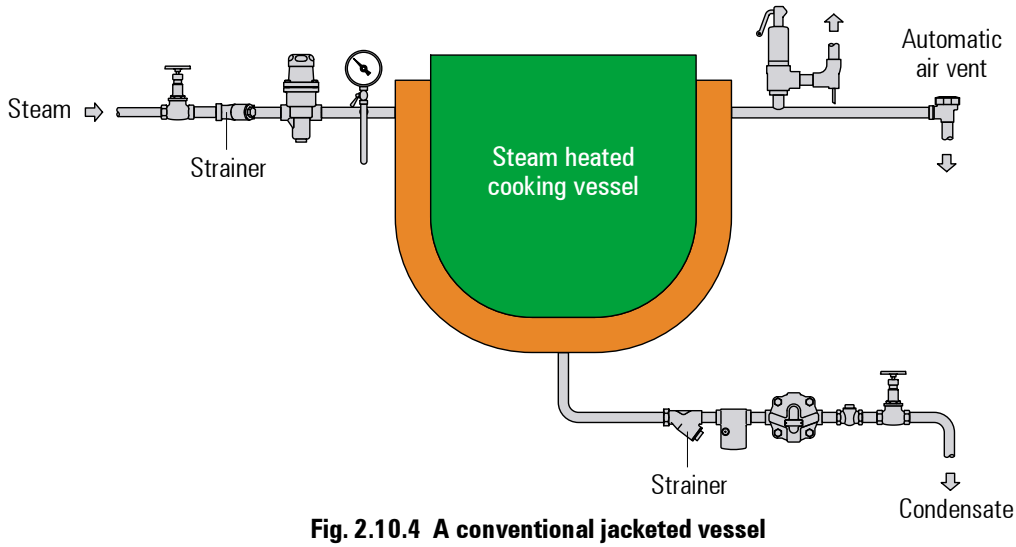


Fig. 2.10.4 A conventional jacketed vessel

The heat transfer area (the vessel wall surface area), can be calculated in the same manner as with a steam coil, using Equation 2.5.3 and the overall heat transfer coefficients provided in Table 2.10.4.

Although steam jackets may generally be less thermally efficient than submerged coils, due to radiation losses to the surroundings, they do allow space for the vessels to be agitated so that heat transfer is promoted. The U values listed in Table 2.10.4. are for moderate non-proximity agitation.

Commonly the vessel walls are made from stainless steel or glass lined carbon steel. The glass lining will offer an additional corrosion resistant layer. The size of the steam jacket space will depend on the size of the vessel, but typically the width may be between 50 mm and 300 mm.

Table 2.10.4 Overall heat transfer coefficients for steam jackets

Process fluid or product	Wall material	U (W/m²°C)
Water	Stainless steel	850 - 1700
	Glass-lined Carbon steel	400 - 570
Aqueous solution	Stainless steel	450 - 1140
	Glass-lined carbon steel	285 - 480
Organics	Stainless steel	285 - 850
	Glass-lined carbon steel	170 - 400
Light oil	Stainless steel	340 - 910
	Glass-lined carbon steel	230 - 425
Heavy oil	Stainless steel	57 - 285
	Glass-lined carbon steel	57 - 230

Questions

1. A tank of water is to be heated by a mild steel coil from 20°C to 80°C in 4 hours. The control valve is supplied with steam at 4 bar g. The mean heat-up steam demand is 98 kg/h and the running demand is 27 kg/h. (Take the 'U' value of the coil to be 550 W/m² °C and disregard fouling for the purpose of this exercise).

Approximately what length of 25 mm coil will be required?

a| 12.5 m ☐

b| 7.6 m ☐

c| 10.4 m ☐

d| 12.2 m ☐

2. What is the disadvantage of heating a tank by direct steam injection?

a| It agitates the solution ☐

b| Some of the enthalpy of water is used ☐

c| Steam traps are not required ☐

d| It dilutes the tank content ☐

3. A published 'U' value from a steam coil to a water based solution is given as 550 - 1 300 W/m² °C.

When would a figure near the lower end of the range be used?

a| When the steam is known to be of good quality ☐

b| For short coils ☐

c| For small diameter coils ☐

d| When scaling or fouling of the coil takes place ☐

4. Steam coils should enter and leave the top of a tank when:

a| The tank contains a corrosive solution ☐

b| When agitation of the tank solution is required ☐

c| When steam locking of the trap draining a base coil could occur ☐

d| When good heat distribution is required ☐

5. What range of 'U' values would you apply for a mild steel jacket around a stainless steel tank containing a water and detergent solution?

a| 285 - 480 ☐

b| 450 - 1140 ☐

c| 850 - 1700 ☐

d| 285 - 850 ☐

6. 20 m of 25 mm stainless steel coil maintains a tank of water based solution at 65°C. Steam pressure is 3 bar g and there is natural circulation in the tank. What will be the approximate steam consumption under this condition (Take the 'U' value of the coil to be 700 W/m²°C)?

a| 256 kg/h ☐

b| 382 kg/h ☐

c| 287 kg/h ☐

d| 195 kg/h ☐

Answers

1: a, 2: d, 3: d, 4: a, 5: b, 6: d

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Module 2.11

Heating Vats and Tanks by Steam Injection

Heating Vats and Tanks by Steam Injection

Direct steam injection involves the discharge of a series of steam bubbles into a liquid at a lower temperature. The steam bubbles condense and give up their heat to the surrounding liquid.

Heat is transferred by direct contact between the steam and the liquid, consequently this method is only used when dilution and an increase in liquid mass is acceptable. Therefore, the liquid being heated is usually water. Direct steam injection is seldom used to heat solutions in which a chemical reaction takes place, as the dilution of the solution would reduce the reaction rate and lower the productivity.

Direct steam injection is the most widely used method for boiler feedtank heating throughout industry. This method is often chosen because of its simplicity. No heat transfer surface or steam trap set is required, and there is no need to consider the condensate return system.

Steam consumption calculations

During direct steam injection, heat is transferred in a different manner to indirect heat exchange. As the heat is not transferred across a surface, and the steam mixes freely with the process fluid being heated, the amount of usable heat in the steam must be calculated in a different way. This can be found using Equation 2.11.1:

$$\dot{m}_s = \frac{\dot{Q}}{h_g - (T\, c_p)} \qquad \textbf{Equation 2.11.1}$$

Where:
$\dot{m}_s$ = Mean steam flowrate (kg/s)
$\dot{Q}$ = Mean heat transfer rate kW (kJ/s)
h_g = Specific enthalpy of steam (taken at the pressure supplying the control valve) (kJ/kg)
T = Final temperature of the water (°C)
c_p = Specific heat capacity of water (kJ/kg °C)

Equation 2.11.1 shows that steam injection utilises all of the enthalpy of evaporation (or latent heat) and a proportion of the liquid enthalpy contained in the steam. The actual proportion of the liquid enthalpy used will depend on the temperature of the water at the end of the injection process.

One major difference between indirect heating and direct steam injection, is that the volume (and mass) of the process fluid is increased as steam is added, by the amount of steam injected.

Another difference is that, when calculating the steam flowrate to a steam coil, the pressure in the coil is considered, but for steam injection, the pressure before the control valve is considered.

In some cases (where the liquid surface is not at the overflow pipe level), this will increase the head of liquid over the injector as time progresses. However, this increase is likely to be small and is rarely taken into account in calculations.

Factors influencing the heat transfer rate

In Equation 2.11.1, the steam consumption rate is directly related to the heat requirement. Unless the steam injection system is designed so that all conditions are conducive to maximum heat transfer, the steam bubbles may simply break the surface of the liquid and escape to the atmosphere; some of the heat contained in the steam will be lost to atmosphere and the actual heat transfer rate to the water will be less than anticipated.

In the case of a submerged coil, the maximum heat transfer rate at the start of the warm-up period will depend on the maximum steam flowrate allowed through the control valve and its associated pipework, and the maximum heat output allowed by the coil surface area.

During direct steam injection, it might be expected that the maximum heat transfer rate at the very start of the warm-up period is dependent on the maximum flowrate through the control valve, and the pipe or injector itself. However, as implied above, it will also depend on other factors such as:

- **Size of the steam bubble -** Condensation of a steam bubble will depend on the heat transfer across the surface of the bubble. To ensure that the steam bubble is completely condensed, the surface area/volume ratio must be as large as possible. Smaller bubbles have a greater surface area per unit volume than larger bubbles, so it is desirable to produce very small bubbles. The differential pressure (between the steam pipe and the point where the steam is discharged into the water) as the bubble emerges will also affect the size of the steam bubble. The specific volume of steam will increase as the pressure is reduced, so that a drop in pressure will increase the size of the steam bubble as it escapes into the liquid. Even if the steam bubble is emitted from a very small hole, the bubble may increase significantly in size if the steam pressure is high. Consequently, a lower pressure in the sparge pipe is better.
- **Head of liquid over the injection point -** The head of liquid over the injection point will create a backpressure so that the differential pressure will be less than the steam pressure. If the head of liquid is large and the steam pressure in the sparge pipe is low, there may only be a very small change in pressure so that the size of the bubbles formed is kept to a minimum.

  A greater head of liquid over the point of injection will give the steam bubbles maximum opportunity to condense before they reach the surface.
- **Velocity of the bubble -** The velocity of the bubble at the point of injection will also depend on the difference between the steam pressure and the liquid head. It is desirable to keep this differential pressure as low as possible, so that bubble velocities are also as low as possible and the bubbles are given the maximum time to condense before they reach the surface.
- **Temperature of the liquid -** The rate at which the steam will condense is directly proportional to the temperature difference between the steam and the liquid being heated. As with all heat transfer processes, the rate of heat exchange is directly proportional to the temperature differential.

  It is always advisable to ensure that the temperature of the liquid is correctly controlled and is kept to the minimum required for the application, so that the maximum heat transfer rate is maintained and there is no wastage of energy.

Sparge pipes

This is simply a pipe mounted inside the tank, with the holes drilled at regular positions (typically 4 o'clock and 8 o'clock) when viewed from the end, equally spaced along the length of the pipe, and with the end blanked off. The steam exits the pipe through the holes as small bubbles, which will either condense as intended or reach the surface of the liquid (see Figure 2.11.1).

Sparge pipes are inexpensive to make and easy to install, but are prone to cause high levels of vibration and noise. A much more effective method is to use a properly designed steam injector.

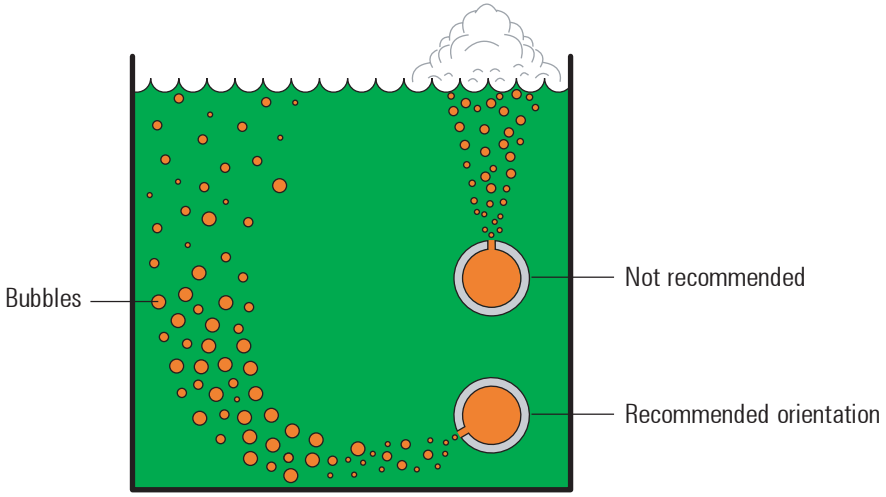


Fig. 2.11.1 Sparge hole orientation

Example 2.11.1 - Determine the steam load to heat a tank of water by steam injection

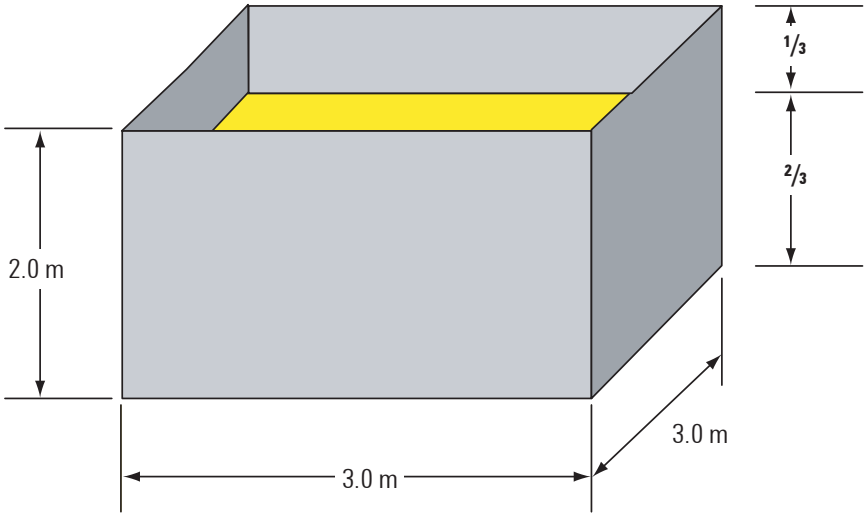


Fig. 2.11.2 The tank used in Example 2.9.1

These calculations (steps 1 to 5) are based on Examples 2.9.1 and 2.10.1 as far as heat losses are concerned, but with the tank containing water (c_p = 4.19 kJ/kg °C), instead of weak acid solution and the water being heated by steam injection rather than a steam coil.

Step 1 - find the energy required to heat up 12 000 kg of water from 8°C to 60°C in 2 hours by using Equation 2.6.1:

$$\dot{Q} = \frac{m\, c_p\, \Delta T}{t} \qquad \textbf{Equation 2.6.1}$$

Where:
$\dot{Q}$ = Mean heat transfer rate to heat the water (kW)
m = 12 000 kg
c_p = 4.19 kJ/kg °C
ΔT = 60 - 8 = 52°C
t = 2 hours x 3 600 = 7 200 seconds

$$\dot{Q}_{(water)} = \frac{12\,000 \text{ kg} \times 4.19 \text{ kJ/kg°C} \times 52\text{°C}}{7200 \text{ seconds}}$$

$$\dot{Q}_{(water)} = \mathbf{363\ kW}$$

Steam is supplied to the control valve at 2.6 bar g. In order to calculate the mean steam flowrate, it is necessary to determine the total enthalpy in the steam (h_g) at this pressure. It can be seen from Table 2.11.1 (an extract from steam tables) that the total enthalpy of steam (h_g) at 2.6 bar g is 2 733.89 kJ/kg.

Table 2.11.1 Extract from steam tables

Pressure bar g	Saturation temperature °C	Specific enthalpy (energy) in kJ/kg			Specific volume of dry saturated steam m³/kg
		Water h_f	Evaporation h_{fg}	Steam h_g	
2.4	138.011	580.741	2 150.53	2 731.27	0.536 766
2.5	139.023	585.085	2 147.51	2 732.60	0.522 409
2.6	140.013	589.333	2 144.55	2 733.89	0.508 820
2.7	140.980	593.490	2 141.65	2 735.14	0.495 939

Step 2 - find the mean steam flowrate to heat the water by using Equation 2.11.1:

$$\dot{m}_s = \frac{\dot{Q}}{h_g - (T\ c_p)} \qquad \textbf{Equation 2.11.1}$$

Where:
$\dot{m}_s$ = Mean steam flowrate to heat the water in the tank (kg/s)
$\dot{Q}$ = $\dot{Q}_{(water)}$ = Mean heat transfer rate to heat the water = 363 kW
h_g = Total enthalpy in the steam supplying the control valve = 2733.89 kJ/kg
T = Final water temperature = 60°C
c_p = Specific heat of water = 4.19 kJ/kg °C

Therefore, from Equation 2.11.1;

$$\text{Mean steam flowrate to heat the water } (\dot{m}_s) = \frac{363}{2\,733.9 - (60°C \times 4.19)} \text{ kg/s}$$

Mean steam flowrate to heat the water ($\dot{m}_s$) = 0.146 kg/s

= 526 kg/h (A)

Step 3 - find the mean steam flowrate to heat the tank material (steel). From Example 2.9.1, the mean heat transfer rate for the tank material = $\dot{Q}_{(tank)}$ = 14 kW

The mean steam flowrate to heat the tank material is calculated by again using Equation 2.11.1:

$$\dot{m}_s = \frac{\dot{Q}}{h_g - (T\ c_p)} \qquad \textbf{Equation 2.11.1}$$

Where:
$\dot{m}_s$ = Mean steam flowrate to heat the tank material (kg/s)
$\dot{Q}$ = $\dot{Q}_{(tank)}$ = Mean heat transfer rate to heat the tank material = 14 kW
h_g = Total enthalpy in the steam supplying the control valve = 2733.89 kJ/kg
T = Final tank temperature = 60°C
c_p = Specific heat of the tank material (steel) = 0.5 kJ/kg °C

Therefore, from Equation 2.11.1

$$\text{Mean steam flowrate to heat the tank material } (\dot{m}_s) = \frac{14}{2\,733.9 - (60°C \times 0.5)} \text{ kg/s}$$

Mean steam flowrate to heat the tank material ($\dot{m}_s$) = 0.0052 kg/s

= 19 kg/h (B)

Step 4 - find the mean steam flowrate to make up for the heat losses from the tank during warm-up. From Example 2.9.1:

The mean heat losses from the tank and water surface = $\dot{Q}_{(sides)}$ + $\dot{Q}_{(surface)}$

The heat losses from the tank and water surface = 7 kW + 8 kW

The heat losses from the tank and water surface = 15 kW

Whilst it is reasonable to accept that the steam's liquid enthalpy will contribute to the rise in temperature of the water and the tank material, it is more difficult to accept how the steam's liquid enthalpy would add to the heat lost from the tank due to radiation. Therefore, the equation to calculate the steam used for heat losses (Equation 2.11.2) considers only the enthalpy of evaporation in the steam at atmospheric pressure.

$$\dot{m}_s = \frac{\dot{Q}}{2\,256.7} \qquad \textbf{Equation 2.11.2}$$

Where:
$\dot{m}_s$ = Mean steam flowrate to provide the heat losses from the tank (kg/s)
$\dot{Q}$ = $\dot{Q}_{(sides)}$ + $\dot{Q}_{(surface)}$ (kW)
2256.7 = Enthalpy of evaporation at atmospheric pressure (kJ/kg)

Therefore, from Equation 2.11.2;

$$\text{Steam load due to heat losses from the tank } (\dot{m}_s) = \frac{15}{2\,256.7} \text{ kg/s}$$

$$\textbf{Steam load due to heat losses from the tank } (\dot{m}_s) = \textbf{0.006 6 kg/s}$$

$$= \textbf{24 kg/h} \quad \textbf{(C)}$$

Step 5 - Determine the steam load to heat a tank of water by steam injection. The total mean steam flowrate can be calculated as follows:

$$\text{The total mean steam flowrate} = (A) + (B) + (C)$$

$$\text{The total mean steam flowrate} = 526 + 19 + 24 \text{ kg/h}$$

$$\textbf{The total mean steam flowrate} = \textbf{569 kg/h} \text{ (For example 2.11.1)}$$

It is important to remember with steam injection systems that the final mass of liquid is equal to the mass of cold liquid, plus the mass of steam added.

In this example, the process started with 12 000 kg of water. During the required heat-up period of 2 hours steam has been injected at the rate of 569 kg/h. The mass of liquid has therefore, increased by 2 h x 569 kg/h = 1 138 kg.

The final mass of the liquid is:

$$12\,000 \text{ kg} + 1\,138 \text{ kg} = 13\,138 \text{ kg}$$

The additional 1 138 kg of condensate has a volume of about 1 138 litres (1.138 m³) and will also have increased the water level by:

$$\frac{1.138 \text{ m}^3}{3 \text{ m} \times 3 \text{ m}} = 0.127 \text{ m} = 127 \text{ mm}$$

Clearly, the process tank needs to have sufficient space above the starting water level to allow for this increase. For safety, an overflow should always be included in the tank construction where steam injection is involved.

Alternatively, if the process requirement had been to **finish** with a mass of 12 000 kg, the mass of water at the beginning of the process would be:

$$\text{Revised mass to finish with 12 000 kg} = \frac{\text{Initial mass}}{\text{Final mass}} \times \text{Initial mass}$$

$$\text{Revised mass to finish with 12 000 kg} = \frac{12\,000}{13\,138} \times 12\,000$$

$$\textbf{Revised mass to finish with 12 000 kg} = \textbf{10 960 kg initial mass of water}$$

Steam injectors

A more effective alternative to the sparge pipe is the steam injector as shown in Figure 2.11.3. The injector draws in cold liquid and mixes it with steam inside the injector, distributing heated liquid to the tank.

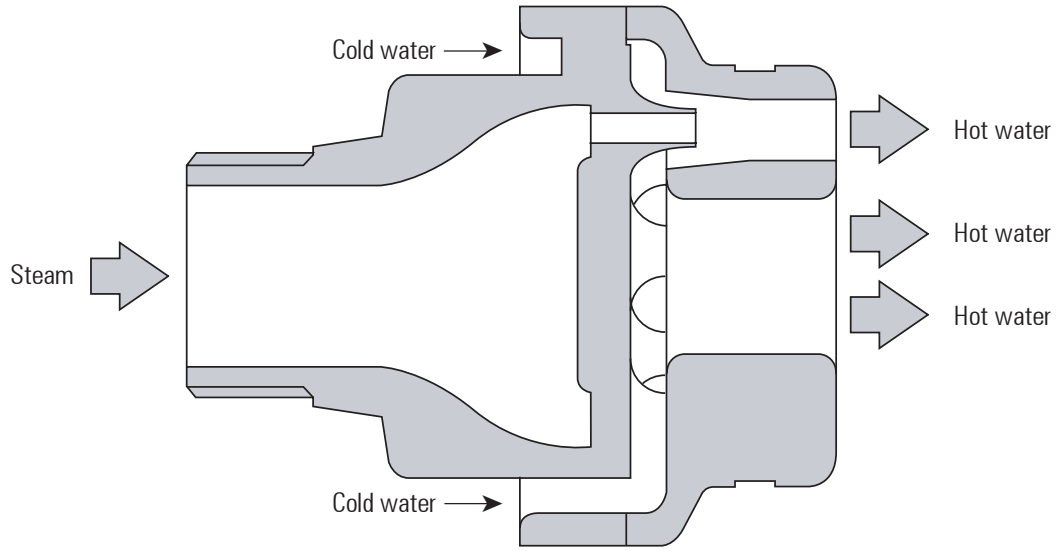


Fig. 2.11.3 A steam injector

The engineered design of the injector body is more sophisticated than the simple sparge pipe, and allows steam at higher pressures to be used. A turbulent zone is created within the body of the injector, which ensures that thorough mixing of the steam and liquid occurs, even at relatively high pressures. This has the effect of agitating and circulating the liquid so that a constant temperature is maintained throughout the tank, without temperature stratification or cold spots.

These injectors are more compact than sparge pipes, consequently any interference with objects that may be dipped in the tank can be avoided. They are more robust and generally quieter than sparge pipes, although noise problems may still be encountered if not installed correctly.

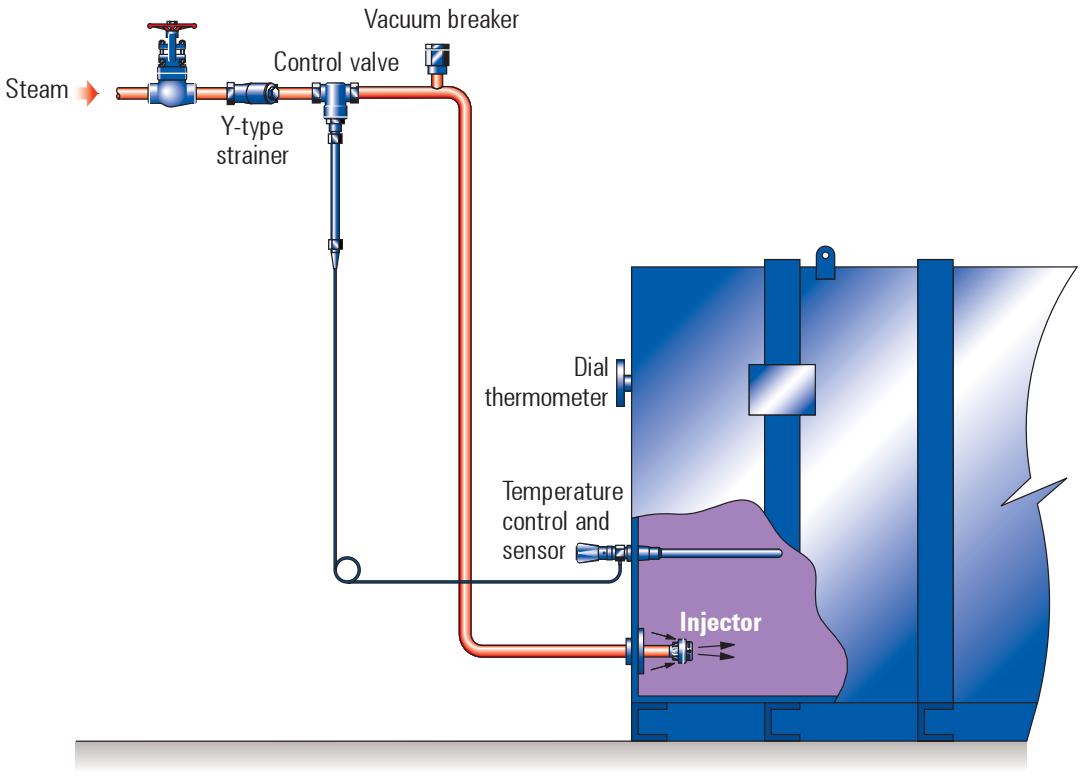


Fig. 2.11.4 Typical steam injector installation

Noises pertaining to steam injectors

When using high pressure steam injectors three distinct noise levels are produced under the following conditions:

- **Normal running -** Where steam pressures at the injector inlet are above 2 bar g, the noise produced during normal running conditions can be described as a soft roar.

  Noise is caused by the condensation of steam inside the discharge tube, as it mixes with recirculating water drawn through the holes into the casting body. Under normal conditions the discharge from the injector tube is approximately 10°C hotter than the incoming water.

  This type of noise increases with steam pressure, water temperature and the number of injectors, but it is rarely objectionable at steam pressures below 8 bar g. Although strong circulation of the tank contents occurs at pressures above 8 bar g, little vibration should be experienced.

- **Incomplete condensation -** This is characterised by a soft bumping noise and is sometimes accompanied by heavy vibration. It occurs when the liquid temperature is too high (usually above 90°C). When the liquid is too hot the injector becomes less efficient and a proportion of the steam escapes from the discharge tube.

  At higher steam pressures, condensation of the steam may cause vibration, which is not recommended for atmospheric tanks. However, in cylindrical pressure vessels of a robust design, this may not cause any problems.

- **Low flowrates -** When the steam pressure at the inlet to the injector falls below 1.5 bar g, a distinctive crackling can be heard. Under these conditions steam is unable to give up its enthalpy of evaporation before it leaves the injector tube.

  At low flowrates the steam is travelling at a lower velocity than in the other modes of operation, and collapsing steam bubbles are found on the body casting and in the connecting pipework, inducing cavitation. This noise is often considered objectionable, and may be found if the steam injector system has been oversized.

  Noise may also be caused by poor installation of the injector. The sides of a rectangular tank may be made from fairly flexible panels. Connecting an injector to the middle of a flexible panel may induce vibration and noise. It may often be better to mount the injector nearer the corner of the tank where the structure is stiffer.

Example 2.11.2

Based on data from Example 2.11.1, propose a steam injection system.

Required steam injection rate = 569 kg/h

The steam injection pressure = 1.0 bar

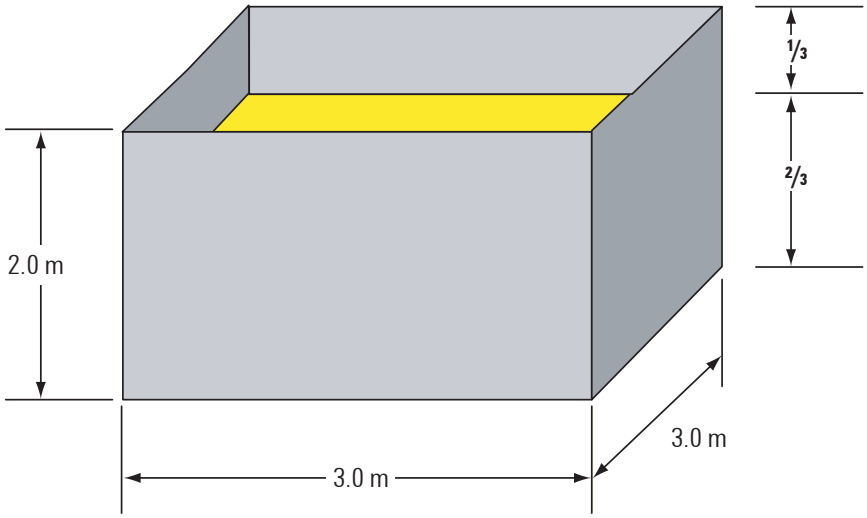


Fig. 2.11.5

Table 2.11.2 Typical steam injector capacity chart

Injector type	IN15	IN25M	IN40M
Steam pressure at inlet of injector (bar g)	**Saturated steam capacity kg/h**		
1	20	135	400
2	48	175	580
3	66	280	805
4	84	350	970
5	102	410	1125
6	120	500	1295
7	138	580	1445
8	156	640	1620
9	174	700	1820
10	192	765	1950
11	210	830	2250
12	228	900	2370
13	246	975	2595
14	264	1045	2710
15	282	1095	2815
16	300	1170	3065
17	318	1225	3200

The largest injector (IN40M) has a capacity of 400 kg/h at 1.0 bar, so this application will require:

$$\frac{569 \text{ kg/h}}{400 \text{ kg/h}} = 1.42 = \textbf{2 steam injectors}$$

Ideally, because of the low pressures involved, the injectors would be installed at opposite ends of the tank to give good mixing.

An alternative would be to use higher pressure steam. This would allow the use of just one, smaller injector, reducing costs and still providing good mixing.

Alternative method of calculating injected steam load

The previous method used in this Module to calculate the mean steam flowrate requires the mean heat load to be calculated first. This is depicted by Equation 2.11.1:

$$\dot{m}_s = \frac{\dot{Q}}{h_g - (T\ c_p)} \qquad \textbf{Equation 2.11.1}$$

Where:
$\dot{Q}$ = Mean heat transfer rate (kW)

If the mean heat transfer rate is not known, another method can be used to determine the mean steam flowrate. This requires the use of a heat balance as described below.

It should be noted that both methods return exactly the same result, so whichever is used depends upon the user's choice.

Calculating the mean steam flowrate by means of a heat balance

A heat balance is considered where the initial heat content in the water plus the heat added by the steam equals the final heat content. The heat balance equation for the water in the tank is shown in Equation 2.11.3:

$$m\ h_1 + m_s\ h_g = (m + m_s)\ h_2 \qquad \textbf{Equation 2.11.3}$$

Where:
m = Initial mass of water in the tank (kg)
h_1 = The heat in the water at the initial temperature (kJ/kg)
m_s = The mass of steam to be injected to raise the water temperature (kg)
h_g = The total enthalpy of the steam onto the control valve (kJ/kg)
h_2 = The heat in the water at the final temperature (kJ/kg)

Mass of steam to be injected

The mass of steam to be injected can be determined more directly from Equation 2.11.4, which is developed from Equation 2.11.3.

$$m_s = \frac{m\ (h_2 - h_1)}{h_g - h_2} \qquad \textbf{Equation 2.11.4}$$

Where:
m_s = The mass of steam to be injected (kg)
m = Initial mass of water in the tank (kg)
h_2 = The heat in the water at the final temperature (kJ/kg)
h_1 = The heat in the water at the initial temperature (kJ/kg)
h_g = The total enthalpy of the steam upstream of the control valve (kJ/kg)

Example 2.11.3
Consider the same conditions as that in Example 2.11.1.

Initial mass of water (m) = 12000 kg
Initial temperature = 8°C
h_1 = 8°C x 4.19 kJ/kg °C
h_1 = 33.5 kJ/kg
Final temperature = 60°C
h_2 = 60°C x 4.19 kJ/kg °C
h_2 = 251.4 kJ/kg
Pressure of steam = 2.6 bar g
m_s = Mass of steam to be injected from 2.6 bar g
h_g = Total enthalpy of steam at 2.6 bar g = 2733.9 kJ/kg

Conducting a heat balance on the water in the tank by using Equation 2.11.4:

$$m_s = \frac{m\,(h_2 - h_1)}{h_g - h_2}$$ **Equation 2.11.4**

Where:
m_s = The mass of steam to be injected to raise the water temperature (kg)
m = 12000 kg
h_2 = 251.4 kJ/kg
h_1 = 33.5 kJ/kg
h_g = 2733.9 kJ/kg

$$m_s = \frac{12\,000\,(251.4 - 33.5)}{2\,733.9 - 251.4}$$

$$m_s = \frac{2\,614\,800}{2\,482.5}$$

$$\mathbf{m_s = 1053\ kg}$$

As the tank is to be heated up in 2 hours:

$$\text{The mean steam flowrate} = \frac{\text{Total mass of steam used}}{\text{Time to heat tank}}$$

$$\text{The mean steam flowrate} = \frac{1053}{2}\ \text{kg/h}$$

The mean steam flowrate = 526 kg/h to heat the water

Conducting a heat balance on the tank material

m = Mass of tank = 3 886 kg
m_s = Mass of steam to be injected to raise the tank temperature
Initial temperature = 8°C
h_1 = 8°C x 0.5 kJ/kg °C
h_1 = 4 kJ/kg
Final temperature = 60°C
h_2 = 60°C x 0.5 kJ/kg °C
h_2 = 30 kJ/kg
Pressure of steam to be injected = 2.6 bar g (2 733.9 kJ/kg)
h_g = 2 733.9 kJ/kg

Using the heat balance Equation 2.11.4 with regard to the steel tank.

$$m_s = \frac{m\,(h_2 - h_1)}{h_g - h_2}$$ **Equation 2.11.4**

Where:
m_s = Mass of steam to be injected to raise the tank temperature
m = 3 886 kg
h_2 = 30 kJ/kg
h_1 = 4 kJ/kg
h_g = 2 733.9 kJ/kg

$$m_s = \frac{3\,886\,(30 - 4)}{2\,733.9 - 30}$$

$$m_s = \frac{101\,036}{2\,703.9}$$

$$m_s = 38 \text{ kg}$$

Over 2 hours heating time $m_s = \frac{38}{2}$ kg/h

$$\mathbf{m_s = 19 \text{ kg/h}}$$

The heat losses from the sides of the tank and the water surface are the same as previously calculated, that is 24 kg/h.

Total mean steam flowrate = Steam to heat water + steam to heat tank + heat losses
Total mean steam flowrate = 526 kg/h + 19 kg/h + 24 kg/h
Total mean steam flowrate = 569 kg/h

This is the same result as that obtained previously in this Module from Equations 2.11.1 and 2.11.2, and proves that either method can be used to calculate the mean steam flowrate to heat the tank and its contents.

Questions

1. A tank is to be heated by direct steam injection. How will the quantity of heat required compare with steam coil heating?

a| It depends on the temperature of the water being heated ☐

b| More heat will be required ☐

c| The same amount of heat will be required ☐

d| Less heat will be required ☐

2. An open topped tank of water measuring 1.5 x 2 x 1.5 m deep is to be heated from 15°C to 75°C in 2 hours using a steam injector supplied with steam at 3 bar g onto the control valve. The tank is well lagged so losses from the sides and base can be ignored. Still air conditions exist above the liquid surface. What is the approximate mean steam flowrate during start-up? (The water depth is 1.3 m).

a| 183 kg/h ☐

b| 156 kg/h ☐

c| 12 kg/h ☐

d| 200 kg/h ☐

3. Referring to Question 2 approximately how much steam would be required if coil heating were used?

a| 230 kg/h ☐

b| 293 kg/h ☐

c| 281 kg/h ☐

d| 248 kg/h ☐

4. Which of the following is an advantage of a steam injector over a sparge pipe?

a| Circulates liquid but is noisier ☐

b| Less robust but quieter ☐

c| Handles higher pressures and is more efficient at mixing the steam and liquid ☐

d| Suitable for lower pressures ☐

Answers

1: c, 2: d, 3: a, 4: c

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Module 2.12

Steam Consumption of Pipes and Air Heaters

Steam Consumption of Pipes and Air Heaters

Steam will condense and give up its enthalpy of evaporation on the walls of any pipe or tube exposed to ambient air. In some cases, such as steam mains, heat transfer is minimised by the lagging of the pipes. In other cases such as air heater batteries, heat transfer may be promoted by the use of fins on the outside of the pipes.

It is not usually possible or necessary to calculate steam consumption exactly. The examples in this Module allow sufficient estimates to be made for most practical purposes.

Steam mains

In any steam system, the condensation of steam caused by the pipe itself must be taken into account. The rate of condensation will be at its highest during the warming up period, and it is this that should govern the size of steam traps used for mains drainage. With the steam main in use, there will also be a smaller (but continual) heat loss from the pipe. Both of these components can be calculated as the 'warming up load' and the 'running load'.

Warm-up load

Heat will initially be required to bring the cold pipe up to working temperature. It is good practice to do this slowly for safety reasons, the pipes also benefit from reduced thermal and mechanical stress. This will result in fewer leaks, lower maintenance costs, and a longer life for the pipe. Slow warm-up can be achieved by fitting a small valve in parallel with the main isolating valve, (Figure 2.12.1). The valve can be sized depending on the warm-up time required. Automating the warm-up valve to open slowly on large pipes can improve safety.

A single main isolating valve can be used successfully, but, as it will be sized to pass the pipeline design flow requirements, it will be oversized during the warm-up period and will consequently operate very close to its seat at this time. A separator placed before the valve will ensure the steam passing through is dry, protecting the trim from premature wear.

The time taken to warm up any steam main should be as long as possible within acceptable limits to minimise mechanical pipework stress, optimise safety and reduce start-up loads.

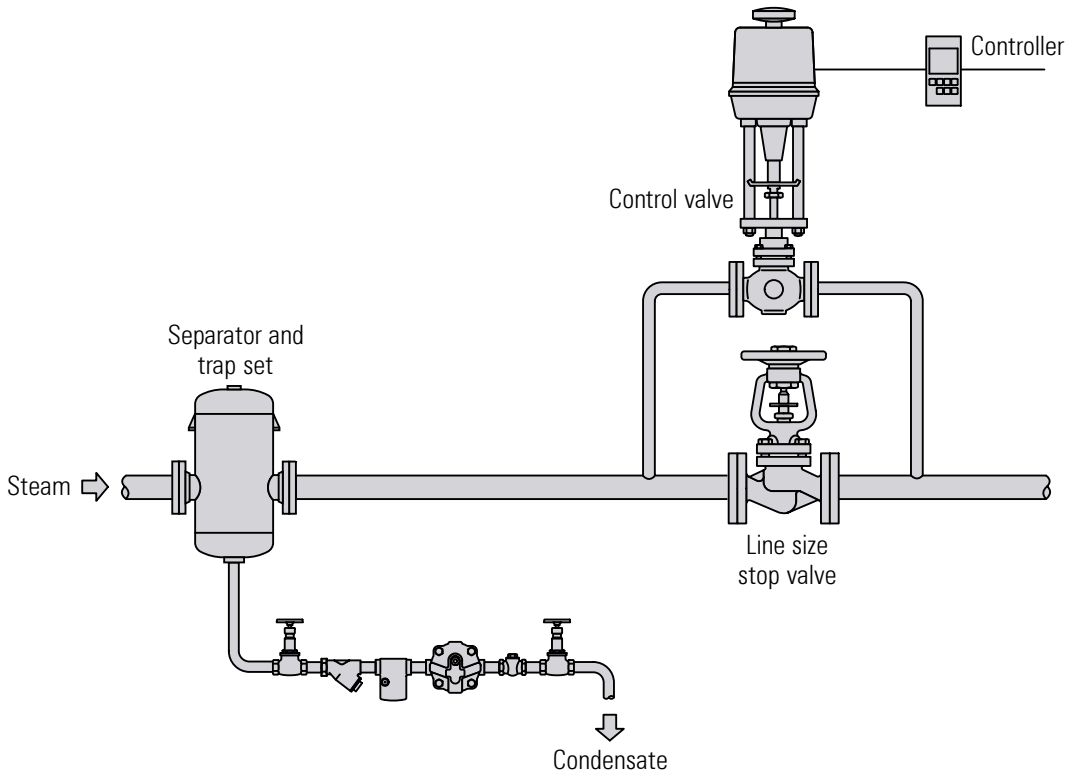


Fig. 2.12.1 Automatic warm-up valve in a Bypass

If 10 minutes can be taken instead of 5 minutes, the initial steam flowrate will be reduced by half. A warm-up time of 20 minutes will reduce the warm-up load even further.

The steam flowrate required to bring a pipework system up to operating temperature is a function of the mass and specific heat of the material, the temperature increase, the enthalpy of evaporation of the steam used, and the allowable time.

This may be expressed by Equation 2.12.1:

$$\dot{m}_s = \frac{60\ W\ (T_s - T_{amb})\ c_p}{h_{fg}\ t}\ kg/h \qquad \textbf{Equation 2.12.1}$$

Where:
$\dot{m}_s$ = Mean rate of condensation of steam (kg/h)
W = Total weight of pipe plus flanges and fittings (kg)
T_s = Steam temperature (°C)
T_{amb} = Ambient temperature (°C)
c_p = Specific heat of pipe material (kJ/kg°C)
h_{fg} = Enthalpy of evaporation at operating pressure (kJ/kg)
t = Time for warming up (minutes)

Note: The constant 60 and time in minutes gives the solution in kg/h

Table 2.12.1 Typical specific heat capacities of metal pipes

Pipe material	Specific heat capacity at 300°C (kJ/kg°C)
Copper	0.385
Carbon steel	0.490
Chromium steel	0.443
AISI 302 Stainless steel	0.480
AISI 304 Stainless steel	0.477
AISI 316 Stainless steel	0.468
AISI 347 Stainless steel	0.480

Example 2.12.1 Heat losses from a steam pipeline

A system consists of 100 m of 100 mm carbon steel main, which includes 9 pairs of PN40 flanged joints, and one isolating valve.

c_p for steel = 0.49 kJ/kg°C

The ambient/starting temperature is 20°C and the steam pressure is 14.0 bar g, 198°C from steam tables (see Table 2.12.2).

Table 2.12.2 Extract from steam tables

Pressure bar g	Saturation temperature °C	Enthalpy (energy) in kJ/kg			Specific volume of dry saturated steam m³/kg
		Water h_f	Evaporation h_{fg}	Steam h_g	
14	198	845	1 947	2 792	0.132

Determine:

Part 1. The warm-up condensing rate for a warm-up time of 30 minutes.
Part 2. The running load if the insulation thickness is 75 mm.

Part 1 Calculate the warm-up load

$$\dot{m}_s = \frac{60\ W\ (T_s - T_{amb})\ c_p}{h_{fg}\ t}\ kg/h \qquad \textbf{Equation 2.12.1}$$

To find W, find the mass of the various steam main items from Table 2.12.3.

100 mm steel main = 16.1 kg/m

100 mm flanges to PN40 = 16.0 kg per pair

100 mm stop valve = 44.0 kg each

Therefore: $W = (100 \times 16.1) + (9 \times 16) + (1 \times 44) = 1\ 798$ kg

So, the mean warming up load:

$$\dot{m}_s = \frac{60 \times 1\ 789\ kg \times (198°C - 20°C) \times 0.49\ kJ/kg°C}{1\ 947\ kJ/kg \times 30\ minutes}\ kg/h$$

Mean warming up load $\dot{m}_s$ = 161 kg/h

Note: This condensing rate will be used to select an appropriate warm-up control valve.

When selecting steam traps, this condensing rate should be multiplied by a factor of two to allow for the lower steam pressure that will occur until warm-up is completed, then divided by the number of traps fitted to give the required capacity of each trap.

Table 2.12.3 Typical weights of steel pipe, flanges and bolts, and isolating valves in kg

Pipe size (mm)	Sch. 40 pipe kg/m	Flange weight per pair			Isolating valve flanged PN40
		PN40	ANSI 150	ANSI 300	
15	1.3	1.7	1.8	2	4
20	1.7	2.3	2.2	3	5
25	2.5	2.6	2.4	4	6
32	3.4	4.0	3.0	6	8
40	4.1	5.0	4.0	8	11
50	5.4	6.0	6.0	9	14
65	8.6	9.0	8.0	12	19
80	11.3	11.0	11.0	15	26
100	16.1	16.0	16.0	23	44
150	28.2	28.0	26.0	32	88

Part 2 Running load

Steam will condense as heat is lost from the pipe to the environment: The rate of condensation depends on the following factors:

- The steam temperature.
- The ambient temperature.
- The efficiency of the lagging.

Table 2.12.4 gives typical heat emission rates expected from unlagged steel pipes in still air at 20°C.

Table 2.12.4 Heat emission from unlagged steel pipes freely exposed in air at 20°C (W/m)

Temperature differential steam to air °C	Pipe size (mm)									
	15	20	25	32	40	50	65	80	100	150
50	56	68	82	100	113	136	168	191	241	332
60	69	85	102	125	140	170	208	238	298	412
70	84	102	124	152	170	206	252	289	360	500
80	100	122	148	180	202	245	299	343	428	594
100	135	164	199	243	272	330	403	464	577	804
120	173	210	256	313	351	426	522	600	746	1 042
140	216	262	319	391	439	533	653	751	936	1 308
160	263	319	389	476	535	651	799	918	1 145	1 603
180	313	381	464	569	640	780	958	1 100	1 374	1 925
200	368	448	546	670	754	919	1 131	1 297	1 623	2 276
220	427	520	634	778	877	1 069	1 318	1 510	1 892	2 655

Distribution mains will normally be lagged however, and is obviously an advantage if flanges and other items of pipeline equipment are lagged too. If the main is flanged, each pair of flanges will have approximately the same surface area as 300 mm of pipe of the same size.

The rate of heat transfer increases when a heat transfer surface is subjected to air movement. In these cases, the multiplication factors, as shown in Table 2.12.5, should be considered.

If finned or corrugated tubing is fitted, then the maker's figures for heat emission should always be used.

In everyday terms, air velocities up to 4 or 5 m/s (approximately 10 mph) represent a gentle breeze, between 5 and 10 m/s (approximately 10 - 20 mph) a strong breeze. Typical air duct velocities are around 3 m/s, in comparison.

Table 2.12.5 Approximate increase in emission due to air movement over pipes with a high emissivity

Air velocity (m/s)	Emission factor
0.00	1.0
0.50	1.0
1.00	1.3
1.50	1.5
2.00	1.7
2.50	1.8
3.00	2.0
4.00	2.3
6.00	2.9
8.00	3.5
10.00	4.0

Note: Exact figures are difficult to determine, as many factors are involved. The factors in Table 2.12.5 are derived and give a rough indication of how much the figures in Table 2.12.4 should be multiplied. Pipes subjected to air movement up to around 1 m/s can be thought of as being in still air, and heat losses are fairly constant up to this point. As a guide, painted pipes will have a high emissivity, oxidised steel a medium emissivity, and polished stainless steel a low emissivity.

The reduction in heat losses will depend on the type and thickness of the lagging material used, and on its general condition. For most practical purposes, the lagging of steam lines will reduce the heat emissions in Table 2.12.4 by the insulation factors (f) shown in Table 2.12.6. Note that these factors are nominal values only. For specific calculations, consult the insulation manufacturer.

Table 2.12.6 Insulation factors 'f'

Pipe size NB (mm)	Steam pressure			
	1 bar g	5 bar g	15 bar g	20 bar g
50 mm insulation				
15	0.16	0.14	0.13	0.12
20	0.15	0.13	0.12	0.11
25	0.14	0.12	0.11	0.10
32	0.13	0.11	0.10	0.10
40	0.12	0.11	0.10	0.09
50	0.12	0.10	0.09	0.08
65	0.11	0.10	0.09	0.08
80	0.10	0.10	0.08	0.07
100	0.10	0.09	0.08	0.07
150	0.10	0.09	0.07	0.07
75 mm insulation				
15	0.14	0.13	0.12	0.11
20	0.13	0.11	0.11	0.10
25	0.13	0.11	0.10	0.09
32	0.11	0.10	0.09	0.08
40	0.10	0.09	0.09	0.08
50	0.10	0.09	0.08	0.07
65	0.10	0.08	0.08	0.07
80	0.09	0.08	0.07	0.07
100	0.08	0.08	0.07	0.06
150	0.08	0.07	0.07	0.06
100 mm insulation				
15	0.12	0.11	0.10	0.08
20	0.11	0.10	0.09	0.07
25	0.10	0.09	0.08	0.07
32	0.10	0.08	0.08	0.06
40	0.09	0.08	0.08	0.06
50	0.08	0.08	0.07	0.06
65	0.08	0.07	0.06	0.05
80	0.07	0.07	0.06	0.05
100	0.07	0.07	0.06	0.05
150	0.07	0.06	0.05	0.04

The heat loss from insulated mains can be expressed as follows in Equation 2.12.2:

$$\dot{m}_s = \frac{3.6\ \dot{Q}\ L\ f}{h_{fg}}\ kg/h \qquad \textbf{Equation 2.12.2}$$

Where:
$\dot{m}_s$ = Rate of condensation (kg/h)
$\dot{Q}$ = Heat emission rate from Table 2.12.4 (W/m)
L = Effective length of pipe allowing for flanges and fittings (m)
f = Insulation factor (from Table 2.12.6)
h_{fg} = Enthalpy of evaporation at operating pressure (kJ/kg)

Note: f = 1.0 if the main is not insulated.
The factor 3.6 in Equation 2.12.2 provides a solution in kg/h

Determine the length, L:
Assuming an allowance equivalent to 0.3 m for each pair of flanges, and 1.2 m for each stop valve, the total effective length (L) of the steam main in this example is:

$$L = 100 + (9 \times 0.3) + (1 \times 1.2)$$
$$\mathbf{L = 103\ m}$$

Determine the heat emission rate, $\dot{Q}$:
The temperature of the steam at 14.0 bar gauge is 198°C and, with the ambient temperature 20°C, the temperature difference is 178°C.

From Table 2.12.4: Heat loss for a 100 mm pipe ≈ **1 374 W/m**

Determine the insulation factor, f:
The insulation factor for 75 mm insulation on 100 mm pipe at 14 bar g (from Table 2.12.6) is approximately 0.07.

h_{fg} at 14 bar g = 1 947 kJ/kg (from steam tables)

$$\dot{m}_s = \frac{3.6 \text{ x } 1\,374 \text{ W/m x } 103 \text{ m x } 0.07}{1\,947 \text{ kJ/kg}}\ kg/h$$

$$\dot{m}_s = 18.3\ kg/h$$

As can be seen from this example, the warm-up load of 161 kg/h (see Example 2.12.1, Part 1) is substantially greater than the running load of 18.3 kg/h, and, in general, steam traps sized on the warm-up duty will automatically cater for the running load.

If the steam line above was unlagged or the lagging was damaged, the running load would have been approximately fourteen times greater.

With an uninsulated pipe, or a poorly insulated pipe, always compare the running and warm-up loads. The higher load should be used to size the steam traps, as described above. Ideally, the quality of insulation should be improved.

Note: When calculating warming up losses, it is sensible to consider the correct pipe specification, as pipe weights can vary between different pipe standards.

Air heating

The density and specific heat of air changes slightly with temperature. For most practical purposes, when heating air for HVAC and process applications with the approach mentioned below, a nominal figure of 1.3 kJ/m³ °C can be used for specific heat and 1.3 kg/m³ for density.

Air heating pipes

Heated air is required for many applications including:

- Space heating.
- Ventilation.
- Process applications.

The equipment required often consists of a matrix of tubes filled with steam, installed across an air stream. As the air passes over the tubes, heat is transferred from the steam to the air. Often, in order to minimise the size and mass of the equipment, and allow it to be installed in confined spaces with reduced support works, and to limit the cost, the rate of heat transfer from the tubes to the air is increased by the addition of fins to the outer wall of the tube.

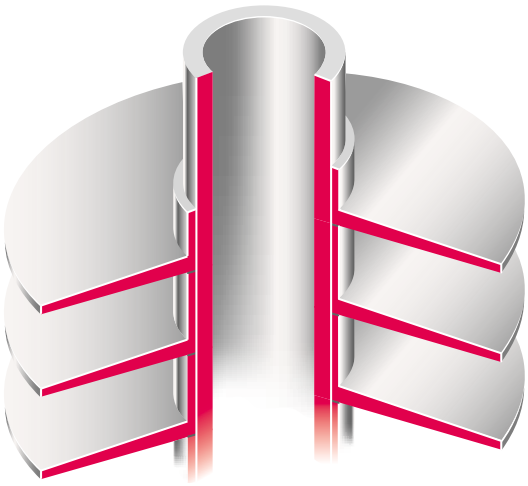


Fig. 2.12.2 Finned tube

This has the effect of increasing the heat transfer area available, and thus reducing the amount of piping required. Figure 2.12.2 shows an example of a finned tube.

Broadly, air heaters may be divided into two categories:

- Unit heaters.
- Air heater batteries.

Unit heaters

These consist of a heater battery and fan in one compact casing (Figure 2.12.3). The primary medium (steam) condenses in the heater battery, and air is warmed as it blows across the coils and is discharged into the space.

Unit heaters can be arranged to have fresh air inlet ducting, but more often operate with recirculated air.

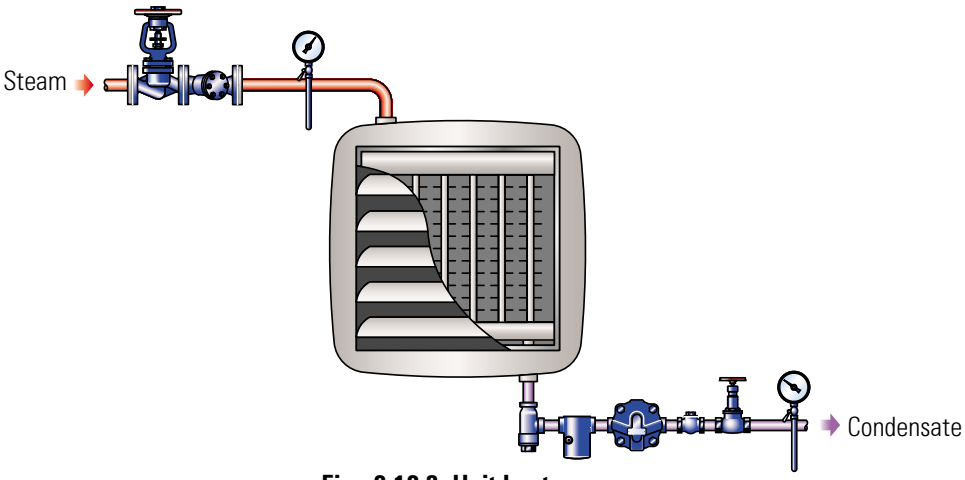


Fig. 2.12.3 Unit heater

The warm air can be discharged vertically downwards or horizontally. Steam pressure, mounting heights, the type of discharge and leaving temperatures are all inter-related and the manufacturer's data should be consulted before selecting the unit heater. Most units are available with low, medium or high speed fans which affect the rated output, and again the manufacturer's data should be consulted, as the noise levels on high speed may be unacceptable.

Air heater batteries

These are really larger and more sophisticated versions of unit heaters, see Figure 2.12.4. They are available in many configurations including roof mounted, or horizontal types, and a fan and filter may also be incorporated. They are usually integrated into a ducted air system.

- Adjustable louvres may be provided to adjust the ratio of fresh to recirculated air.
- A number of heater banks may be incorporated to provide frost protection.

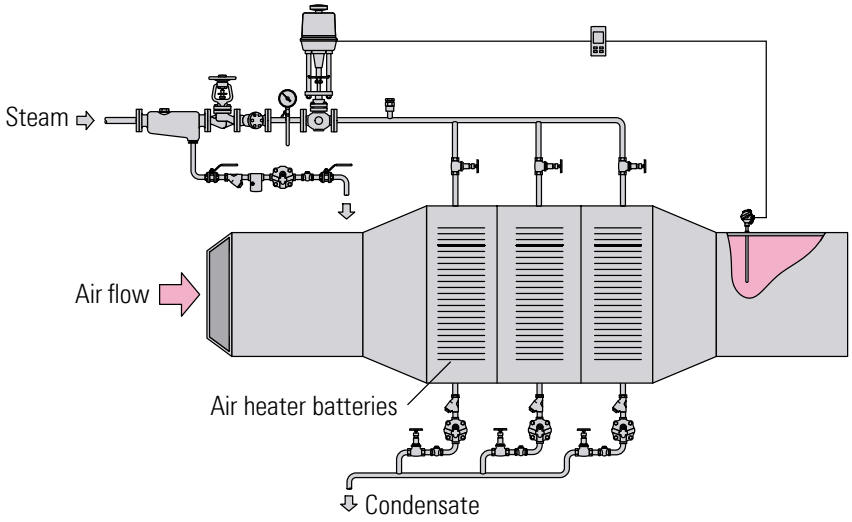


Fig. 2.12.4 Ducted air system with air heater batteries

Manufacturers of unit heaters and air heater batteries usually give the output of their heaters in kW at a working pressure. From this, the condensing rate can be calculated by dividing the heat output by the enthalpy of evaporation of steam at this pressure. The solution will be in kg/s; multiplying by 3 600 (seconds in an hour) will give the solution in kg/h.

Thus a 44 kW unit heater working at 3.5 bar g (h_{fg} = 2 120 kJ/kg from steam tables) will condense:

$$\dot{m}_s = \frac{\text{kW x 3 600 s/h}}{h_{fg}}$$

$$\dot{m}_s = \frac{44 \text{ x } 3\,600}{2\,120}$$

$$\dot{m}_s = \mathbf{75\ kg/h}$$

Note: The constant 3 600 is included in the formula to give flowrate in kg/h rather than kg/s.

If the manufacturer's figures are not available but the following are known:

- The volumetric flowrate of air being heated.
- The temperature rise of the air being heated.
- The steam pressure in the heater.

Then the approximate rate of condensation can be calculated using Equation 2.12.3:

$$\dot{m}_s = \frac{3\,600\ \dot{V}\ \Delta T\ c_p}{h_{fg}} \text{ kg/h} \qquad \textbf{Equation 2.12.3}$$

Where:
$\dot{m}_s$ = Rate of steam condensation (kg/h)
$\dot{V}$ = Volumetric flowrate of air being heated (m^3/s)
ΔT = Air temperature rise (°C)
c_p = Specific heat of air at constant pressure (1.3 kJ/m^3 °C)
h_{fg} = Enthalpy of evaporation of steam in the coils (kJ/kg)

Note: The constant 3 600 gives the solution in kg/h rather than kg/s.

Horizontal pipes assembled into coils with several rows of pipes one above the other, and relying upon natural convection, become less effective as the number of pipes is increased. When calculating the rate of condensation for such coils, the figures given in Table 2.12.5 should be multiplied by the emission factors in Table 2.12.7.

Vertically installed heating pipes are also less effective than horizontal pipes. The condensation rate of such pipes can be determined by multiplying the figures in Table 2.12.4 by the factors in Table 2.12.6.

Table 2.12.7 can also be used to find the rate of condensation in horizontal pipes used for heating still air. In this instance use the Equation 2.12.4:

$$\dot{m}_s = \frac{3.6\ \dot{Q}\ L}{h_{fg}}\ kg/h \qquad \textbf{Equation 2.12.4}$$

Where:
$\dot{m}_s$ = Rate of steam condensation (kg/h)
$\dot{Q}$ = Heat emission from Table 2.12.4 (W/m)
L = Effective length of pipe (metres)
h_{fg} = Enthalpy of evaporation at the working pressure (kJ/kg)

Note: The constant 3.6 has been included in the Equation to give $\dot{m}_s$ in kg/h.

Table 2.12.7 Approximate reduction in emission of banked horizontal pipes

Number of pipes	1	2	3	4	5	6	7	8	9	10
Emission factor	1.00	0.96	0.91	0.86	0.82	0.78	0.74	0.70	0.67	0.63

Table 2.12.8 Approximate reduction in emission of banked vertical pipes

Pipe size mm	15	20	25	32	40	50	65	80	100	150
Emission factor	0.76	0.80	0.82	0.84	0.86	0.88	0.91	0.93	0.95	1.00

Effects of air flowrate

When a fan is used to increase the flow of air over pipe coils, the rate of condensation will increase. The figures for heat emission from bare steel pipes (Table 2.12.4), can be used when multiplied in accordance with the factors in Tables 2.12.5, 2.12.7 and 2.12.8 where appropriate.

If finned tubing is being considered, then the maker's figures for heat emission should be used in all cases.

Example 2.12.2 Calculate the steam load on an air heater battery

An air heater battery raises the temperature of air flowing at 2.3 m³/s from 18°C to 82°C (ΔT = 64°C) with steam at 3.0 bar g in the coils.

Table 2.12.9 Extract from steam tables

Pressure bar g	Saturation temperature °C	Enthalpy (energy) in kJ/kg			Specific volume of dry saturated steam m³/kg
		Water h_f	Evaporation h_{fg}	Steam h_g	
3	144	605	2133	2738	0.461
4	152	641	2108	2794	0.374

The rating of the battery is unknown, but the condensing rate of steam can be calculated using Equation 2.12.3:

$$\dot{m}_s = \frac{3600\ \dot{V}\ \Delta T\ c_p}{h_{fg}}\ kg/h \qquad \textbf{Equation 2.12.3}$$

Where:
$\dot{m}_s$ = Rate of condensation (kg/h)
$\dot{V}$ = Air flowrate 2.3 m³/s
ΔT = Air temperature 82 - 18°C = 64°C
c_p = Specific heat of air at constant pressure (1.3 kJ/m³°C)
h_{fg} = Enthalpy of evaporation of steam in the coils 2 133 kJ/kg (from steam tables)

Note: The constant 3 600 is included in the Equation to give flowrate in kg/h rather than kg/s.

$$\dot{m} = \frac{3600 \text{ x } 2.3\ m^3/s \text{ x } 64°C \text{ x } 1.3\ kJ/m^3°C}{2133\ kJ/kg}\ kg/h$$

$$\dot{m} = \mathbf{323\ kg/h}$$

Questions

1. A process air heater battery raises 4 m³/s of air from 20°C to 50°C. It is necessary to size steam traps for the heater. The specific heat of air is 1.3 kJ/m³°C. The steam supply pressure is 4 bar g upstream of the control valve.
What will be the approximate condensate flowrate from the heater battery ?

a| 147 kg/h ☐
b| 218 kg/h ☐
c| 252 kg/h ☐
d| 272 kg/h ☐

2. A 40 m section of 80 mm steam main has been left unlagged. It incorporates two pairs of flanges and an isolation valve. The surrounding air is still and at 20°C. Steam is at 7 bar g.
What would be the approximate saving in the heat emission if the main was lagged with 75 mm insulation ?

a| 3.5 kW ☐
b| 31 kW ☐
c| 35 kW ☐
d| 33 kW ☐

3. An air heater heating air from 5°C to 35°C has a rating of 25 kW when supplied with steam at 7 bar g onto the control valve.
What will be the approximate steam consumption rate of the heater ?

a| 33 kg/h ☐
b| 57 kg/h ☐
c| 51 kg/h ☐
d| 44 kg/h ☐

4. What will be the approximate mean rate of condensation during a 30 minute warm-up of a 100 m length of 65 mm schedule 40 carbon steel pipe ?

Incorporated in the pipe are 4 pairs of ANSI 150 flanges and two isolating valves. The main is well insulated with 75 mm of insulation and the surrounding air can be considered as still and at -5°C. Steam is at 10 bar g. The specific heat of steel is 0.434 kJ/kg°C.

a| 75 kg/h ☐
b| 55 kg/h ☐
c| 45 kg/h ☐
d| 150 kg/h ☐

5. With reference to Question 4, what will be the approximate mean radiation losses during start-up?

a| 18 kW ☐
b| 26 kW ☐
c| 32 kW ☐
d| 2 kW ☐

6. A process air heater battery whose control valve is supplied with steam at 4 bar g delivers 4 m³/s of air, heated from 20°C to 50°C. It is necessary to size a steam trap for the battery on the running load.
What will be the condensate flowrate from the heater ? The specific heat of air is 1.0 kJ/kg°C and the density of air is about 1.3 kg/m³.

a| 187 kg/h ☐

b| 228 kg/h ☐

c| 252 kg/h ☐

d| 266 kg/h ☐

Answers

1: c, 2: b, 3: d, 4: a, 5: d, 6: c

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Module 2.13

Steam Consumption of Heat Exchangers

Steam Consumption of Heat Exchangers

The term heat exchanger strictly applies to all types of equipment in which heat transfer is promoted from one medium to another. A domestic radiator, where hot water gives up its heat to the ambient air, may be described as a heat exchanger. Similarly, a steam boiler where combustion gases give up their heat to water in order to achieve evaporation, may be described as a fired heat exchanger.

However, the term is often more specifically applied to shell and tube heat exchangers or plate heat exchangers, where a primary fluid such as steam is used to heat a process fluid. A shell and tube heat exchanger used to heat water for space heating (using either steam or water) is often referred to as a non-storage calorifier. (A storage calorifier, as shown in Figure 2.13.1, is constructed differently, it usually consists of a hot water storage vessel with a primary heating coil inside).

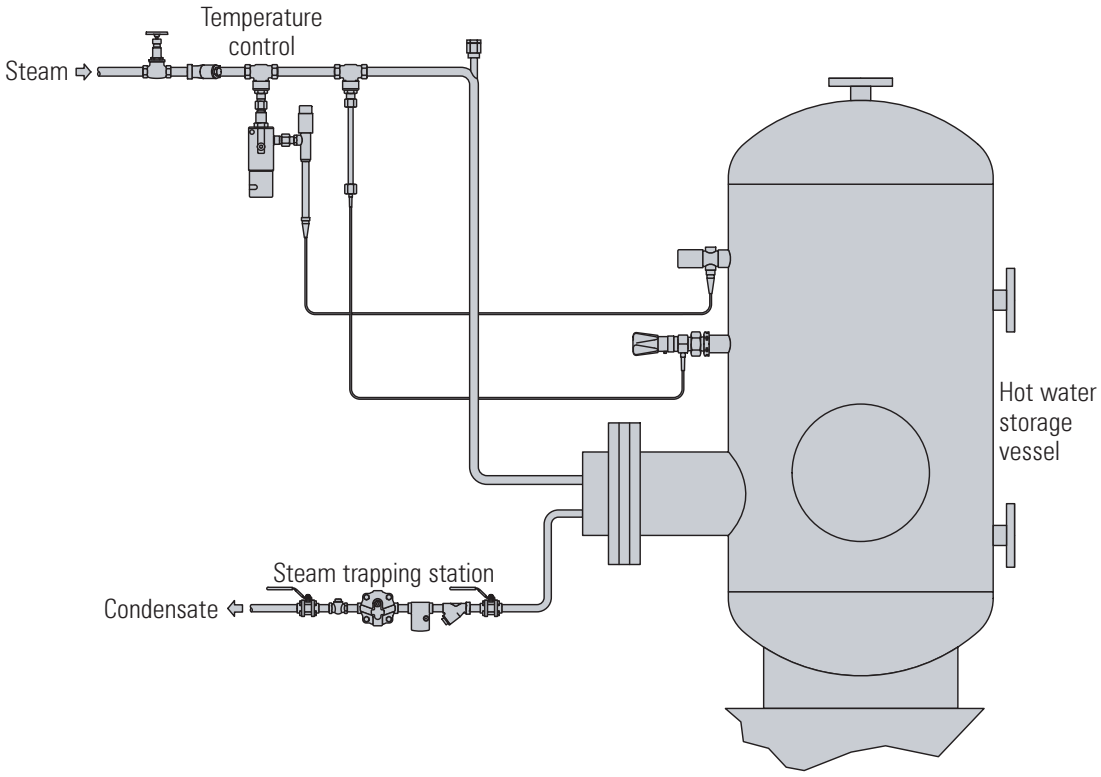


Fig. 2.13.1 A storage calorifier installation

Manufacturers often provide a thermal rating for their heat exchangers in kW, and from this the steam consumption may be determined, as for air heater batteries. However, heat exchangers (particularly shell and tube) are frequently too large for the systems which they are required to serve.

A non-storage calorifier (as shown in Figure 2.13.2) will normally be selected from a standard range of sizes, and may often have a much larger capacity than the design figure. For the hot water heating of buildings there may also be certain safety factors included in the heat load calculations.

Plate heat exchangers may also be chosen from a standard range of sizes if the units are brazed or welded. However, there is more flexibility in the sizing of gasketed plate heat exchangers, where plates can often be added or removed to achieve the desired heat transfer area. In many cases, plate heat exchangers are oversized simply to reduce the pressure drop for the secondary fluid.

On existing plant, an indication of actual load may be obtained if the flow and return temperatures and the pumping rate are known. However, it is important to note that throughput as given on the pump maker's plate will probably relate to a pressure head, which may or may not be present in practice.

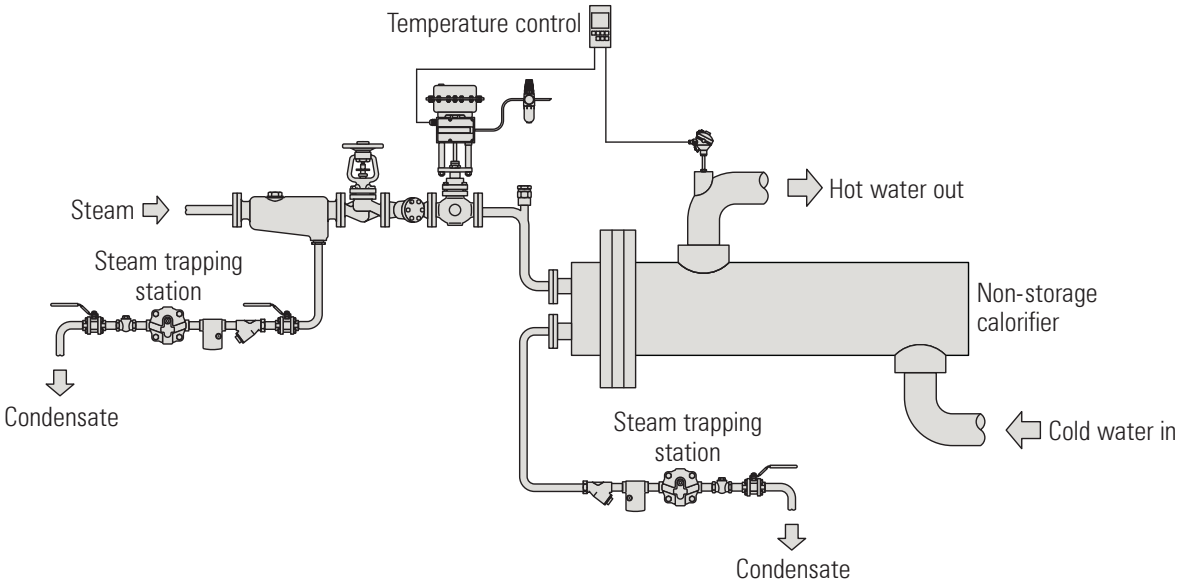


Fig. 2.13.2 A non-storage calorifier installation

Steam consumption calculations for heat exchangers

Shell and tube heat exchangers and plate heat exchangers are typical examples of flow type applications. Therefore, when determining the steam consumption for these applications, Equation 2.6.5 should be used.

The start-up load may be ignored if it occurs rarely, or if the time taken to reach full-load output is not too important. Heat exchangers are more often sized on the full running load, with the possible addition of safety factors.

Heat losses are rarely taken into account with these flow type applications, as they are significantly less than the full running load. Shell and tube heat exchangers are usually lagged to prevent heat loss, and to prevent possible injury to personnel. Plate heat exchangers tend to be more compact and have a much smaller surface area exposed to the ambient air, in relation to the size of the unit.

Example 2.13.1
Determine the heat load and steam load of the following non-storage heating calorifier

A heating calorifier is designed to operate at full-load with steam at 2.8 bar g in the primary steam space.

The secondary water flow and return temperatures are 82°C and 71°C respectively, at a pumped water rate of 7.2 kg/s.

c_p for water = 4.19 kJ/kg°C

Table 2.13.1 Extract from steam tables

Pressure bar g	Saturation temperature °C	Enthalpy (energy) in kJ/kg			Specific volume of dry saturated steam m³/kg
		Water h_f	Evaporation h_{fg}	Steam h_g	
2	134	562	2163	2725	0.603
2.8	142	596	2139	2735	0.489
3	144	605	2133	2738	0.461

Part 1 Determine the heat load

The full-load may be calculated using Equation 2.6.5:

$$\dot{Q} = \dot{m}\ c_p\ \Delta T \qquad \textbf{Equation 2.6.5}$$

Where:

$\dot{Q}$ = Quantity of heat energy (kW) kJ/s
$\dot{m}$ = Secondary fluid flowrate = 7.2 kg/s
c_p = Specific heat capacity of the water = 4.19 kJ/kg°C
ΔT = Temperature rise of the substance (82 - 71) = 11°C

$$\dot{Q} = 7.2\ \text{kg/s} \times 4.19\ \text{kJ/kg°C} \times 11°\text{C}$$

$$\dot{Q} = \mathbf{332\ kW}$$

Part 2 Determine the steam load

The full-load condensing rate can be determined using the left hand side of the heat balance Equation 2.6.6:

$$\dot{m}_s\ h_{fg} = \dot{Q} = \dot{m}\ c_p\ \Delta T \qquad \textbf{Equation 2.6.6}$$

Where:

$\dot{m}_s$ = Steam consumption (kg/s)
h_{fg} = Specific enthalpy of evaporation (kJ/kg)
$\dot{Q}$ = Heat transfer rate (kW)

Rearranging: a 332 kW calorifier working at 2.8 bar g (h_{fg} = 2 139 kJ/kg from steam tables) will condense:

$$\dot{m}_s = \frac{\dot{Q}}{h_{fg}}$$

$$\dot{m}_s = \frac{332}{2\,139}\ \text{kg/s}$$

$$\dot{m}_s = 0.155\,2\ \text{kg/s}$$

$$\dot{m}_s = \mathbf{558\ kg/h}$$

Plate heat exchangers

A plate heat exchanger consists of a series of thin corrugated metal plates between which a number of channels are formed, with the primary and secondary fluids flowing through alternate channels. Heat transfer takes place from the primary fluid steam to the secondary process fluid in adjacent channels across the plate. Figure 2.13.3 shows a schematic representation of a plate heat exchanger.

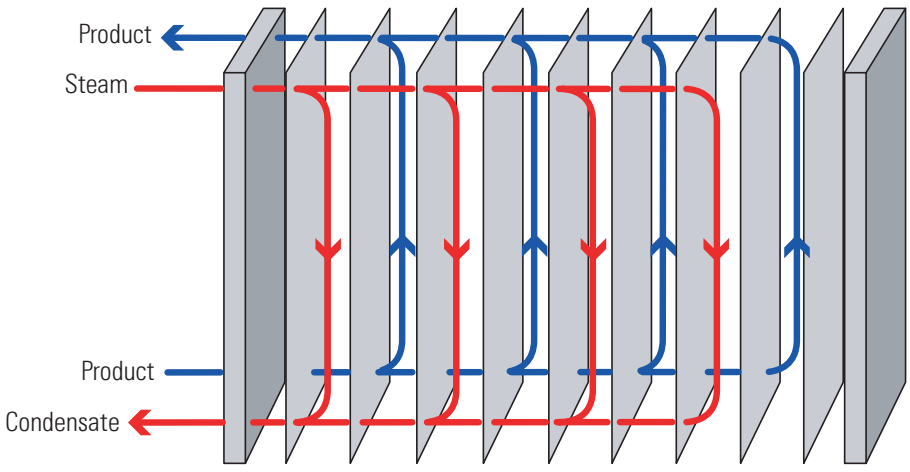


Fig. 2.13.3 Schematic diagram of a plate heat exchanger

A corrugated pattern of ridges increases the rigidity of the plates and provides greater support against differential pressures. This pattern also creates turbulent flow in the channels, improving heat transfer efficiency, which tends to make the plate heat exchanger more compact than a traditional shell and tube heat exchanger. The promotion of turbulent flow also eliminates the presence of stagnant areas and thus reduces fouling. The plates will usually be coated on the primary side, in order to promote the dropwise condensation of steam.

The steam heat exchanger market was dominated in the past by the shell and tube heat exchanger, whilst plate heat exchangers have often been favoured in the food processing industry and used water heating. However, recent design advances mean that plate heat exchangers are now equally suited to steam heating applications.

A plate heat exchanger may permit both the condensing and sub-cooling of condensate within a single unit. If the condensate is drained to an atmospheric receiver, by reducing the condensate temperature, the amount of flash steam lost to the atmosphere through the receiver vent is also reduced. This can eliminate the need for a separate sub-cooler or flash steam recovery system.

Although a nominal heat transfer area may theoretically be calculated using Equation 2.5.3, plate heat exchangers are proprietary designs and will normally be specified in consultation with the manufacturers.

Gasketed plate heat exchangers (plate and frame heat exchangers) - In a gasketed plate heat exchanger the plates are clamped together in a frame, and a thin gasket (usually a synthetic polymer) seals each plate around the edge. Tightening bolts fitted between the plates are used to compress the plate pack between the frame plate and the pressure plate. This design allows easy dismantling of the unit for cleaning, and allows the capacity of the unit to be modified by the simple addition or removal of plates.

The use of gaskets gives a degree of flexibility to the plate pack, offering some resistance to thermal fatigue and sudden pressure variations. This makes some types of gasketed plate heat exchanger an ideal choice as a steam heater for instantaneous hot water supply, where the plates will be exposed to a certain amount of thermal cycling.

The limitation in the use of the gasketed plate heat exchanger lies in the operating temperature range of the gaskets, which places a restriction on the steam pressure that may be used on these units.

Brazed plate heat exchangers - In a brazed plate heat exchanger all the plates are brazed together (normally using copper or nickel) in a vacuum furnace. It is a development of the gasketed plate heat exchanger, and was developed to provide more resistance to higher pressures and temperatures at a relatively low cost.

However, unlike the gasketed unit, the brazed plate heat exchanger cannot be dismantled. If cleaning is required it must be either back-flushed or chemically cleaned. It also means that these units come in a standard range of sizes, consequently oversizing is common.

While the brazed heat exchanger has a more robust design than the gasketed type, it is also more prone to thermal fatigue due to its more rigid construction. Any sudden or frequent changes in temperature and load should therefore be avoided, and greater attention should be paid to the control on the steam side to avoid thermal stress.

Brazed heat exchangers are more suitable (and primarily used) for applications where temperature variations are slow, such as in space heating. They may also successfully be used with secondary fluids which expand gradually, such as thermal oil.

Welded plate heat exchangers - In a welded plate heat exchanger the plate pack is held together by welded seams between the plates. The use of laser welding techniques allows the plate pack to be more flexible than a brazed plate pack, enabling the welded unit to be more resistant to pressure pulsation and thermal cycling. The high temperature and pressure operating limits of the welded unit mean that these heat exchangers normally have a higher specification, and are more suited to heavy duty process industry applications. They are often used where a high pressure or temperature performance is required, or when viscous media such as oil and other hydrocarbons are to be heated.

Shell and tube heat exchangers

The shell and tube heat exchanger is probably the most common method of providing indirect heat exchange in industrial process applications. A shell and tube heat exchanger consists of a bundle of tubes enclosed in a cylindrical shell. The ends of the tubes are fitted into tube sheets, which separate the primary and the secondary fluids.

Where condensing steam is used as the heating medium, the heat exchanger is usually horizontal with condensation taking place inside the tubes. Sub-cooling may also be used as a means to recover some extra heat from the condensate in the heat exchanger. However, if the degree of sub-cooling required is relatively large it is often more convenient to use a separate condensate cooler.

Steam heated non-storage calorifiers

A common design for a steam to water non-storage calorifier is shown in Figure 2.13.4. This is known as a 'one shell pass two tube pass' type of shell and tube heat exchanger and consists of a U-tube bundle fitted into a fixed tube sheet.

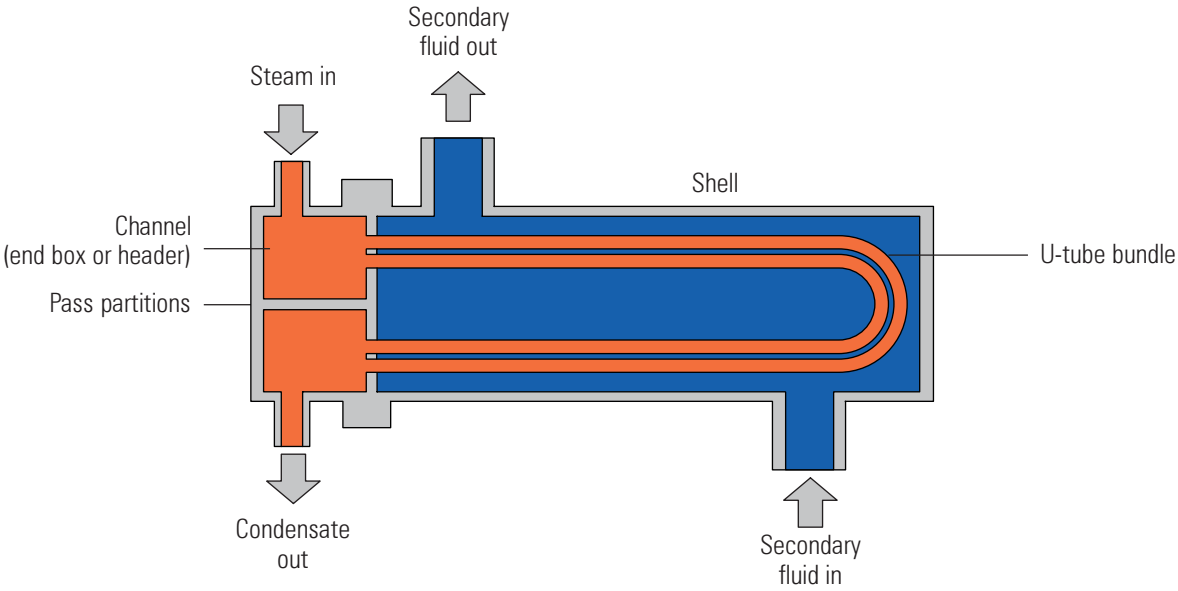


Fig. 2.13.4 Schematic diagram of a shell and tube heat exchanger

It is said to have 'one shell pass' because the secondary fluid inlet and outlet connections are at different ends of the heat exchanger, consequently the shell side fluid passes the length of the unit only once. It is said to have two tube passes because the steam inlet and outlet connections are at the same end of the exchanger, so that the tube-side fluid passes the length of the unit twice.

A pass partition (also called a partition plate or a feather plate) divides up the exchanger header, so that the tube-side fluid is diverted down the U-tube bundle rather than straight through the header.

This is a comparatively simple and inexpensive design because only one tube sheet is required, but it is limited in use to relatively clean fluids as the tubes are more difficult to clean. Note; it is more difficult to replace a tube with these types of heat exchanger.

Baffles are usually provided in the shell, to direct the shell-side fluid stream across the tubes, improving the rate of heat transfer, and to support the tubes.

Starting from cold

As mentioned in Module 2.7, the start-up load can often be ignored if it seldom occurs or if the time taken to reach full-load output is not critical. For this reason, control valves and heat exchangers will often be found to be sized on full-load plus the usual safety factors.

With systems that shut down at night and weekends, secondary water temperature can be low at start-up on a cold winter morning, and condensing rates in heating calorifiers will be higher than the full-load condition. Consequently, pressure in the steam space may be considerably below the pressure at which the heat exchanger normally operates, until the secondary inlet temperature rises to its design figure.

From a thermal viewpoint, this may not pose a problem - the system simply takes longer to heat up. However, if the designer has not taken this situation into consideration, an inadequate steam trapping and condensate removal system can cause condensate to accumulate in the steam space.

This can cause:

- Internal corrosion.
- Mechanical stress due to distortion.
- Noise, due to waterhammer.

These will cause problems for heat exchangers not designed to withstand such conditions.

Estimating heating loads

Buildings - A practical, subjective method to estimate a heating load is to look at the building itself. Calculations can be complicated, involving factors such as the number of air changes and heat transfer rates through cavity walls, windows and roofs. However, a reasonable estimate can usually be obtained by taking the total building volume and simply allowing 30 - 40 W/m³ of space up to 3 000 m³, and 15 - 30 W/m³ if above 3 000 m³. This will give a reasonable estimate of the heating load when the outside temperature is around a design condition of -1°C.

A practical way to establish steam consumption for an existing installation is to use an accurate reliable steam flowmeter.

Example 2.13.2
Determine the design rating of a heating calorifier from actual measured conditions
The design rating of a heating calorifier is unknown, but the steam load is measured at 227 kg/h when the outside temperature is 7°C and the inside temperature is 19°C, a difference of 12°C.

The calorifier is also designed to provide 19°C inside temperature when the outside temperature is -1°C, a difference of 20°C.

The steam load at the design condition can be estimated simply by the ratio of the temperature differences:

$$\text{Design steam load} = \frac{\text{Design } \Delta T}{\text{Measured } \Delta T} \times \text{measured steam load}$$

$$\text{Design steam load} = \frac{20}{12} \times 227 \text{ kg/h}$$

$$\textbf{Design steam load} = \textbf{378 kg/h}$$

Hot water storage calorifiers

Hot water storage calorifiers are designed to raise the temperature of the entire contents from cold to the storage temperature within a specified period.

The mean rate at which steam is condensed during the heat up or recovery period can be calculated using Equation 2.13.1

$$\dot{m}_s = \frac{m\, c_p\, \Delta T}{h_{fg}\, t} \qquad \textbf{Equation 2.13.1}$$

Where:
$\dot{m}_s$ = Mean rate of condensation (kg/h)
m = Mass of water heated (kg)
c_p = Specific heat of water (kJ/kg°C)
ΔT = Temperature rise (°C)
h_{fg} = Enthalpy of evaporation of steam (kJ/kg)
t = Recovery time (hours)

Example 2.13.2 Calculate the mean steam load of a storage calorifier

A storage calorifier has a capacity of 2 272 litres (2 272 kg), and is designed to raise the temperature of this water from 10°C to 60°C in ½ hour with steam at 2 bar g.

c_p for water = 4.19 kJ/kg °C

Table 2.13.2 Extract from steam tables

Pressure bar g	Saturation temperature °C	Enthalpy (energy) in kJ/kg			Specific volume of dry saturated steam m³/kg
		Water h_f	Evaporation h_{fg}	Steam h_g	
2	134	562	2163	2725	0.603

What is the mean rate at which steam is condensed?

$$\dot{m}_s = \frac{m\ c_p\ \Delta T}{h_{fg}\ t} \qquad \textbf{Equation 2.13.1}$$

Where:
m = 2 272 kg
ΔT = 60°C - 10°C = 50°C
h_{fg} = 2 163 kJ/kg
t = ½ hour

$$\dot{m}_s = \frac{2\,272\ \text{kg} \times 4.19\ \text{kJ/kg°C} \times 50°\text{C}}{2163\ \text{kJ/kg} \times 0.5\ \text{hours}}\ \text{kg/h}$$

$$\dot{m}_s = \mathbf{440\ kg/h}$$

This mean value can be used to size the control valve. However, when the temperature of water may be at its lowest value, for example 10°C, the high condensing rate of steam may be more than the fully open control valve can pass, and the coil will be starved of steam. The pressure in the coil will drop significantly, with the net effect of reducing the capacity of the steam trapping device. If the trapping device is wrongly sized or selected, condensate may back up into the coil, reducing its ability to transfer heat and achieve the required heat up time. Waterhammer may result, causing severe noise and mechanical stresses to the coil. However, if condensate is not allowed to back up into the coil the system should still maintain the correct heat up time.

The solution is to ensure proper condensate drainage. This could be achieved either by a steam trap or automatic pump-trap depending on the system needs. (Refer to Module 13.1 - Condensate Removal from Heat Exchangers).

Other shell and tube steam heaters

In other heat exchangers using steam an internal floating head may be used, which is generally more versatile than the fixed head of the U-tube exchangers. They are more suitable for use on applications with higher temperature differences between the steam and secondary fluid. As the tube bundle can be removed they can be cleaned more easily. The tube-side fluid is often directed to flow through a number of passes to increase the length of the flow path.

Exchangers are normally built with between one and sixteen tube passes, and the number of passes is selected to achieve the designed tube-side velocity. The tubes are arranged into the number of passes required by dividing up the header using a number of partition plates. Two shell passes are occasionally created by fitting a longitudinal shell-side baffle down the centre of the exchanger, where the temperature difference would be unsuitable for a single pass. Divided flow and split flow arrangements are also used where the pressure drop rather than the heat transfer rate is the controlling factor in the design, to reduce the shell-side pressure drop.

Steam may also be used to evaporate (or vaporise) a liquid, in a type of shell and tube heat exchanger known as a reboiler. These are used in the petroleum industry to vaporise a fraction of the bottom product from a distillation column. These tend to be horizontal, with vaporisation in the shell and condensation in the tubes (see Figure 2.13.5).

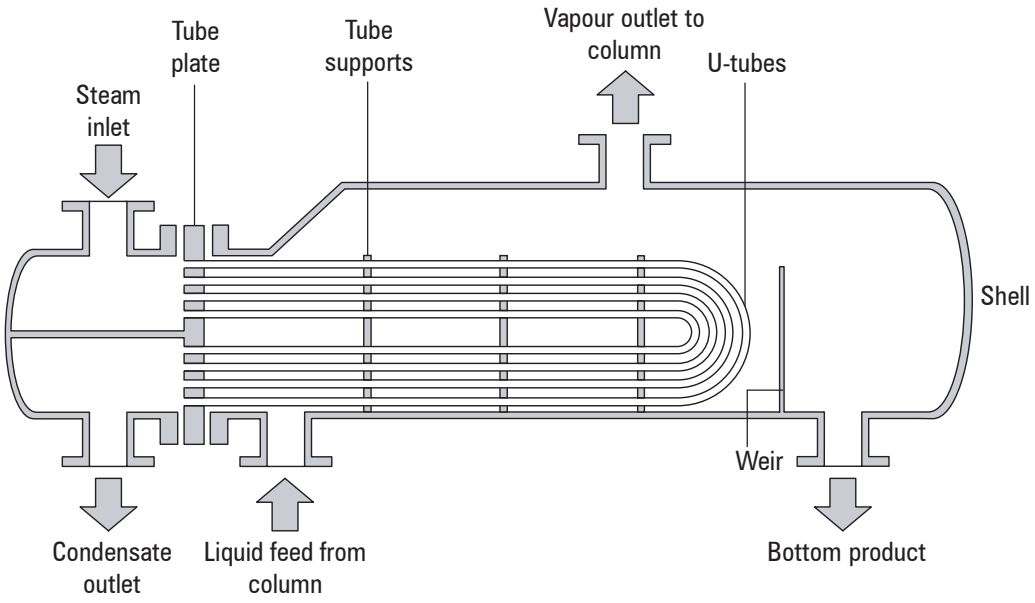


Fig. 2.13.5 A kettle reboiler

In forced circulation reboilers the secondary fluid is pumped through the exchanger, whilst in thermosyphon reboilers natural circulation is maintained by differences in density. In kettle reboilers there is no circulation of the secondary fluid, and the tubes are submerged in a pool of liquid.

Table 2.13.3 Typical heat transfer coefficients for some shell and tube heat exchangers

Secondary Fluid	U (W/m² °C)
Water	1 500 - 4 000
Organic solvents	500 - 1 000
Light oils	300 - 900
Heavy oils	60 - 450
Gases	30 - 300
Aqueous solutions (vaporising)	1 000 - 1 500
Light organics (vaporising)	900 - 1 200
Heavy organics (vaporising)	600 - 900

Although it is desirable to achieve dropwise condensation in all these applications, it is often difficult to maintain and is unpredictable. To remain practical, design calculations are generally based on the assumption of filmwise condensation.

The heat transfer area for a shell and tube heat exchanger may be estimated using Equation 2.5.3. Although these units will also normally be specified in consultation with the manufacturers, some typical overall heat transfer coefficients where steam is used as the heating medium (and which include an allowance for fouling) are provided in Table 2.13.3, as a guide.

Corrugated tube heat exchangers

One evolution in the design of the traditional shell and tube heat exchanger, is the recent development of the corrugated tube heat exchanger. This is a single passage fixed plate heat exchanger with a welded shell, and rectilinear corrugated tubes that are suitable for low viscosity fluids. In a similar manner to the plate heat exchangers, the corrugated tubes promote turbulent operating conditions that maximise heat transfer and reduce fouling. Like the traditional shell and tube heat exchangers, these units are commonly installed horizontally. However, in the corrugated tube heat exchanger the steam should always be on the shell side.

Spiral heat exchangers

Spiral heat exchangers share many similar characteristics with shell and tube and plate heat exchangers and are used on many of the same applications. They consist of fabricated metal sheets that are cold worked and welded to form a pair of concentric spiral channels, which are closed by gasketed end-plates bolted to an outer case.

Turbulence in the channels is generally high, with identical flow characteristics being obtained for both fluids. They are also relatively easy to clean and can be used for very heavy fouling fluids and slurries. The use of only a single pass for both fluids, combined with the compactness of the unit, means that pressure drops across the connections are usually quite low.

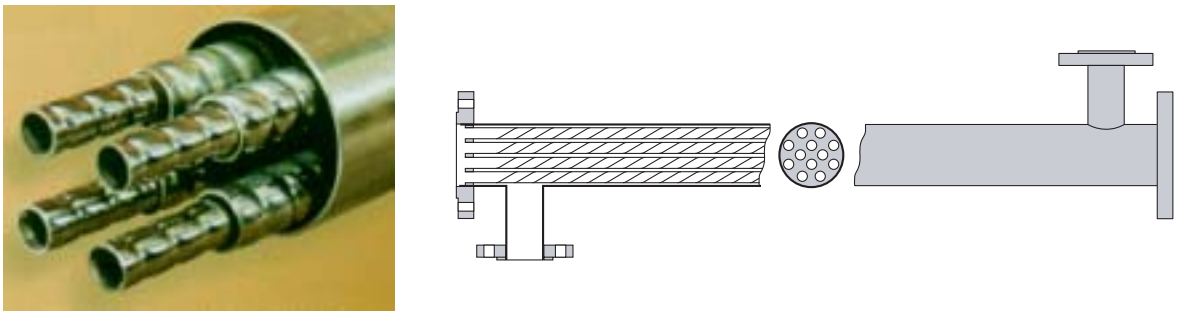


Fig. 2.13.6 Corrugated tube heat exchangers

Questions

1. The thermal output or rating of a non-storage calorifier is unknown. What would be the preferred method of calculating the steam requirement of the unit?

a| From the secondary pump duty ☐

b| From the size of the steam supply and its carrying capacity ☐

c| From the size and capacity of the secondary pipework ☐

d| By using the connected load details ☐

2. Which of the following is an advantage of a plate heat exchanger over a shell and tube heat exchanger?

a| More turbulent flow in a plate heat exchanger improves control ☐

b| In a plate heat exchanger, the rating can be easily changed ☐

c| Plate heat exchangers are less susceptible to fouling of the heat transfer surface ☐

d| In a plate heat exchanger package, sub-cooling of the condensate can be arranged ☐

3. A non-storage calorifier rated at 120 kW when charged with steam at 5 bar g, is used in a space heating system, raising water from 71°C to 82°C.

What will be the approximate design steam flowrate?

a| 193 kg/h ☐

b| 217 kg/h ☐

c| 207 kg/h ☐

d| 187 kg/h ☐

4. What is a disadvantage of using the thermal rating of a calorifier to calculate its steam consumption?

a| If the return water temperature rises above 71°C the calorifier output will rise, as will the steam consumption ☐

b| Actual connected heat load may be different to the calorifier published thermal rating ☐

c| If the return water temperature drops below 71°C the calorifier output will drop below the published figure ☐

d| The published rating would only be true of a new calorifier ☐

5. A 1 000 litre capacity storage calorifier is required to raise water from 10 °C to 60 °C in 30 minutes using steam at 4 bar g. What will be its steam flowrate?

a| 116 kg/h ☐

b| 184 kg/h ☐

c| 198 kg/h ☐

d| 212 kg/h ☐

6. Which of the following is a limitation of a gasketted plate heat exchanger compared with a brazed or welded exchanger?

a| They cannot handle thermal cycling ☐

b| They are difficult to clean ☐

c| The operating temperature of the gasket ☐

d| They are susceptible to fatigue ☐

Answers

1: d, 2: d, 3: c, 4: b, 5: c, 6: c

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Module 2.14

Steam Consumption of Plant Items

Steam Consumption of Plant Items

The examples in the following sections within this Module are a revision of previously mentioned equipment, and indicate the steam consumption of other common plant items.

Heater batteries

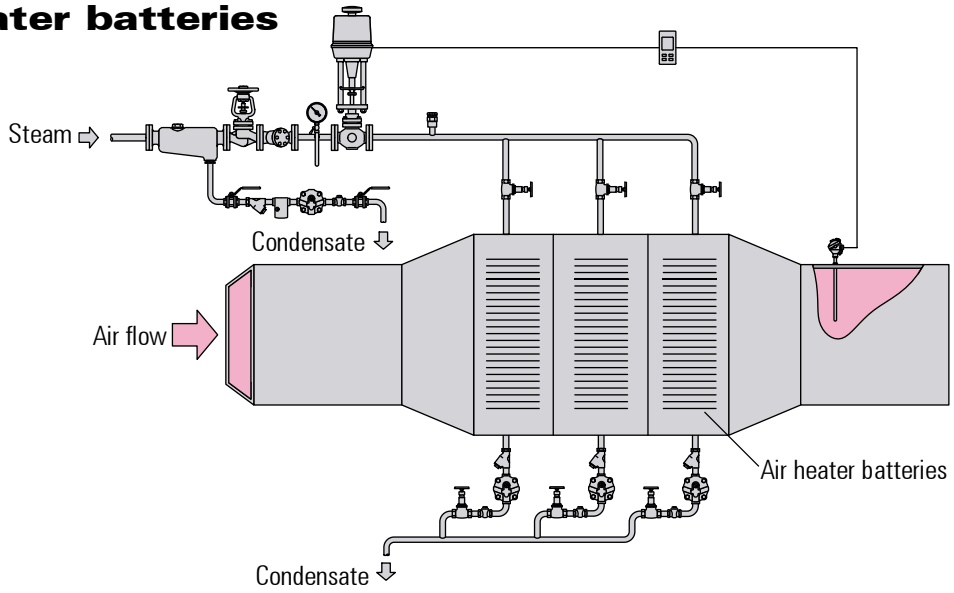


Fig. 2.14.1 Typical air heater battery installation

Most manufacturers of unit heaters and air heater batteries give the output of their equipment in kW. The condensing rate may be determined from this by dividing the equipment rating (in kW) by the enthalpy of evaporation of the steam at the operating pressure (in kJ/kg) to give a steam flowrate in kg/s. Multiplying the result by 3 600 will give kg/h.

$$\text{Steam flowrate } (\dot{m}_s) = \frac{\text{Load in kW}}{h_{fg} \text{ at operating pressure}} \text{ kg/s} \qquad \textbf{Equation 2.8.1}$$

Thus a unit heater rated at 44 kW when supplied with steam at 3.5 bar g (h_{fg} = 2 210 kJ/kg) will condense:

$$\dot{m}_s = \frac{44}{2\,120}$$

$$\dot{m}_s = 0.02 \text{ kg/s}$$

$$\mathbf{\dot{m}_s = 75 \text{ kg/h}}$$

If the manufacturer's figures are not available, but the following is known:

- The volumetric air flowrate.
- The temperature rise.
- The steam pressure.

Then the condensing rate can be determined by using Equation 2.12.3:

$$\dot{m}_s = \frac{3\,600\,\dot{V}\,\Delta T\,c_p}{h_{fg}} \text{ kg/h} \qquad \textbf{Equation 2.12.3}$$

Where:
$\dot{m}_s$ = Condensing rate (kg/h)
$\dot{V}$ = Volumetric air flowrate (m³/hour)
ΔT = Temperature rise (°C)
c_p = Specific heat of air at constant pressure (kJ/m³ °C)
h_{fg} = Enthalpy of evaporation at operating steam pressure (kJ/kg)
Note: The factor 3 600 gives the answer in kg/h

Without more formal data, the following figures may be used as an approximation:

- Density of air ≈ 1.3 kg/m³
- Specific heat of air c_p (by volume) ≈ 1.3 kJ/m³ °C
- Specific heat of air c_p (by mass) ≈ 1.0 kJ/kg °C

Example 2.14.1
An air heater designed to raise air temperature from -5 to 30°C is fitted in a duct 2 m x 2 m. The air velocity in the duct is 3 m/s, steam is supplied to the heater battery at 3 bar g, and the specific heat of air is taken as 1.3 kJ/m³°C.

Determine the steam condensing rate ($\dot{m}_s$):

$$\text{Volumetric air flowrate } \dot{V} = \text{Velocity x area}$$

$$\dot{V} = 3 \text{ m/s} \times 2 \text{ m} \times 2 \text{ m}$$

$$\mathbf{\dot{V} = 12 \text{ m}^3/\text{s}}$$

$$\text{Temperature rise } \Delta T = -5 + 30$$

$$\mathbf{\Delta T = 35°C}$$

$$\mathbf{c_p = 1.3 \text{ kJ/m}^3\,°C}$$

$$\mathbf{h_{fg} \text{ at } 3 \text{ bar g} = 2\,133 \text{ kJ/kg}}$$

$$\dot{m}_s = \frac{3\,600 \times 12 \text{ m}^3/\text{s} \times 35°C \times 1.3 \text{ kJ/m}^3\,°C}{2\,133 \text{ kJ/kg}}$$

$$\mathbf{\dot{m}_s = 2\,073 \text{ kg/h}}$$

Heating calorifiers

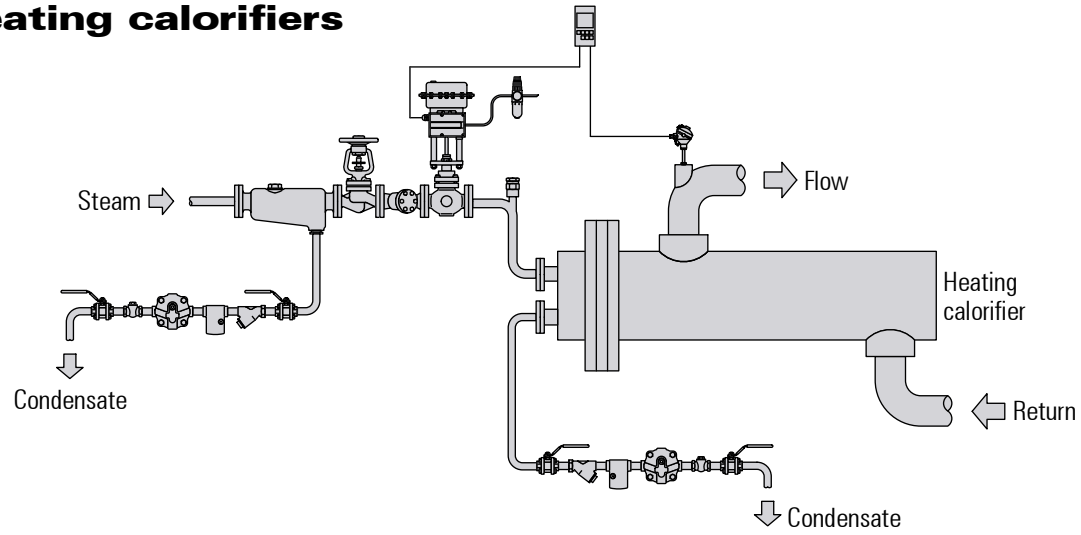


Fig. 2.14.2 Typical heating calorifier installation

As with air heaters, most heating calorifier manufacturers will usually provide a rating for their equipment, and the steam consumption may be determined by dividing the kW rating by the enthalpy of steam at the operating pressure to produce a result in kg/s (see Equation 2.8.1). However, calorifiers are frequently too large for the systems they serve because:

- The initial heat load calculations on the building they serve will have included numerous and over-cautious safety factors.
- The calorifier itself will have been selected from a standard range, so the first size up from the calculated load will have been selected.
- The calorifier manufacturer will have included his own safety factor on the equipment.

An estimate of the actual load at any point in time may be obtained if the flow and return temperatures and the pumping rate are known. Note however that the pressure head on the discharge side affects the throughput of the pump, and this may or may not be constant.

Example 2.14.2
4 l/s of low temperature hot water (flow/return = 82/71°C) is pumped around a heating system.

Determine the heat output:

Heat output = Water flowrate x specific heat of water x temperature change

Heat output = 4 l/s x 4.19 kJ/kg °C x (82 - 71°C)

Heat output = 184 kW

An alternative method of estimating the load on a heating calorifier is to consider the building being heated. The calculations of heat load can be complicated by factors including:

- Air changes.
- Heat transfer rates through walls, windows and roofs.

However, a reasonable estimate may be obtained by taking the volume of the building and allowing a heating capacity of 30 W/m³. This will give the running load for an inside temperature of about 20°C when the outside temperature is about -1°C.

Typical flow and return temperatures for:

- Low temperature hot water (LTHW) systems are 82°C and 71°C (ΔT = 11°C).
- Medium temperature hot water (MTHW) systems are 94°C and 72°C (ΔT = 22°C).

Figures for high temperature hot water (HTHW) systems vary considerably, and must be checked for each individual application.

Example 2.14.3
The steam flow to a heating calorifier has been measured as 227 kg/h when the outside temperature is 7°C and the inside temperature is 18°C.

If the outside temperature falls to -1°C, and the inside temperature is 19°C, determine the approximate steam flowrate. This can be calculated by proportionality.

Temperature difference at initial condition = 18 - 7 = 11°C

Temperature difference at second condition = 19 - (-1) = 20°C

$$\text{Approximate steam flowrate} = \frac{20}{11} \times 227$$

Approximate steam flowrate = 413 kg/h

Hot water storage calorifiers

Hot water storage calorifiers are designed to raise the temperature of their entire contents from cold to storage temperature within a specified time period.

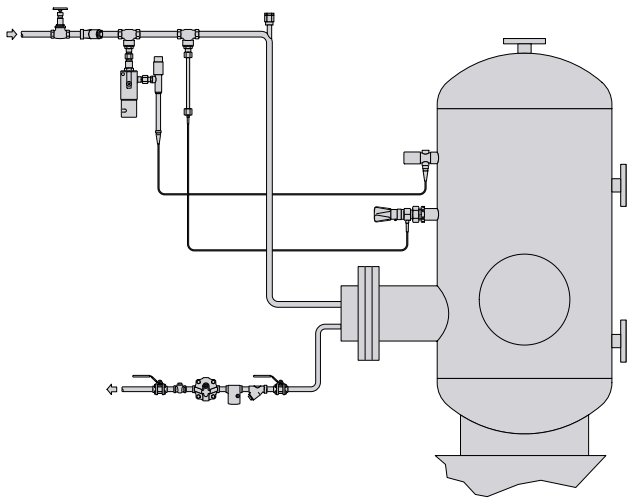


Fig. 2.14.3 Typical hot water storage calorifier installation

Typical UK values are:

- Cold water temperature 10°C
- Hot water temperature 60°C

Heat up time (also referred to as 'recovery time') = 1 hour.

The mass of water to be heated may be determined from the volume of the vessel.

(For water, density ρ = 1 000 kg/m³, and specific heat (c_p) = 4.19 kJ/kg °C).

Example 2.14.4
A storage calorifier comprises of a cylindrical vessel, 1.5 m diameter and 2 m high. The contents of the vessel are to be heated to 60°C in 1 hour.

The incoming water temperature is 10°C, and the steam pressure is 7 bar g.

Determine the steam flowrate:

$$\text{Volume of vessel} = \frac{\pi \times D^2}{4} \times \text{height} = \frac{\pi \times 1.5^2}{4} \times 2 = \mathbf{3.53\ m^3}$$

$$\text{Mass of water} = 3.53\ m^3 \times 1000\ kg/m^3 = \mathbf{3\,530\ kg}$$

$$\text{Temperature rise} = 60 - 10 = \mathbf{50°C}$$

$$\text{Energy required} = \frac{3\,530\ kg \times 4.19\ kJ/kg°C \times 50°C}{1\ hour} = \mathbf{739\,535\ kJ/h}$$

Enthalpy of evaporation of steam at 7 bar g = 2 048 kJ/kg (from steam tables).

$$\text{Steam consumption rate} = \frac{739\,535\ kg/h}{2\,048\ kJ/kg}$$

$$\textbf{Steam consumption rate} = \mathbf{361\ kg/h}$$

Drying cylinders

Drying cylinders vary significantly in layout and application and, consequently, in steam consumption.

Apart from wide variations in size, steam pressure, and running speed, cylinders may be drained through the frame of the machines, as in textile can dryers, or by means of a blow-through system in the case of high speed paper machines. Conversely, film dryers and slow speed paper machines may use individual steam traps on each cylinder. Demand will vary from small standing losses from a cylinder drying sized cotton thread, to the heavy loads at the wet end of a paper machine or in a film dryer.

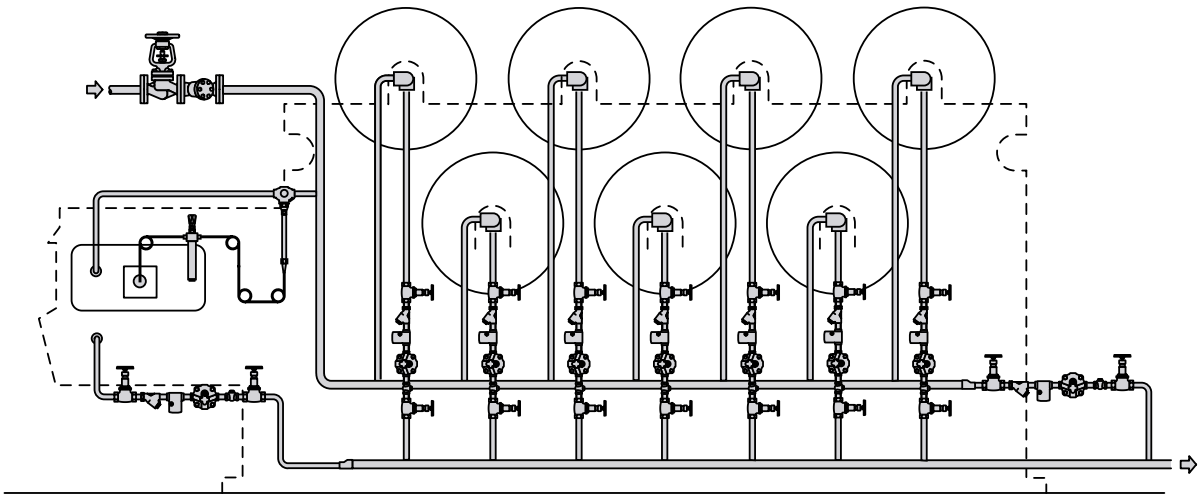


Fig. 2.14.4 Drying cylinders

Because of this, accurate figures can only be obtained by measurement. However, certain trusted formulae are in use, which enable steam consumption to be estimated within reasonable limits.

In the case of textile cylinder drying machines, counting the number of cylinders and measuring the circumference and width of each will lead to the total heating surface area. The two ends of each cylinder should be included and 0.75 m² per cylinder should be added to cover doll heads and frames except where individual trapping is used. The radiation loss from the machine, while standing, measured in kg of steam per hour, can be estimated by multiplying the total area by a factor of 2.44. The running load in kg per hour will be obtained by using a factor of 8.3. (In imperial units the area will be measured in square feet and the corresponding factors will be 0.5 and 1.7 respectively). This is based on a machine drying piece goods at a rate of 64 to 73 metres per minute, (70 to 80 yards per minute), but by making allowances, it can be used for machines working under different conditions.

Where the amount of moisture to be removed is known, steam consumption can be calculated using the empirical Equation 2.14.1, assuming that the wet and dry weights of the material being handled are known.

$$\dot{m}_s = \frac{1.5\left([W_w - W_d]\,2550 + 1.26\,W_d\,[T_2 - T_1]\right)}{h_{fg}} \qquad \textbf{Equation 2.14.1}$$

Where:
$\dot{m}_s$ = Mass flowrate of steam (kg/h)
W_w = Throughput of wet material (kg/h)
W_d = Throughput of dry material (kg/h)
T_2 = Temperature of material leaving the machine (°C)
T_1 = Temperature of material entering the machine (°C)
h_{fg} = Enthalpy of evaporation of steam in cylinders (kJ/kg)

The factors in the equation above are empirically derived constants:

1.5 = Factor applied to cylinder dryers.

2550 = Average water enthalpy + enthalpy of evaporation required to evaporate moisture.

1.26 = Average specific heat of material.

Drying cylinders tend to have a heavy start-up load due to the huge volume of the steam space and the mass of metal to be heated, and a factor of three times the running load should be allowed in sizing steam traps. It must also be remembered that air can cause particular difficulties, such as prolonged warming up times and uneven surface temperature. Special provision must therefore be made for venting air from the cylinders.

Presses

Presses, like drying cylinders, come in all shapes, sizes and working pressures, and are used for many purposes, such as moulding plastic powders, preparing laminates, producing car tyres (see Figure 2.14.4), and manufacturing plywood. They sometimes also incorporate a cooling cycle.

Clearly, it would be difficult to calculate steam loads with any accuracy and the only way of getting credible results is by measurement.

This type of equipment may be 'open', allowing a radiation loss to atmosphere, or 'closed', when the two heating surfaces are in effect insulated from each other by the product. Although some heat is absorbed by the product, the net result is that the steam consumption is much the same whether the plant is working or standing idle, although fluctuations will occur during opening and closing.

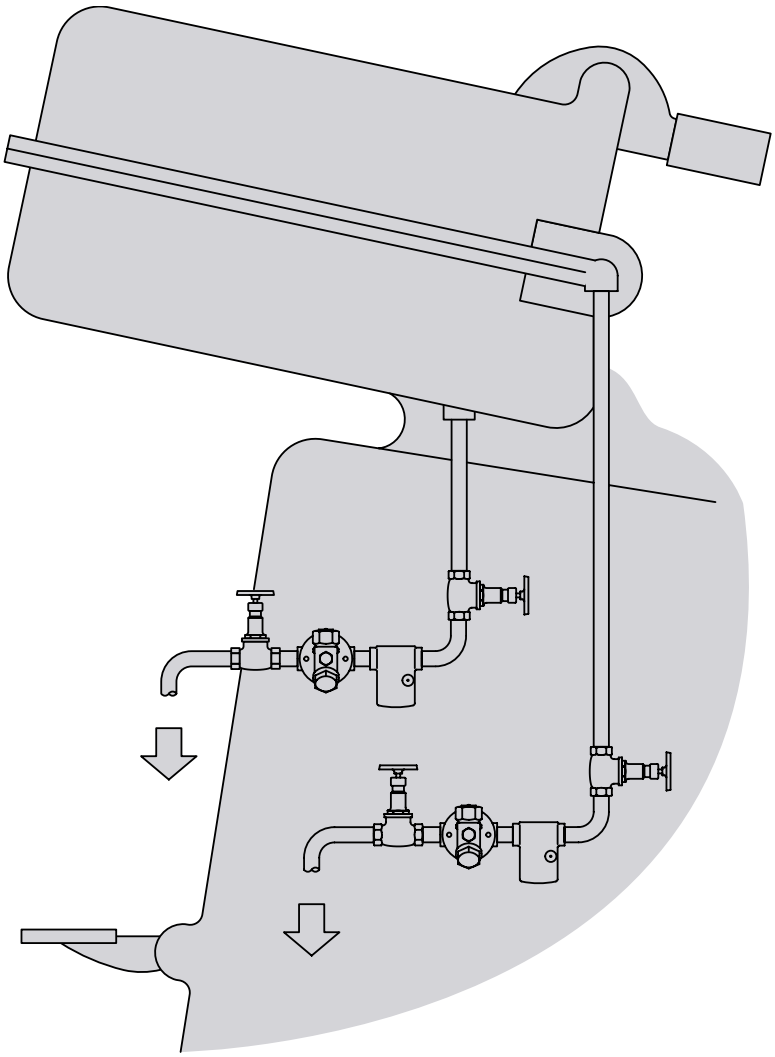


Fig. 2.14.5 Tyre press

Steam consumption can sometimes be estimated using the basic heat transfer Equation 2.5.3:

$$\dot{Q} = U\,A\,\Delta T \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
U = Overall heat transfer coefficient (W/m² K or W/m² °C)
A = Heat transfer area (m²)
ΔT = Temperature difference between the steam and the product (K or °C)

The U values shown in Figure 2.9.1 may sometimes be used. They can give reasonable results in the case of large platen presses but are less accurate when small numbers of intricately shaped moulds are considered, mainly due to the difficulty of estimating the surface area.

A feature of this type of plant is the small steam space, and a relatively high steam load when warming up from cold. To account for this and the load fluctuations, steam traps should be sized with a factor of 2 times the running load. Temperature control can be very accurate using pilot operated direct acting reducing valves, giving a constant and consistent steam pressure corresponding to the required surface temperature. These are sized simply on the designed steam load.

Tracer lines

Pipelines carrying viscous fluids are frequently maintained at an elevated temperature by means of steam tracers. These usually consist of one or more small bore steam lines running alongside the product line, the whole being covered in insulation.

In theory, the exact calculation of steam consumption is difficult, as it depends on:

- The degree of contact between the two lines, and whether heat conducting pastes are used.
- The temperature of the product.
- The length, temperature and pressure drop along the tracer lines.
- The ambient temperature.
- Wind speed.
- The emissivity of the cladding.

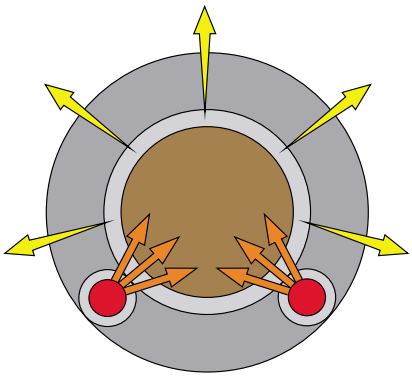


Fig. 2.14.6 A steam tracer

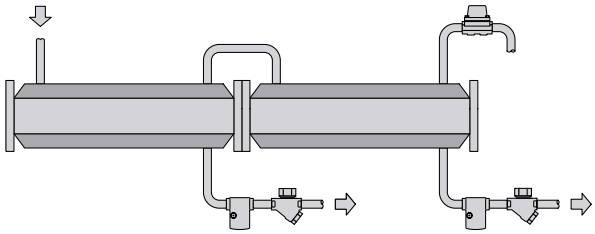


Fig. 2.14.7 Jacketed pipeline

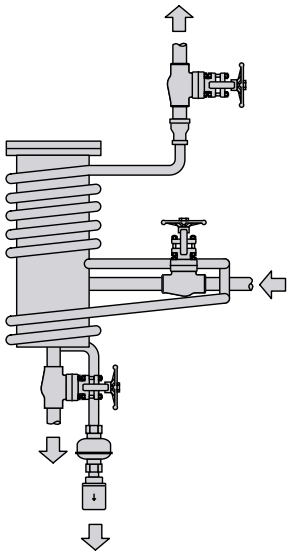


Fig. 2.14.8 Heated sampling point

In practice, it is usually safe to assume that the tracer line simply replaces radiation losses from the product line itself. On this basis, the steam consumption of the tracer line may be taken as a running load being equal to the radiation loss from the product lines.

Table 2.14.1 provides heat losses from insulated pipes with either 50 or 100 mm of insulation.

Table 2.14.1 Typical heat losses from insulated pipes (W/m) with wind speed of 10 m/s (36 km/h)

Pipe diameter (mm)	Insulation thickness	Product/ambient temperature difference (°C) 25	75	100	125	150	175	200
100	50	14	43	58	71	86	100	115
	100	9	26	36	45	54	62	71
150	50	20	59	77	97	116	136	155
	100	12	35	46	58	69	81	92
200	50	24	72	97	120	144	168	192
	100	14	41	55	70	84	98	112
250	50	29	87	116	145	174	202	231
	100	16	49	66	82	99	115	131
300	50	33	101	135	168	201	235	268
	100	18	56	75	94	113	131	151
400	50	41	123	164	206	246	288	329
	100	23	68	91	113	136	158	181
500	50	51	151	201	252	301	352	403
	100	28	82	109	136	163	191	217

Once the heat loss has been determined, steam consumption can be calculated using Equation 2.12.4:

$$\dot{m}_s = \frac{3.6\ \dot{Q}\ L}{h_{fg}} \qquad \textbf{Equation 2.12.4}$$

Where:
$\dot{m}_s$ = Steam demand (kg/h)
$\dot{Q}$ = Heat loss from Table 2.14.1 (W/m)
L = Length of traced product line (m)
h_{fg} = Enthalpy of evaporation at operating pressure (kJ/kg)
Note: The factor 3.6 gives the answer in kg/h

Example 2.14.5
A 50 m long x 200 mm pipe contains a liquid product at 120°C. The ambient temperature is 20°C, the pipe has 50 mm of insulation, and steam is supplied at 7 bar g to the tracer(s). Determine the steam consumption:

Pipe length (L) = 50 m

Temperature difference between product and ambient = 120°C - 20°C = 100°C

Heat loss per metre from the pipe ($\dot{Q}$) = 97 W/m (from Figure 2.14.1)

h_{fg} of steam at 7 bar g = 2048 kJ/kg (steam tables)

Equation 2.14.3

$$\dot{m}_s = \frac{3.6\ \dot{Q}\ L}{h_{fg}}$$

$$\dot{m}_s = \frac{3.6 \text{ x } 97 \text{ W/m x } 50 \text{ m}}{2\,048 \text{ kJ/kg}}$$

$$\dot{m}_s = \mathbf{8.53\ kg/h}$$

For jacketed lines, the heat loss may be assumed to be the same as that from a steam main which has a diameter equal to that of the jacket; also taking any insulation into account.

When sizing the steam traps, a factor of 2 times the running load should be used to cover start-up conditions, but any temperature control valve can be sized to handle the design load only.

Sizing the tracer line

Example 2.14.5 calculates the steam tracer load on the basis of the heat loss from the pipe.

In practice, the tracer line will not be exactly sized to match this heat loss. Table 2.14.2 shows the useful heat output from 15 mm and 20 mm steel and copper tracer lines operating at different pressures alongside product lines at different temperatures. The Table accounts for heat losses from the tracer lines to the surrounding air through the insulation.

Table 2.14.2 Useful heat outputs from steel and copper tracer lines

		Steel (NB)								Copper (OD)							
Steam pressure		3 bar g		5 bar g		7 bar g		9 bar g		3 bar g		5 bar g		7 bar g		9 bar g	
Tracer dia. (mm)		15	20	15	20	15	20	15	20	15	20	15	20	15	20	15	20
Product temperature	10°C	113	145	125	161	135	174	143	184	80	197	89	119	96	129	102	135
	25°C	16	20	29	37	38	49	46	59	11	20	29	37	38	49	46	59
	50°C	79	101	92	118	101	130	109	141	56	75	65	87	72	97	78	104
	75°C	58	74	71	91	80	103	88	114	41	55	50	67	57	77	63	84
	100°C	37	47	50	64	59	76	67	86	26	35	35	47	42	56	48	64
	125°C	16	20	29	37	38	49	46	59	11	20	29	37	38	49	46	59
	150°C	-	-	8	10	17	22	25	32	-	-	5	7	12	16	18	24

In Example 2.14.5, the heat loss from the pipe was 97 W/m. The tracer line has to be able to supply at least this rate of heat transfer.

Table 2.14.2 shows that, by interpolation, the useful heat output from a 15 mm steel tracer line is 33 W/m for a product temperature of 120°C and a steam pressure of 5 bar g.

The number of tracers required to maintain the product temperature of 120°C are therefore:

$$\text{Number of tracer lines} = \frac{\text{Rate of heat loss from process line}}{\text{Heat output from tracer line}}$$

$$= \frac{97 \text{ W/m}}{33 \text{ W/m}}$$

$$= 2.94 \text{ tracer lines}$$

Therefore three 15mm steel tracer lines will be required for this application as shown in Figure 2.14.9.

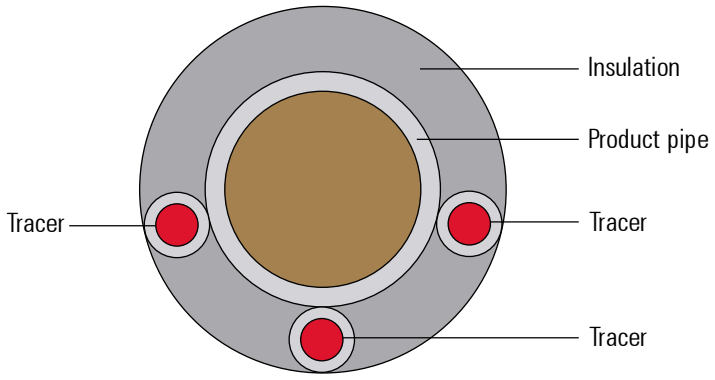


Fig. 2.14.9 Three 15 mm tracer lines fitted to a 200 mm process pipe

Questions

1. A 30 kW air heater unit, 700 mm x 700 mm, is supplied with steam at 3 bar g. If the air velocity is 2.5 m/s, and the incoming air temperature is 10°C, what is the temperature of the air leaving the heater unit (Specific heat of air ≈ 1.3 kJ/m³°C)?

a| 25.7°C ☐

b| 23.9°C ☐

c| 28.8°C ☐

d| 35.6°C ☐

2. A building has an internal volume of approximately 5 000 m³. Determine the approximate heating load if the outside temperature is -5°C.

a| 175.5 kW ☐

b| 178.6 kW ☐

c| 180.4 kW ☐

d| 150.0 kW ☐

3. What is the approximate heat loss from a 100 mm bore carrying oil pipe surrounded by a 150 mm bore steam jacket? Steam to the jacket is 4 bar g. The oil being carried is at 65°C. Assume an ambient temperature of 20°C and still air conditions. The lagging around the jacket is 50 mm thick.

a| 60 W/m ☐

b| 97 W/m ☐

c| 120 W/m ☐

d| 112 W/m ☐

4. A drying cylinder is designed for material entering the machine at 30°C. The temperature is now 22°C. What will be the effect on the steam consumption of the cylinder?

a| None ☐

b| The steam flowrate will be higher ☐

c| The steam flowrate will be lower ☐

d| The steam flowrate will be lower and the machine speed must be reduced ☐

5. A textile drying cylinder operates at 70 m/minute and is supplied with steam at 4 bar. The cylinder is 1.5 m diameter and 3 m long. What will be the approximate running load steam consumption of the cylinder?

a| 147 kg/h ☐

b| 153 kg/h ☐

c| 43 kg/h ☐

d| 87 kg/h ☐

6. Temperature control of a drying cylinder is best achieved by:

a| A pneumatic control sensing the temperature of the condensate leaving the cylinder ☐

b| Supplying steam at a pressure corresponding to the required temperature ☐

c| Manual control of the steam supply to give the required degree of drying ☐

d| A temperature control on the steam supply coupled to a sensor strapped to the cylinder surface ☐

Answers

1: c, 2: b, 3: d, 4: b, 5: b, 6: b

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Module 2.15

Entropy - A Basic Understanding

Entropy - A Basic Understanding

What is entropy?

In some ways, it is easier to say what it is not! It is not a physical property of steam like pressure or temperature or mass. A sensor cannot detect it, and it does not show on a gauge. Rather, it must be calculated from things that can be measured. Entropy values can then be listed and used in calculations; in particular, calculations to do with steam flow, and the production of power using turbines or reciprocating engines.

It is, in some ways, a measure of the lack of quality or availability of energy, and of how energy tends always to spread out from a high temperature source to a wider area at a lower temperature level. This compulsion to spread out has led some observers to label entropy as 'time's arrow'. If the entropy of a system is calculated at two different conditions, then the condition at which the entropy is greater occurs at a later time. The increase of entropy in the overall system always takes place in the same direction as time flows.

That may be of some philosophical interest, but does not help very much in the calculation of actual values. A more practical approach is to define entropy as energy added to or removed from a system, divided by the mean absolute temperature over which the change takes place.

To see how this works, perhaps it is best to start off with a diagram showing how the enthalpy content of a kilogram of water increases as it is heated to different pressures and evaporated into steam.

Since the temperature and pressure at which water boils are in a fixed relationship to each other, Figure 2.15.1 could equally be drawn to show enthalpy against temperature, and then turned so that temperature became the vertical ordinates against a base of enthalpy, as in Figure 2.15.2.

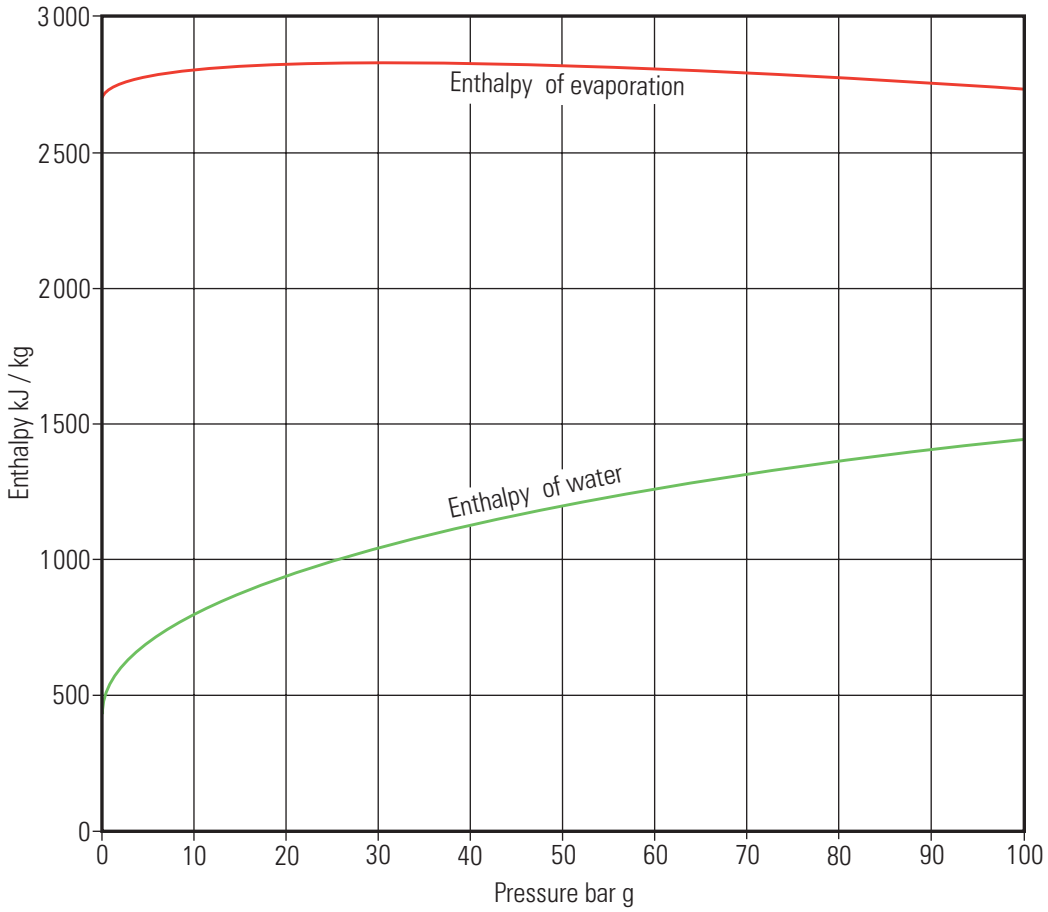


Fig. 2.15.1 The enthalpy/pressure diagram

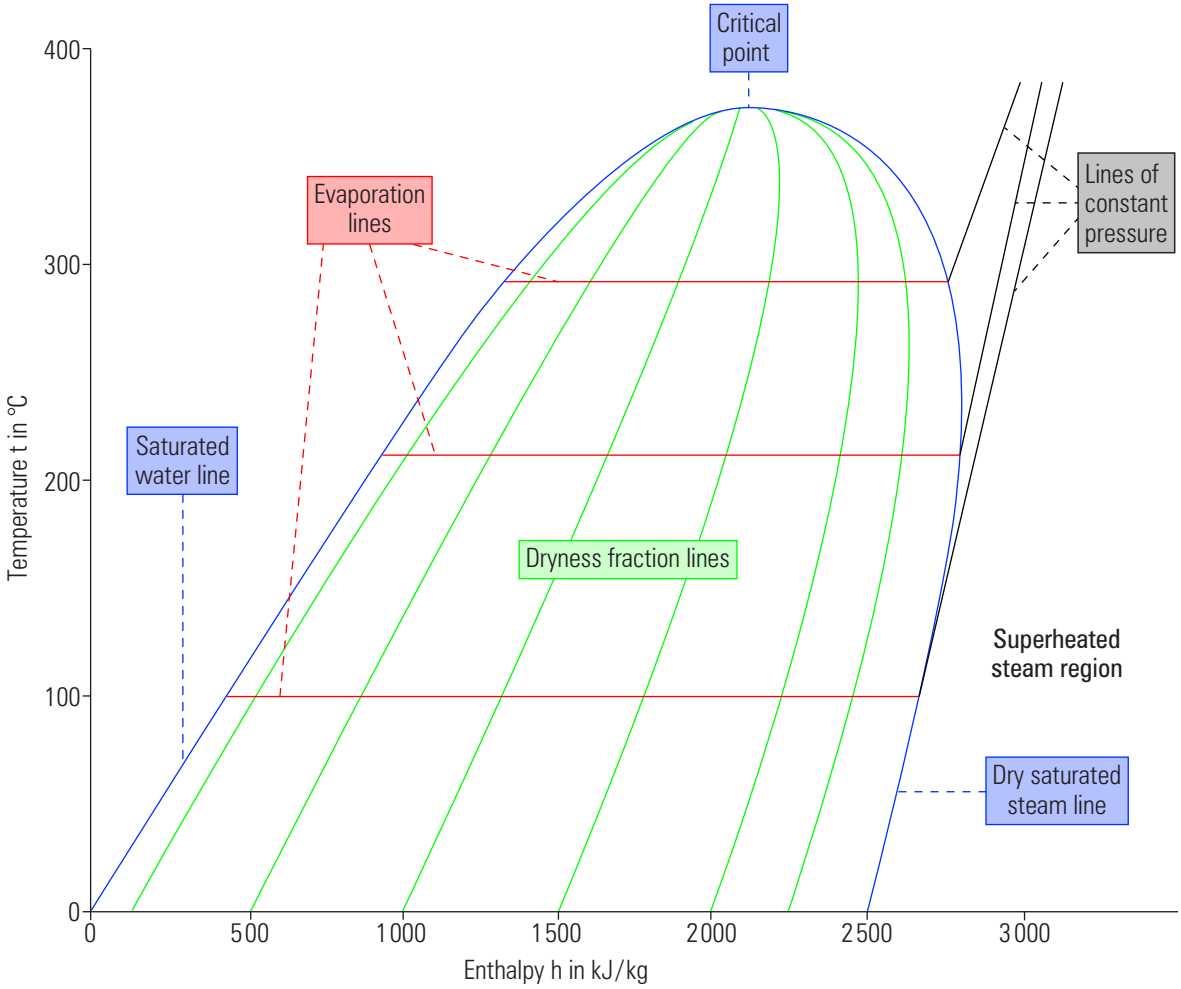


Fig. 2.15.2 The temperature/enthalpy diagram

Lines of constant pressure originate on the saturated water line. The horizontal distance between the saturated water line and the dry saturated steam line represents the amount of latent heat or enthalpy of evaporation, and is called the evaporation line; (enthalpy of evaporation decreases with rising pressure). The area to the right of the dry saturated steam line is the superheated steam region, and lines of constant pressure now curve upwards as soon as they cross the dry saturated steam line.

A variation of the diagram in Figure 2.15.2, that can be extremely useful, is one in which the horizontal axis is not enthalpy but instead is enthalpy divided by the mean temperature at which the enthalpy is added or removed. To produce such a diagram, the entropy values can be calculated. By starting at the origin of the graph at a temperature of 0°C at atmospheric pressure, and by adding enthalpy in small amounts, the graph can be built. As entropy is measured in terms of absolute temperature, the origin temperature of 0°C is taken as 273.15 K. The specific heat of saturated water at this temperature is 4.228 kJ/kg K. For the purpose of constructing the diagram in Figure 2.15.3 the base temperature is taken as 273 K not 273.15 K.

By assuming a kilogram of water at atmospheric pressure, and by adding 4.228 kJ of energy, the water temperature would rise by 1 K from 273 K to 274 K. The mean temperature during this operation is 273.5 K, see Figure 2.15.3.

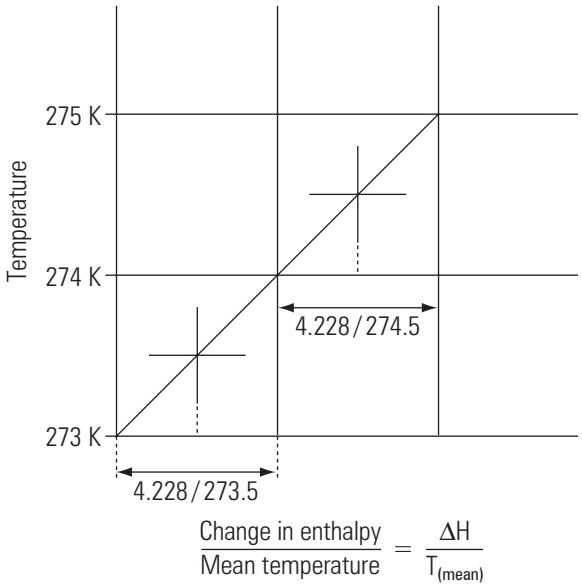


Fig. 2.15.3 The cumulative addition of 4.228 kJ of energy to water from 0°C

The width of the element representing the added enthalpy $= \frac{4.228}{273.5} = 0.015459$ kJ/kg K.

This value represents the change in enthalpy per degree of temperature rise for one kilogram of water and is termed the change in specific entropy. The metric units for specific entropy are therefore kJ/kg K.

This process can be continued by adding another 4.228 kJ of energy to produce a series of these points on a state point line. In the next increment, the temperature would rise from 274 K to 275 K, and the mean temperature is 274.5 K.

The width of this element representing the added enthalpy $= \frac{4.228}{274.5} = 0.015403$ kJ/kg K.

It can be seen from these simple calculations that, as the temperature increases, the change in entropy for each equal increment of enthalpy reduces slightly. If this incremental process were continuously repeated by adding more heat, it would be noticed that the change in entropy would continue to decrease. This is due to each additional increment of heat raising the temperature and so reducing the width of the elemental strip representing it. As more heat is added, so the state point line, in this case the saturated water line, curves gently upwards.

At 373.14 K (99.99°C), the boiling point of water is reached at atmospheric pressure, and further additions of heat begin to boil off some of the water at this constant temperature. At this position, the state point starts to move horizontally across the diagram to the right, and is represented on Figure 2.15.4 by the horizontal evaporation line stretching from the saturated water line to the dry saturated steam line. Because this is an evaporation process, this added heat is referred to as enthalpy of evaporation,

At atmospheric pressure, steam tables state that the amount of heat added to evaporate 1 kg of water into steam is 2256.71 kJ. As this takes place at a constant temperature of 373.14 K, the mean temperature of the evaporation line is also 373.14 K.

The change in specific entropy from the water saturation line to the steam saturation line is therefore:

$$\frac{2256.71}{373.14} = 6.047891 \text{ kJ/kg K}$$

The diagram produced showing temperature against entropy would look something like that in Figure 2.15.4, where:

- 1 is the saturated water line.
- 2 is the dry saturated steam line.
- 3 are constant dryness fraction lines in the wet steam region.
- 4 are constant pressure lines in the superheat region.

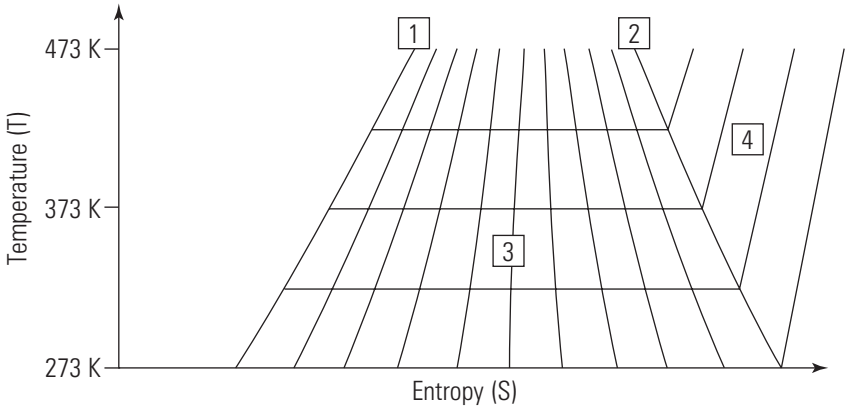


Fig. 2.15.4 The temperature - entropy diagram

What use is the temperature - entropy diagram (or T - S diagram)?

One potential use of the T - S diagram is to follow changes in the steam condition during processes occurring with no change in entropy between the initial and final state of the process. Such processes are termed Isentropic (constant entropy).

Unfortunately, the constant total heat lines shown in a T - S diagram are curved, which makes it difficult to follow changes in such free and unrestricted expansions as those when steam is allowed to flow through and expand after a control valve. In the case of a control valve, where the velocities in the connecting upstream and downstream pipes are near enough the same, the overall process occurs with constant enthalpy (isenthalpic). In the case of a nozzle, where the final velocity remains high, the overall process occurs with constant entropy.

To follow these different types of processes, a new diagram can be drawn complete with pressures and temperatures, showing entropy on the horizontal axis, and enthalpy on the vertical axis, and is called an enthalpy - entropy diagram, or H - S diagram, Figure 2.15.5.

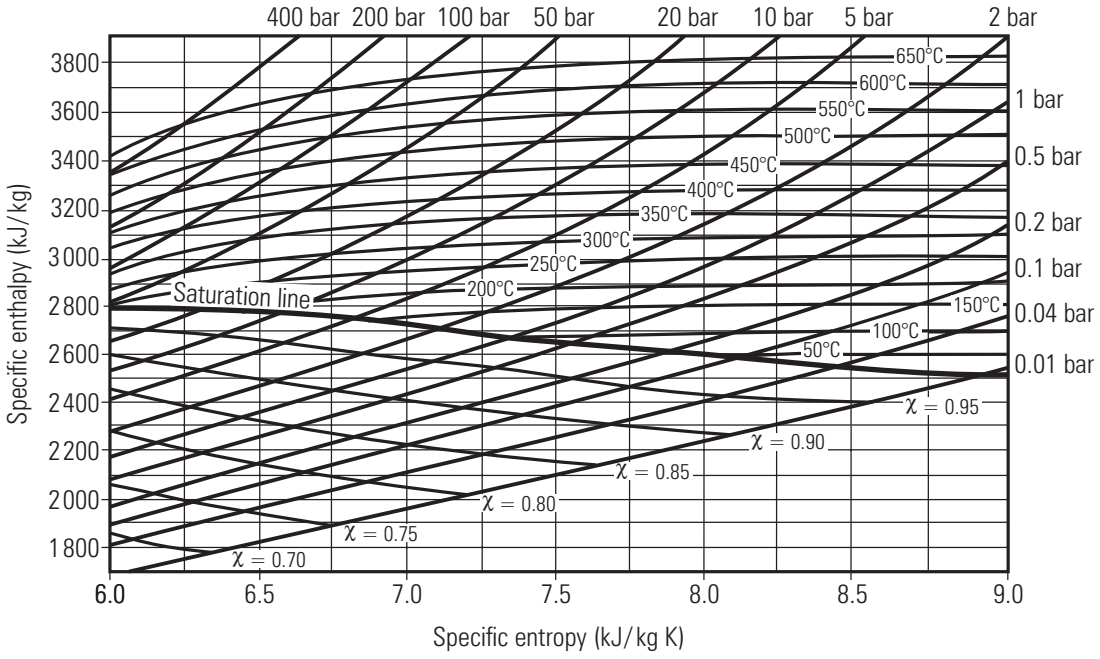


Fig. 2.15.5 The H - S diagram

The H - S diagram is also called the Mollier diagram or Mollier chart, named after Dr. Richard Mollier of Dresden who first devised the idea of such a diagram in 1904.

Now, the isenthalpic expansion of steam through a control valve is simply represented by a straight horizontal line from the initial state to the final lower pressure to the right of the graph, see Figure 2.15.6; and the isentropic expansion of steam through a nozzle is simply a line from the initial state falling vertically to the lower final pressure, see Figure 2.15.7.

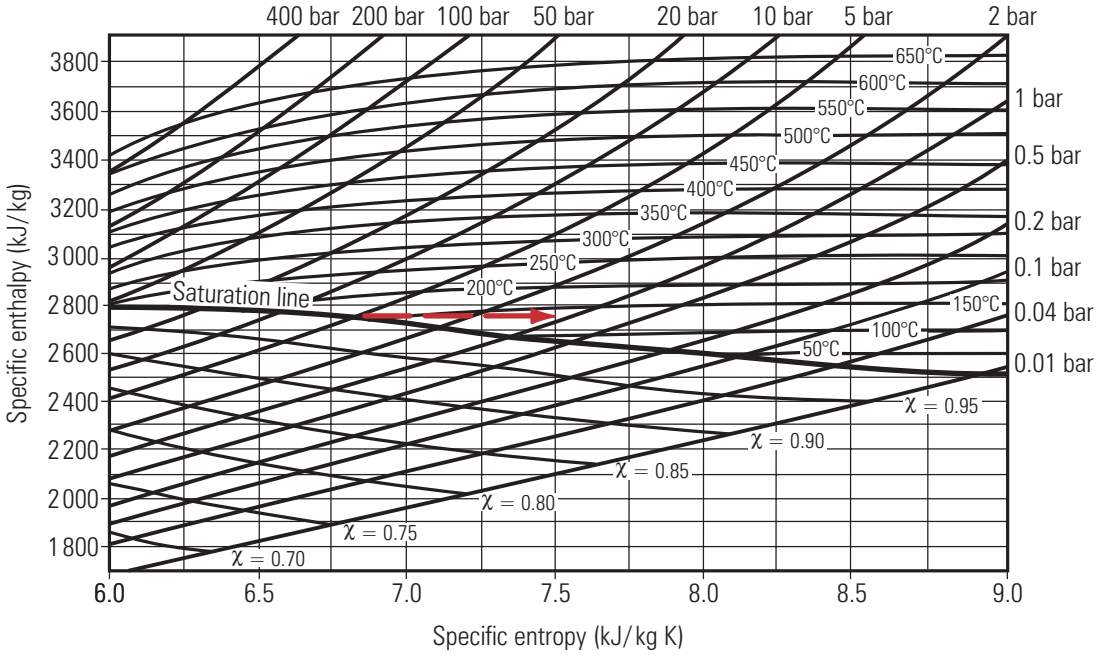


Fig. 2.15.6 Isenthalpic expansion, as through a control valve

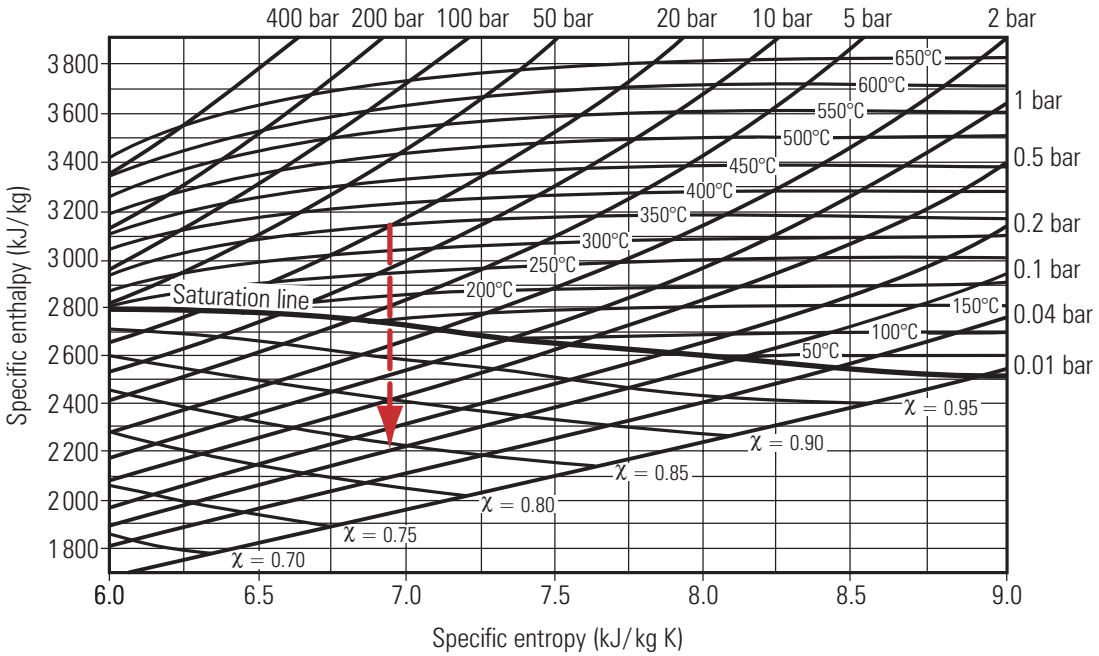


Fig. 2.15.7 Isentropic expansion, as through a nozzle

An isentropic expansion of steam is always accompanied by a decrease in enthalpy, and this is referred to as the 'heat drop' (H) between the initial and final condition. The H values can be simply read at the initial and final points on the Mollier chart, and the difference gives the heat drop. The accuracy of the chart is sufficient for most practical purposes.

As a point of interest, as the expansion through a control valve orifice is an isenthalpic process, it is assumed that the state point moves directly to the right; as depicted in Figure 2.15.6. In fact, it does not do so directly. For the steam to squeeze through the narrow restriction it has to accelerate to a higher speed. It does so by borrowing energy from its enthalpy and converting it to kinetic energy. This incurs a heat drop. This part of the process is isentropic; the state point moves vertically down to the lower pressure.

Having passed through the narrow restriction, the steam expands into the lower pressure region in the valve outlet, and eventually decelerates as the volume of the valve body increases to connect to the downstream pipe. This fall in velocity requires a reduction in kinetic energy which is mostly re-converted back into heat and re-absorbed by the steam. The heat drop that caused the initial increase in kinetic energy is reclaimed (except for a small portion lost due to the effects of friction), and on the H - S chart, the state point moves up the constant pressure line until it arrives at the same enthalpy value as the initial condition.

The path of the state point is to be seen in Figure 2.15.8, where pressure is reduced from 5 bar at saturation temperature to 1 bar via, for example, a pressure reducing valve. Steam's enthalpy at the upstream condition of 5 bar is 2748 kJ/kg.

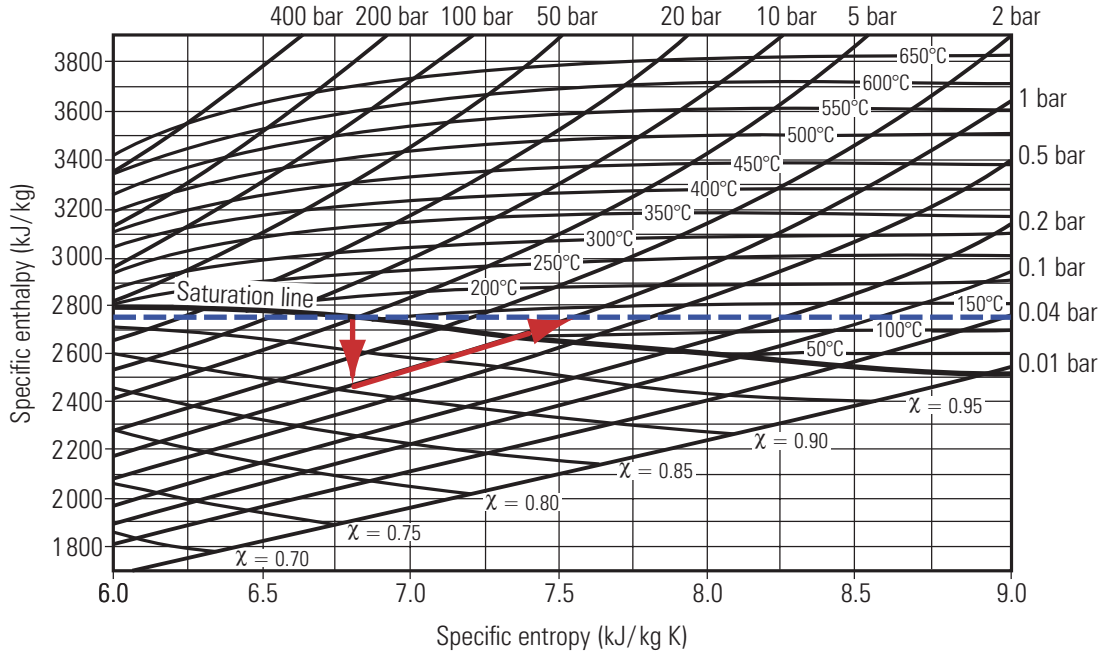


Fig. 2.15.8 The actual path of the state point in a control valve expansion

It is interesting to note that, in the example dicussed above and shown in Figure 2.15.8, the final condition of the steam is above the saturation line and is therefore superheated. Whenever such a process (commonly called a throttling process) takes place, the final condition of the steam will, in most cases, be drier than its initial condition. This will either produce drier saturated steam or superheated steam, depending on the respective positions of the initial and final state points. The horizontal distance between the initial and final state points represents the change in entropy. In this example, although there was no overall change in enthalpy (ignoring the small effects of friction), the entropy increased from about 6.8 kJ/kg K to about 7.6 kJ/kg K.

Entropy always increases in a closed system

In any closed system, the overall change in entropy is always positive, that is, it will always increase. It is worth considering this in more detail, as it is fundamental to the concept of entropy. Whereas energy is always conserved (the first law of thermodynamics states that energy cannot be created or destroyed), the same cannot be said about entropy. The second law of thermodynamics says that whenever energy is exchanged or converted from one form to another, the potential for energy to do work gets less. This really is what entropy is all about. It is a measure of the lack of potential or quality of energy; and once that energy has been exchanged or converted, it cannot revert back to a higher state. The ultimate truth of this is that it is nature's duty for all processes in the Universe to end up at the same temperature, so the entropy of the Universe is always increasing.

Example 2.15.1

Consider a teapot on a kitchen table that has just been filled with a certain quantity of water containing 200 kJ of heat energy at 100°C (373 K) from an electric kettle. Consider next that the temperature of the air surrounding the mug is at 20°C, and that the amount of heat in the teapot water would be 40 kJ at the end of the process. The second law of thermodynamics also states that heat will always flow from a hot body to a colder body, and in this example, it is certain that, if left for sufficient time, the teapot will cool to the same temperature as the air that surrounds it. What are the changes in the entropy values for the overall process?

For the teapot:

$$\text{Initial enthalpy in the teapot} = 200 \text{ kJ}$$

$$\text{Initial teapot temperature} = 373 \text{ K} \ (100°C)$$

$$\text{Final teapot temperature (the air temperature)} = 293 \text{ K} \ (20°C)$$

$$\text{Mean teapot temperature } T_{(mean)} = \frac{373 \text{ K} + 293 \text{ K}}{2}$$

$$\textbf{Mean teapot temperature } \mathbf{T_{(mean)}} = \mathbf{333 \text{ K} \ (60°C)}$$

$$\text{Final enthalpy in the teapot} = 40 \text{ kJ}$$

$$\text{Enthalpy delivered by the teapot to its surroundings} = 200 - 40 \text{ kJ}$$

$$\textbf{Enthalpy delivered by the teapot to its surroundings} = \mathbf{160 \text{ kJ}}$$

$$\text{Entropy delivered by the teapot to its surroundings} = \frac{\text{Enthalpy change}}{T_{(mean)}}$$

$$\text{Entropy delivered by the teapot to its surroundings} = \frac{160 \text{ kJ}}{333 \text{ K}}$$

$$\textbf{Entropy delivered by the teapot to its surroundings} = \mathbf{-0.48 \text{ kJ/K}}$$

Because the heat is lost from the teapot, convention states that its change in entropy is negative.

For the air:

$$\text{Initial air temperature} = 293 \text{ K} \ (20°C)$$

At the end of the process, the water in the teapot would have lost 160 kJ and the air would have gained 160 kJ; however, the air temperature would not have changed because of its large volume, therefore:

$$T_{(mean)} \text{ for the air} = 293 \text{ K} \ (20°C)$$

$$\text{Entropy received by the air} = \frac{160 \text{ kJ}}{293 \text{ K}}$$

$$\textbf{Entropy received by the air} = \mathbf{+0.546 \text{ kJ/K}}$$

Because the heat is received by the air, convention states that its change in enthalpy is positive.

Therefore:

$$\text{The overall change in entropy of the teapot and surroundings} = -0.48 + 0.546 \text{ kJ/K}$$

$$\textbf{The overall change in entropy of the teapot and surroundings} = \mathbf{+0.066 \text{ kJ/K}}$$

Practical applications - Heat exchangers

In a heat exchanger using saturated steam in the primary side to heat water from 20°C to 60°C in the secondary side, the steam will condense as it gives up its heat. This is depicted on the Mollier chart by the state point moving to the left of its initial position. For steady state conditions, dry saturated steam condenses at constant pressure, and the steam state point moves down the constant pressure line as shown in Figure 2.15.9.

Example 2.15.2

This example considers steam condensing from saturation at 2 bar at 120°C with an entropy of 7.13 kJ/kg K, and an enthalpy of about 2700 kJ/kg. It can be seen that the state point moves from right to left, not horizontally, but by following the constant 2 bar pressure line. The chart is not big enough to show the whole condensing process but, if it were, it would show that the steam's final state point would rest with an entropy of 1.53 kJ/kg K and an enthalpy of 504.8 kJ/kg, at 2 bar and 120°C on the saturated water line.

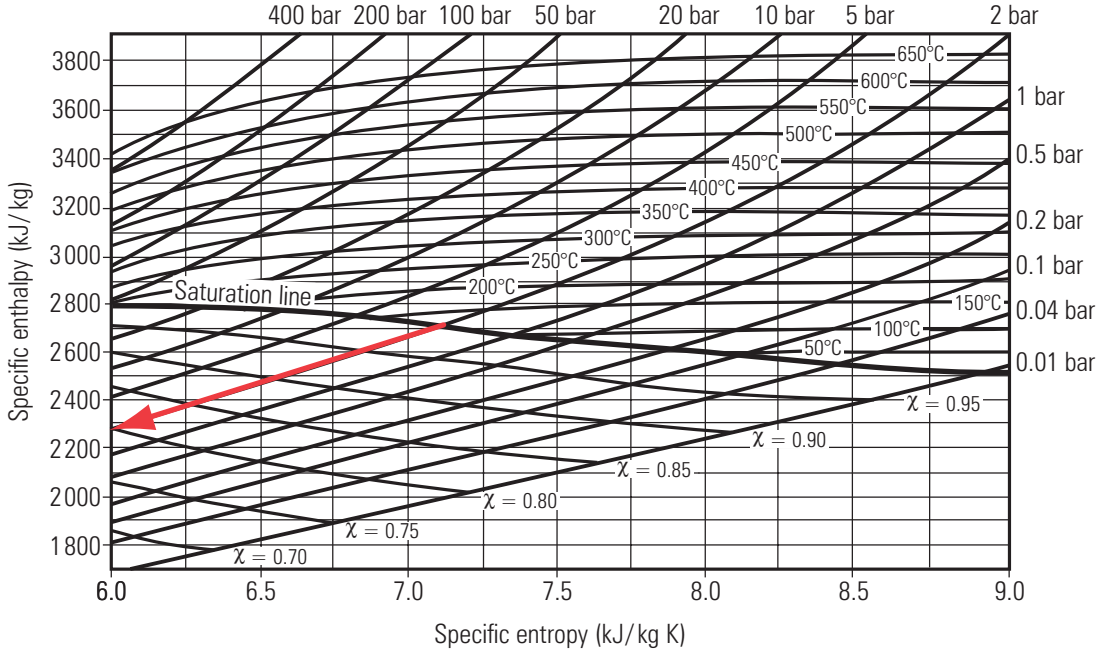


Fig. 2.15.9 The initial path of the state point for condensing steam

It can be seen from Figure 2.15.9 that, when steam condenses, the state point moves down the evaporation line and the entropy is lowered. However, in any overall system, the entropy must increase, otherwise the second law of thermodynamics is violated; so how can this decrease in entropy be explained?

As for the teapot in the Example 2.15.1, this decrease in entropy only reflects what is happening in one part of the system. It must be remembered that any total system includes its surroundings, in Example 2.15.2, the water, which receives the heat imparted by the steam.

In Example 2.15.2, the water receives exactly the same amount of heat as the steam imparts (it is assumed there are no heat losses), but does so at a lower temperature than the steam; so, as entropy is given by enthalpy/temperature, dividing the same quantity of heat by a lower temperature means a greater gain in entropy by the water than is lost by the steam. There is therefore an overall gain in the system entropy, and an overall spreading out of energy.

Table 2.15.1 Relative densities/specific heat capacities of various solids

Material	Relative density	Specific heat capacity kJ/kg °C
Aluminium	2.55 - 2.80	0.92
Andalusite		0.71
Antimony		0.20
Apatite		0.83
Asbestos	2.10 - 2.80	0.83
Augite		0.79
Bakelite, wood filler	1.38	
Bakelite, asbestos filler		1.59
Barite	4.50	0.46
Barium	3.50	2.93
Basalt rock	2.70 - 3.20	0.83
Beryl		0.83
Bismuth	9.80	0.12
Borax	1.70 - 1.80	1.00
Boron	2.32	1.29
Cadmium	8.65	0.25
Calcite 0 - 37°C		0.79
Calcite 0 - 100°C		0.83
Calcium	4.58	0.62
Carbon	1.80 - 2.100	0.71
Carborundum		0.66
Cassiterite		0.37
Cement, dry		1.54
Cement, powder		0.83
Charcoal		1.00
Chalcopyrite		0.54
Chromium	7.10	0.50
Clay	1.80 - 2.60	0.92
Coal	0.64 - 0.93	1.08 - 1.54
Cobalt	8.90	0.46
Concrete, stone		0.79
Concrete, cinder		0.75
Copper	8.80 - 8.95	0.37

Material	Relative density	Specific heat capacity kJ/kg °C
Corundum		0.41
Diamond	3.51	0.62
Dolomite rock	2.90	0.92
Fluorite		0.92
Fluorspar		0.87
Galena		0.20
Proxylin plastics	1.42 - 1.59	
Quartz, 12.8 - 100°C	2.50 - 2.80	0.79
Quartz, 0°C		0.71
Rock salt		0.92
Rubber		2.00
Sandstone	2.00 - 2.60	0.92
Serpentine	2.70 - 2.80	1.08
Silk		1.38
Silver	10.40 - 10.60	0.25
Sodium	0.97	1.25
Steel	7.80	0.50
Stone		0.83
Stoneware		0.79
Talc	2.60 - 2.80	0.87
Tar	1.20	1.46
Tellurium	6.00 - 6.24	0.20
Tin	7.20 - 7.50	0.20
Tile, hollow		0.62
Titanium	4.50	0.58
Topaz		0.87
Tungsten	19.22	0.16
Vanadium	5.96	0.50
Vulcanite		1.38
Wood	0.35 - 0.99	1.33 - 2.00
Wool	1.32	1.38
Zinc blend	3.90 - 4.20	0.46
Zinc	6.90 - 7.20	0.37

Table 2.15.2 Relative densities/specific heat capacities of various liquids

Liquid	Relative density	Specific heat capacitiy kJ/kg °C
Acetone	0.7900	2.13
Alcohol, ethyl, 0°C	0.7890	2.30
Alcohol, ethyl, 40°C	0.7890	2.72
Alcohol, methyl, 4 - 10°C	0.7960	2.46
Alcohol, methyl, 15 - 21°C	0.7960	2.51
Ammonia 0°C	0.6200	4.60
Ammonia 40°C		4.85
Ammonia 80°C		5.39
Ammonia 100°C		6.19
Ammonia 114°C		6.73
Anilin	1.0200	2.17
Benzol		1.75
Calcium chloride	1.2000	3.05
Castor oil		1.79
Citron oil		1.84
Diphenylamine	1.1600	1.92
Ethyl ether		2.21
Ethylene glycol		2.21
Fuel oil	0.9600	1.67
Fuel oil	0.9100	1.84
Fuel oil	0.8600	1.88
Fuel oil	0.8100	2.09
Gasoline		2.21
Glycerine	1.2600	2.42
Kerosene		2.00
Mercury	19.6000	1.38
Naphthalene	1.1400	1.71
Nitrobenzole		1.50
Olive oil	0.91 - 0.9400	1.96
Petroleum		2.13
Potassium hydrate	1.2400	3.68
Sea water	1.0235	3.93
Sesame oil		1.63
Sodium chloride	1.1900	3.30
Sodium hydrate	1.2700	3.93
Soybean oil		1.96
Toluol	0.8660	1.50
Turpentine	0.8700	1.71
Water	1.0000	4.18
Xylene	0.861 - 0.8810	1.71

Table 2.15.3 Specific heat capacities of gases and vapours

Gas or vapour	Specific heat capacity kJ/kg °C (constant pressure)
Acetone	1.31
Air, dry, 0°C	1.00
Air, dry, 100°C	1.01
Air, dry, 200°C	1.03
Air, dry, 300°C	1.05
Air, dry, 400°C	1.07
Air, dry, 500°C	1.09
Alcohol, C2 H5 OH	1.66
Alcohol, CH3 OH	1.53
Ammonia	1.76
Argon	0.30
Benzene, C6 H6	0.98
Bromine	0.19
Carbon dioxide	0.62
Carbon monoxide	0.71
Carbon disulphide	0.55
Chlorine	3.43
Chloroform	0.54
Ether	1.95
Hydrochloric acid	0.56
Hydrogen	10.00
Hydrogen sulphide	0.79
Methane	1.86
Nitrogen	0.71
Nitric oxide	0.69
Nitrogen tetroxide	4.59
Nitrous oxide	0.69
Oxygen	0.65
Steam, 0.5 bar a saturated	1.99
Steam, 2 bar a saturated	2.13
Steam, 10 bar a saturated	2.56
Steam, 0.5 bar a 150°C	1.95
Steam, 2 bar a 200°C	2.01
Steam, 10 bar a 250°C	2.21
Sulphur dioxide	0.49

Questions

1. What is true of entropy?

a| It is not a physical property of steam ☐

b| It reflects the quality of energy during a process ☐

c| It is energy change divided by the mean temperature of the change ☐

d| It is all of the above ☐

2. From the Mollier diagram in Figure 2.15.8, if the initial state point was saturated steam at 2 bar, the final state point was at 0.01 bar, and the expansion was isentropic, what is the approximate heat drop in one kilogram of steam?

a| 2 000 kJ ☐

b| 2 700 kJ ☐

c| 700 kJ ☐

d| 1 000 kJ ☐

3. What always accompanies an isentropic expansion of steam?

a| An increase in entropy ☐

b| An increase in enthalpy ☐

c| A decrease in entropy ☐

d| A decrease in enthalpy ☐

4. What always accompanies an isenthalpic expansion of steam?

a| An increase in entropy ☐

b| An increase in enthalpy ☐

c| A decrease in entropy ☐

d| A decrease in enthalpy ☐

5. What is true about steam as it condenses?

a| It does so at constant entropy and temperature ☐

b| It does so at constant enthalpy and reducing temperature ☐

c| Both enthalpy and entropy reduce and temperature remains constant ☐

d| Both enthalpy and entropy increase ☐

6. From the Mollier diagram in Figure 2.15.8, if the initial state point was saturated steam at 2 bar, the final state point was at 0.01 bar, and the expansion was isenthalpic, what is the approximate heat drop in one kilogram of steam?

a| None ☐

b| 1.5 kJ ☐

c| 700 kJ ☐

d| 2 000 kJ ☐

Answer

1: d, 2:c, 3:d, 4:a, 5:c, 6:a,

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Module 2.16

Entropy - Its Practical Use

Entropy - Its Practical Use

Practical use of entropy

It can be seen from Module 2.15 that entropy can be calculated. This would be laborious in practice, consequently steam tables usually carry entropy values, based on such calculations. Specific entropy is designated the letter 's' and usually appears in columns signifying specific values for saturated liquid, evaporation, and saturated steam, s_f, s_{fg} and s_g respectively. These values may equally be found in charts, and both Temperature - Entropy (T - S) and Enthalpy - Entropy (H - S) charts are to be found, as mentioned in Module 2.15. Each chart has particular use in specific circumstances.

The T - S chart is often used to determine the properties of steam during its expansion through a nozzle or an orifice. The seat of a control valve would be a typical example.

To understand how a T - S chart is applied, it is worth sketching such a chart and plotting the steam properties at the start condition, reading these from the steam tables.

Example 2.16.1

Steam is expanded from 10 bar a and a dryness fraction of 0.9 to 6 bar a through a nozzle, and no heat is removed or supplied during this expansion process. Calculate the final condition of the steam at the nozzle outlet? Specific entropy values quoted are in units of kJ/kg °C.

At 10 bar a, steam tables state that for dry saturated steam:

Specific entropy of saturated water (s_f) = 2.138 9

Specific entropy of evaporation of dry saturated steam (s_{fs}) = 4.447 1

At the inlet condition, as the dryness fraction is 0.9:

Specific entropy of evaporation present = 0.9 x 4.447 1

= 4.002 4

Specific entropy of the inlet steam = 2.138 9 + 4.002 4

Specific entropy of the inlet steam = 6.141 3

As no heat is added or removed during the expansion, the process is described as being adiabatic and isentropic, that is, the entropy does not change. It must still be 6.141 3 kJ/kg K at the very moment it passes the throat of the nozzle.

At the outlet condition of 6 bar a, steam tables state that:

Specific entropy of saturated water (s_f) = 1.931 6

Specific entropy of evaporation of dry saturated steam (s_{fg}) = 4.828 5

But, in this example, since the total entropy is fixed at 6.141 3 kJ/kg K:

Specific entropy of evaporation present = 6.1413 - 1.9316

= 4.209 7

Therefore:

$$\text{Dryness fraction} = \frac{4.209\,7}{4.828\,5}$$

Dryness fraction = 0.871 8

By knowing that this process is isentropic, it has been possible to calculate the dryness fraction at the outlet condition. It is now possible to consider the outlet condition in terms of specific enthalpy (units are in kJ/kg).

From steam tables, at the inlet pressure of 10 bar a:

Specific enthalpy of saturated water (h_f) = 762.9

Specific enthalpy of evaporation of dry saturated steam (h_{fg}) = 2014.83

As the dryness fraction is 0.9 at the inlet condition:

Specific enthalpy of evaporation present = 0.9 x 2014.83

= 1813.35

Specific enthalpy of the inlet steam = 762.9 + 1813.35

Specific enthalpy of the inlet steam = 2576.25

From steam tables, at the outlet condition of 6 bar a:

Specific enthalpy of saturated water (h_f) = 670.74

Specific enthalpy of evaporation of dry saturated steam (h_{fg}) = 2085.98

But as the dryness fraction is 0.8718 at the outlet condition:

Specific enthalpy of evaporation present = 0.8718 x 2085.98

= 1818.56

Total specific enthalpy of the outlet steam = 670.74 + 1818.56

Total specific enthalpy of the outlet steam = 2489.30

It can be seen that the specific enthalpy of the steam has dropped in passing through the nozzle from 2576.25 to 2489.30 kJ/kg, that is, a heat drop of 86.95 kJ/kg.

This seems to contradict the adiabatic principle, which stipulates that no energy is removed from the process. But, as seen in Module 2.15, the explanation is that the steam at 6 bar a has just passed through the nozzle throat at high velocity, consequently it has gained kinetic energy. As energy cannot be created or destroyed, the gain in kinetic energy in the steam is at the expense of its own heat drop.

The above entropy values in Example 2.16.1 can be plotted on a T - S diagram, see Figure 2.16.1.

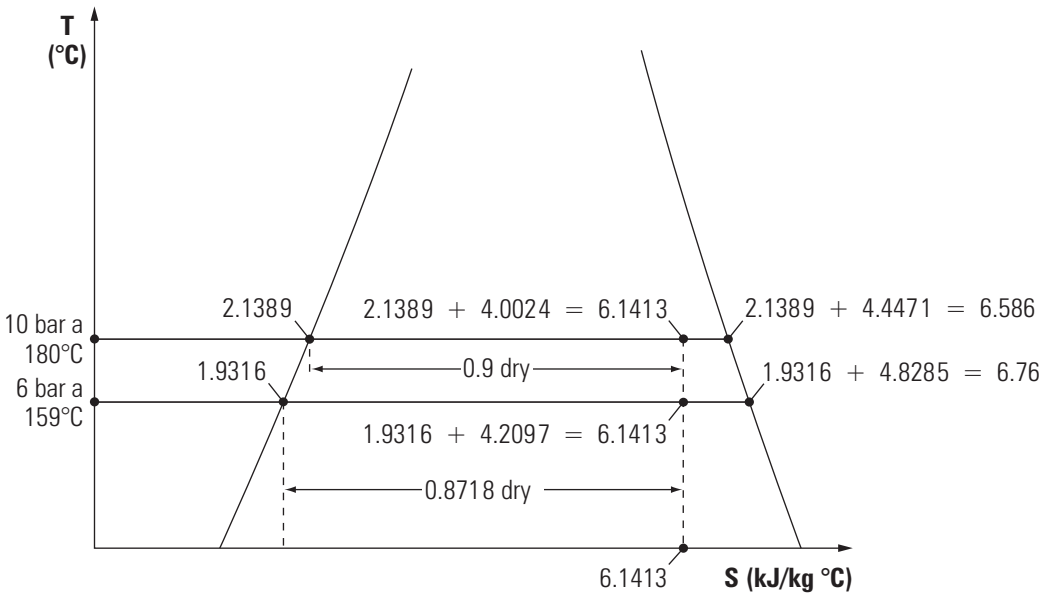


Fig. 2.16.1 The T - S diagram for Example 2.16.1

Further investigation of kinetic energy in steam

What is the significance of being able to calculate the kinetic energy of steam? By knowing this value, it is possible to predict the steam velocity and therefore the mass flow of steam through control valves and nozzles.

Kinetic energy is proportional to mass and the square of the velocity.

It can be further shown that, when incorporating Joule's mechanical equivalent of heat, kinetic energy can be written as Equation 2.16.1:

$$\text{Kinetic energy } (E) = \frac{m\ u^2}{2\ g\ J} \qquad \textbf{Equation 2.16.1}$$

Where:
E = Kinetic energy (kJ)
m = Mass of the fluid (kg)
u = Velocity of the fluid (m/s)
g = Acceleration due to gravity (9.806 65 m/s²)
J = Joule's mechanical equivalent of heat (101.972 m kg/kJ)

By transposing Equation 2.16.1 it is possible to find velocity as shown by Equation 2.16.2:

$$u^2 = \frac{2\ E\ g\ J}{m} \qquad \textbf{Equation 2.16.2}$$

For each kilogram of steam, and by using Equation 2.16.2

$u^2 = 2\ E\ g\ J$
$u^2 = 2\ \times\ E\ \times\ 9.806\,65\ \times\ 101.972$
$u^2 = 2\,000\ \times\ E$
$u = \sqrt{2\,000}\ \times\ \sqrt{E}$
$u = 44.72\ \sqrt{E}$

As the gain in kinetic energy equals the heat drop, the equation can be written as shown by Equation 2.16.3:

$$u = 44.72\ \sqrt{h} \qquad \textbf{Equation 2.16.3}$$

Where:
h = Heat drop in kJ/kg

By calculating the adiabatic heat drop from the initial to the final condition, the velocity of steam can be calculated at various points along its path; especially at the throat or point of minimum pass area between the plug and seat in a control valve.

This could be used to calculate the orifice area required to pass a given amount of steam through a control valve. The pass area will be greatest when the valve is fully open. Likewise, given the valve orifice area, the maximum flowrate through the valve can be determined at the stipulated pressure drop. See Examples 2.16.2 and 2.16.3 for more details.

Example 2.16.2

Consider the steam conditions in Example 2.16.1 with steam passing through a control valve with an orifice area of 1 cm². Calculate the maximum flow of steam under these conditions.

The downstream steam is at 6 bar a, with a dryness fraction of 0.8718.

Specific volume of dry saturated steam at 6 bar a (s_g) equals 0.315 6 m³/kg.

Specific volume of saturated steam at 6 bar a and a dryness fraction of 0.8718 equals 0.3156 m³/kg x 0.8718 which equates to 0.2751 m³/kg.

The heat drop in Example 2.16.1 was 86.95 kJ/kg, consequently the velocity can be calculated using Equation 2.16.3:

$$u = 44.72 \sqrt{h} \qquad \textbf{Equation 2.16.3}$$

$$u = 44.72 \sqrt{86.95}$$

$$u = 44.72 \times 9.32$$

$$\mathbf{u = 417\ m/s}$$

The mass flow is calculated using Equation 2.16.4:

$$\text{The mass flow} = \frac{\text{Velocity m/s} \times \text{Orifice area m}^2}{\text{Specific volume m}^3/\text{kg}}\ \text{kg/s} \qquad \textbf{Equation 2.16.4}$$

An orifice area of 1 cm² equals 0.0001 m²

$$\text{The mass flow} = \frac{417 \times 0.0001}{0.2751}\ \text{kg/s}$$

$$\textbf{The mass flow} = \mathbf{0.152\ kg/s}\ (547\ \text{kg/h})$$

Point of interest

Thermodynamic textbooks will usually quote Equation 2.16.3 in a slightly different way as shown in Equation 2.16.5:

$$u = \sqrt{2\,h} \qquad \textbf{Equation 2.16.5}$$

Where:
u = Velocity of the fluid in m/s
h = Heat drop in J/kg
2 = Constant of proportionality incorporating the gravitational constant 'g'.

Considering the conditions in Example 2.16.3:

$$\text{Heat drop (h)} = 86.95\ \text{kJ/kg}$$

$$\textbf{Heat drop (h)} = \mathbf{86\,950\ J/kg}$$

$$u = \sqrt{2\,h}$$

$$u = \sqrt{2 \times 86950}$$

$$\mathbf{u = 417\ m/s}$$

This velocity is exactly the same as that calculated from Equation 2.16.3, and the user is free to practise either equation according to preference.

The above calculations in Example 2.16.2 could be carried out for a whole series of reduced pressures, and, if done, would reveal that the flow of saturated steam through a fixed opening increases quite quickly at first as the downstream pressure is lowered.

The increases in flow become progressively smaller with equal increments of pressure drops and, with saturated steam, these increases actually become zero when the downstream pressure is 58% of the absolute upstream pressure. (If the steam is initially superheated, CPD will occur at just below 55% of the absolute upstream pressure).

This is known as the 'critical flow' condition and the pressure drop at this point is referred to as critical pressure drop (CPD). After this point has been reached, any further reduction of downstream pressure will not give any further increase in mass flow through the opening.

In fact if, for saturated steam, the curves of steam velocity (u) and sonic velocity (s) were drawn for a convergent nozzle (Figure 2.16.2), it would be found that the curves intersect at the critical pressure. P_1 is the upstream pressure, and P is the pressure at the throat.

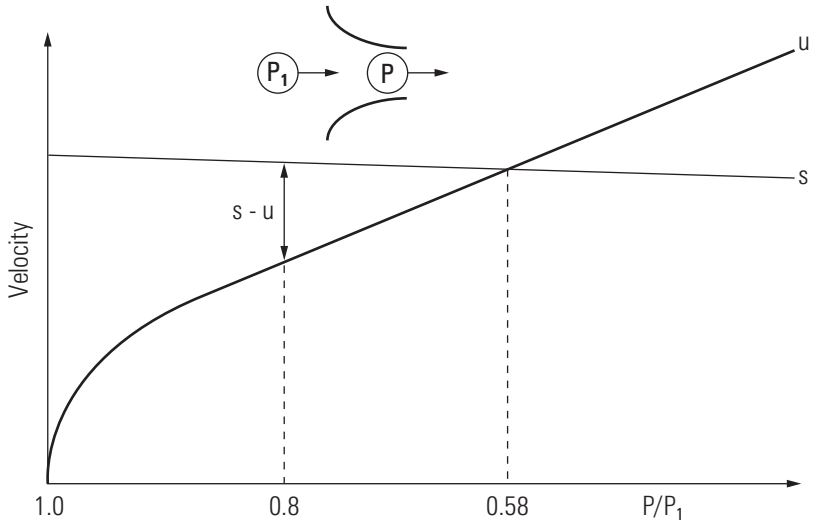


Fig. 2.16.2 Steam and acoustic velocities through a nozzle

The explanation of this, first put forward by Professor Osborne Reynolds (1842 - 1912) of Owens College, Manchester, UK, is as follows:

Consider steam flowing through a tube or nozzle with a velocity u, and let s be the speed of sound (sonic velocity) in the steam at any given point, s being a function of the pressure and density of the steam. Then the velocity with which a disturbance such as, for example, a sudden change of pressure P, will be transmitted back through the flowing steam will be s - u.

Referring to Figure 2.16.2, let the final pressure P at the nozzle outlet be 0.8 of its inlet pressure P_1. Here, as the sonic velocity s is greater than the steam velocity u, s - u is clearly positive. Any change in the pressure P would produce a change in the rate of mass flow.

When the pressure P has been reduced to the critical value of 0.58 P_1, s - u becomes zero, and any further reduction of pressure after the throat has no effect on the pressure at the throat or the rate of mass flow.

When the pressure drop across the valve seat is greater than critical pressure drop, the critical velocity at the throat can be calculated from the heat drop in the steam from the upstream condition to the critical pressure drop condition, using Equation 2.16.5.

Control valves

The relationship between velocity and mass flow through a restriction such as the orifice in a control valve is sometimes misunderstood.

Pressure drop greater than critical pressure drop

It is worth reiterating that, if the pressure drop across the valve is equal to or greater than critical pressure drop, the mass flow through the throat of the restriction is a maximum and the steam will travel at the speed of sound (sonic velocity) in the throat. In other words, the critical velocity is equal to the local sonic velocity, as described above.

For any control valve operating under critical pressure drop conditions, at any reduction in throat area caused by the valve moving closer to its seat, this constant velocity will mean that the mass flow is simultaneously reduced in direct proportion to the size of the valve orifice.

Pressure drop less than critical pressure drop

For a control valve operating such that the downstream pressure is greater than the critical pressure (critical pressure drop is not reached), the velocity through the valve opening will depend on the application.

Pressure reducing valves

If the valve is a pressure reducing valve, (its function is to achieve a constant downstream pressure for varying mass flowrates) then, the heat drop remains constant whatever the steam load. This means that the velocity through the valve opening remains constant whatever the steam load and valve opening. Constant upstream steam conditions are assumed.

It can be seen from Equation 2.16.4 that, under these conditions, if velocity and specific volume are constant, the mass flowrate through the orifice is directly proportional to the orifice area.

$$\text{The mass flow} = \frac{\text{velocity m/s x orifice area m}^2}{\text{Specific volume m}^3/\text{kg}} \text{ kg/s} \qquad \textbf{Equation 2.16.4}$$

Temperature control valves

In the case of a control valve supplying steam to a heat exchanger, the valve is required to reduce the mass flow as the heat load falls. The downstream steam pressure will then fall with the heat load, consequently the pressure drop and heat drop across the valve will increase. Thus, the velocity through the valve must increase as the valve closes.

In this case, Equation 2.16.4 shows that, as the valve closes, a reduction in mass flow is not directly proportional to the valve orifice, but is also modified by the steam velocity and its specific volume.

Example 2.16.3

Find the critical velocity of the steam at the throat of the control valve for Example 2.16.2, where the initial condition of the steam is 10 bar a and 90% dry, and assuming the downstream pressure is lowered to 3 bar a.

Specific enthalpy at 10 bar a, 0.9 dryness fraction = 2576.26 kJ/kg

Specific entropy at 10 bar a, 0.9 dryness fraction = 6.141 29 kJ/kg K

For wet steam, critical pressure can be taken as 58% of the absolute upstream pressure, therefore:

Pressure of steam at the throat = 0.58 x 10 bar a

= 5.8 bar a

At the throat condition of 5.8 bar a, and from steam tables:

Specific entropy of saturated water (s_f) = 1.918 36

Specific entropy of evaporation of dry saturated steam (s_{fg}) = 4.853 8

But, in this example, since the total entropy is fixed at 6.141 29 kJ/kg K:

Specific entropy of evaporation present = 6.141 29 - 1.918 36

= 4.222 93

Therefore, the dryness fraction at the throat at the throat $= \frac{4.222\,93}{4.853\,8}$

Dryness fraction = 0.870 1

From steam tables, at the throat condition of 5.8 bar a:

Specific enthalpy of saturated water (h_f) = 665.008

Specific enthalpy of evaporation of dry saturated steam (h_{fg}) = 2090.23

But as the dryness fraction is 0.8701 at the throat condition:

Specific enthalpy of evaporation present = 0.8701 x 2090.23 = 1818.71

Total specific enthalpy of the outlet steam = 665.008 + 1818.71

= 2483.72

Therefore, the heat drop at critical pressure drop = 2576.72 - 2483.72

Heat drop at critical pressure drop = 92.54 kJ/kg (92 540 J/kg)

The velocity of the steam through the throat of the valve can be calculated using Equation 2.16.5:

$$u = \sqrt{2\,h}$$ **Equation 2.16.5**

$$u = \sqrt{2 \times 92\,540}$$

$$\mathbf{u = 430\ m/s}$$

The critical velocity occurs at the speed of sound, consequently 430 m/s is the sonic velocity for the Example 2.16.3.

Noise in control valves

If the pressure in the outlet of the valve body is lower than the critical pressure, the heat drop at a point immediately after the throat will be greater than at the throat. As velocity is directly related to heat drop, the steam velocity will increase after the steam passes the throat of the restriction, and supersonic velocities can occur in this region.

In a control valve, steam, after exiting the throat, is suddenly confronted with a huge increase in space in the valve outlet, and the steam expands suddenly. The kinetic energy gained by the steam in passing through the throat is converted back into heat; the velocity falls to a value similar to that on the upstream side of the valve, and the pressure stabilises in the valve outlet and connecting pipework.

For the reasons mentioned above, valves operating at and greater than critical pressure drop will incur sonic and supersonic velocities, which will tend to produce noise. As noise is a form of vibration, high levels of noise will not only cause environmental problems, but may actually cause the valve to fail. This can sometimes have an important bearing when selecting valves that are expected to operate under critical flow conditions.

It can be seen from previous text that the velocity of steam through control valve orifices will depend on the application of the valve and the pressure drop across it at any one time.

Reducing noise in control valves

There are some practical ways to deal with the effects of noise in control valves.

Perhaps the simplest way to overcome this problem is to reduce the working pressure across the valve. For instance, where there is a need to reduce pressure, by reducing pressure with two valves instead of one, both valves can share the total heat drop, and the potential for noise in the pressure reducing station can be reduced considerably.

Another way to reduce the potential for noise is by increasing the size of the valve body (but retaining the correct orifice size) to help ensure that the supersonic velocity will have dissipated by the time the flow impinges upon the valve body wall.

In cases where the potential for noise is extreme, valves fitted with a noise attenuator trim may need to be used.

Steam velocities in control valve orifices will reach, typically, 500 m/s. Water droplets in the steam will travel at some slightly lower speed through a valve orifice, but, being incompressible, these droplets will tend to erode the valve and its seat as they squeeze between the two.

It is always sensible to ensure that steam valves are protected from wet steam by fitting separators or by providing adequate line drainage upstream of them.

Summing up of Modules 2.15 and 2.16

The T - S diagram, shown in Figure 2.16.1, and reproduced below in Figure 2.16.3, shows clearly that the steam becomes wetter during an isentropic expansion (0.9 at 10 bar a to 0.8718 at 6 bar a) in Example 2.16.1.

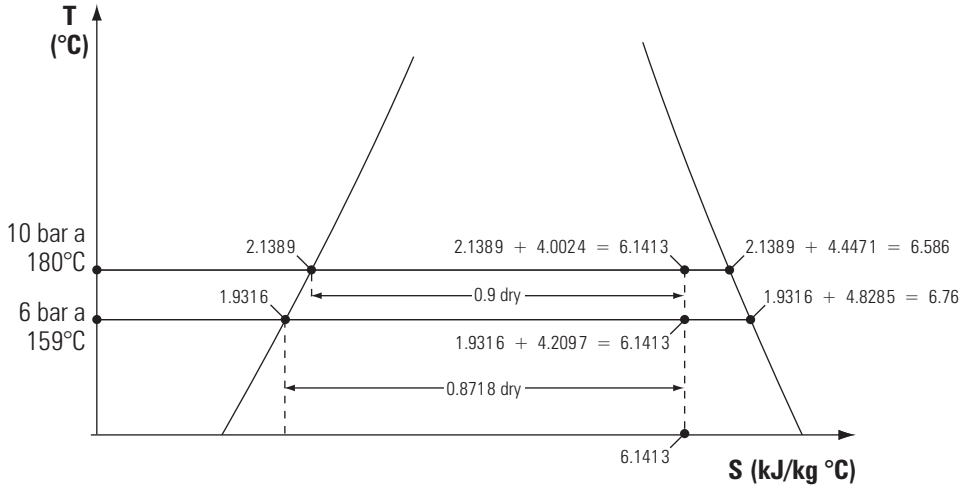


Fig. 2.16.3 A T-S diagram showing wetter steam from an isentropic expansion

At first, this seems strange to those who are used to steam getting drier or becoming superheated during an expansion, as happens when steam passes through, for example, a pressure reducing valve.

The point is that, during an adiabatic expansion, the steam is accelerating up to high speed in passing through a restriction, and gaining kinetic energy. To provide this energy, a little of the steam condenses (if saturated steam), (if superheated, drops in temperature and may condense) providing heat for conversion into kinetic energy.

If the steam is flowing through a control valve, or a pressure reducing valve, then somewhere downstream of the valve's seat, the steam is slowed down to something near its initial velocity. The kinetic energy is destroyed, and must reappear as heat energy that dries out or superheats the steam depending on the conditions.

The T - S diagram is not at all convenient for showing this effect, but the Mollier diagram (the H - S diagram) can do so quite clearly.

The Mollier diagram can depict both an isenthalpic expansion as experienced by a control valve, (see Figure 2.15.6) by moving horizontally across the graph to a lower pressure; and an isentropic expansion as experienced by steam passing through a nozzle, (see Figure 2.15.7) by moving horizontally down to a lower pressure. In the former, the steam is usually either dried or superheated, in the latter, the steam gets wetter.

This perhaps begs the question, 'How does the steam know if it is to behave in an isenthalpic or isentropic fashion?' Clearly, as the steam accelerates and rushes through the narrowest part of the restriction (the throat of a nozzle, or the adjustable gap between the valve and seat in a control valve) it must behave the same in either case.

The difference is that the steam issuing from a nozzle will next meet a turbine wheel and gladly give up its kinetic energy to turn the turbine. In fact, a nozzle could be thought of as a device to convert heat energy into kinetic energy for this very purpose.

In a control valve, instead of doing such work, the steam simply slows down in the valve outlet passages and its connecting pipework, when the kinetic energy appears as heat energy, and unwittingly goes on its way to give up this heat at a lower pressure.

It can be seen that both the T - S diagram and H - S diagram have their uses, but neither would have been possible had the concept of entropy not been utilised.

Questions

1. From the T - S diagram shown in Figure 2.16.1, had the initial state point been 100% dry saturated steam at 10 bar, what would have been its specific entropy?

a| 6.586 kJ/kg K ☐
b| 2.1389 kJ/kg K ☐
c| 6.1413 kJ/kg K ☐
d| 6.76 kJ/kg K ☐

2. From the T - S diagram shown in Figure 2.16.1, had the initial state point been 100% dry saturated steam at 10 bar, and the final pressure 6 bar, in which region would the final state point have been?

a| The superheated region ☐
b| On the saturated steam line ☐
c| The wet steam region ☐
d| On the saturated water line ☐

3. In a steam control valve, the heat drop from the initial condition to that at the valve throat is calculated to be 50 kJ/kg. What is the velocity of steam passing through the valve orifice?

a| 416.65 m/s ☐
b| 316.23 m/s ☐
c| Sonic velocity ☐
d| Supersonic velocity ☐

4. In Question 3, the orifice area is known to be 50 mm², and the specific volume of steam at the downstream pressure is 0.3 m³/kg. What is the mass flowrate?

a| Critical flow ☐
b| 200.01 kg/h ☐
c| 189.74 kg/h ☐
d| 40 kg/h ☐

5. A pressure control valve is set to reduce and maintain pressure from 10 bar g to 7 bar g. The velocity through the valve orifice at full-load is 400 m/s. What is the velocity through the orifice at half-load?

a| 200 m/s ☐
b| 800 m/s ☐
c| 282.8 m/s ☐
d| 400 m/s ☐

6. What can be done to reduce noise in valves operating under critical conditions?

a| Use two valves in series instead of one ☐
b| Use a valve with the same size seat but having a larger body ☐
c| Use a valve with a noise attenuation trim ☐
d| Any of the above ☐

Answers

1: a, 2: c, 3: b, 4: c, 5: d, 6: d,

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Module 3.1

Introduction

Introduction

The Boiler House Block of the *Steam and Condensate Loop* will concentrate on the design and contents of the boiler house, and the applications within it.

A well designed, operated and maintained boiler house is the heart of an efficient steam plant. However, a number of obstacles can prevent this ideal. The boiler house and its contents are sometimes viewed as little more than a necessary inconvenience and even in today's energy-conscious environment, accurate steam flow measurement and the correct allocation of costs to the various users, is not universal. This can mean that efficiency improvements and cost-saving projects related to the boiler house may be difficult to justify to the end user.

In many cases, the boiler house and the availability of steam are the responsibility of the Engineering Manager, consequently any efficiency problems are seen to be his.

It is important to remember that the steam boiler is a pressurised vessel containing scalding hot water and steam at more than 100°C, and its design and operation are covered by a number of complex standards and regulations.

These standards vary as follows:

- **Location -** For example, the UK, Australia, and New Zealand all have individual standards. The variations between standards may seem small but can sometimes be quite significant.
- **Over time -** For example, technology is changing at a tremendous rate, and improvements in the capabilities of equipment, together with the frequent adjustment of operating standards demanded by the relevant legislative bodies, are resulting in increases in the safety of boiler equipment.
- **Environmental terms -** Many governments are insisting on increasingly tight controls, including emission standards and the overall efficiency of the plant. Users who chose to ignore these (and pending controls) do so with an increasing risk of higher penalties being imposed on them.
- **Cost terms -** Fuel costs are continually increasing, and organisations should constantly review alternative steam raising fuels, and energy waste management.

For the reasons listed above, the user must confirm national and local and current legislation.

The objective of this Module is to provide the designer, operator, and maintainer of the boiler house with an insight into the considerations required in the development of the boiler and its associated equipment.

Modern steam boilers come in all sizes to suit both large and small applications. Generally, where more than one boiler is required to meet the demand, it becomes economically viable to house the boiler plant in a centralised location, as installation and operating costs can be significantly lower than with decentralised plant.

For example, centralisation offers the following benefits over the use of dispersed, smaller boilers:

- More choices of fuel and tariff.
- Identical boilers are frequently used in centralised boiler rooms reducing spares, inventory and costs.
- Heat recovery is easy to implement for best returns.
- A reduction in manual supervision releases labour for other duties on site.
- Economic sizing of boiler plant to suit diversified demand.
- Exhaust emissions are more easily monitored and controlled.
- Safety and efficiency protocols are more easily monitored and controlled.

Fuel for boilers

The three most common types of fuel used in steam boilers, are coal, oil, and gas. However, industrial or commercial waste is also used in certain boilers, along with electricity for electrode boilers.

Coal

Coal is the generic term given to a family of solid fuels with a high carbon content. There are several types of coal within this family, each relating to the stages of coal formation and the amount of carbon content. These stages are:

- Peat.
- Lignite or brown coals.
- Bituminous.
- Semi bituminous.
- Anthracite.

The bituminous and anthracite types tend to be used as boiler fuel.

In the UK, the use of lump coal to fire shell boilers is in decline. There are a number of reasons for this including:

- **Availability and cost -** With many coal seams becoming exhausted, smaller quantities of coal are produced in the UK than formerly, and its decline must be expected to continue.
- **Speed of response to changing loads -** With lump coal, there is a substantial time lag between:
  - Demand for heat occurring.
  - Stoking of coal into the boiler.
  - Ignition of the coal.
  - Steam being generated to satisfy the demand.

To overcome this delay, boilers designed for coal firing need to contain more water at saturation temperature to provide the reserve of energy to cover this time lag. This, in turn, means that the boilers are bigger, and hence more expensive in purchase cost, and occupy more valuable product manufacturing space.

Ash - Ash is produced when coal is burned.

The ash may be awkward to remove, usually involving manual intervention and a reduction in the amount of steam available whilst de-ashing takes place.

The ash must then be disposed of, which in itself may be costly.

Stoking equipment - A number of different arrangements exist including stepper stokers, sprinklers and chain-grate stokers. The common theme is that they all need substantial maintenance.

Emissions - Coal contains an average of 1.5% sulphur (S) by weight, but this level may be as high as 3% depending upon where the coal was mined.

During the combustion process:

- Sulphur will combine with oxygen (O_2) from the air to form SO_2 or SO_3.
- Hydrogen (H) from the fuel will combine with oxygen (O_2) from the air to form water (H_2O).

After the combustion process is completed, the SO_3 will combine with the water (H_2O) to produce sulphuric acid (H_2SO_4), which can condense in the flue causing corrosion if the correct flue temperatures are not maintained. Alternatively, it is carried over into the atmosphere with the flue gases. This sulphuric acid is brought back to earth with rain, causing:

- Damage to the fabric of buildings.
- Distress and damage to plants and vegetation.

The ash produced by coal is light, and a proportion will inevitably be carried over with the exhaust gases, into the stack and expelled as particulate matter to the environment.

Coal, however, is still used to fire many of the very large water-tube boilers found in power stations.

Because of the large scale of these operations, it becomes economic to develop solutions to the problems mentioned above, and there may also be governmental pressure to use domestically produced fuels, for national security of electrical supply.

The coal used in power stations is milled to a very fine powder, generally referred to as 'pulverised fuel', and usually abbreviated to 'pf'.

- The small particle size of pf means that its surface area-to-volume ratio is greatly increased, making combustion very rapid, and overcoming the rate of response problem encountered when using lump coal.
- The small particle size also means that pf flows very easily, almost like a liquid, and is introduced into the boiler furnace through burners, eliminating the stokers used with lump coal.
- To further enhance the flexibility and turndown of the boiler, there may be 30+ pf burners around the walls and roof of the boiler, each of which may be controlled independently to increase or decrease the heat in a particular area of the furnace. For example, to control the temperature of the steam leaving the superheater.

With regard to the quality of the gases released into the atmosphere:

- The boiler gases will be directed through an electrostatic precipitator where electrically charged plates attract ash and other particles, removing them from the gas stream.
- The sulphurous material will be removed in a gas scrubber.
- The final emission to the environment is of a high quality.

Approximately 8 kg of steam can be produced from burning 1 kg of coal.

Oil

Oil for boiler fuel is created from the residue produced from crude petroleum after it has been distilled to produce lighter oils like gasoline, paraffin, kerosene, diesel or gas oil. Various grades are available, each being suitable for different boiler ratings; the grades are as follows:

- **Class D -** Diesel or gas oil.
- **Class E -** Light fuel oil.
- **Class F -** Medium fuel oil.
- **Class G -** Heavy fuel oil.

Oil began to challenge coal as the preferred boiler fuel in the UK during the 1950s. This came about in part from the then Ministry of Fuel and Power's sponsorship of research into improving boiler plant.

The advantages of oil over coal include:

- A shorter response time between demand and the required amount of steam being generated.
- This meant that less energy had to be stored in the boiler water. The boiler could therefore be smaller, radiating less heat to the environment, with a consequent improvement in efficiency.
- The smaller size also meant that the boiler occupied less production space.
- Mechanical stokers were eliminated, reducing maintenance workload.
- Oil contains only traces of ash, virtually eliminating the problem of ash handling and disposal.
- The difficulties encountered with receiving, storing and handling coal were eliminated.

Approximately 15 kg of steam can be produced from 1 kg of oil, or 14 kg of steam from 1 litre of oil.

Gas

Gas is a form of boiler fuel that is easy to burn, with very little excess air. Fuel gases are available in two different forms:

- **Natural gas -** This is gas that has been produced (naturally) underground. It is used in its natural state, (except for the removal of impurities), and contains a high proportion of methane.
- **Liquefied petroleum gases (LPG) -** These are gases that are produced from petroleum refining and are then stored under pressure in a liquid state until used. The most common forms of LPG are propane and butane.

In the late 1960s the availability of natural gas (such as from the North Sea) led to further developments in boilers.

The advantages of gas firing over oil firing include:

- Storage of fuel is not an issue; gas is piped right into the boiler house.
- Only a trace of sulphur is present in natural gas, meaning that the amount of sulphuric acid in the flue gas is virtually zero.

Approximately 42 kg of steam can be produced from 1 Therm of gas (equivalent to 105.5 MJ) for a 10 bar g boiler, with an overall operating efficiency of 80%.

Waste as the primary fuel

There are two aspects to this:

- **Waste material** - Here, waste is burned to produce heat, which is used to generate steam. The motives may include the safe and proper disposal of hazardous material. A hospital would be a good example:

  - In these circumstances, it may be that proper and complete combustion of the waste material is difficult, requiring sophisticated burners, control of air ratios and monitoring of emissions, especially particulate matter. The cost of this disposal may be high, and only some of the cost is recovered by using the heat generated to produce steam. However, the overall economics of the scheme, taking into consideration the cost of disposing of the waste by other means, may be attractive.

  - Using waste as a fuel may involve the economic utilisation of the combustible waste from a process. Examples include the bark stripped from wood in paper plants, stalks (bagasse) in sugar cane plants and sometimes even litter from a chicken farm.

  The combustion process will again be fairly sophisticated, but the overall economics of the cost of waste disposal and generation of steam for other applications on site, can make such schemes attractive.

- **Waste heat** - here, hot gases from a process, such as a smelting furnace, may be directed through a boiler with the objective of improving plant efficiency. Systems of this type vary in their level of sophistication depending upon the demand for steam within the plant. If there is no process demand for steam, the steam may be superheated and then used for electrical generation.

  This type of technology is becoming popular in Combined Heat and Power (CHP) plants:

  - A gas turbine drives an alternator to produce electricity.

  - The hot (typically 500°C) turbine exhaust gases are directed to a boiler, which produces saturated steam for use on the plant.

  Very high efficiencies are available with this type of plant. Other benefits may include either security of electrical supply on site, or the ability to sell the electricity at a premium to the national electricity supplier.

Which fuel to use?

The choice of fuel(s) is obviously very important, as it will have a significant impact on the costs and flexibility of the boiler plant.

Factors that need consideration include:

- **Cost of fuel** - For comparison purposes the cost of fuel is probably most conveniently expressed in £/kg of steam generated.
- **Cost of firing equipment** - The cost of the burner(s) and associated equipment to suit the fuel(s) selected, and the emission standards which must be observed.

Security of supply

What are the consequences of having no steam available for the plant ? Gas, for example, may be available at advantageous rates, provided an interruptible supply can be accepted. This means that the gas company will supply fuel while they have a surplus. However, should demand for fuel approach the limits of supply, perhaps due to seasonal variation, then supply may be cut, maybe at very short notice.

As an alternative, boiler users may elect to specify dual fuel burners which may be fired on gas when it is available at the lower tariff, but have the facility to switch to oil firing when gas is not available. The dual fuel facility is obviously a more expensive capital option, and the likelihood of gas not being available may be small. However, the cost of plant downtime due to the non-availability of steam is usually significantly greater than the additional cost.

Fuel storage

This is not an issue when using a mains gas supply, except where a dual fuel system is used. However it becomes progressively more of an issue if bottled gas, light oils, heavy oils and solid fuels are used.

The issues include:

- How much is to be stored, and where.
- How to safely store highly combustible materials.
- How much it costs to maintain the temperature of heavy oils so that they are at a suitable viscosity for the equipment.
- How to measure the fuel usage rate accurately.
- Allowance for storage losses.

Boiler design

The boiler manufacturer must be aware of the fuel to be used when designing a boiler. This is because different fuels produce different flame temperatures and combustion characteristics.

For example:

- Oil produces a luminous flame, and a large proportion of the heat is transferred by radiation within the furnace.
- Gas produces a transparent blue flame, and a lower proportion of heat is transferred by radiation within the furnace.

On a boiler designed only for use with oil, a change of fuel to gas may result in higher temperature gases entering the first pass of fire-tubes, causing additional thermal stresses, and leading to early boiler failure.

Boiler types

The objectives of a boiler are:

- To release the energy in the fuel as efficiently as possible.
- To transfer the released energy to the water, and to generate steam as efficiently as possible.
- To separate the steam from the water ready for export to the plant, where the energy can be transferred to the process as efficiently as possible.

A number of different boiler types have been developed to suit the various steam applications.

Questions

1. What is one advantage of an interruptible gas supply compared to a non-interruptible supply?

a | The gas is cheaper ☐
b | The boiler efficiency is normally higher ☐
c | The gas is cleaner ☐
d | Easier to obtain ☐

2. Which of the following is a harmful by-product of coal combustion?

a | H_2SO_4 ☐
b | O_2 ☐
c | SO_2 ☐
d | SO_3 ☐

3. What type of coal is generally used in a power station?

a | Lignite ☐
b | Brown lump coal ☐
c | Peat ☐
d | Pulverised fuel ☐

4. Which one of the following is probably true of decentralised boiler plant?

a | Reduction in manual supervision possible ☐
b | Safety and efficiency protocols more easily monitored ☐
c | Reduction in overall steam main losses ☐
d | More choices of fuel and tariffs ☐

5. What is used in a power station to remove sulphurous material?

a | Filters ☐
b | Chain grate stoker ☐
c | Electrostatic precipitator ☐
d | Gas scrubber ☐

6. What is the disadvantage of an interruptible gas supply arrangement?

a | Greater storage of gas is necessary ☐
b | The gas costs more ☐
c | Interruptions can occur at short notice ☐
d | The need to use heavy fuel oil as a reserve ☐

Answers

1: a, 2: a, 3: d, 4: c, 5: d, 6: b

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Module 3.2

Shell Boilers

Shell Boilers

Shell boilers may be defined as those boilers in which the heat transfer surfaces are all contained within a steel shell. Shell boilers may also be referred to as 'fire tube' or 'smoke tube' boilers because the products of combustion pass through the boiler tubes, which in turn transfer heat to the surrounding boiler water.

Several different combinations of tube layout are used in shell boilers, involving the number of passes the heat from the boiler furnace will usefully make before being discharged.

Figures 3.2.1a and 3.2.1b show a typical two-pass boiler configuration.

Figure 3.2.1a shows a dry back boiler where the hot gases are reversed by a refractory lined chamber on the outer plating of the boiler.

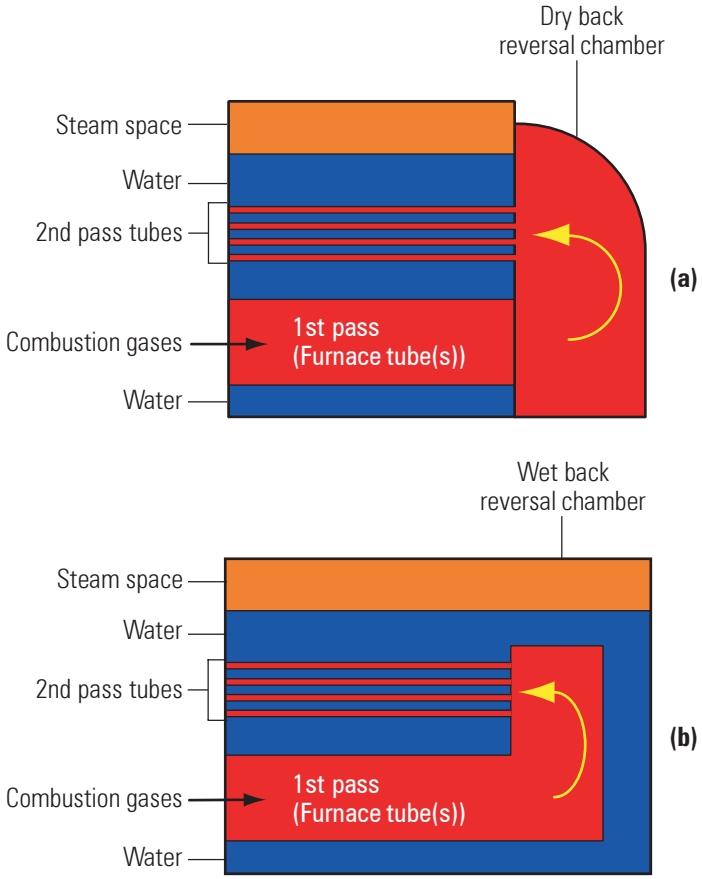


Fig. 3.2.1 Shell boiler - Wet and dry back configurations

Figure 3.2.1b shows a more efficient method of reversing the hot gases through a wet back boiler configuration. The reversal chamber is contained entirely within the boiler. This allows for a greater heat transfer area, as well as allowing the boiler water to be heated at the point where the heat from the furnace will be greatest - on the end of the chamber wall.

It is important to note that the combustion gases should be cooled to at least 420°C for plain steel boilers and 470°C for alloy steel boilers before entering the reversal chamber. Temperatures in excess of this will cause overheating and cracking of the tube end plates. The boiler designer will have taken this into consideration, and it is an important point if different fuels are being considered.

Several different types of shell boilers have been developed, which will now be looked at in more detail.

Lancashire boiler

Sir William Fairbairn developed the Lancashire boiler in 1844 from Trevithick's single flue Cornish boiler. Although only a few are still in operation, they were ubiquitous and were the predecessors of the sophisticated and highly efficient boilers used today.

The Lancashire boiler comprised a large steel shell usually between 5 - 9 m long through which passed two large-bore furnace tubes called flues. Part of each flue was corrugated to take up the expansion when the boiler became hot, and to prevent collapse under pressure. A furnace was installed at the entrance to each flue, at the front end of the boiler. Typically, the furnace would be arranged to burn coal, being either manually or automatically stoked.

The hot gaseous products of combustion passed from the furnace through the large-bore corrugated flues. Heat from the hot flue gases was transferred into the water surrounding these flues.

The boiler was in a brickwork setting which was arranged to duct the hot gases emerging from the flues downwards and beneath the boiler, transferring heat through the bottom of the boiler shell, and secondly back along the sides of the boiler before exiting through the stack.

These two side ducts met at the back of the boiler and fed into the chimney.

These passes were an attempt to extract the maximum amount of energy from the hot product gases before they were released to atmosphere.

Later, the efficiency was improved by the addition of an economiser. The gas stream, after the third pass, passed through the economiser into the chimney. The economiser heated the feedwater and resulted in an improvement in thermal efficiency.

One of the disadvantages of the Lancashire boiler was that repeated heating and cooling of the boiler, with the resultant expansion and contraction that occurred, upset the brickwork setting and ducting. This resulted in the infiltration of air, which upset the furnace draught.

These boilers would now be very expensive to produce, due to the large amounts of material used and the labour required to build the brick setting.

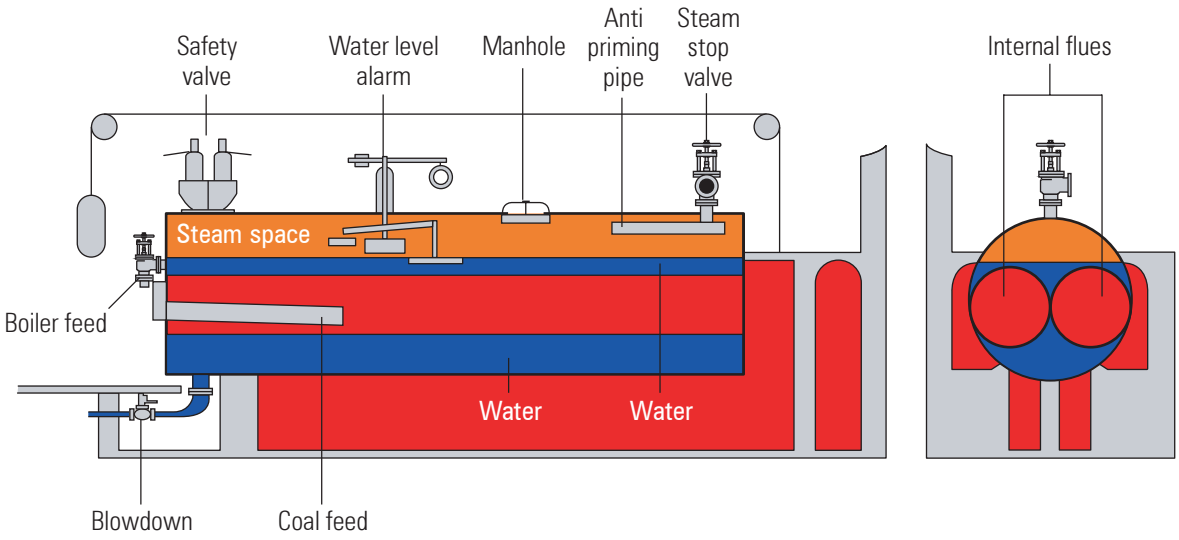


Fig. 3.2.2 Lancashire boiler

Table 3.2.1 Size range of Lancashire boilers

Capacity	Small	Large
Dimensions	5.5 m long x 2 m diameter	9 m long x 3 m diameter
Output	1 500 kg/h	6 500 kg/h
Pressure	Up to 12 bar g	up to 12 bar g

The large size and water capacity of these boilers had a number of significant advantages:

- Sudden large steam demands, such as a pit-winding engine being started, could easily be tolerated because the resulting reduction in boiler pressure released copious amounts of flash steam from the boiler water held at saturation temperature.

  These boilers may well have been manually stoked, consequently the response to a decrease in boiler pressure and the demand for more fuel would have been slow.

- The large volume of water meant that although the steaming rate might vary widely, the rate of change of the water level was relatively slow.

  Water level control would again have been manual, and the operator would either start a reciprocating, steam powered feedwater pump, or adjust a feedwater valve to maintain the desired water level.

- The low level alarm was simply a float that descended with the water level, and opened a port to a steam whistle when a pre-determined level was reached.

- The large water surface area in relation to the steaming rate meant that the rate at which steam was released from the surface (expressed in terms of kg per square metre) was low.

  This low velocity meant that, even with water containing high concentrations of Total Dissolved Solids (TDS), there was plenty of opportunity for the steam and water particles to separate and dry steam to be supplied to the plant.

As control systems, materials, and manufacturing techniques have become more sophisticated, reliable and cost effective, the design of boiler plant has changed.

Economic boiler (two-pass, dry back)

The two-pass economic boiler was only about half the size of an equivalent Lancashire boiler and it had a higher thermal efficiency. It had a cylindrical outer shell containing two large-bore corrugated furnace flues acting as the main combustion chambers. The hot flue gases passed out of the two furnace flues at the back of the boiler into a brickwork setting (dry back) and were deflected through a number of small-bore tubes arranged above the large-bore furnace flues. These small bore tubes presented a large heating surface to the water. The flue gases passed out of the boiler at the front and into an induced draught fan, which passed them into the chimney.

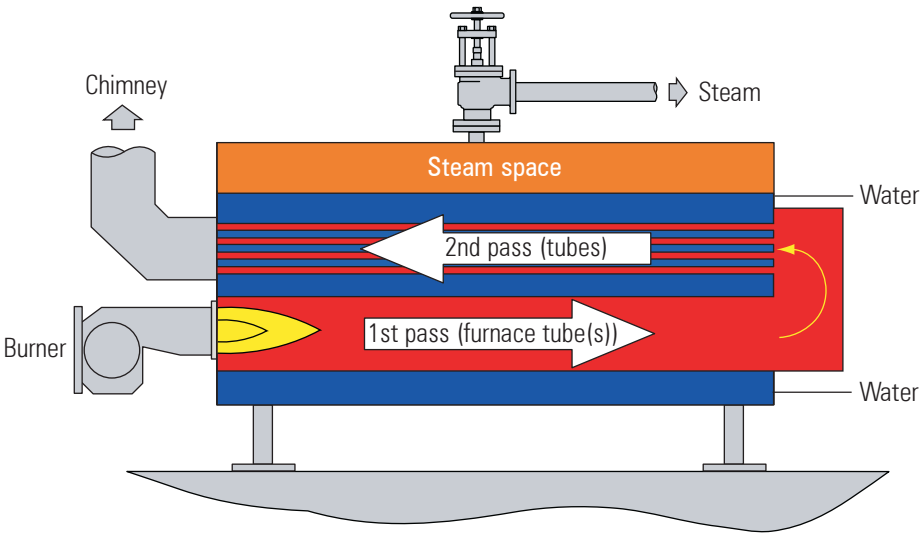


Fig. 3.2.3 Economic boiler (two-pass, dry back)

Table 3.2.2 Size range of two-pass, dry back economic boilers

Capacity	Small	Large
Dimensions	3 m long x 1.7 m diameter	7 m long x 4 m diameter
Output	1 000 kg/h	15 000 kg/h
Pressure	Up to 17 bar g	up to 17 bar g

Economic boiler (three-pass, wet back)

A further development of the economic boiler was the creation of a three-pass wet back boiler which is a standard configuration in use today, (see Figure 3.2.4).

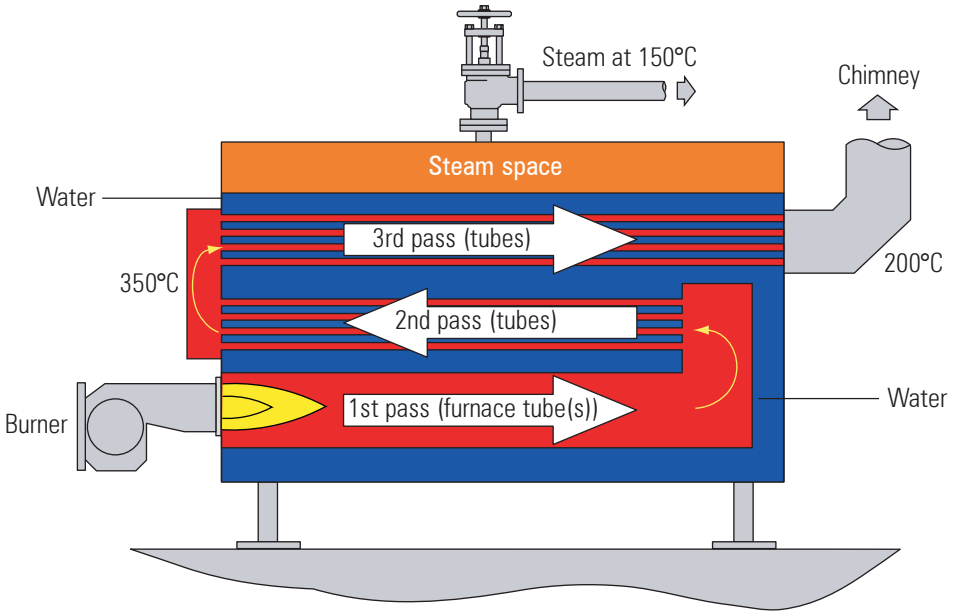


Fig. 3.2.4 Economic boiler (three-pass, wet back)

This design has evolved as materials and manufacturing technology has advanced: thinner metal tubes were introduced allowing more tubes to be accommodated, the heat transfer rates to be improved, and the boilers themselves to become more compact.

Typical heat transfer data for a three-pass, wet back, economic boiler is shown in Table 3.2.3.

Table 3.2.3 Heat transfer details of a modern three pass, wet back, economic boiler

	Area of tubes	Temperature	Proportion of total heat transfer
1st pass	11 m²	1 600°C	65%
2nd pass	43 m²	400°C	25%
3rd pass	46 m²	350°C	10%

Packaged boiler

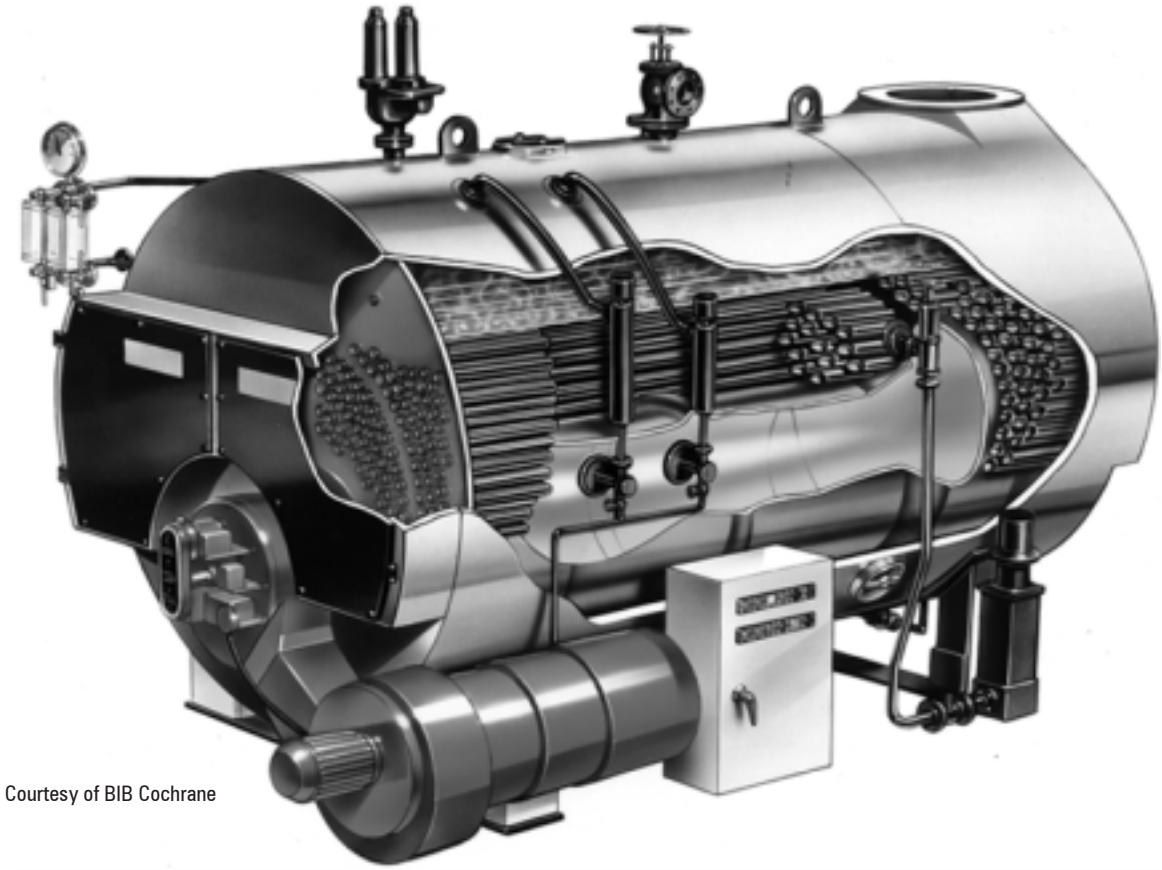
In the early 1950s, the UK Ministry of Fuel and Power sponsored research into improving boiler plant. The outcome of this research was the packaged boiler, and its a further development on the three-pass economic wet back boiler. Mostly, these boilers were designed to use oil rather than coal.

The packaged boiler is so called because it comes as a complete package with burner, level controls, feedpump and all necessary boiler fittings and mountings. Once delivered to site it requires only the steam, water, and blowdown pipework, fuel supply and electrical connections to be made for it to become operational.

Development has also had a significant effect on the physical size of boilers for a given output:

- Manufacturers wanted to make the boilers as small as possible to save on materials and hence keep their product competitive.
- Efficiency is aided by making the boiler as small as it is practical; the smaller the boiler and the less its surface area, the less heat is lost to the environment.

  To some extent the universal awareness of the need for insulation, and the high performance of modern insulating materials, reduces this issue.
- Consumers wanted the boilers to be as small as possible to minimise the amount of floor space needed by the boiler house, and hence increase the space available for other purposes.



Courtesy of BIB Cochrane

Fig. 3.2.5 Modern packaged boiler

- Boilers with smaller dimensions (for the same steam output) tend to be lower in capital cost. Table 3.2.4 demonstrates this, and other factors.

Table 3.2.4 Comparison of 5 000 kg/h boilers

Boiler type	Fuel	Length (m)	Diameter (m)	Efficiency (%)	Volumetric heat release (kW/m³)	Steam release rate from water Surface (kg/m² s)
Lancashire	Coal	9.0	2.75	74	340	0.07
Economic	Coal	6.0	3.00	76	730	0.12
Packaged	Oil	3.9	2.50	82	2 330	0.20
Packaged	Gas	3.9	2.50	80	2 600	0.20

Volumetric heat release (kW/m³)

This factor is calculated by dividing the total heat input by the volume of water in the boiler. It effectively relates the quantity of steam released under maximum load to the amount of water in the boiler. The lower this number, the greater the amount of reserve energy in the boiler.

Note that the figure for a modern boiler relative to a Lancashire boiler, is larger by a factor of almost eight, indicating a reduction in stored energy by a similar amount. This means that a reduced amount of stored energy is available in a modern boiler. This development has been made possible by control systems which respond quickly and with appropriate actions to safeguard the boiler and to satisfy demand.

Steam release rate (kg/m² s)

This factor is calculated by dividing the amount of steam produced per second by the area of the water plane. The lower this number, the greater the opportunity for water particles to separate from the steam and produce dry steam.

Note the modern boiler's figure is larger by a factor of almost three. This means that there is less opportunity for the separation of steam and water droplets.

This is made much worse by water with a high TDS level, and accurate control is essential for efficiency and the production of dry steam.

At times of rapidly increasing load, the boiler will experience a reduction of pressure, which, in turn, means that the density of the steam is reduced, and even higher steam release rates will occur, and progressively wetter steam is exported from the boiler.

Four-pass boilers

Four-pass units are potentially the most thermally efficient, but fuel type and operating conditions may prevent their use. When this type of unit is fired at low demand with heavy fuel oil or coal, the heat transfer from the combustion gases can be very large. As a result, the exit flue gas temperature can fall below the acid dew point, causing corrosion of the flues and chimney and possibly of the boiler itself. The four-pass boiler unit is also subject to higher thermal stresses, especially if large load swings suddenly occur; these can lead to stress cracks or failures within the boiler structure. For these reasons, four-pass boilers are unusual.

Reverse flame/thimble boiler

This is a variation on conventional boiler design. The combustion chamber is in the form of a thimble, and the burner fires down the centre. The flame doubles back on itself within the combustion chamber to come to the front of the boiler. Smoke tubes surround the thimble and pass the flue gases to the rear of the boiler and the chimney.

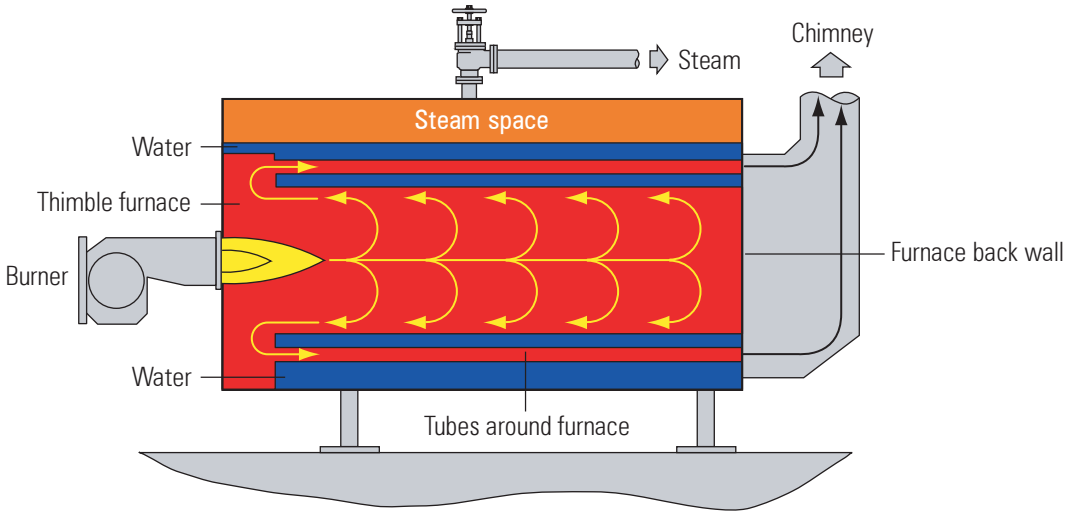


Fig. 3.2.6 Thimble or reverse flame boiler

Pressure and output limitations of shell type boilers

The stresses that may be imposed on the boiler are limited by national standards. Maximum stress will occur around the circumference of a cylinder. This is called 'hoop' or 'circumferential' stress. The value of this stress can be calculated using Equation 3.2.1:

$$\sigma = \frac{P\,D}{2\,\varkappa} \qquad \textbf{Equation 3.2.1}$$

Where:
σ = Hoop stress (N/m²)
P = Boiler pressure (N/m² = bar x 10^5)
D = Diameter of cylinder (m)
$\varkappa$ = Plate thickness (m)

From this it can be deduced that hoop stress increases as diameter increases. To compensate for this the boiler manufacturer will use thicker plate. However, this thicker plate is harder to roll and may need stress relieving with a plate thickness over 32 mm.

One of the problems in manufacturing a boiler is in rolling the plate for the shell. Boilermakers' rolls, as shown in Figures 3.2.7 and 3.2.8, cannot curve the ends of the plate and will, hence, leave a flat:

- Roll **A** is adjusted downwards to reduce radius of the curvature.
- Rolls **B** and **C** are motorised to pull the plate through the rolls.
- The rolls cannot curve the ends of the plate.

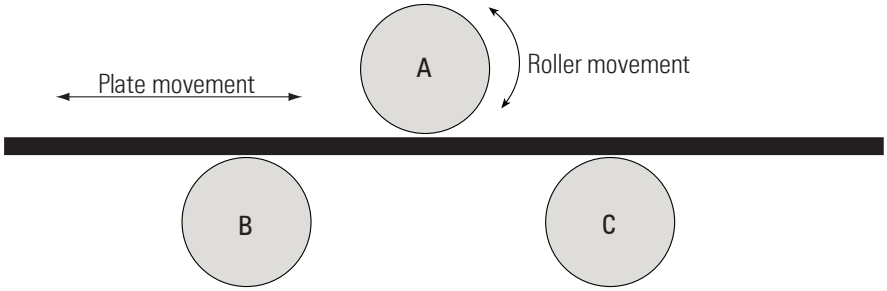


Fig. 3.2.7 Rolling the boiler shell using boilermakers' rolls

When the plates are welded together and the boiler is pressurised, the shell will assume a circular cross section. When the boiler is taken off-line, the plates will revert to the 'as rolled' shape. This cycling can cause fatigue cracks to occur some distance away from the shell welds. It is a cause for concern to boiler inspectors who will periodically ask for all the boiler lagging to be removed and then use a template to determine the accuracy of the boiler shell curvature.

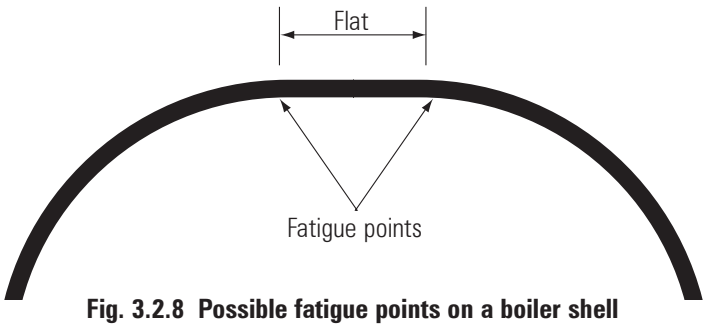


Fig. 3.2.8 Possible fatigue points on a boiler shell

Obviously, this problem is of more concern on boilers that experience a lot of cycling, such as being shutdown every night, and then re-fired every morning.

Pressure limitation

Heat transfer through the furnace tubes is by conduction. It is natural that thick plate does not conduct heat as quickly as thin plate. Thicker plate is also able to withstand more force.

This is of particular importance in the furnace tubes where the flame temperature may be up to 1 800°C, and a balance must be struck between:

- A thicker plate, which has the structural strength to withstand the forces generated by pressure in the boiler.
- A thinner plate, which has the ability to transfer heat more quickly.

The equation that connects plate thickness to structural strength is Equation 3.2.1:

$$\sigma = \frac{P\ D}{2\ \varkappa} \qquad \textbf{Equation 3.2.1}$$

Where:
σ = Hoop stress (N/m²)
P = Boiler pressure (N/m² = bar x 10^5)
D = Diameter of cylinder (m)
$\varkappa$ = Plate thickness (m)

Equation 3.2.1 shows that as the plate thickness gets less, the stress increases for the same boiler pressure.

The equation that connects plate thickness to heat transfer is Equation 2.5.1:

$$\dot{Q} = k\ A \frac{\Delta T}{\varkappa} \qquad \textbf{Equation 2.5.1}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
A = Heat transfer area (m²)
k = Thermal conductivity of the material (W/m K or W/m°C)
ΔT = Temperature difference across the material (K or °C)
$\varkappa$ = Material thickness (m)

Equation 2.5.1 shows that as the plate thickness gets less, the heat transfer increases. By transposing both equations to reflect the plate thickness.

$$\varkappa = \frac{P\ D}{2\ \sigma}$$

$$\varkappa = \frac{k\ A\ \Delta T}{\dot{Q}}$$

By equating Equation 3.2.1 to Equation 3.5.1:

$$\frac{P\ D}{2\ \sigma} = \frac{k\ A\ \Delta T}{\dot{Q}}$$

$$P = \frac{2\ \sigma\ k\ A\ \Delta T}{\dot{Q}\ D}$$

For the same boiler, σ; k; A; and D are constant and, as ΔT is directly proportional to P, it can be said that:

$$P \propto \frac{1}{\dot{Q}}$$ **Equation 3.2.2**

Where:
P = Boiler pressure (N/m^2 = bar x 10^5)
$\dot{Q}$ = Heat transfer rate (kW)

For any one boiler, if the heat transfer rate ($\dot{Q}$) is increased, the maximum allowable boiler pressure is reduced.

A compromise is reached with a furnace tube wall thickness of between 18 mm and 20 mm. This translates to a practical pressure limit for shell boilers of around 27 bar.

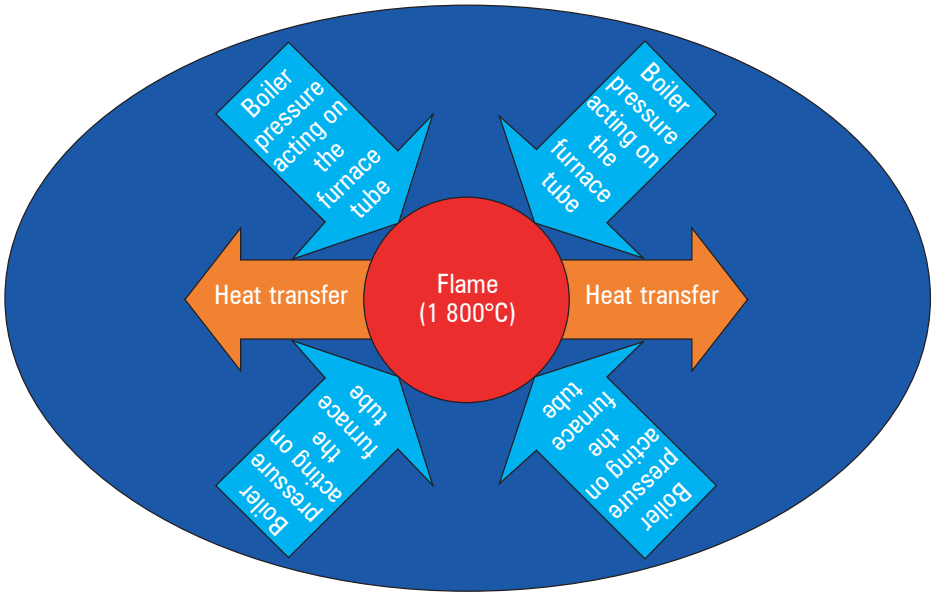


Fig. 3.2.9 Heat transfer from the furnace tube

Output limitation

Shell boilers are manufactured as packaged units with all the ancillary equipment fixed into position. After manufacture, the packaged boiler must be transported to site and the largest boiler which can be transported by road in the UK has an output of around 27 000 kg / h.

If more than 27 000 kg / h is required, then multi-boiler installations are used. However, this has the advantage of providing better security of supply and improved plant turndown.



Fig. 3.2.10 Road transportation

Summary

Today's highly efficient and responsive shell boiler is the result of more than 150 years of development in:

- Boiler and burner design.
- Material science.
- Boiler manufacturing techniques.
- Control systems.

To guarantee its successful and efficient operation, the user must:

- Know the conditions, environment, and demand characteristics of the plant, and accurately specify these conditions to the boiler manufacturer.
- Provide a boiler house layout and installation that promotes good operation and maintenance.
- Select the control systems that allow the boiler to operate safely and efficiently.
- Select the control systems that will support the boiler in supplying dry steam to the plant at the required pressure(s) and flowrate(s).
- Identify the fuel to be used and, if necessary, where and how the fuel reserve is to be safely stored.

Advantages of shell boilers:

- The entire plant may be purchased as a complete package, only needing securing to basic foundations, and connecting to water, electricity, fuel and steam systems before commissioning. This means that installation costs are minimised.
- This package arrangement also means that it is simple to relocate a packaged shell boiler.
- A shell boiler contains a substantial amount of water at saturation temperature, and hence has a substantial amount of stored energy which can be called upon to cope with short term, rapidly applied loads.

  This can also be a disadvantage in that when the energy in the stored water is used, it may take some time before the reserve is built up again.
- The construction of a shell boiler is generally straight forward, which means that maintenance is simple.
- Shell boilers often have one furnace tube and burner. This means that control systems are fairly simple.
- Although shell boilers may be designed and built to operate up to 27 bar, the majority operate at 17 bar or less. This relatively low pressure means that the associated ancillary equipment is easily available at competitive prices.

Disadvantages of shell boilers:

- The package principle means that approximately 27 000 kg / h is the maximum output of a shell boiler. If more steam is required, then several boilers need to be connected together.
- The large diameter cylinders used in the construction of shell boilers effectively limit their operating pressure to approximately 27 bar. If higher pressures are needed, then a water-tube boiler is required.

Questions

1. What is one advantage of a Lancashire boiler over a modern packaged boiler?

a| It has a higher efficiency ☐

b| Manual control of the boiler means closer control ☐

c| The larger size means it can respond faster to load changes ☐

d| It can tolerate sudden demands for steam more easily because of the formation of flash steam ☐

2. Typically, which type of boiler gives the greatest efficiency?

a| Lancashire ☐

b| Packaged boiler oil fired ☐

c| Economic ☐

d| Packaged boiler gas fired ☐

3. Why is the largest packaged boiler limited to 27 000 kg/h?

a| Above this the efficiency is reduced ☐

b| Above this the road transport becomes impractical ☐

c| Above this the control becomes difficult ☐

d| Stress limitations prevent the use of larger boilers ☐

4. What proportion of total heat is transferred in the first pass of a three-pass economic boiler?

a| 25% ☐

b| 55% ☐

c| 65% ☐

d| 80% ☐

5. A lower steam release rate (kg/m² s) means:

a| A greater opportunity for dry steam ☐

b| Wetter steam ☐

c| Greater energy reserves in the boiler ☐

d| The blowdown rate can be lower ☐

6. Boilers need to be brought slowly up to working conditions from cold to:

a| Produce drier steam ☐

b| Reduce TDS in the boiler ☐

c| Reduce hoop stress ☐

d| Reduce fatigue cracks in the boiler shell ☐

Answers

1: d, 2: d, 3: b, 4: c, 5: a, 6: d

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Module 3.3

Water-tube Boilers

Water-tube Boilers

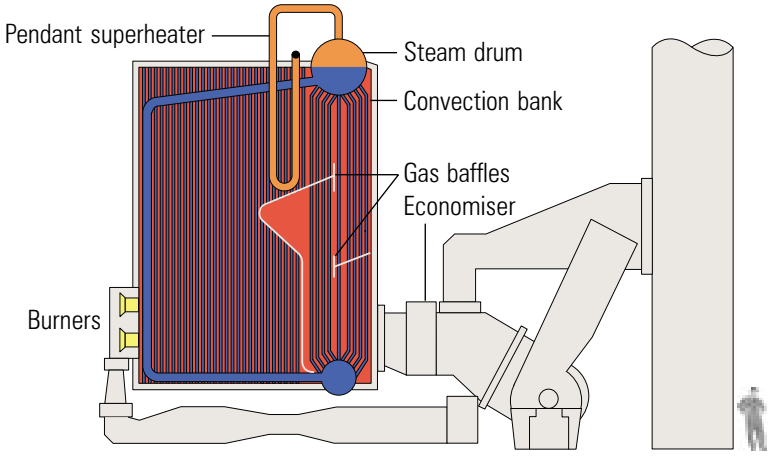


Fig. 3.3.1 Water-tube boiler

Water-tube boilers differ from shell type boilers in that the water is circulated inside the tubes, with the heat source surrounding them. Referring back to the equation for hoop stress (Equation 3.2.1), it is easy to see that because the tube diameter is significantly smaller, much higher pressures can be tolerated for the same stress.

Water-tube boilers are used in power station applications that require:

- A high steam output (up to 500 kg/s).
- High pressure steam (up to 160 bar).
- Superheated steam (up to 550°C).

However, water-tube boilers are also manufactured in sizes to compete with shell boilers.

Small water-tube boilers may be manufactured and assembled into a single unit, just like packaged shell boilers, whereas large units are usually manufactured in sections for assembly on site.

Many water-tube boilers operate on the principle of natural water circulation (also known as 'thermo-siphoning'). This is a subject that is worth covering before looking at the different types of water-tube boilers that are available. Figure 3.3.2 helps to explain this principle:

- Cooler feedwater is introduced into the steam drum behind a baffle where, because the density of the cold water is greater, it descends in the 'downcomer' towards the lower or 'mud' drum, displacing the warmer water up into the front tubes.
- Continued heating creates steam bubbles in the front tubes, which are naturally separated from the hot water in the steam drum, and are taken off.

However, when the pressure in the water-tube boiler is increased, the difference between the densities of the water and saturated steam falls, consequently less circulation occurs. To keep the same level of steam output at higher design pressures, the distance between the lower drum and the steam drum must be increased, or some means of forced circulation must be introduced.

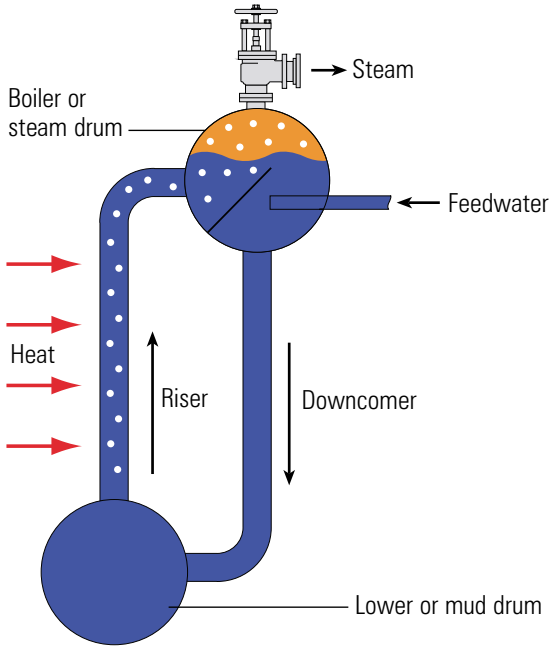


Fig. 3.3.2
Natural water circulation in a water-tube boiler

Water-tube boiler sections

The energy from the heat source may be extracted as either radiant or convection and conduction.

The furnace or radiant section

This is an open area accommodating the flame(s) from the burner(s). If the flames were allowed to come into contact with the boiler tubes, serious erosion and finally tube failure would occur.

The walls of the furnace section are lined with finned tubes called membrane panels, which are designed to absorb the radiant heat from the flame.

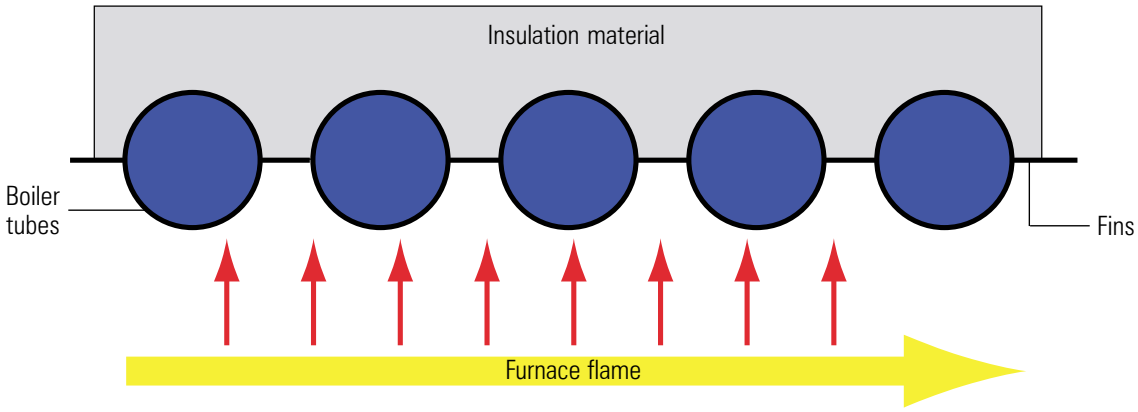


Fig. 3.3.3 Heat transfer in the furnace or radiant section

Convection section

This part is designed to absorb the heat from the hot gases by conduction and convection.

Large boilers may have several tube banks (also called pendants) in series, in order to gain maximum energy from the hot gases.

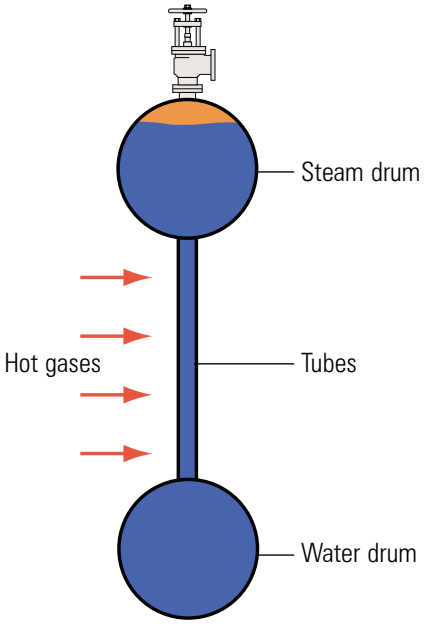


Fig. 3.3.4 Heat transfer in the convection section

Water-tube boiler designation

Water-tube boilers are usually classified according to certain characteristics, see Table 3.3.1.

Table 3.3.1 Water-tube boiler classifications

Reservoir drum position	For example, longitudinal or cross drum
Water circulation	For example, natural or forced
Number of drums	For example, two, three
Capacity	For example, 25 500 kg/h, 7 kg/s, 55 000 lb/h

Alternative water-tube boiler layouts

The following layouts work on the same principles as other water-tube boilers, and are available with capacities from 5 000 kg/h to 180 000 kg/h.

Longitudinal drum boiler

The longitudinal drum boiler was the original type of water-tube boiler that operated on the thermo-siphon principle (see Figure 3.3.5).

Cooler feedwater is fed into a drum, which is placed longitudinally above the heat source. The cooler water falls down a rear circulation header into several inclined heated tubes. As the water temperature increases as it passes up through the inclined tubes, it boils and its density decreases, therefore circulating hot water and steam up the inclined tubes into the front circulation header which feeds back to the drum. In the drum, the steam bubbles separate from the water and the steam can be taken off.

Typical capacities for longitudinal drum boilers range from 2 250 kg/h to 36 000 kg/h.

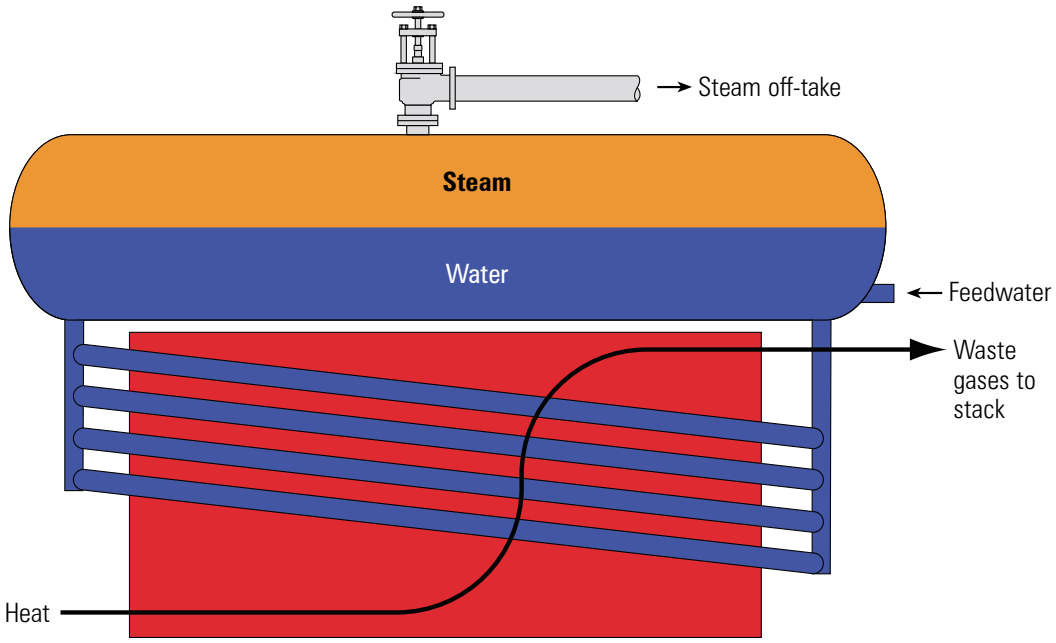


Fig. 3.3.5 Longitudinal drum boiler

Cross drum boiler

The cross drum boiler is a variant of the longitudinal drum boiler in that the drum is placed cross ways to the heat source as shown in Figure 3.3.6. The cross drum operates on the same principle as the longitudinal drum except that it achieves a more uniform temperature across the drum. However it does risk damage due to faulty circulation at high steam loads; if the upper tubes become dry, they can overheat and eventually fail.

The cross drum boiler also has the added advantage of being able to serve a larger number of inclined tubes due to its cross ways position.

Typical capacities for a cross drum boiler range from 700 kg/h to 240 000 kg/h.

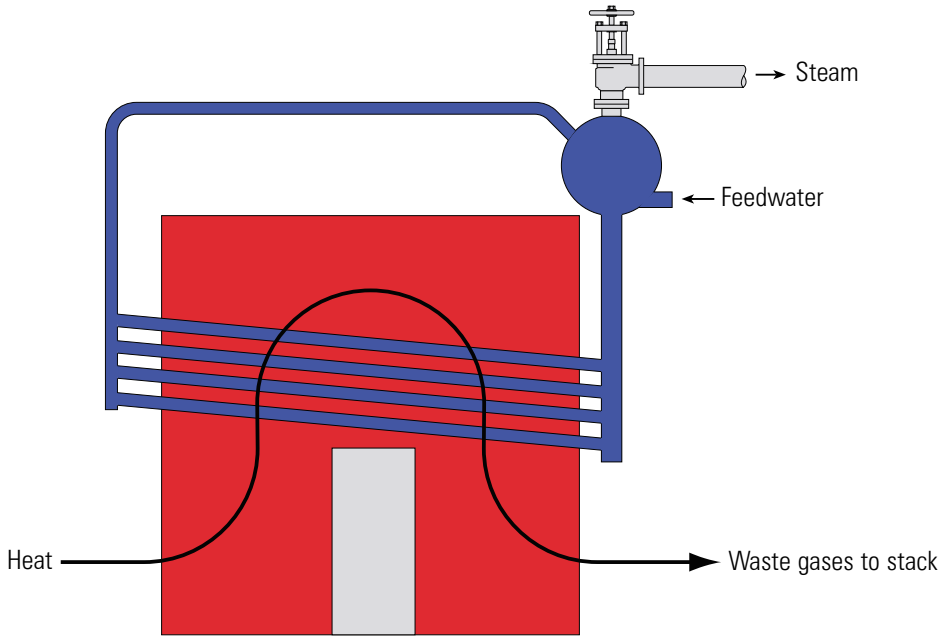


Fig. 3.3.6 Cross drum boiler

Bent tube or Stirling boiler

A further development of the water-tube boiler is the bent tube or Stirling boiler shown in Figure 3.3.7. Again this operates on the principle of the temperature and density of water, but utilises four drums in the following configuration.

Cooler feedwater enters the left upper drum, where it falls due to greater density, towards the lower, or water drum. The water within the water drum, and the connecting pipes to the other two upper drums, are heated, and the steam bubbles produced rise into the upper drums where the steam is then taken off.

The bent tube or Stirling boiler allows for a large surface heat transfer area, as well as promoting natural water circulation.

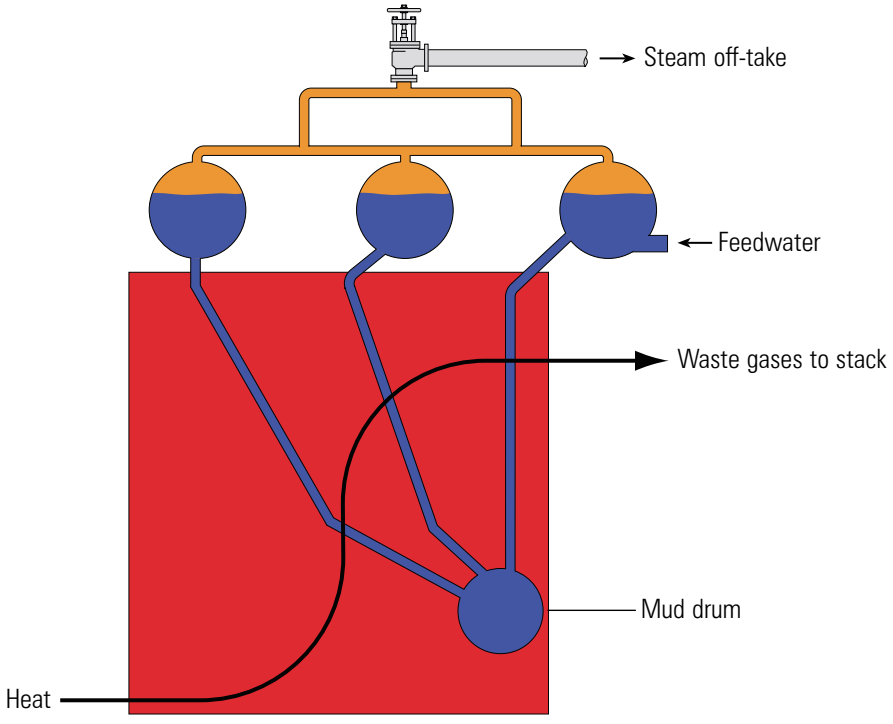


Fig. 3.3.7 Bent tube or Stirling boiler

Advantages of water-tube boilers:

- They have a small water content, and therefore respond rapidly to load change and heat input.
- The small diameter tubes and steam drum mean that much higher steam pressures can be tolerated, and up to 160 bar may be used in power stations.
- The design may include many burners in any of the walls, giving horizontal, or vertical firing options, and the facility of control of temperature in various parts of the boiler. This is particularly important if the boiler has an integral superheater, and the temperature of the superheated steam needs to be controlled.

Disadvantages of water-tube boilers:

- They are not as simple to make in the packaged form as shell boilers, which means that more work is required on site.
- The option of multiple burners may give flexibility, but the 30 or more burners used in power stations means that complex control systems are necessary.

Combined heat and power (CHP) plant

The water-tube boilers described above are usually of a large capacity. However, small, special purpose, smaller waste heat boilers to be used in conjunction with land based gas turbine plants are in increasing demand.

Several types of steam generating land based gas turbine plant are used:

- **Combined heat and power -** These systems direct the hot exhaust gases from a gas turbine (approximately 500°C) through a boiler, where saturated steam is generated and used as a plant utility.

  Typical applications for these systems are on plant or sites where the demands for electricity and steam are in step and of proportions which can be matched to a CHP system.

  Efficiencies can reach 90%.

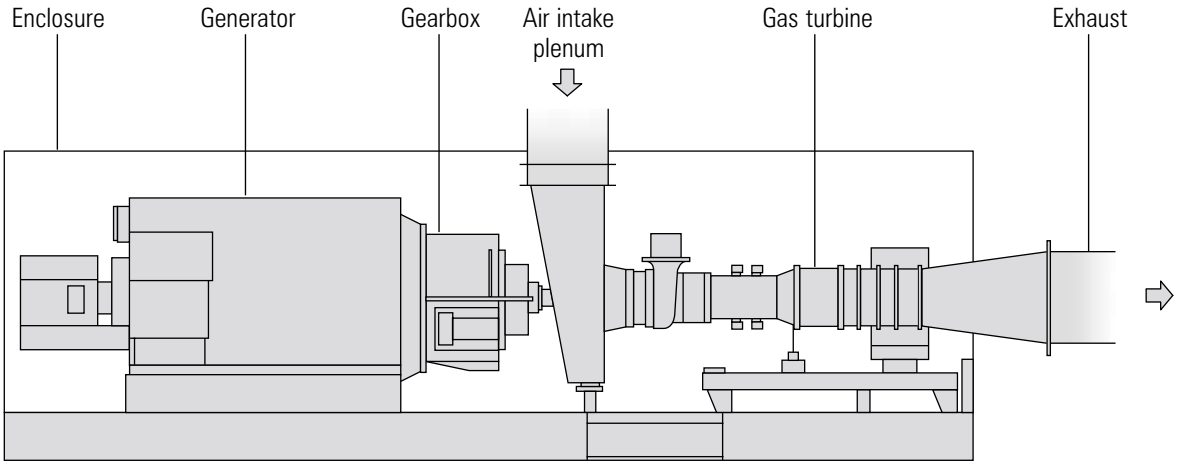


Fig. 3.3.8 Gas turbine/alternator set

- **Combined cycle plant -** These are extensions to CHP systems, and the saturated steam is taken through a superheater to produce superheated steam. The superheater may be separately fired because of the comparatively low temperature of the gas turbine exhaust. The superheated steam produced is directed to steam turbines which drive additional alternators, and generate electricity.

  The turndown ratio of these plants is poor, because of the need for the turbine to rotate at a speed synchronised to the electrical frequency. This means that it is only practical to run these plants at full-load, providing the base load of steam to the plant.

  Because of the relatively low temperature of the gas turbine exhaust, compared to the burner flame in a conventional boiler, a much greater boiler heat transfer area is required for a given heat load. Also, there is no need to provide accommodation for burners. For these reasons, water-tube boilers tend to provide a better and more compact solution. Because efficiency is a major factor with CHP decision-makers, the design of these boilers may well incorporate an economiser (feedwater heater).

  If the plant is 'combined cycle' the design may also include a superheater. However, the relatively low temperatures may mean that additional burners are required to bring the steam up to the specification required for the steam turbines.

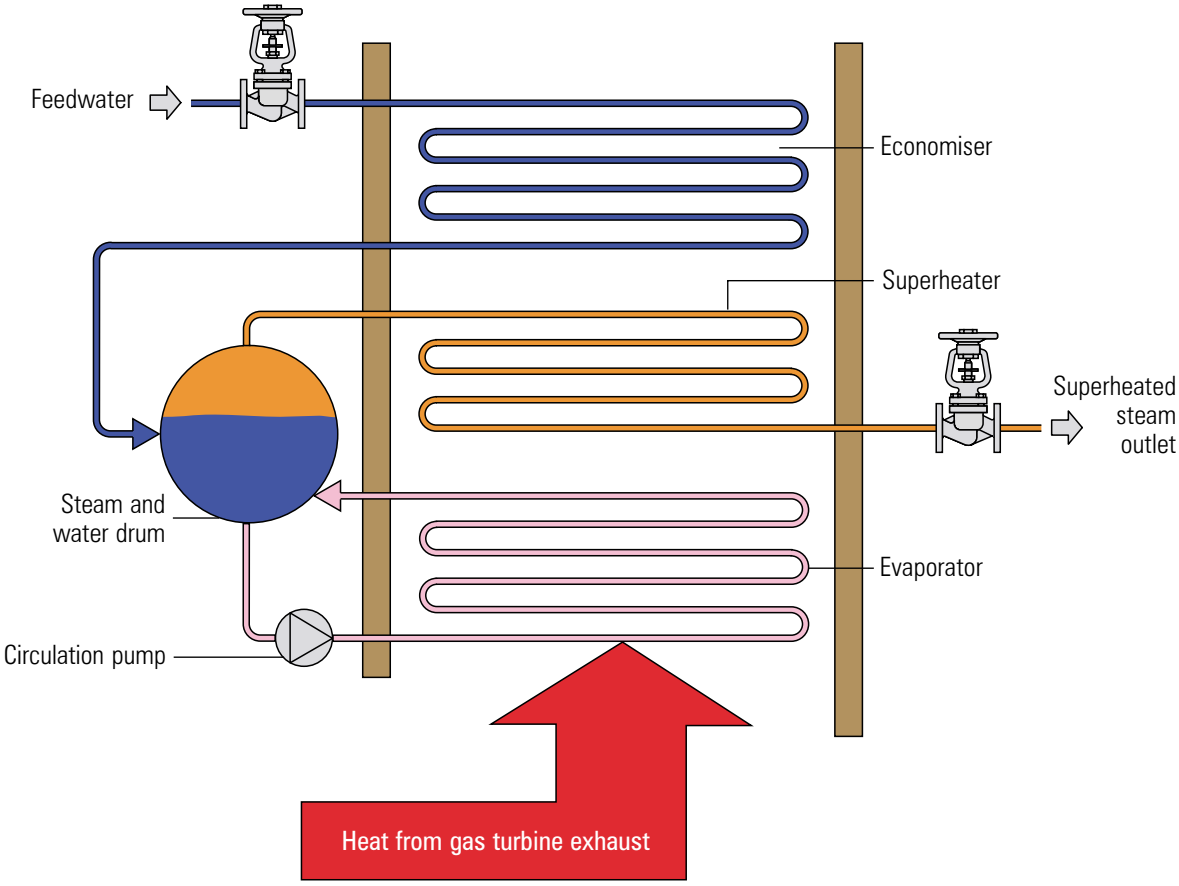


Fig. 3.3.9 A forced circulation water-tube boiler as used on CHP plant

Questions

1. Why can higher pressure steam be produced in a water-tube boiler compared with a shell boiler ?

a| A superheater is incorporated in a water-tube boiler ☐

b| Water-tube boilers incorporate a radiant and convection section ☐

c| In a water-tube boiler the water is in tubes and a higher stress and pressure can be accepted ☐

d| Water-tube boilers have a greater heat transfer surface ☐

2. Which of the following is a disadvantage of a water-tube boiler compared to a shell boiler ?

a| They have a lower water content ☐

b| They are more difficult to control because of the number of burners ☐

c| They are physically much larger ☐

d| It is more difficult to produce superheated steam in a water-tube boiler ☐

3. Why are water-tube boilers typically used in power stations ?

a| Ease of temperature turndown as load changes ☐

b| They are flexible to rapid load changes ☐

c| Because of their pressure, capacity and the degree of superheat ☐

d| Because the body of a water-tube boiler can accept a higher stress than a shell boiler ☐

4. Which of the following is a disadvantage of a cross drum boiler ?

a| It does not permit superheating ☐

b| It doesn't incorporate a mud drum ☐

c| Due to having an external steam drum steam quality can be poor ☐

d| Faulty circulation can occur at high steam loads ☐

5. What is the advantage of a CHP system ?

a| Saturated steam is produced from waste gases ☐

b| The system is at least 90% efficient ☐

c| The steam produced is a by-product of power generation ☐

d| All of the above ☐

6. Which of the following is a disadvantage of a gas turbine/alternator set ?

a| The turndown ratio is poor ☐

b| The superheater always needs separate firing ☐

c| Because of the low gas temperature only low pressure steam can be produced ☐

d| The superheated steam produced is unsuitable for driving another generator ☐

Answers

1: c, 2: b, 3: c, 4: d, 5: d, 6: a

Module 3.4

Miscellaneous Boiler Types, Economisers and Superheaters

Miscellaneous Boiler Types, Economisers and Superheaters

Steam generators

In many applications:

- The amount of steam required is too small to warrant a shell boiler, i.e. Less than 1 000 kg/h.
- The small process requiring steam operates on a day shift only, meaning that the plant would be started every morning and shut down every night.
- The capital cost of a conventional shell boiler would adversely affect the economic viability of the process.
- The level of expertise on site, as far as boilers are concerned, is not as high as would be required on a larger steam system.

To meet these specific demands two types of boiler have been developed.

Coil boiler

These are a 'once through' type of water tube boiler, and referred to in some regulations as, 'boilers with no discernible water level'.

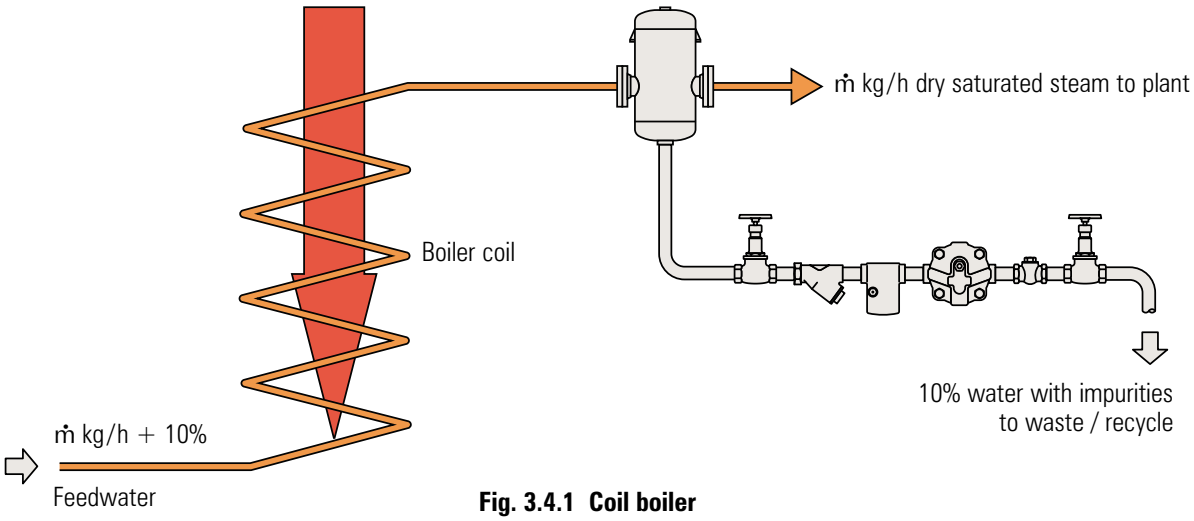


Fig. 3.4.1 Coil boiler

Water supply to the boiler will usually be at 10 to 15% above the steaming rate to:

- Ensure that all the water is not evaporated, thus ensuring that superheated steam is not produced.
- Provide a vehicle for the feedwater TDS to be carried through. If this vehicle was not available, the salts in the feedwater would be deposited on the insides of the tubes and impair heat transfer, leading to over heating and eventually to tube failure. Clearly, a separator is an essential component of this type of boiler to remove this contaminated water.

Being of the water tube type, they can produce steam at very high pressures.

Typical applications for steam generators and coil boilers include laundries and garment manufacture, where the demand is small and the rate of change in load is slow.

Vertical tubeless packaged steam boiler

Various models are available with outputs in the range 50 to 1 000 kg/h, and pressures up to 10 bar g. Boiler heights vary typically from 1.7 m to 2.4 m for outputs of about 100 kg/h to 1 000 kg/h respectively.

A cross section of the design is shown in Figure 3.4.2. Note the downward path of the flame, and the swirling action. The heat path is reversed at the bottom of the boiler and the hot gases rise, releasing heat to the fins.

Also note the small quantity of water in the boiler. This allows the boiler to be brought up to operating temperature very quickly, typically 15 minutes. However, this small quantity of water means that only a small amount of energy is stored in the boiler, consequently it is not easily able to cope with sudden and maintained changes in load. If the load change occurs faster than the boiler can respond, then the pressure inside the boiler will drop and ultimately the boiler will prime with feedwater. This is aggravated by the small water surface area, which gives high steam release velocities. However, the path of the steam is vertically up and away from the water surface as opposed to horizontally over the water surface (as in a shell boiler), and this minimises the effect

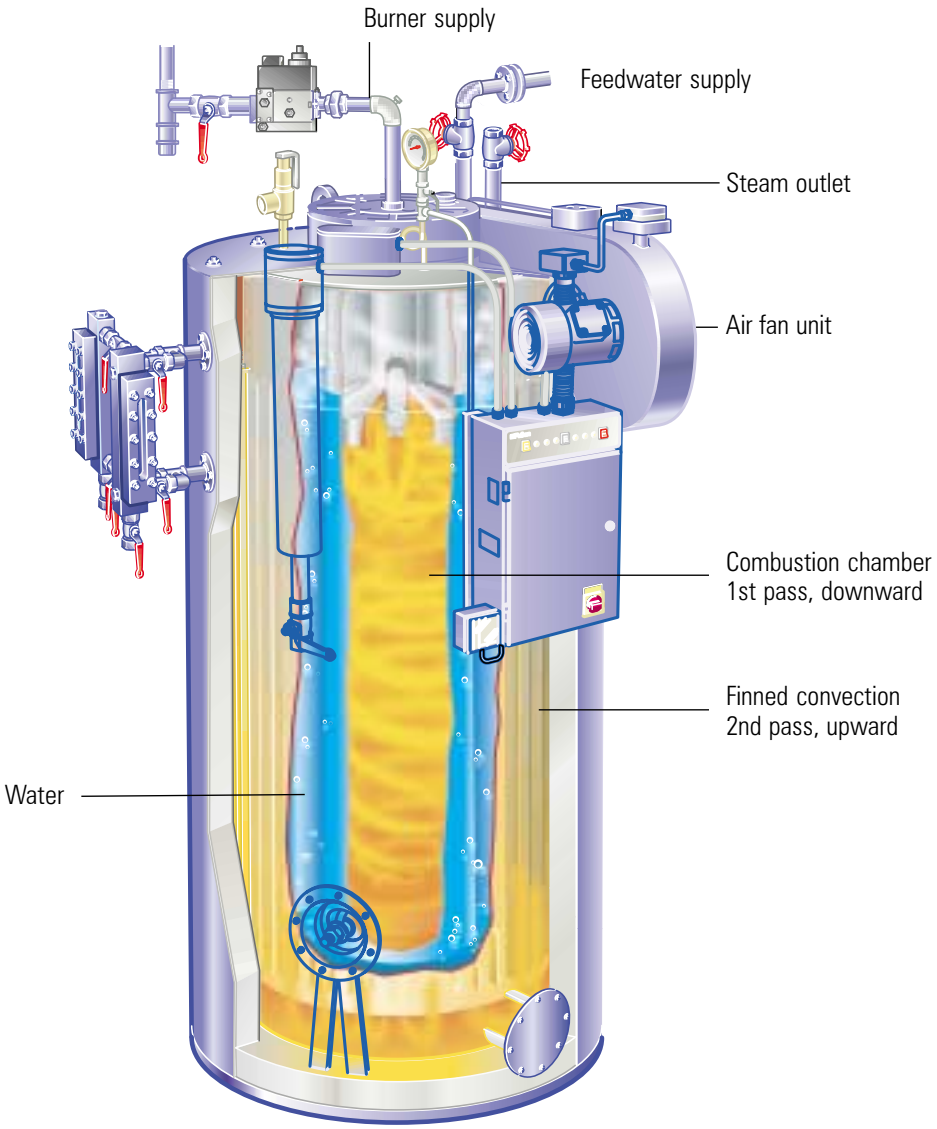


Fig. 3.4.2 Vertical tubeless packaged steam boiler

Economisers

The flue gases, having passed through the main boiler and the superheater, will still be hot. The energy in these flue gases can be used to improve the thermal efficiency of the boiler. To achieve this the flue gases are passed through an economiser.

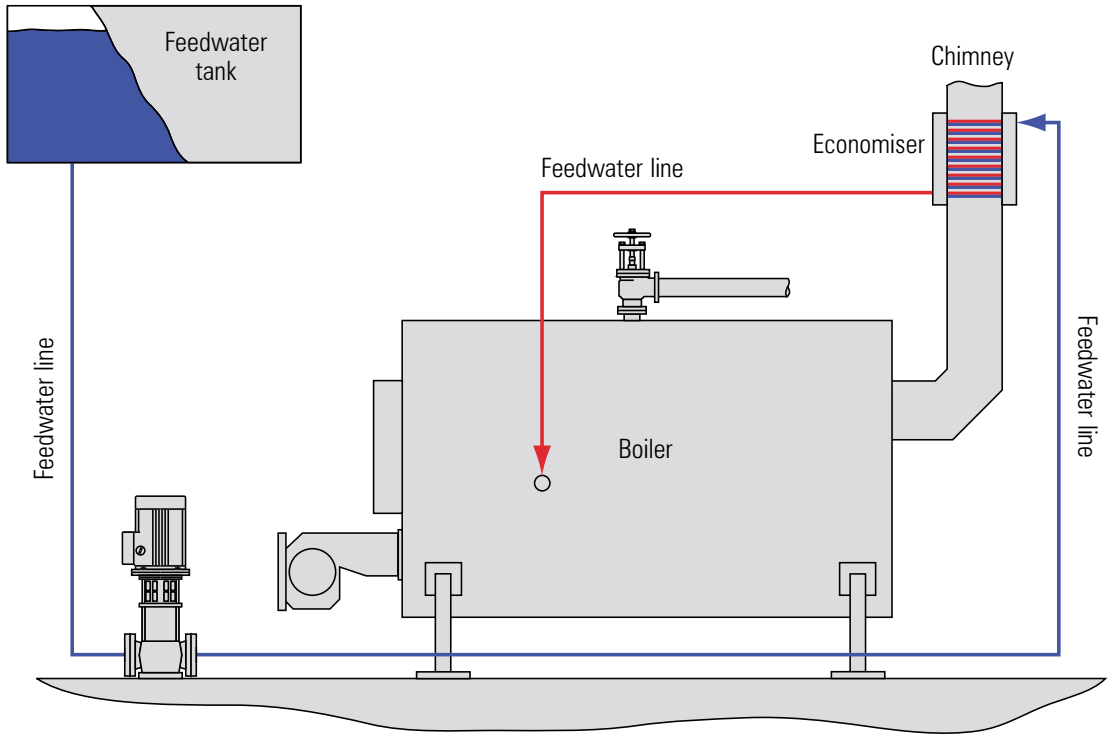


Fig. 3.4.3 A shell boiler with an economiser

The economiser is a heat exchanger through which the feedwater is pumped. The feedwater thus arrives in the boiler at a higher temperature than would be the case if no economiser was fitted. Less energy is then required to raise the steam. Alternatively, if the same quantity of energy is supplied, then more steam is raised. This results in a higher efficiency. In broad terms a 10°C increase in feedwater temperature will give an efficiency improvement of 2%.

Note:

- Because the economiser is on the high-pressure side of the feedpump, feedwater temperatures in excess of 100°C are possible. The boiler water level controls should be of the 'modulating' type, (i.e. not 'on-off') to ensure a continuous flow of feedwater through the heat exchanger.
- The heat exchanger should not be so large that:
  - The flue gases are cooled below their dew point, as the resulting liquor may be acidic and corrosive.
  - The feedwater boils in the heat exchanger.

Superheaters

Whatever type of boiler is used, steam will leave the water at its surface and pass into the steam space. Steam formed above the water surface in a shell boiler is always saturated and cannot become superheated in the boiler shell, as it is constantly in contact with the water surface.

If superheated steam is required, the saturated steam must pass through a superheater. This is simply a heat exchanger where additional heat is added to the saturated steam.

In water-tube boilers, the superheater may be an additional pendant suspended in the furnace area where the hot gases will provide the degree of superheat required (see Figure 3.4.4). In other cases, for example in CHP schemes where the gas turbine exhaust gases are relatively cool, a separately fired superheater may be needed to provide the additional heat.

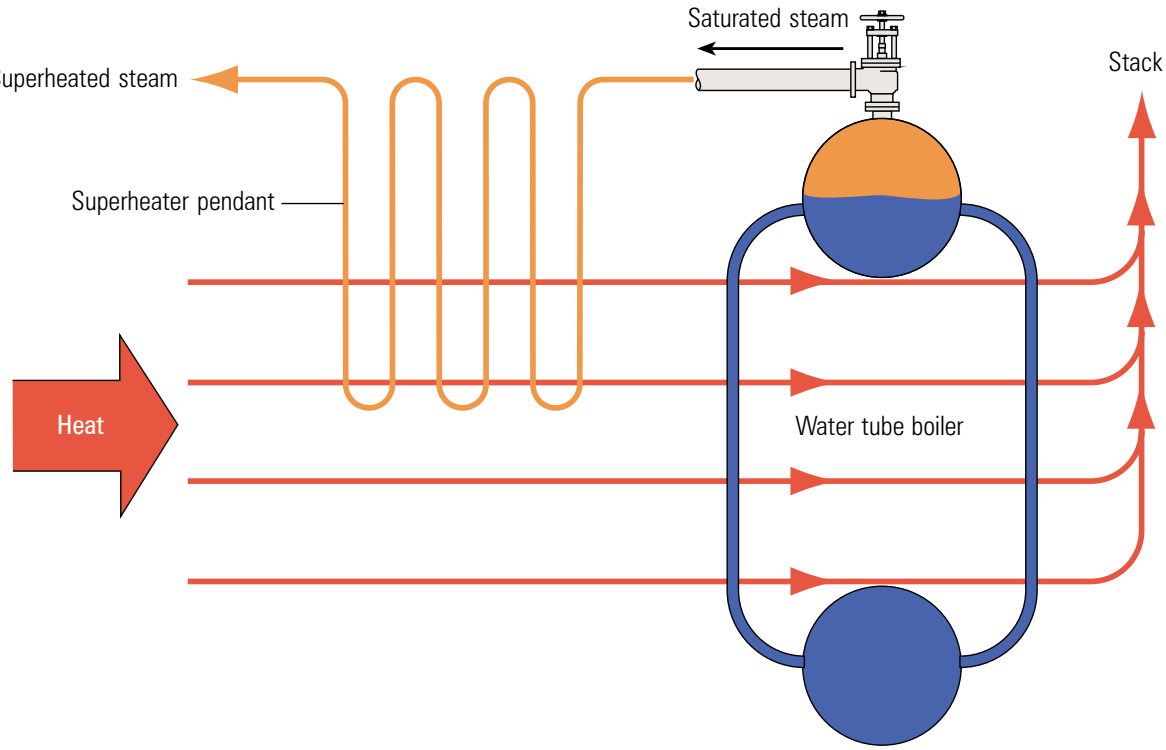


Fig. 3.4.4 A water tube boiler with a superheater

If accurate control of the degree of superheat is required, as would be the case if the steam is to be used to drive turbines, then an attemperator (desuperheater) is fitted. This is a device installed after the superheater, which injects water into the superheated steam to reduce its temperature.

Questions

1. What is the main advantage of a vertical tubeless packaged steam boiler when compared with a shell boiler ?

a| There is little water stored in the boiler ☐

b| Water level controls are not required ☐

c| Steam can be raised in 15 minutes ☐

d| It is quick to respond to steam load changes ☐

2. From the following identify a reason why the water supply rate to a coil boiler is 10% greater than the steam requirement ?

a| The excess water is a vehicle for the feedwater TDS to be carried through ☐

b| To even out stresses within the boiler ☐

c| It is easier to control the degree of superheat in the steam produced ☐

d| It is easier to control the water flowrate ☐

3. How is a dry steam supply assured from a coil boiler package ?

a| Through an intermittent water supply ☐

b| It isn't, the steam will be wet ☐

c| By using a superheater ☐

d| By using a separator ☐

4. What effect can a rapid load change have on a vertical tubeless packaged steam boiler ?

a| No effect ☐

b| The water level will drop ☐

c| The boiler will react quickly ☐

d| The boiler will prime with feedwater ☐

5. What is the purpose of an economiser ?

a| To cool boiler exhaust gases to below dew point ☐

b| To reduce the amount of energy required in the production of steam ☐

c| To enable more steam to be produced from a boiler ☐

d| To utilise heat from boiler exhaust gases ☐

6. Why are superheaters normally associated with water tube boilers rather than with shell boilers ?

a| Control of the degree of superheat is easier with a water tube boiler than a shell boiler ☐

b| Water tube boilers always incorporate superheaters ☐

c| Turbines need high pressure superheated steam and this is more readily available from water tube boilers ☐

d| Because water tube boilers produce wet steam and superheating is therefore usual ☐

Answers

1: c, 2: a, 3: d, 4: d, 5: d, 6: c

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Module 3.5

Boiler Ratings

Boiler Ratings

Three types of boiler ratings are commonly used:

- 'From and at' rating.
- kW rating.
- Boiler horsepower (BoHP).

'From and at' rating

The 'from and at' rating is widely used as a datum by shell boiler manufacturers to give a boiler a rating which shows the amount of steam in kg/h which the boiler can create 'from and at 100°C', at atmospheric pressure. Each kilogram of steam would then have received 2 257 kJ of heat from the boiler.

Shell boilers are often operated with feedwater temperatures lower than 100°C. Consequently the boiler is required to supply enthalpy to bring the water up to boiling point.

Most boilers operate at pressures higher than atmospheric, because steam at an elevated pressure carries more heat energy than does steam at 100°C. This calls for additional enthalpy of saturation of water. As the boiler pressure rises, the saturation temperature is increased, needing even more enthalpy before the feedwater is brought up to boiling temperature.

Both these effects reduce the actual steam output of the boiler, for the same consumption of fuel. The graph in Figure 3.5.1 shows feedwater temperatures plotted against the percentage of the 'from and at' figure for operation at pressures of 0, 5, 10 and 15 bar g.

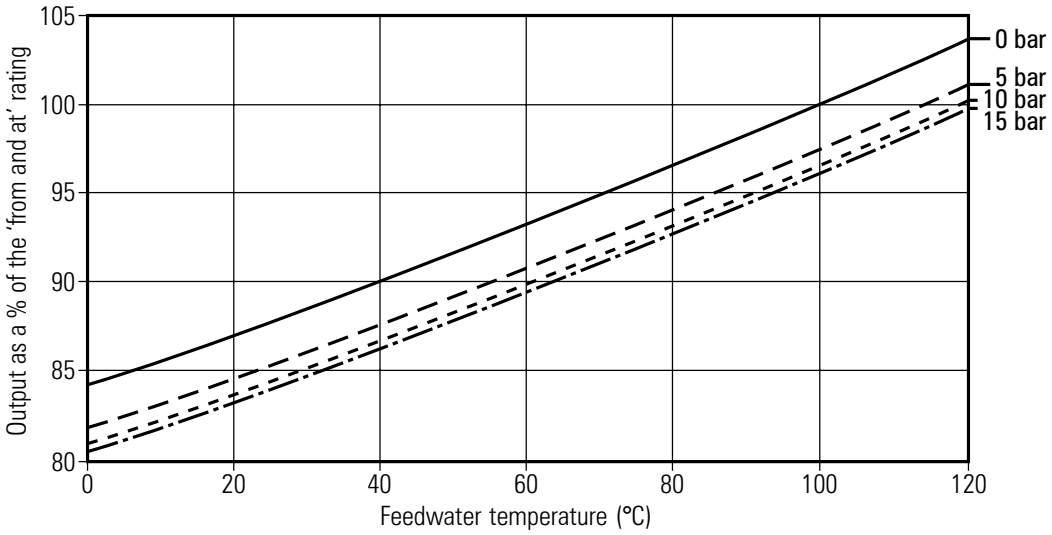


Fig. 3.5.1 'From and at' graph

The application of the 'from and at' rating graph (Figure 3.5.1) is shown in Example 3.5.1, as well as a demonstration of how the values are determined.

Example 3.5.1
A boiler has a 'from and at' rating of 2 000 kg/h and operates at 15 bar g. The feedwater temperature is 68°C.

Using the graph:

The percentage 'from and at' rating ≈ 90%

Therefore actual output = 2 000 kg/h x 90%

Boiler evaporation rate = 1 800 kg/h

The use of Equation 3.5.1 will determine a factor to produce the same result:

$$\text{Evaporation factor} = \frac{A}{B - C} \qquad \textbf{Equation 3.5.1}$$

Where:
A = Specific enthalpy of evaporation at atmospheric pressure.
B = Specific enthalpy of steam at operating pressure.
C = Specific enthalpy of water at feedwater temperature.

Note: These values are all from steam tables.

Using the information from Example 3.5.1 and the Equation 3.5.1 the evaporation factor can be calculated:

$$\text{Evaporation factor} = \frac{2\ 257\ \text{kJ/kg}}{2\ 794\ \text{kJ/kg} - 284.9\ \text{kJ/kg}}$$

$$\textbf{Evaporation factor} = \mathbf{0.9}$$

Therefore: boiler evaporation rate = 2 000 kg/h x 0.9

$$\textbf{Boiler evaporation rate} = \mathbf{1\ 800\ kg/h}$$

kW rating

Some manufacturers will give a boiler rating in kW. This is not an evaporation rate, and is subject to the same 'from and at' factor.

To establish the actual evaporation by mass, it is first necessary to know the temperature of the feedwater and the pressure of the steam produced, in order to establish how much energy is added to each kg of water. Equation 3.5.2 can then be used to calculate the steam output:

$$\text{Steam output (kg/h)} = \text{Boiler rating (kW)} \times \frac{3\ 600\ \text{s/h}}{\text{Energy to be added (kJ/kg)}} \qquad \textbf{Equation 3.5.2}$$

Example 3.5.2
A boiler is rated at 3 000 kW rating and operates at 10 bar g with a feedwater temperature of 50°C. How much steam can be generated ?

Where, using steam tables:
Feedwater h_f = 4.19 kJ/kg°C
Steam h_g = 2 782 kJ/kg

$$\text{Energy content of feedwater at 50°C} = 50°C \times 4.19\ \text{kJ/kg°C}$$

$$\textbf{Energy content of feedwater at 50°C} = \mathbf{209.5\ kJ/kg}$$

$$\textbf{Energy content of steam at 10 bar g} = \mathbf{2\ 782\ kJ/kg}$$

$$\text{Consequently, the boiler must provide} = 2\ 782 - 209.5$$

$$\textbf{Energy provided by the boiler} = \mathbf{2\ 572.5\ kJ/kg}$$

$$\text{Steam output } (\dot{m}_s) = 3\ 000\ \text{kJ/s} \times \frac{3\ 600\ \text{s/h}}{2\ 572.5\ \text{kJ/kg}}$$

$$\textbf{Steam output} = \mathbf{4\ 198\ kg/h}$$

Boiler horsepower (BoHP)

This unit tends to be used only in the USA, Australia, and New Zealand. A boiler horsepower is not the commonly accepted 550 ft lbf/s and the generally accepted conversion factor of 746 Watts = 1 horsepower does not apply.

In New Zealand, boiler horsepower is a function of the heat transfer area in the boiler, and a boiler horsepower relates to 17 ft² of heating surface, as depicted in Equation 3.5.3:

$$\text{Heat transfer area (ft}^2\text{)} \times \frac{1}{17} = \text{BoHP} \qquad \textbf{Equation 3.5.3}$$

New Zealand

Example 3.5.3

A boiler has a heat transfer area of 2 500 square feet, how many BoHP is this ?

$$2\,500\ \text{ft}^2 \times \frac{1}{17} = 147\ \text{BoHP}$$

USA and Australia

In the USA and Australia the readily accepted definition of a boiler horsepower is the amount of energy required to evaporate 34.5 lb of water at 212°F atmospheric conditions.

Example 3.5.4

A boiler is rated at 500 BoHP, what is its steam output ?

$$500\ \text{BoHP} \times 34.5\ \text{lb/h} = 17\,250\ \text{lb/h}$$

Important: This is essentially the same as a 'from and at' rating, so using feedwater at lower temperatures and steam at higher pressures will reduce the amount of steam generated. **In practice:** A BoHP figure of 28 to 30 lb/h would be a more realistic maximum continuous rating, taking into account the steam pressure and average feedwater temperatures.

A more practical result would then be:

$$500\ \text{BoHP} \times 28 = 14\,000\ \text{lb/h}$$

Consequently: If 17 250 lb/h of steam is required, a 500 BoHP boiler would be too small, and the user would need to specify a boiler with a rating of:

$$17\,250 \times \frac{1}{28} = 616\ \text{BoHP}$$

Questions

1. A boiler with a 'from and at' rating of 10 000 kg/h operates at 10 bar g and is supplied with feedwater at 85°C.
Which of the following will be the nearest to the actual evaporation rate of the boiler ?

a| 8 210 kg/h ☐
b| 9 320 kg/h ☐
c| 8 240 kg/h ☐
d| 12 166 kg/h ☐

2. A boiler has a 'from and at' rating of 8 000 kg/h and operates at 7 bar g with a feedwater temperature of 70°C.
What is the effect on the actual output if the feedwater temperature is 85°C ?

a| Output remains the same ☐
b| Output reduces ☐
c| Output increases and pressure increases ☐
d| Output increases ☐

3. Referring to Question 2, what change, if any, will there be in the overall energy required to produce the steam ?

a| Overall energy required will remain the same ☐
b| Energy required reduces ☐
c| Energy required increases ☐

4. A boiler is rated at 4 000 kW and operates at 7 bar g with a feedwater temperature of 80°C.
Which of the following will be its actual steam output ?

a| 5 916 kg/h ☐
b| 6 824 kg/h ☐
c| 3 726 kg/h ☐
d| 4 310 kg/h ☐

Answers

1: b, 2: d, 3: a, 4: a

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Module 3.6

Boiler Efficiency and Combustion

Boiler Efficiency and Combustion

This Module is intended to give a very broad overview of the combustion process, which is an essential component of overall boiler efficiency. Readers requiring a more in-depth knowledge are directed towards specialist textbooks and burner manufacturers.

Boiler efficiency simply relates energy output to energy input, usually in percentage terms:

$$\text{Boiler efficiency (\%)} = \frac{\text{Heat exported in steam}}{\text{Heat provided by the fuel}} \times \frac{100}{1} \qquad \textbf{Equation 3.6.1}$$

'Heat exported in steam' and 'Heat provided by the fuel' is covered more fully in the following two Sections.

Heat exported in steam

This is calculated (using the steam tables) from knowledge of:

- The feedwater temperature.
- The pressure at which steam is exported.
- The steam flowrate.

Heat provided by the fuel

Calorific value
This value may be expressed in two ways 'Gross' or 'Net' calorific value.

Gross calorific value
This is the theoretical total of the energy in the fuel. However, all common fuels contain hydrogen, which burns with oxygen to form water, which passes up the stack as steam.

The gross calorific value of the fuel includes the energy used in evaporating this water. Flue gases on steam boiler plant are not condensed, therefore the actual amount of heat available to the boiler plant is reduced.

Accurate control of the amount of air is essential to boiler efficiency:

- Too much air will cool the furnace, and carry away useful heat.
- Too little air and combustion will be incomplete, unburned fuel will be carried over and smoke may be produced.

Table 3.6.1 Fuel oil data

Oil Type - Grade	Gross calorific value (MJ/l)
Light - E	40.1
Medium - F	40.6
Heavy - G	41.1
Bunker - H	41.8

Table 3.6.2 Gas data

Gas Type	Gross calorific value (MJ/m³ at NTP)
Natural	38.0
Propane	93.0
Butane	122.0

Net calorific value

This is the calorific value of the fuel, excluding the energy in the steam discharged to the stack, and is the figure generally used to calculate boiler efficiencies. In broad terms:

$$\text{Net calorific value} \approx \text{Gross calorific value} - 10\%$$

The combustion process:

Fuel		Air	Combustion	Heat
C + H	+	$O_2 + N_2$	⟶	$CO_2 + H_2O + N_2$

Where:
C = Carbon
H = Hydrogen
O = Oxygen
N = Nitrogen

Accurate control of the amount of air is essential to boiler efficiency:

- Too much air will cool the furnace, and carry away useful heat.
- Too little air and combustion will be incomplete, unburned fuel will be carried over and smoke may be produced.

In practice, however, there are a number of difficulties in achieving perfect (stoichiometric) combustion:

- The conditions around the burner will not be perfect, and it is impossible to ensure the complete matching of carbon, hydrogen, and oxygen molecules.
- Some of the oxygen molecules will combine with nitrogen molecules to form nitrogen oxides (NO_x).

To ensure complete combustion, an amount of 'excess air' needs to be provided. This has an effect on boiler efficiency.

The control of the air/fuel mixture ratio on many existing smaller boiler plants is 'open loop'. That is, the burner will have a series of cams and levers that have been calibrated to provide specific amounts of air for a particular rate of firing.

Clearly, being mechanical items, these will wear and sometimes require calibration. They must, therefore, be regularly serviced and calibrated.

On larger plants, 'closed loop' systems may be fitted which use oxygen sensors in the flue to control combustion air dampers.

Air leaks in the boiler combustion chamber will have an adverse effect on the accurate control of combustion.

Legislation

Presently, there is a global commitment to a Climate Change Programme, and 160 countries have signed the Kyoto Agreement of 1997. These countries agreed to take positive and individual actions to:

- **Reduce the emission of harmful gases to the atmosphere -** Although carbon dioxide (CO_2) is the least potent of the gases covered by the agreement, it is by far the most common, and accounts for approximately 80% of the total gas emissions to be reduced.
- **Make quantifiable annual reductions in fuel used -** This may take the form of using either alternative, non-polluting energy sources, or using the same fuels more efficiently.

In the UK, the commitment is referred to as 'The UK National Air Quality Strategy', and this is having an effect via a number of laws and regulations. Other countries will have similar strategies.

Technology

Pressure from legislation regarding pollution, and from boiler users regarding economy, plus the power of the microchip have considerably advanced the design of both boiler combustion chambers and burners.

Modern boilers with the latest burners may have:

- Re-circulated flue gases to ensure optimum combustion, with minimum excess air.
- Sophisticated electronic control systems that monitor all the components of the flue gas, and make adjustments to fuel and air flows to maintain conditions within specified parameters.
- Greatly improved turndown ratios (the ratio between maximum and minimum firing rates) which enable efficiency and emission parameters to be satisfied over a greater range of operation.

Heat losses

Having discussed combustion in the boiler furnace, and particularly the importance of correct air ratios as they relate to complete and efficient combustion, it remains to review other potential sources of heat loss and inefficiency.

Heat losses in the flue gases
This is probably the biggest single source of heat loss, and the Engineering Manager can reduce much of the loss.

The losses are attributable to the temperature of the gases leaving the furnace. Clearly, the hotter the gases in the stack, the less efficient the boiler.

The gases may be too hot for one of two reasons:

1. **The burner is producing more heat than is required for a specific load on the boiler:**
   - This means that the burner(s) and damper mechanisms require maintenance and re-calibration.
2. **The heat transfer surfaces within the boiler are not functioning correctly, and the heat is not being transferred to the water:**
   - This means that the heat transfer surfaces are contaminated, and require cleaning.

Some care is needed here - Too much cooling of the flue gases may result in temperatures falling below the 'dew point' and the potential for corrosion is increased by the formation of:

- Nitric acid (from the nitrogen in the air used for combustion).
- Sulphuric acid (if the fuel has a sulphur content).
- Water.

Radiation losses

Because the boiler is hotter than its environment, some heat will be transferred to the surroundings. Damaged or poorly installed insulation will greatly increase the potential heat losses.

A reasonably well-insulated shell or water-tube boiler of 5 MW or more will lose between 0.3 and 0.5% of its energy to the surroundings.

This may not appear to be a large amount, but it must be remembered that this is 0.3 to 0.5% of the boiler's full-load rating, and this loss will remain constant, even if the boiler is not exporting steam to the plant, and is simply on stand-by.

This indicates that to operate more efficiently, a boiler plant should be operated towards its maximum capacity. This, in turn, may require close co-operation between the boiler house personnel and the production departments.

Table 3.6.3 Typical net boiler efficiencies

Type of Boiler	Net efficiency (%)
Packaged, three pass	87
Water-tube boiler with economiser	85
Economic, two pass	78
Lancashire boiler	65
Lancashire boiler with economiser	75

Burners and controls

Burners are the devices responsible for:

- Proper mixing of fuel and air in the correct proportions, for efficient and complete combustion.
- Determining the shape and direction of the flame.

Burner turndown

An important function of burners is turndown. This is usually expressed as a ratio and is based on the maximum firing rate divided by the minimum controllable firing rate.

The turndown rate is not simply a matter of forcing differing amounts of fuel into a boiler, it is increasingly important from an economic and legislative perspective that the burner provides efficient and proper combustion, and satisfies increasingly stringent emission regulations over its entire operating range.

As has already been mentioned, coal as a boiler fuel tends to be restricted to specialised applications such as water-tube boilers in power stations. The following Sections within this Module will review the most common fuels for shell boilers.

Oil burners

The ability to burn fuel oil efficiently requires a high fuel surface area-to-volume ratio. Experience has shown that oil particles in the range 20 and 40 μm are the most successful. Particles which are:

- Bigger than 40 μm tend to be carried through the flame without completing the combustion process.
- Smaller than 20 μm may travel so fast that they are carried through the flame without burning at all.

A very important aspect of oil firing is viscosity. The viscosity of oil varies with temperature: the hotter the oil, the more easily it flows. Indeed, most people are aware that heavy fuel oils need to be heated in order to flow freely. What is not so obvious is that a variation in temperature, and hence viscosity, will have an effect on the size of the oil particle produced at the burner nozzle. For this reason the temperature needs to be accurately controlled to give consistent conditions at the nozzle.

Pressure jet burners

A pressure jet burner is simply an orifice at the end of a pressurised tube. Typically the fuel oil pressure is in the range 7 to 15 bar.

In the operating range, the substantial pressure drop created over the orifice when the fuel is discharged into the furnace results in atomisation of the fuel. Putting a thumb over the end of a garden hosepipe creates the same effect.

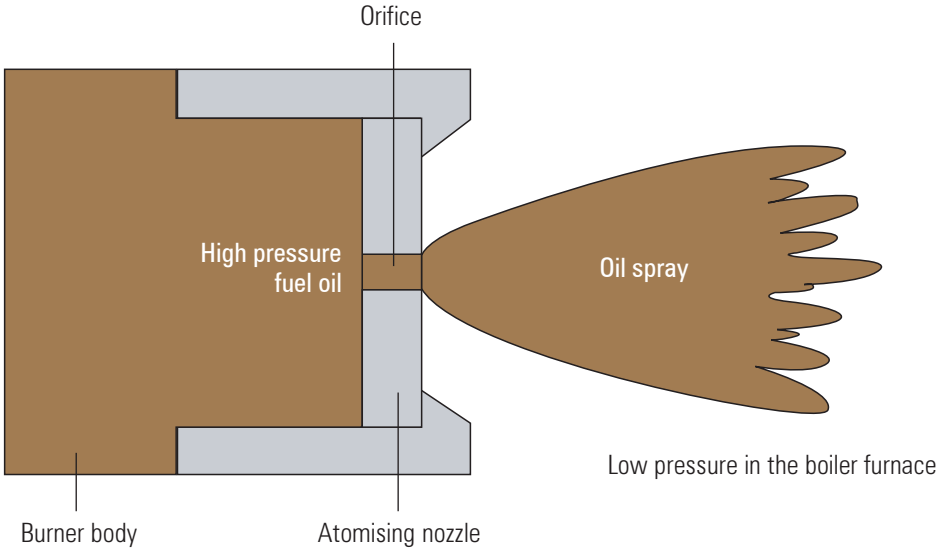


Fig. 3.6.1 Pressure jet burner

Varying the pressure of the fuel oil immediately before the orifice (nozzle) controls the flowrate of fuel from the burner.

However, the relationship between pressure (P) and flow (F) has a square root characteristic, $\sqrt{P} \propto F$, or knowing the flowrate $P \propto F^2$.

For example if: $F_2 = 0.5\ F_1$

$P_2 = (0.5)^2\ P_1$

$P_2 = 0.25\ P_1$

If the fuel flowrate is reduced to 50%, the energy for atomisation is reduced to 25%.

This means that the turndown available is limited to approximately 2:1 for a particular nozzle. To overcome this limitation, pressure jet burners are supplied with a range of interchangeable nozzles to accommodate different boiler loads.

Advantages of pressure jet burners:
- Relatively low cost.
- Simple to maintain.

Disadvantages of pressure jet burners:
- If the plant operating characteristics vary considerably over the course of a day, then the boiler will have to be taken off-line to change the nozzle.
- Easily blocked by debris. This means that well maintained, fine mesh strainers are essential.

Rotary cup burner

Fuel oil is supplied down a central tube, and discharges onto the inside surface of a rapidly rotating cone. As the fuel oil moves along the cup (due to the absence of a centripetal force) the oil film becomes progressively thinner as the circumference of the cap increases. Eventually, the fuel oil is discharged from the lip of the cone as a fine spray.

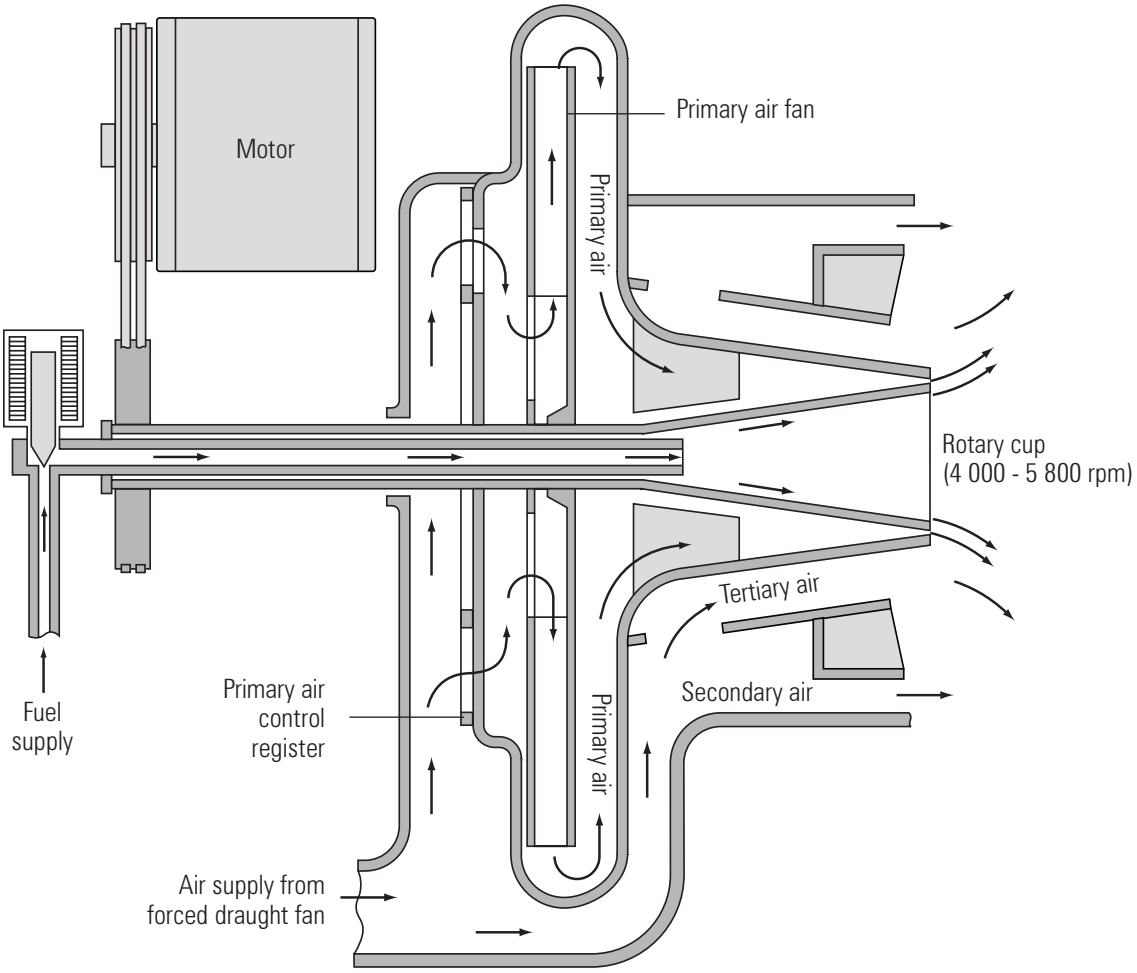


Fig. 3.6.2 Rotary cup burner

Because the atomisation is produced by the rotating cup, rather than by some function of the fuel oil (e.g. pressure), the turndown ratio is much greater than the pressure jet burner.

Advantages of rotary cup burners:
- Robust.
- Good turndown ratio.
- Fuel viscosity is less critical.

Disadvantages of rotary cup burners:
- More expensive to buy and maintain.

Gas burners

At present, gas is probably the most common fuel used in the UK.

Being a gas, atomisation is not an issue, and proper mixing of gas with the appropriate amount of air is all that is required for combustion.

Two types of gas burner are in use 'Low pressure' and 'High pressure'.

Low pressure burner

These operate at low pressure, usually between 2.5 and 10 mbar. The burner is a simple venturi device with gas introduced in the throat area, and combustion air being drawn in from around the outside.

Output is limited to approximately 1 MW.

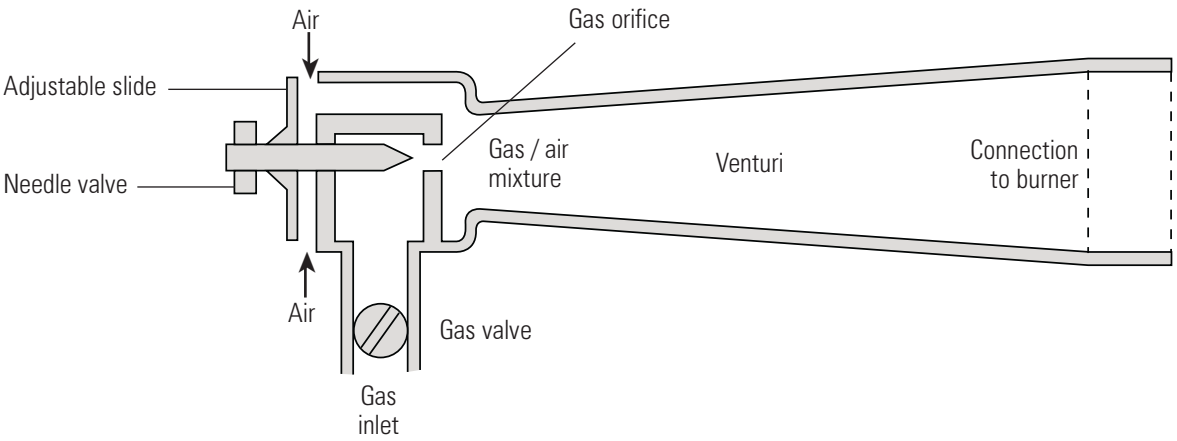


Fig. 3.6.3 Low pressure gas burner

High pressure burner

These operate at higher pressures, usually between 12 and 175 mbar, and may include a number of nozzles to produce a particular flame shape.

Dual fuel burners

The attractive 'interruptible' gas tariff means that it is the choice of the vast majority of organisations in the UK. However, many of these organisations need to continue operation if the gas supply is interrupted.

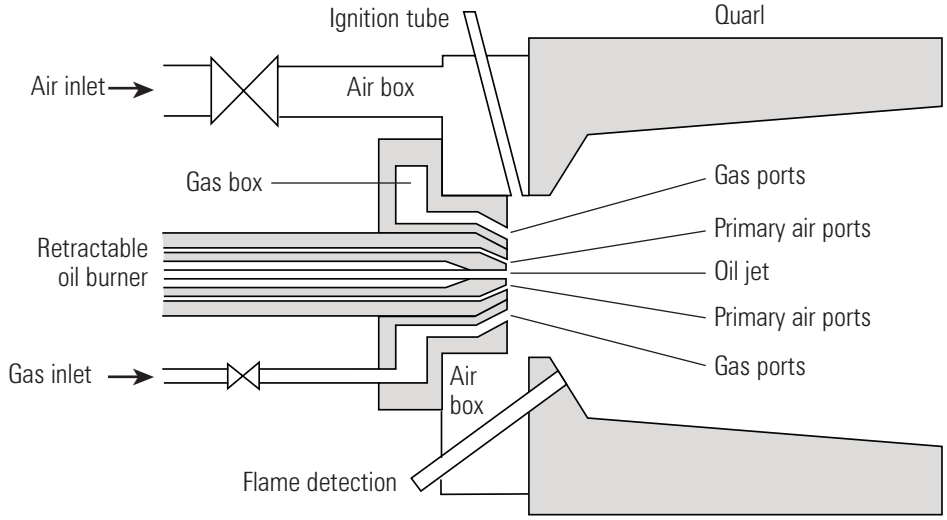


Fig. 3.6.4 Dual fuel burner

The usual arrangement is to have a fuel oil supply available on site, and to use this to fire the boiler when gas is not available. This led to the development of 'dual fuel' burners.

These burners are designed with gas as the main fuel, but have an additional facility for burning fuel oil.

The notice given by the Gas Company that supply is to be interrupted may be short, so the change over to fuel oil firing is made as rapidly as possible, the usual procedure being:

- Isolate the gas supply line.
- Open the oil supply line and switch on the fuel pump.
- On the burner control panel, select 'oil firing'.
  (This will change the air settings for the different fuel).
- Purge and re-fire the boiler.

This operation can be carried out in quite a short period. In some organisations the change over may be carried out as part of a periodic drill to ensure that operators are familiar with the procedure, and any necessary equipment is available.

However, because fuel oil is only 'stand-by', and probably only used for short periods, the oil firing facility may be basic.

On more sophisticated plants, with highly rated boiler plant, the gas burner(s) may be withdrawn and oil burners substituted.

Table 3.6.4 Typical turndown ratio available with different types of burner

Burner type	Turndown ratio
Pressure jet	2:1
Rotary cup	4:1
Gas	5:1

Burner control systems

The reader should be aware that the burner control system cannot be viewed in isolation. The burner, the burner control system, and the level control system should be compatible and work in a complementary manner to satisfy the steam demands of the plant in an efficient manner.

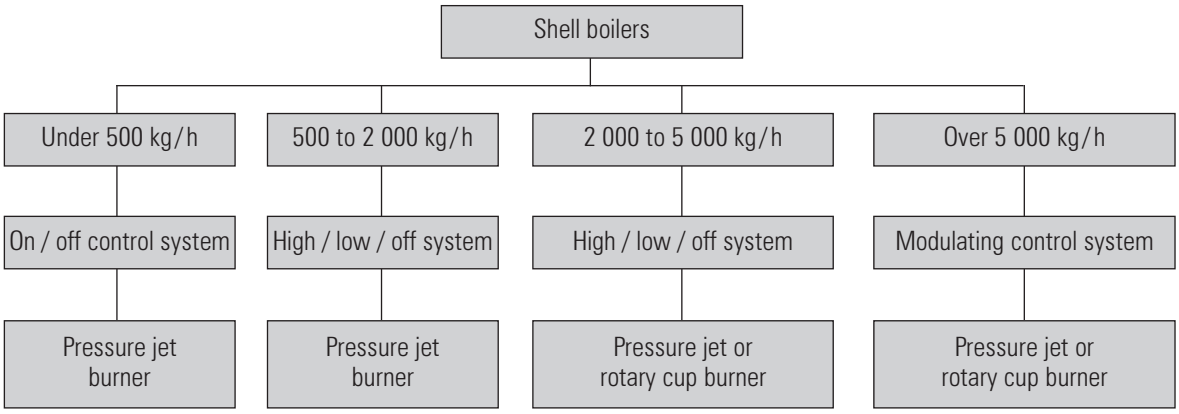


Fig. 3.6.5 Relating boiler output to controls and burner type

The next few paragraphs broadly outline the basic burner control systems.

On/off control system

This is the simplest control system, and it means that either the burner is firing at full rate, or it is off. The major disadvantage to this method of control is that the boiler is subjected to large and often frequent thermal shocks every time the boiler fires. Its use should therefore be limited to small boilers up to 500 kg/h.

Advantages of an on/off control system:

- Simple.
- Least expensive.

Disadvantages of an on/off control system:

- If a large load comes on to the boiler just after the burner has switched off, the amount of steam available is reduced. In the worst cases this may lead to the boiler priming and locking out.
- Thermal cycling.

High/low/off control system

This is a slightly more complex system where the burner has two firing rates. The burner operates first at the lower firing rate and then switches to full firing as needed, thereby overcoming the worst of the thermal shock. The burner can also revert to the low fire position at reduced loads, again limiting thermal stresses within the boiler. This type of system is usually fitted to boilers with an output of up to 5 000 kg/h.

Advantages of a high/low/off control:

- The boiler is better able to respond to large loads as the 'low fire' position will ensure that there is more stored energy in the boiler.
- If the large load is applied when the burner is on 'low fire', it can immediately respond by increasing the firing rate to 'high fire', for example the purge cycle can be omitted.

Disadvantages of a high/low/off control system:

- More complex than on-off control.
- More expensive than on-off control.

Modulating control system

A modulating burner control will alter the firing rate to match the boiler load over the whole turndown ratio. Every time the burner shuts down and re-starts, the system must be purged by blowing cold air through the boiler passages. This wastes energy and reduces efficiency. Full modulation, however, means that the boiler keeps firing over the whole range to maximise thermal efficiency and minimise thermal stresses. This type of control can be fitted to any size boiler, but should always be fitted to boilers rated at over 10 000 kg/h.

Advantages of a modulating control system:

The boiler is even more able to tolerate large and fluctuating loads. This is because:

- The boiler pressure is maintained at the top of its control band, and the level of stored energy is at its greatest.
- Should more energy be required at short notice, the control system can immediately respond by increasing the firing rate, without pausing for a purge cycle.

Disadvantages of a modulating control system:

- Most expensive.
- Most complex.
- Burners with a high turndown capability are required.

Safety

A considerable amount of energy is stored in fuel, and it burns quickly and easily. It is therefore essential that:

- Safety procedures are in place, and rigorously observed.
- Safety interlocks, for example purge timers, are in good working order and never compromised.

Questions

1. With an oil burner, what is the effect of insufficient combustion air ?

a | The burner turndown ratio is reduced ☐
b | Excessive CO_2 is produced ☐
c | The boiler output is reduced ☐
d | All of the above ☐

2. What is the likely cause of a slow increase in flue temperature with the burner at a maximum firing rate ?

a | High TDS ☐
b | The pressure thermostats have failed ☐
c | No water in the boiler ☐
d | Scaling in the boiler ☐

3. Which one of the following applies to a rotary cup burner ?

a | The fuel viscosity is less critical than with a pressure jet ☐
b | They are prone to being blocked by debris ☐
c | Their turndown ratio is typically 2:1 ☐
d | To cater for large load variations nozzle changes are required ☐

4. What is the disadvantage of an on/off burner control ?

a | They are of complex operation ☐
b | Thermal cycling ☐
c | Suitable only for oil burners ☐
d | Can be difficult to modulate the burner ☐

5. What is the advantage of modulating burner control ?

a | Inexpensive ☐
b | Simple ☐
c | It can be applied to any size boiler ☐
d | Able to tolerate large and fluctuating loads ☐

6. What is the advantage of interruptible tariff ?

a | Quick and easy to change to heavy fuel oil when required ☐
b | Price of fuel ☐
c | Convenience of supply ☐
d | Price of interruptible gas lower than fixed supply ☐

Answers

1:d, 2: d, 3: a, 4: b, 5: d, 6: d

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Module 3.7

Boiler Fittings and Mountings

Boiler Fittings and Mountings

A number of items must be fitted to steam boilers, all with the objective of improving:

- Operation.
- Efficiency.
- Safety.

While this Module can offer advice on this subject, definitive information should always be sought from the appropriate standard. In the UK, the standard relating to the specification of valves, mountings and fittings in connection with steam boilers is BS 759: Part 1. BS 6759 also refers to safety valves for steam and process fluids.

Several key boiler attachments will now be explained, together with their associated legislation where appropriate.

Boiler name-plate

In the latter half of the 19th century explosions of steam boilers were commonplace. As a consequence of this, a company was formed in Manchester with the objective of reducing the number of explosions by subjecting steam boilers to independent examination. This company was, in fact, the beginning of today's Safety Federation (SAFed), the body whose approval is required for boiler controls and fittings in the UK.

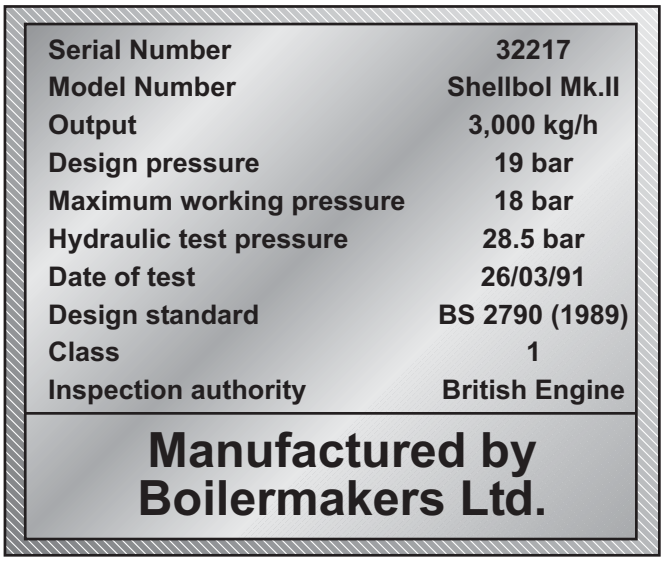


Fig. 3.7.1 Boiler name-plate

After a comparatively short period, only eight out of the 11 000 boilers examined exploded. This compared to 260 steam boiler explosions in boilers not examined by the scheme. This success led to the Boiler Explosions Act (1882) which included a requirement for a boiler name-plate. An example of a boiler name-plate is shown in Figure 3.7.1.

The serial number and model number uniquely identify the boiler and are used when ordering spares from the manufacturer and in the main boiler log book.

The output figure quoted for a boiler may be expressed in several ways, as discussed in previous Modules within this Block.

Safety valves

An important boiler fitting is the safety valve. Its function is to protect the boiler shell from over pressure and subsequent explosion. In the UK:

- BS 6759 (related to but not equivalent to ISO 4126) is concerned with the materials, design and construction of safety valves on steam boilers.
- BS 2790 relates to the specification for the design and manufacture of shell boilers of welded construction, with Section 8 specifically referring to safety valves, fittings and mountings.

Many different types of safety valves are fitted to steam boiler plant, but they must all meet the following criteria:

- The total discharge capacity of the safety valve(s) must be at least equal to the 'from and at 100°C' capacity of the boiler. If the 'from and at' evaporation is used to size the safety valve, the safety valve capacity will always be higher than the actual maximum evaporative boiler capacity.
- The full rated discharge capacity of the safety valve(s) must be achieved within 110% of the boiler design pressure.
- The minimum inlet bore of a safety valve connected to a boiler shall be 20 mm.
- The maximum set pressure of the safety valve shall be the design (or maximum permissible working pressure) of the boiler.
- There must be an adequate margin between the normal operating pressure of the boiler and the set pressure of the safety valve.

Safety valve regulations (UK)

A boiler shall be fitted with at least one safety valve sized for the rated output of the boiler. (Refer to BS 278, Section 8.1 for details.)

The discharge pipework from the safety valve must be unobstructed and drained at the base to prevent the accumulation of condensate. It is good practice to ensure that the discharge pipework is kept as short as possible with the minimum number of bends to minimise any backpressure, which should be no more than 12% of the safety valve set pressure. It will be quite normal for the internal diameter of the discharge pipework to be more than the internal diameter of the safety valve outlet connection, but under no circumstances should it be less.



Fig. 3.7.2 Boiler safety valve

Boiler stop valves

A steam boiler must be fitted with a stop valve (also known as a crown valve) which isolates the steam boiler and its pressure from the process or plant. It is generally an angle pattern globe valve of the screw-down variety. Figure 3.7.3 shows a typical stop valve of this type.

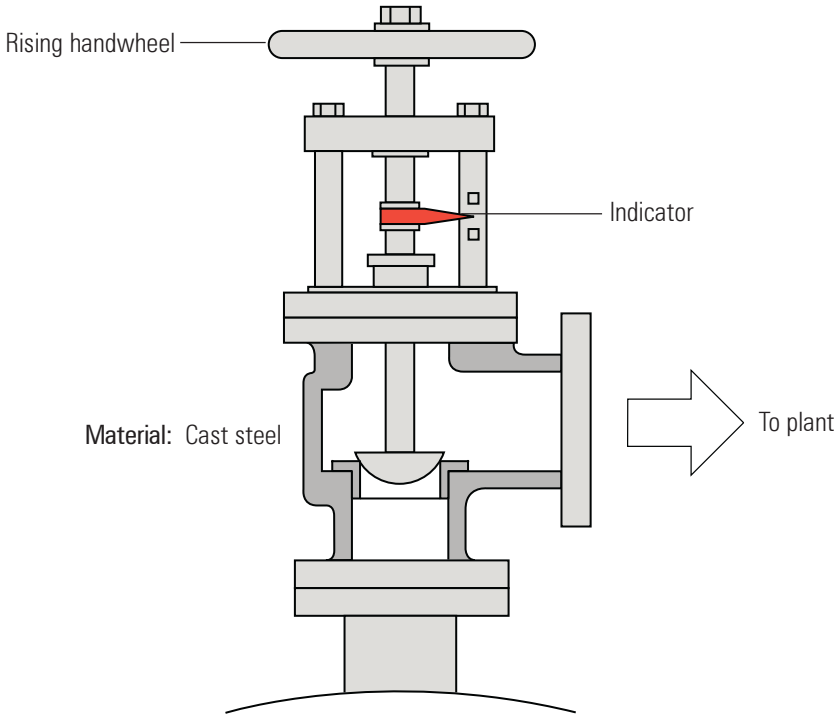


Fig. 3.7.3 Boiler stop valve

In the past, these valves have often been manufactured from cast iron, with steel and bronze being used for higher pressure applications. In the UK, BS 2790 states that cast iron valves are no longer permitted for this application on steam boilers. Nodular or spheroidal graphite (SG) iron should not be confused with grey cast iron as it has mechanical properties approaching those of steel. For this reason many boilermakers use SG iron valves as standard.

The stop valve is not designed as a throttling valve, and should be fully open or closed. It should always be opened slowly to prevent any sudden rise in downstream pressure and associated waterhammer, and to help restrict the fall in boiler pressure and any possible associated priming.

To comply with UK regulations, the valve should be of the 'rising handwheel' type. This allows the boiler operator to easily see the valve position, even from floor level. The valve shown is fitted with an indicator that makes this even easier for the operator.

On multi-boiler applications an additional isolating valve should be fitted, in series with the crown valve. At least one of these valves should be lockable in the closed position. The additional valve is generally a globe valve of the screw-down, non-return type which prevents one boiler pressurising another. Alternatively, it is possible to use a screw-down valve, with a disc check valve sandwiched between the flanges of the crown valve and itself.

Feedwater check valves

The feedwater check valve (as shown in Figures 3.7.4 and 3.7.5) is installed in the boiler feedwater line between the feedpump and boiler. A boiler feed stop valve is fitted at the boiler shell.

The check valve includes a spring equivalent to the head of water in the elevated feedtank when there is no pressure in the boiler. This prevents the boiler being flooded by the static head from the boiler feedtank.

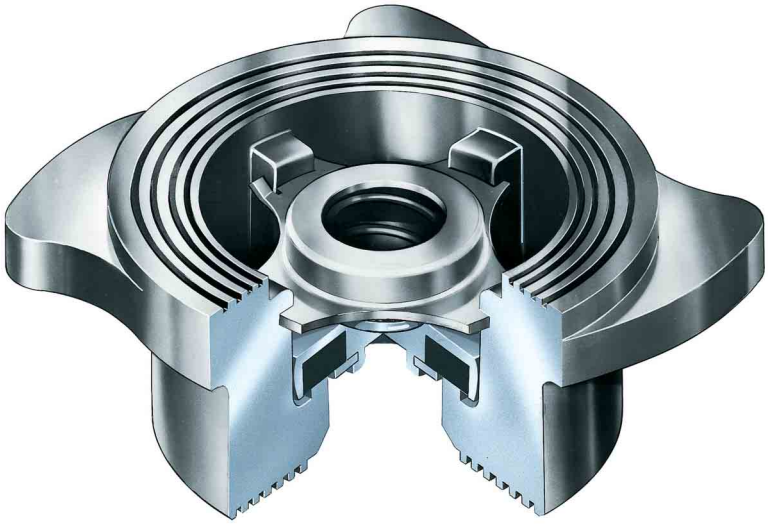


Fig. 3.7.4 Boiler check valve

Under normal steaming conditions the check valve operates in a conventional manner to stop return flow from the boiler entering the feedline when the feedpump is not running. When the feedpump is running, its pressure overcomes the spring to feed the boiler as normal.

Because a good seal is required, and the temperatures involved are relatively low (usually less than 100°C) a check valve with a EPDM (Ethylene Propylene) soft seat is generally the best option.

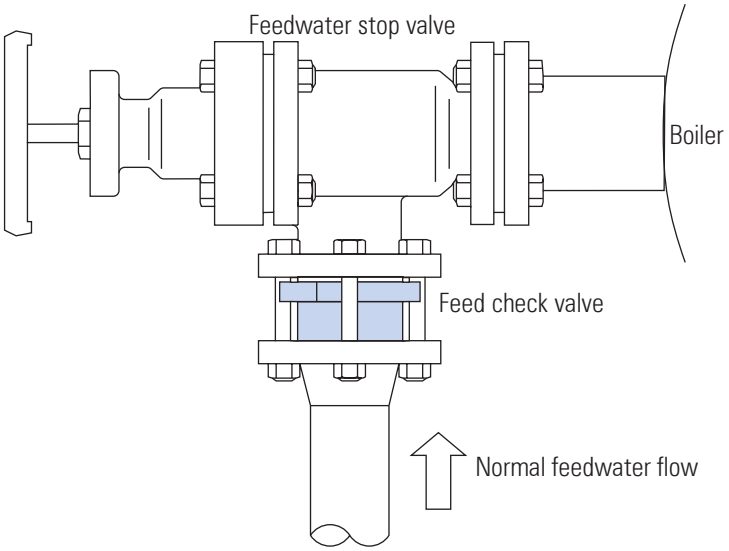


Fig. 3.7.5 Location of feed check valve

Boiler water quality control

The maintenance of water quality is essential to the safe and efficient operation of a steam boiler. The measurement and control of the various parameters is a complex topic, which is also covered by a number of regulations. It is therefore covered in detail later in this Block. The objective of the next few Sections is simply to identify the fittings to be seen on a boiler.

TDS control

This controls the amount of Total Dissolved Solids (TDS) in the boiler water, and is sometimes also referred to as 'continuous blowdown'. The boiler connection is typically DN15 or 20. The system may be manual or automatic. Whatever system is used, the TDS in a sample of boiler water is compared with a set point; if the TDS level is too high, a quantity of boiler water is released to be replaced by feedwater with a much lower TDS level. This has the effect of diluting the water in the boiler, and reducing the TDS level.

On a manually controlled TDS system, the boiler water would be sampled every shift.

A typical automatic TDS control system is shown in Figure 3.7.6

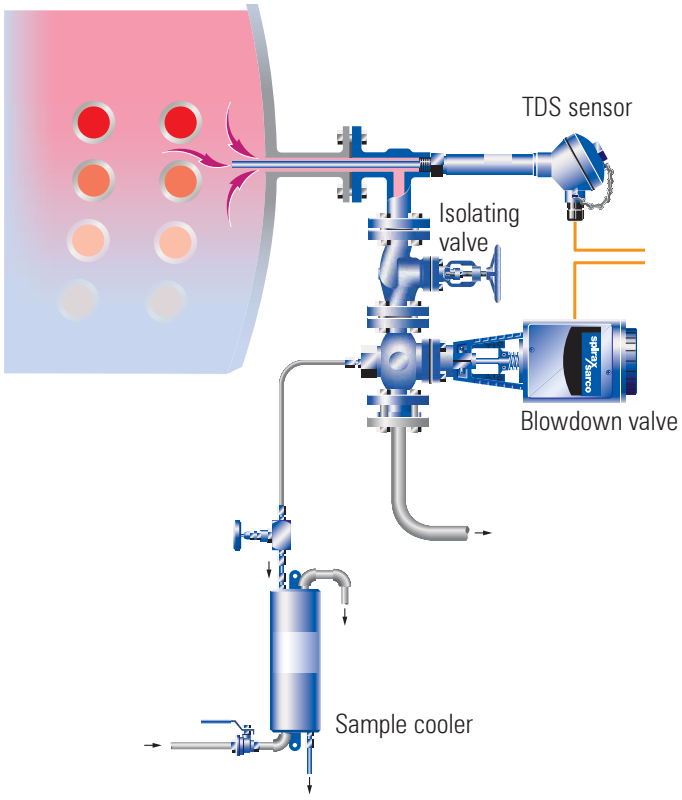


Fig. 3.7.6 Typical automatic TDS control system

Bottom blowdown

This ejects the sludge or sediment from the bottom of the boiler. The control is a large (usually 25 to 50 mm) key operated valve. This valve might normally be opened for a period of about 5 seconds, once per shift.

Figure 3.7.7 and Figure 3.7.8 illustrate a bottom blowdown valve and its typical position in a blowdown system.

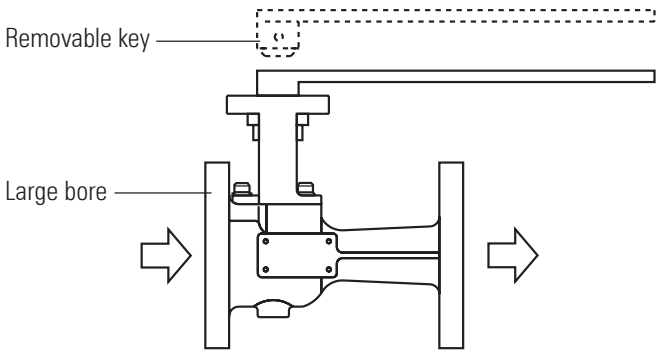


Fig. 3.7.7 Key operated bottom blowdown valve

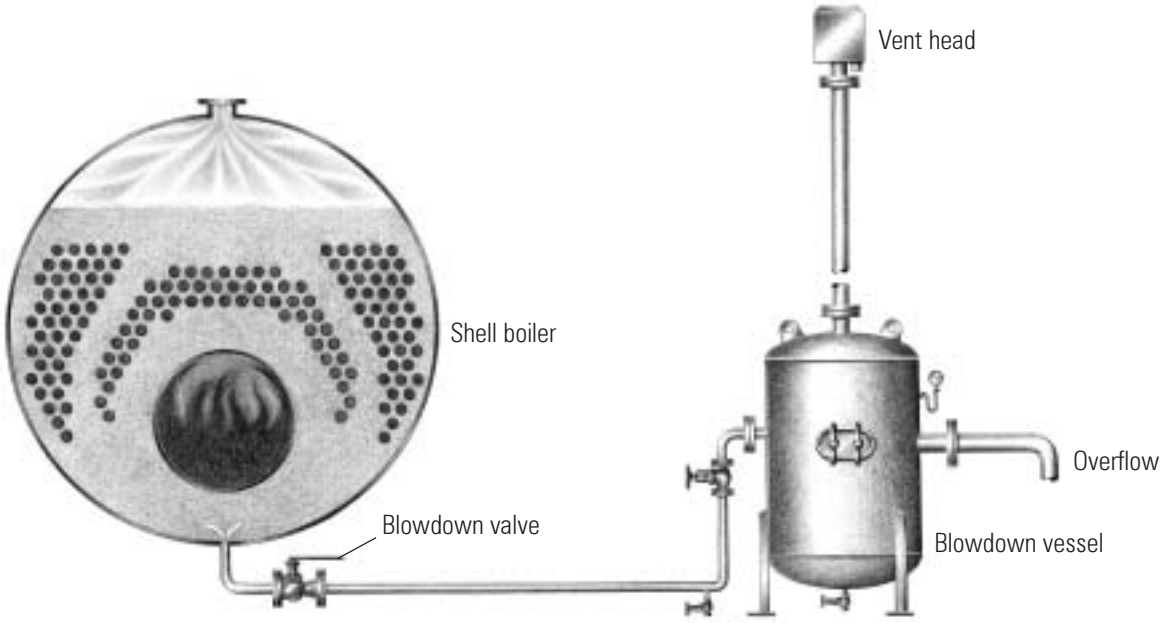


Fig. 3.7.8 Typical position for a bottom blowdown valve

Pressure gauge

All boilers must be fitted with at least one pressure indicator.

The usual type is a simple pressure gauge constructed to BS 1780 Part 2 - Class One.

The dial should be at least 150 mm in diameter and of the Bourdon tube type, it should be marked to indicate the normal working pressure and the maximum permissible working pressure/design pressure.

Pressure gauges are connected to the steam space of the boiler and usually have a ring type siphon tube which fills with condensed steam and protects the dial mechanism from high temperatures.

Pressure gauges may be fitted to other pressure containers such as blowdown vessels, and will usually have smaller dials as shown in Figure 3.7.9.

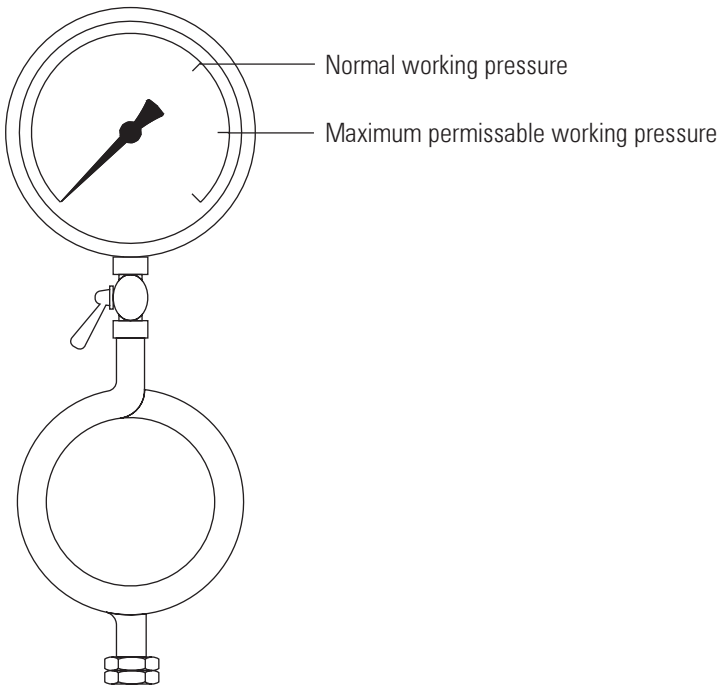


Fig. 3.7.9 Typical pressure gauge with ring siphon

Gauge glasses and fittings

All steam boilers are fitted with at least one water level indicator, but those with a rating of 100 kW or more should be fitted with two indicators. The indicators are usually referred to as gauge glasses complying with BS 3463.

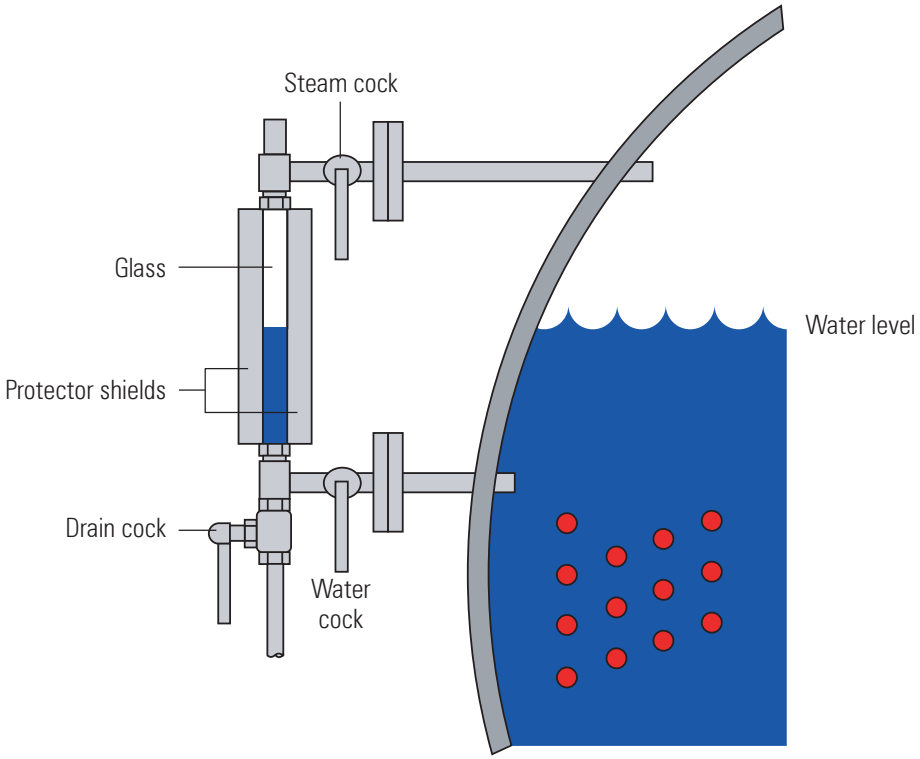


Fig. 3.7.10 Gauge glass and fittings

A gauge glass shows the current level of water in the boiler, regardless of the boiler's operating conditions. Gauge glasses should be installed so that their lowest reading will show the water level at 50 mm above the point where overheating will occur. They should also be fitted with a protector around them, but this should not hinder visibility of the water level. Figure 3.7.10 shows a typical gauge glass.

Gauge glasses are prone to damage from a number of sources, such as corrosion from the chemicals in boiler water, and erosion during blowdown, particularly at the steam end. Any sign of corrosion or erosion indicates that a new glass is required.

When testing the gauge glass steam connection, the water cock should be closed. When testing the gauge glass water connections, the steam cock pipe should be closed.

To test a gauge glass, the following procedure should be followed:

1. Close the water cock and open the drain cock for approximately 5 seconds.

2. Close the drain cock and open the water cock

Water should return to its normal working level relatively quickly. If this does not happen, then a blockage in the water cock could be the reason, and remedial action should be taken as soon as possible.

3. Close the steam cock and open the drain cock for approximately 5 seconds.

4. Close the drain cock and open the steam cock.

If the water does not return to its normal working level relatively quickly, a blockage may exist in the steam cock. Remedial action should be taken as soon as possible.

The authorised attendant should systematically test the water gauges at least once each day and should be provided with suitable protection for the face and hands, as a safeguard against scalding in the event of glass breakage.

Note: that all handles for the gauge glass cocks should point downwards when in the running condition.

Gauge glass guards

The gauge glass guard should be kept clean. When the guard is being cleaned in place, or removed for cleaning, the gauge should be temporarily shut-off.

Make sure there is a satisfactory water level before shutting off the gauge and take care not to touch or knock the gauge glass. After cleaning, and when the guard has been replaced, the gauge should be tested and the cocks set in the correct position.

Maintenance

The gauge glass should be thoroughly overhauled at each annual survey. Lack of maintenance can result in hardening of packing and seizure of cocks. If a cock handle becomes bent or distorted special care is necessary to ensure that the cock is set full open. A damaged fitting should be renewed or repaired immediately. Gauge glasses often become discoloured due to water conditions; they also become thin and worn due to erosion. Glasses, therefore, should be renewed at regular intervals.

A stock of spare glasses and cone packing should always be available in the boiler house. Remember:

- If steam passes are choked a false high water level may be given in the gauge glass. After the gauge has been tested a false high water level may still be indicated.
- If the water passages are choked an artificially high water level may be observed due to steam condensing in the glass. After testing, the glass will tend to remain empty unless the water level in the boiler is higher than the top connection, in which case water might flow into the glass from this connection.
- Gauge glass levels must be treated with the utmost respect, as they are the only visual indicator of water level conditions inside the boiler. Any water level perceived as abnormal must be investigated as soon as it is observed, with immediate action taken to shut down the boiler burner if necessary.

Water level controls

The maintenance of the correct water level in a steam boiler is essential to its safe and efficient operation. The methods of sensing the water level, and the subsequent control of water level is a complex topic that is covered by a number of regulations. The following few Sections will provide a brief overview, and the topic will be discussed in much greater detail later.

External level control chambers

Level control chambers are fitted externally to boilers for the installation of level controls or alarms, as shown in Figure 3.7.11.

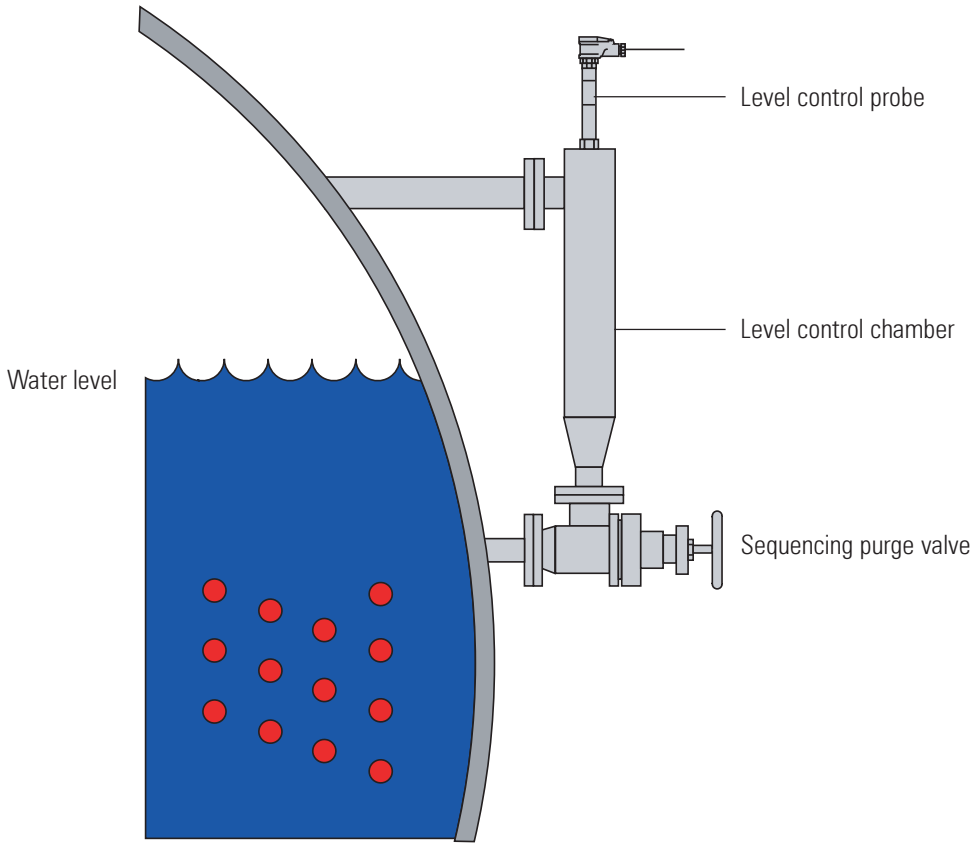


Fig. 3.7.11 External level control chamber

The function of the level controls or alarms is checked daily using the sequencing purge valves. With the handwheel turned fully anticlockwise the valve is in the 'normal working' position and a back seating shuts off the drain connection. The handwheel dial may look similar to that shown in Figure 3.7.12. Some handwheels have no dial, but rely on a mechanism for correct operation.

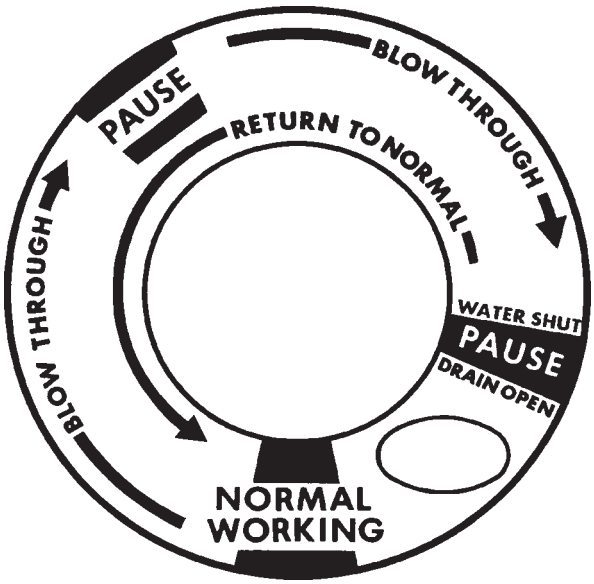


Fig. 3.7.12 Purge valve handwheel

The following is a typical procedure that may be used to test the controls when the boiler is under pressure, and the burner is firing:

- Slowly turn the handwheel clockwise until the indicating pointer is at the first 'pause' position. The float chamber connection is baffled, the drain connection is opened, and the water connection is blown through.
- Pause for 5 to 8 seconds.
- Slowly move the handwheel further clockwise to full travel. The water connection is shut-off, the drain valve remains open, and the float chamber and steam connections are blown through. The boiler controls should operate as for lowered water level in boiler i.e. pump running and / or audible alarm sounding and burner cut-out. Alternatively if the level control chamber is fitted with a second or extra low water alarm, the boiler should lock-out.
- Pause for 5 to 8 seconds.
- Slowly turn the handwheel fully anticlockwise to shut-off against the back seating in the 'normal working' position.

Sequencing purge valves are provided by a number of different manufacturers. Each may differ in operating procedure. It is essential that the manufacturer's instructions be followed regarding this operation.

Internally mounted level controls

Level control systems with sensors (or probes) which fit inside the boiler shell (or steam drum) are also available. These provide a higher degree of safety than those fitted externally. The level alarm systems may also provide a self-checking function on system integrity.

Because they are mounted internally, they are not subject to the procedures required to blow down external chambers. System operation is tested by an evaporation test to '1st low' position, followed by blowing down to '2nd low' position.

Protection tubes are fitted to discourage the movement of water around the sensor.

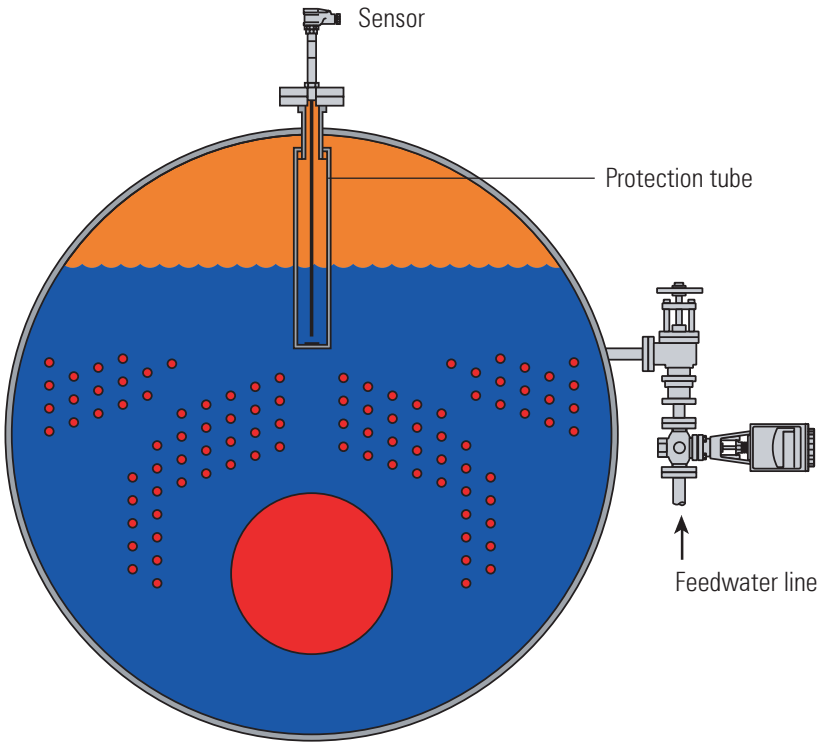


Fig. 3.7.13 Internally mounted level controls

Air vents and vacuum breakers

When a boiler is started from cold, the steam space is full of air. This air has no heat value, and will adversely affect steam plant performance due to its effect of blanketing heat exchange surfaces. The air can also give rise to corrosion in the condensate system, if not removed adequately.

The air may be purged from the steam space using a simple cock; normally this would be left open until a pressure of about 0.5 bar is showing on the pressure gauge. An alternative to the cock is a balanced pressure air vent which not only relieves the boiler operator of the task of manually purging air (and hence ensures that it is actually done), it is also much more accurate and will vent gases which may accumulate in the boiler. Typical air vents are shown in Figure 3.7.14.

When a boiler is taken off-line, the steam in the steam space condenses and leaves a vacuum. This vacuum causes pressure to be exerted on the boiler from the outside, and can result in boiler inspection doors leaking, damage to the boiler flat plates and the danger of overfilling a shutdown boiler. To avoid this, a vacuum breaker (see Figure 3.7.14) is required on the boiler shell.

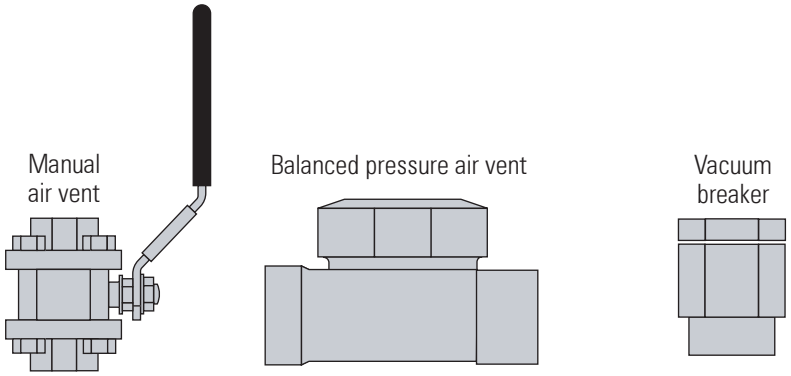


Fig. 3.7.14 Typical air vents and vacuum breakers

Questions

1. At what pressure should a boiler safety valve be set?

a| Maximum working pressure ☐
b| Normal working pressure ☐
c| Hydraulic test pressure ☐
d| Feedpump maximum pressure ☐

2. What is the purpose of a bottom blowdown valve?

a| To control water level ☐
b| To drain the boiler ☐
c| To maintain TDS ☐
d| To remove sludge ☐

3. How often, as a minimum, should gauge glasses be tested?

a| Once a shift ☐
b| Twice a day ☐
c| Once a day ☐
d| Once a week ☐

4. Why are two gauge glasses often fitted?

a| One is a check against the other ☐
b| One is a reserve ☐
c| It is a legal requirement ☐
d| To increase periods between maintenance ☐

5. What is the advantage of an internal water level control over an external one?

a| The external control is in a 'dead' area ☐
b| It is less likely to scale up ☐
c| It will respond more quickly to changes in water level ☐
d| Daily testing of the level control chamber is not required ☐

6. What is the purpose of testing gauge glasses?

a| To ensure the gauge cocks are operative ☐
b| To ensure there is sufficient water over the top fire tube ☐
c| To ensure the boiler water level is being properly sensed ☐
d| To check the boiler 1st and 2nd low water level alarms ☐

Answers

1: a, 2: d, 3: c, 4: c, 5: d, 6: a

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Module 3.8

Steam Headers and Off-takes

Steam Headers and Off-takes

Shell boilers are made for capacities up to around 27 000 kg/h of steam. When loads in excess of this are required, two or more boilers are connected in parallel, with an installation of four or more boilers not being uncommon. The design of the interconnecting steam header is highly important.

Figure 3.8.1 shows a common method of connecting four boilers: a method that is frequently a source of problems.

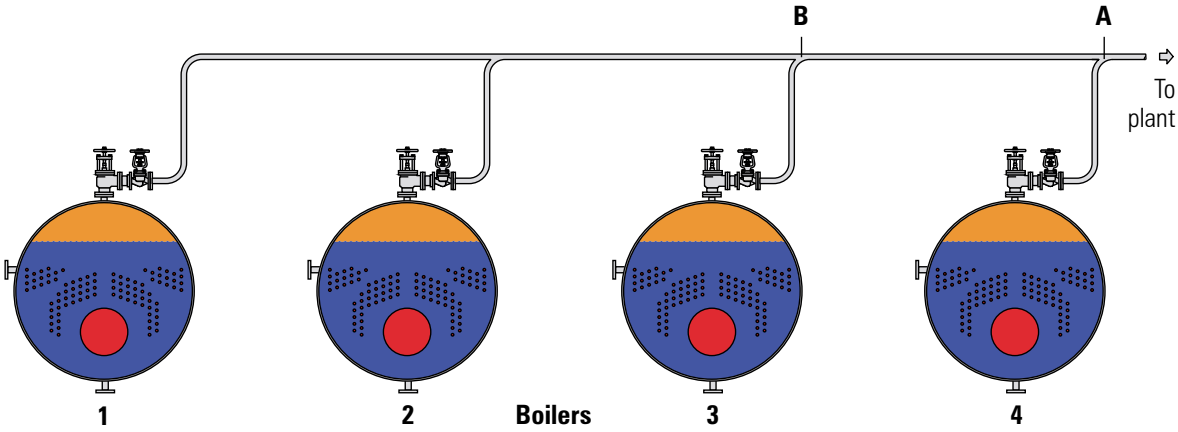


Fig. 3.8.1 Common four boiler layout - not recommended

Referring to Figure 3.8.1, with all boilers operating at the same pressure, the pressure at point **A** has to be less than that at point **B** for steam to flow from boiler number 3 to the plant. Consequently, there must be a greater pressure drop between boiler number 4 and point **A** than boiler number 3 and point **A**.

Flow depends on pressure drop, it follows then, that boiler number 4 will discharge more steam than boiler number 3. Likewise, boiler number 3 will discharge more than number 2, and so on. The net effect is that if boiler number 1 is fully loaded, the other boilers are progressively overloaded, the effect worsening nearer to the final off-take.

It can be shown that, typically, if boiler number 1 is fully loaded, number 2 will be around 1% overloaded, number 3 around 6%, and number 4 around 15% overloaded. Whilst shell boilers are able to cope with occasional overload conditions of 5%, an overload of 15% is undesirable. The increased steam outlet velocity from the boiler creates an extremely volatile water surface, and the level control system might fail to control.

At high loads, in this example, boiler number 4 would lock-out, throwing an already unstable system onto the three remaining boilers, which would soon also lock-out.

The main observation is that this design of distribution header does not allow the boilers to share the load equally.

The aim should be that the pressure drops between each boiler outlet and the header off-take to the plant should be within 0.1 bar. This will minimise carryover and help to prevent overload and lockout of boilers.

The layout shown in Figure 3.8.2 shows an improved design of a new header.

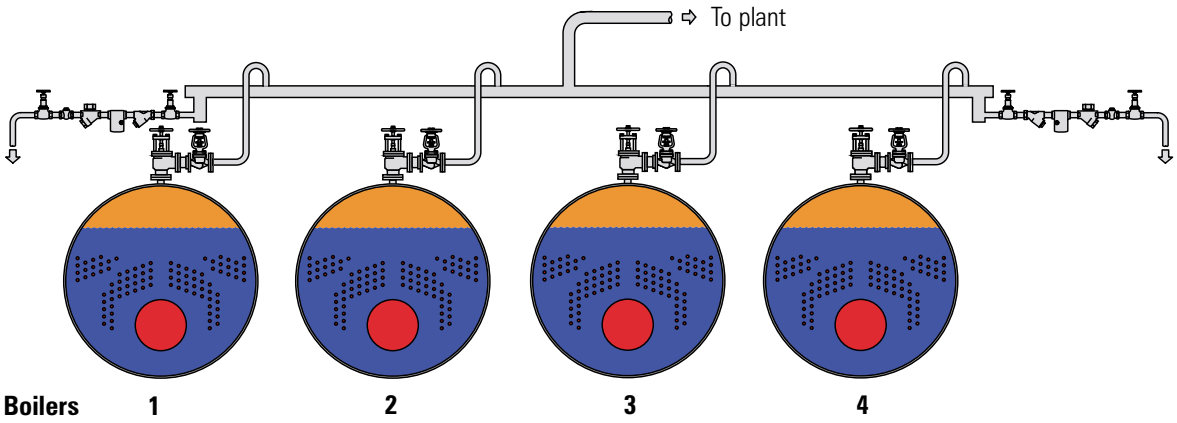


Fig. 3.8.2 Four boiler header design - improved layout

The header is arranged to discharge from the centre, rather than at one end. In this way, no boiler will be overloaded by the header by more than 1%, providing the header pipework is properly sized.

A better arrangement is shown in Figure 3.8.3 for an installation of four or more boilers, rather like a family tree, where the load on each boiler is spread equally. This arrangement is recommended for heavily loaded boilers, with sequencing control where one or more is regularly off-line.

It is emphasised that correct header design will save much trouble and expense later.

Correct boiler header design on multi-boiler applications will always result in a well-balanced operation.

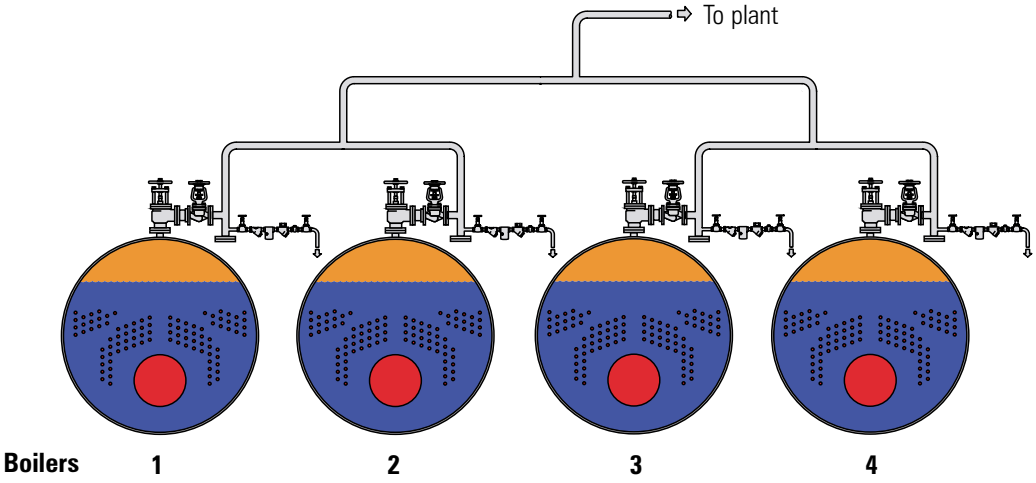


Fig. 3.8.3 Four boiler header design - recommended layout

Steam off-takes

Having considered the general arrangement of the steam header, the following conditions need to be ensured:

- That dry steam is exported to the plant.
- That the warm-up operation is properly controlled.
- That steam is properly distributed to the plant.
- That one boiler cannot accidentally pressurise another.

Water carryover

When a well-designed boiler generates steam under steady load conditions, the dryness fraction of the steam will be high, approximately 96 to 99%. Changes in load that occur faster than the boiler can respond will adversely affect the dryness fraction. Poor control of boiler water TDS, or contamination of boiler feedwater, will result in wet steam being discharged from the boiler.

A number of problems are associated with this:

- Water in a steam system gives the potential for dangerous waterhammer.
- Water in steam does not contain the enthalpy of evaporation that the plant has been designed to use, so transporting it to the plant is inefficient.
- Water carried over with steam from a boiler will inevitably contain dissolved and suspended solids, which can contaminate controls, heat transfer surfaces, steam traps and the product.

For these reasons, a separator close to the boiler is recommended. Separators work by forcing the steam to rapidly change direction. This results in the much denser water particles being separated from the steam due to their inertia, and then encouraged to gravitate to the bottom of the separator body, where they collect and drain away via a steam trap.

Warm-up

It is essential that when a boiler is brought on line, it is done in a slow, safe and controlled manner to avoid:

- **Waterhammer -** Where large quantities of condensate lie inside the pipe and are then pushed along the pipe at steam velocities. This can result in damage when the water impacts with an obstruction in the pipe, for example a control valve.
- **Thermal shock -** Where the pipework is being heated so rapidly that the expansion is uncontrolled, setting up stresses in the pipework and causing large movement on the pipe supports.
- **Priming -** Where a sudden reduction of steam pressure caused by a large, suddenly applied load may result in boiler water being pulled into the pipework. Not only is this bad for plant operation, the boiler can often go to 'lock-out' and it will take some time to return the boiler to operating status. The discharged water can also give rise to waterhammer in the pipework.

The warm-up period for every plant will be different and will depend on many factors. A small low-pressure boiler in a compact plant such as a laundry, for example, could be brought up to operating pressure in less than 15 minutes. A large industrial complex may take many hours. The starting point, when safely bringing a small boiler on line, is the main stop valve, which should be opened slowly.

On larger plants, however, the rate of warm-up is difficult to control using the main stop valve. This is because the main stop valve is designed to provide good isolation; it has a flat seat that means that all the force exerted by turning the handwheel acts directly onto the seat, thus ensuring a good seal when under pressure. It also means that the valve is not characterised and will pass approximately 80% of its capacity in the first 10% of its movement.

For this reason it is good practice to install a control valve after the main stop valve. A control valve has a profiled plug, which means that the relationship between an increase in flow and the movement of the plug is much less severe. Consequently the flowrate, and hence warm-up rate, is better controlled.

An example of a control valve fitted after the boiler main stop valve is shown in Figure 3.8.4.

A typical warm-up arrangement may be that the control valve is closed until the boiler is required. At this point a pulse timer slowly opens the control valve over a predetermined time period. This arrangement also has the advantage that it does not require manpower (unless the boiler is heated up from cold) over the boiler warm-up period, which may be during twilight hours.

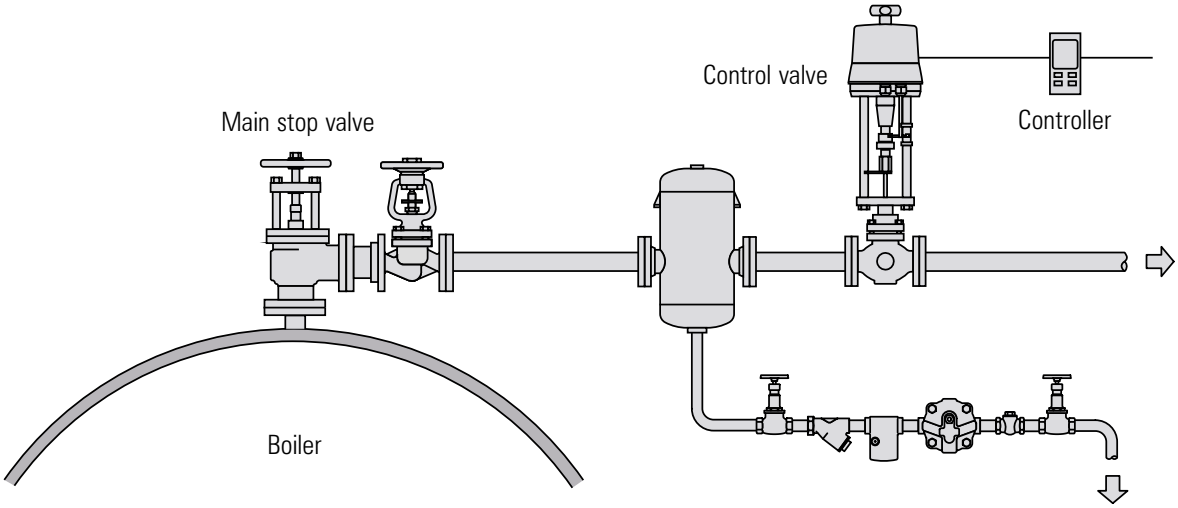


Fig. 3.8.4 Control valve after main stop valve

The subject of bringing boilers on-line is covered by the HSE guidelines in the UK.

On large distribution systems, a line size control valve is still often too coarse to provide the required slow warm-up. In these circumstances a small control valve in a loop around an isolation valve could be used. This also has the advantage that where parallel slide valves are used for isolation, the pressure can be equalised either side of the valve prior to opening. This will make them easier to open, and reduces wear.

Preventing one boiler pressurising another

From BS 2790, Section 8.8.3.

Where two or more boilers are connected to a common header, in addition to the boiler main stop valve, a second valve shall be incorporated in the steam connection, and this valve shall be capable of being locked in the closed position. This allows better protection for a decommissioned boiler when isolated from the distribution header.

Unless a separate non-return valve is fitted in the steam connection, one of the two stop valves must incorporate a non-return facility.

The objective of this section of the British Standard is to provide safe working conditions when the boiler is shut down for repair or inspection.

Simple flap-type non-return valves are not suitable for this purpose, because small changes in boiler pressures can cause them to oscillate, placing excess load on to one boiler or the other alternately. This can, under severe conditions, cause cyclical overloading of the boilers.

Many cases of instability with two-boiler installations are caused in this way. Main stop valves with integral non-return valves tend to suffer less from this phenomenon. Alternatively, spring loaded disc check valves can provide a dampening effect which tends to reduce the problems caused by oscillation (Figure 3.8.5).

BS 2790 states that a non-return valve must be fitted in this line together with the main stop valve, alternatively, the main stop valve must incorporate an integral non-return valve.

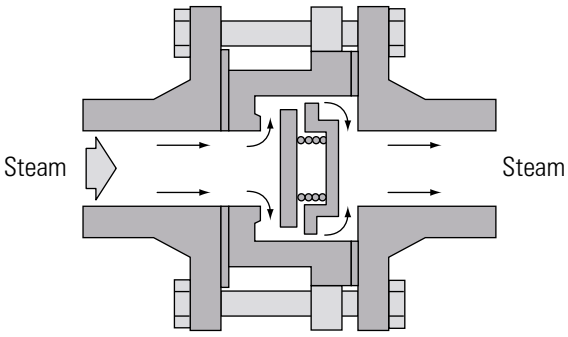


Fig. 3.8.5 Typical disc type non-return valve

Boiler related standards (UK)

Statutory instrument 1989 No. 2169 (The pressure systems and transportable gas containers regulations 1989) with the associated guide and approved code of practice.

BS 2790	Specification for design and manufacture of shell boilers of welded construction.
BS 1113	Specification for design and manufacture of water-tube steam generating plant.
BS 6759 (related to but not equivalent to ISO 4126) Part 1	Specification for safety valves for steam and hot water.
BS 759 Part 1	Specification for valves, mountings and fittings for steam boilers above 1 bar g.

Ensuring proper steam distribution

The starting point for the distribution system is the boiler house, where it is often convenient for the boiler steam lines to converge at a steam manifold usually referred to as the main distribution header. The size of the header will depend upon the number and size of boilers and the design of the distribution system. In a large plant, the most practical approach is to distribute steam via a high pressure main around the site.

High pressure distribution is generally preferred as it reduces pipe sizes relative to capacities and velocities. Heat losses may also be reduced due to lower overall pipe diameters. This allows steam supplies to be taken from the main, either direct to high pressure users, or to pressure reducing stations providing steam to local users at reduced pressure. A steam header at the boiler house provides a useful centralised starting point.

It provides an extra separating function if the boiler separator is overwhelmed, and a means of allowing the attached boilers to share the distribution system load.

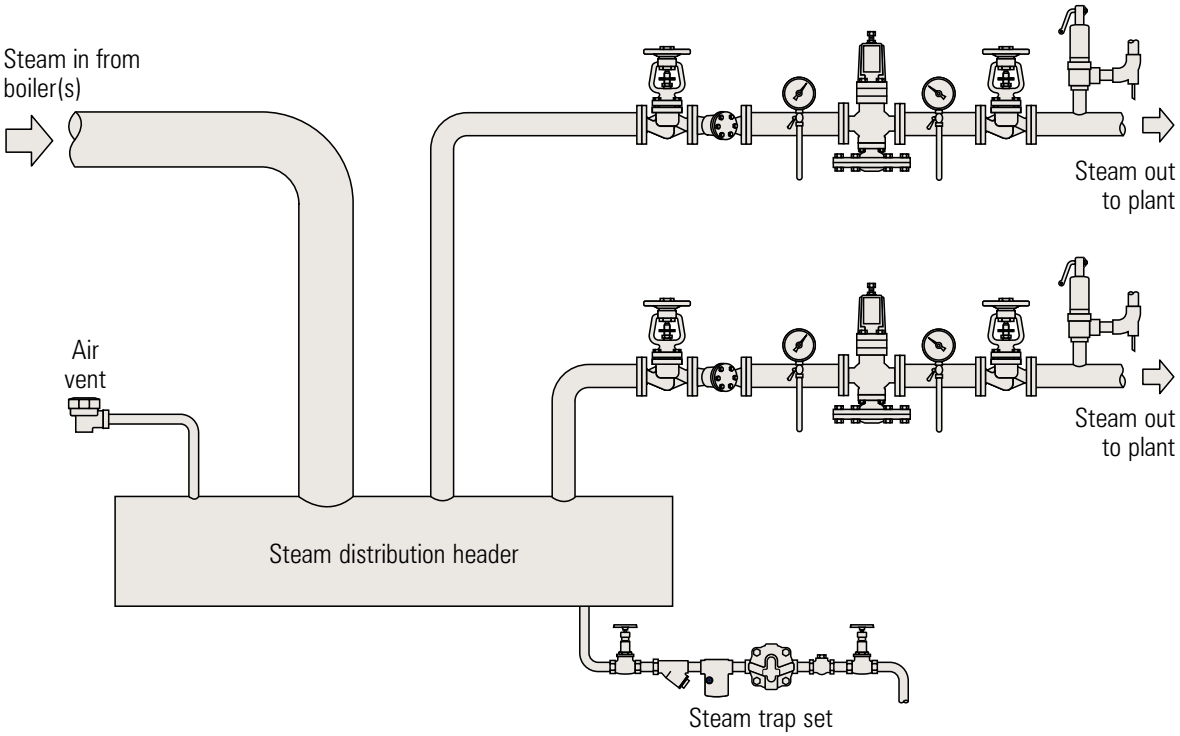


Fig. 3.8.6 Steam distribution manifold

Operating pressure

The header should be designed for the boiler operating pressure and to conform to the Pressure Systems Regulations. It is important to remember that flange standards are based on temperature and pressure and that the allowable pressure reduces as the operating temperature increases. For example, a PN16 rating is 16 bar at 120°C, but is only suitable for up to 13.8 bar saturated steam (198°C).

Diameter

The header diameter should be calculated with a maximum steam velocity of 15 m/s under full-load conditions. Low velocity is important as it helps any entrained moisture to fall out.

Off-takes

These should always be from the top of the distribution header.

Gravity and the low velocity will ensure that any condensate falls to and drains from the bottom of the header. This ensures that only dry steam is exported.

Steam trapping

It is important that condensate is removed from the header as soon as it forms. For this reason a mechanical trap, for instance a float trap, is the best choice. If the header is the first trapping point after the boiler off-takes, the condensate can contain carryover particles and it may be useful to drain this steam trap into the boiler blowdown vessel, rather than the boiler feedtank.

Related reading:

1. *The Steam and Condensate Loop,* Block 11, 'Steam Trapping'

2. *The Steam and Condensate Loop,* Block 10, 'Steam Distribution'

Questions

1. In Figure 3.8.1 which boiler works the hardest ?

a| 1 ☐
b| 2 ☐
c| 3 ☐
d| 4 ☐

2. What is one effect of an overloaded boiler ?

a| Water level rises and lock-out occurs ☐
b| Reduced steam production ☐
c| Water level drops and lock-out occurs ☐
d| Steam velocity reduces and separator efficiency drops ☐

3. Why is slow, controlled warm-up of a steam system essential ?

a| To make it easier to open the boiler main stop valve ☐
b| To minimise undue stresses and eliminate damage ☐
c| To permit separators to remove more water ☐
d| To prevent stress on the boiler ☐

4. Which of the following is the main purpose of the steam distribution manifold ?

a| It replaces the need for a separator after the boiler ☐
b| To remove air from the steam system ☐
c| To provide an extra separating function ☐
d| It is a requirement of the pressure systems regulations ☐

**5. Four boilers are connected to a common header as in Figure 3.8.2.
Why is a second isolation valve after each main stop valve a recommendation ?**

a| For slow opening and warm-up ☐
b| To balance the boiler loading in a multi-boiler arrangement ☐
c| In place of a check valve ☐
d| To double isolate against reverse flow ☐

6. Priming of a boiler is:

a| Getting a boiler prepared for start-up ☐
b| A reduction in boiler pressure and carryover of water ☐
c| Occurrence of excessive TDS and carryover of water ☐
d| Balancing of boilers in a multi-boiler installation ☐

Answers

1: d, 2: c, 3: b, 4: c, 5: d, 6: b

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Module 3.9

Water Treatment, Storage and Blowdown for Steam Boilers

Water Treatment, Storage and Blowdown for Steam Boilers

Before boiler blowdown can be discussed and understood it is necessary to establish a definition of water along with its impurities and associated terms such as hardness, pH etc.

Water is the most important raw material on earth. It is essential to life, it is used for transportation, and it stores energy. It is also called the 'universal solvent'.

Pure water (H_20) is tasteless, odourless, and colourless in its pure state; however, pure water is very uncommon. All natural waters contain various types and amounts of impurities.

Good drinking water does not necessarily make good boiler feedwater. The minerals in drinking water are readily absorbed by the human body, and essential to our well being. Boilers, however, are less able to cope, and these same minerals will cause damage in a steam boiler if allowed to remain.

Of the world's water stock, 97% is found in the oceans, and a significant part of that is trapped in the polar glaciers - only 0.65% is available for domestic and industrial use.

This small proportion would soon be consumed if it were not for the water cycle (see Figure 3.9.1). After evaporation, the water turns into clouds, which are partly condensed during their journey and then fall to earth as rain. However, it is wrong to assume that rainwater is pure; during its fall to earth it will pick up impurities such as carbonic acid, nitrogen and, in industrial areas, sulphur dioxide.

Charged with these ingredients, the water percolates through the upper layers of the earth to the water table, or flows over the surface of the earth dissolving and collecting additional impurities.

These impurities may form deposits on heat transfer surfaces that may:

- Cause metal corrosion.
- Reduce heat transfer rates, leading to overheating and loss of mechanical strength.

Table 3.9.1 shows the technical and commonly used names of the impurities, their chemical symbols, and their effects.

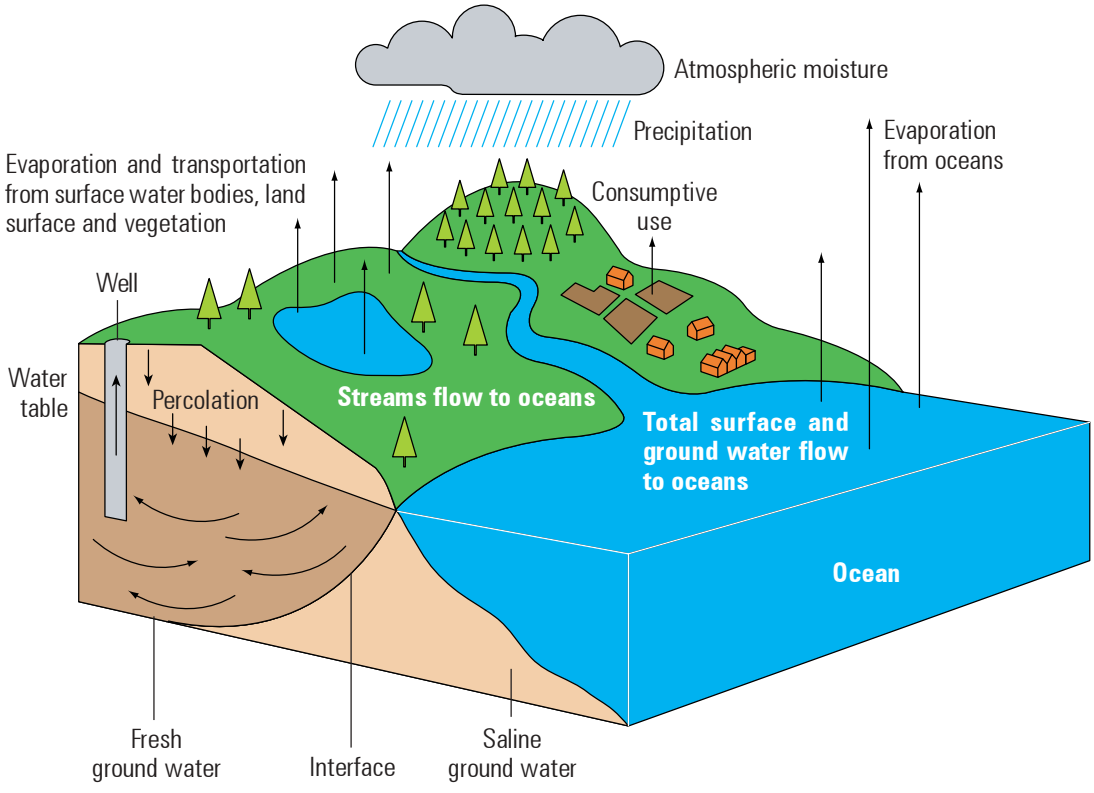


Fig. 3.9.1 Typical water cycle

Table 3.9.1 Impurities in water

Name	Symbol	Common name	Effect
Calcium carbonate	$CaCO_3$	Chalk, limestone	Soft scale
Calcium bicarbonate	$Ca(HCO_3)_2$		Soft scale + CO_2
Calcium sulphate	$CaSO_4$	Gypsum, plaster of paris	Hard scale
Calcium chloride	$CaCl_2$		Corrosion
Magnesium carbonate	$MgCO_3$	Magnesite	Soft scale
Magnesium sulphate	$MgSO_4$	Epsom salts	Corrosion
Magnesium bicarbonate	$Mg(HCO_3)_2$		Scale, corrosion
Sodium chloride	NaCl	Common salt	Electrolysis
Sodium carbonate	Na_2CO_3	Washing soda or soda	Alkalinity
Sodium bicarbonate	$NaHCO_3$	Baking soda	Priming, foaming
Sodium hydroxide	NaOH	Caustic soda	Alkalinity, embrittlement
Sodium sulphate	Na_2SO_4	Glauber salts	Alkalinity
Silicon dioxide	SiO_2	Silica	Hard scale

Raw water quality and regional variations

Water quality can vary tremendously from one region to another depending on the sources of water, local minerals (see Figure 3.9.2). Table 3.9.2 gives some typical figures for different areas in a relatively small country like the UK.

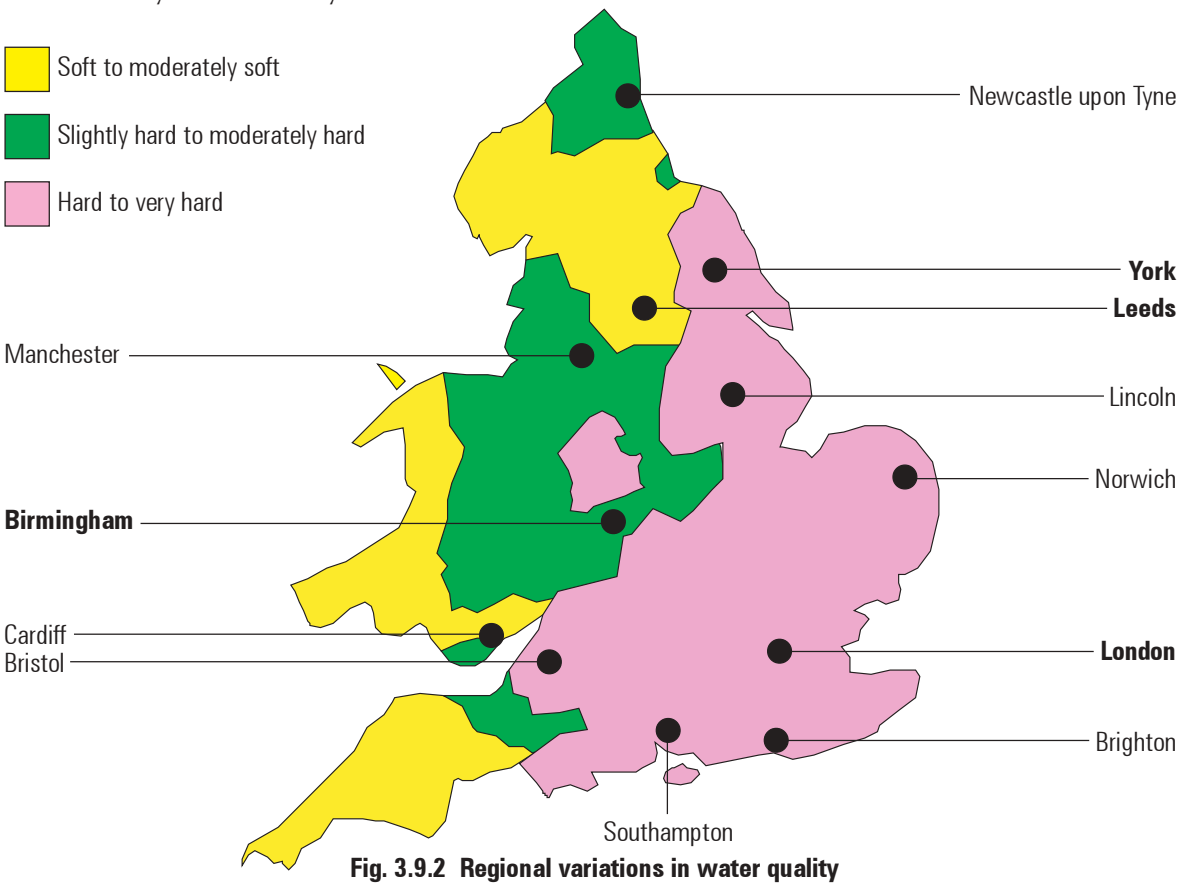


Fig. 3.9.2 Regional variations in water quality

Table 3.9.2 Water variation within the UK - All impurities expressed in mg/l calcium carbonate equivalents

Area	Alkaline hardness (temporary)	Non-alkaline hardness (permanent)	Total hardness	Non-hardness salts	Total dissolved solids (TDS)
Leeds	12	10	22	24	46
York	156	92	248	62	310
Birmingham	28	72	100	130	230
London	180	192	372	50	422

The common impurities in raw water can be classified as follows:

- **Dissolved solids** - These are substances that will dissolve in water.

  The principal ones are the carbonates and sulphates of calcium and magnesium, which are scale-forming when heated.

  There are other dissolved solids, which are non-scale forming.

  In practice, any salts forming scale within the boiler should be chemically altered so that they produce suspended solids, or sludge rather than scale.

- **Suspended solids** - These are substances that exist in water as suspended particles.

  They are usually mineral, or organic in origin.

  These substances are not generally a problem as they can be filtered out.

- **Dissolved gases** - Oxygen and carbon dioxide can be readily dissolved by water.

  These gases are aggressive instigators of corrosion.

- **Scum forming substances** - These are mineral impurities that foam or scum.

  One example is soda in the form of a carbonate, chloride, or sulphate.

The amount of impurities present is extremely small and they are usually expressed in any water analysis in the form of parts per million (ppm), by weight or alternatively in milligrams per litre (mg/l).

The following sections within this Module describe the characteristics of water.

Hardness

Water is referred to as being either 'hard' or 'soft'. Hard water contains scale-forming impurities while soft water contains little or none. The difference can easily be recognised by the effect of water on soap. Much more soap is required to make a lather with hard water than with soft water.

Hardness is caused by the presence of the mineral salts of calcium and magnesium and it is these same minerals that encourage the formation of scale.

There are two common classifications of hardness:

- **Alkaline hardness (also known as temporary hardness)** - Calcium and magnesium bicarbonates are responsible for alkaline hardness. The salts dissolve in water to form an alkaline solution. When heat is applied, they decompose to release carbon dioxide and soft scale or sludge.

  The term 'temporary hardness' is sometimes used, because the hardness is removed by boiling. This effect can often be seen as scale on the inside of an electric kettle.

  See Figures 3.9.3 and 3.9.4 - the latter representing the situation within the boiler.

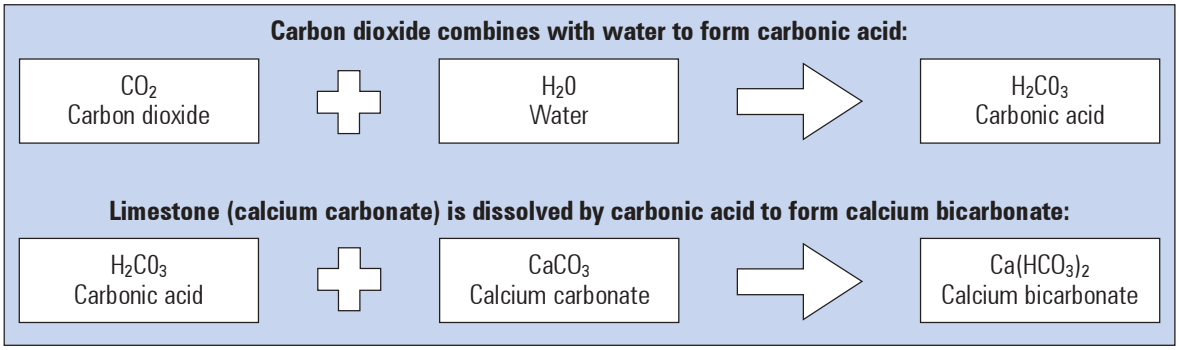


Fig. 3.9.3 Alkaline or temporary hardness

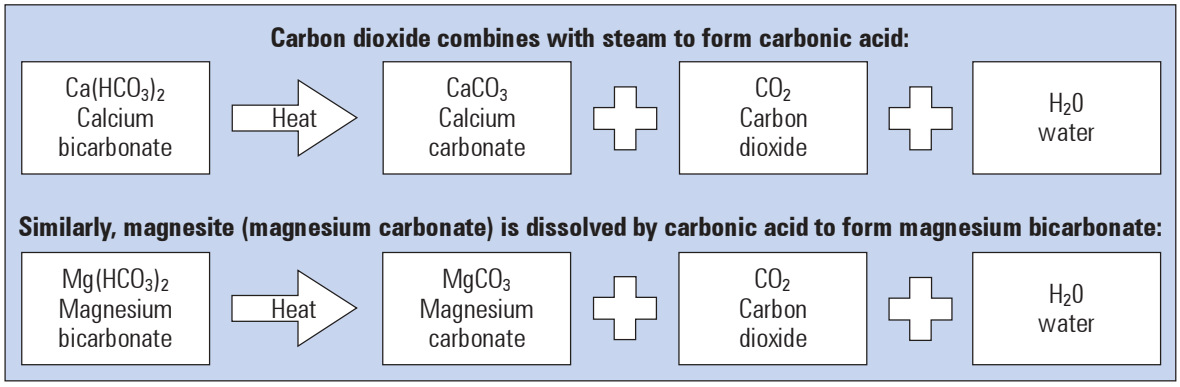


Fig. 3.9.4 Non-alkaline or permanent hardness (scale + carbonic acid)

- **Non-alkaline hardness and carbonates (also known as permanent hardness) -** This is also due to the presence of the salts of calcium and magnesium but in the form of sulphates and chlorides. These precipitate out of solution, due to their reduced solubility as the temperature rises, and form hard scale, which is difficult to remove.

  In addition, the presence of silica in boiler water can also lead to hard scale, which can react with calcium and magnesium salts to form silicates which can severely inhibit heat transfer across the fire tubes and cause them to overheat.

Total hardness

Total hardness is not to be classified as a type of hardness, but as the sum of concentrations of calcium and magnesium ions present when these are both expressed as $CaC0_3$. If the water is alkaline, a proportion of this hardness, equal in magnitude to the total alkalinity and also expressed as $CaC0_3$, is considered as alkaline hardness, and the remainder as non-alkaline hardness. (See Figure 3.9.5)

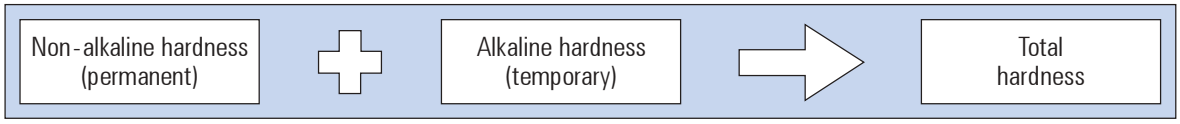


Fig. 3.9.5 Total hardness

Non-scale forming salts

Non-hardness salts, such as sodium salts are also present, and are far more soluble than the salts of calcium or magnesium and will not generally form scale on the surfaces of a boiler, as shown in Figure 3.9.6.

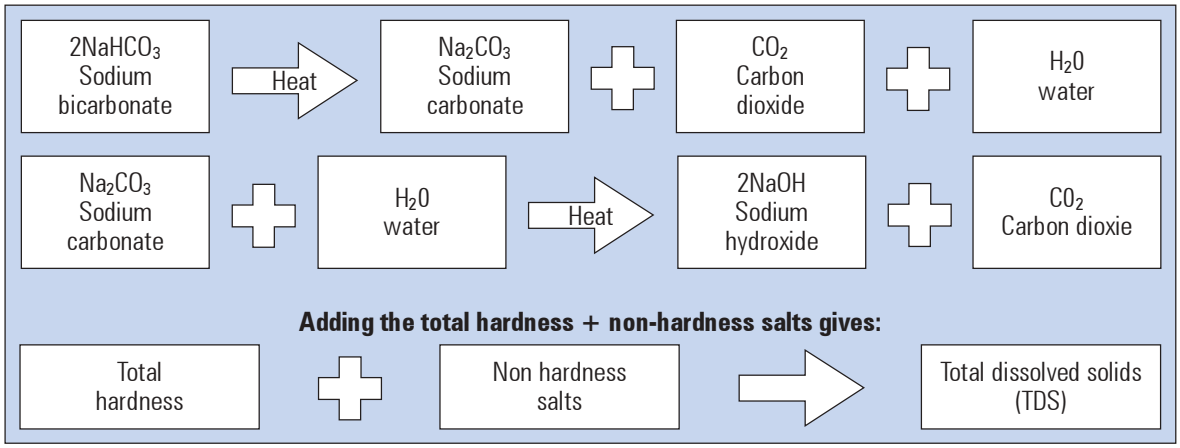


Fig. 3.9.6 The effects of heat

Comparative units

When salts dissolve in water they form electrically charged particles called ions.

The metallic parts (calcium, sodium, magnesium) can be identified as cations because they are attracted to the cathode and carry positive electrical charges.

Anions are non-metallic and carry negative charges - bicarbonates, carbonate, chloride, sulphate, are attracted to the anode.

Each impurity is generally expressed as a chemically equivalent amount of calcium carbonate, which has a molecular weight of 100.

pH value

Another term to be considered is the pH value; this is not an impurity or constituent but merely a numerical value representing the potential hydrogen content of water - which is a measure of the acidic or alkaline nature of the water. Water, H_2O, has two types of ions - hydrogen ions (H^+) and hydroxyl ions (OH^-).

If the hydrogen ions are predominant, the solution will be acidic with a pH value between 0 and 6. If the hydroxyl ions are predominant, the solution will be alkaline, with a pH value between 8 and 14. If there are an equal number of both hydroxyl and hydrogen ions, then the solution will be neutral, with a pH value of 7.

Acids and alkalis have the effect of increasing the conductivity of water above that of a neutral sample. For example, a sample of water with a pH value of 12 will have a higher conductivity than a sample that has a pH value of 7.

Table 3.9.3 shows the pH chart and Figure 3.9.7 illustrates the pH values already mentioned both numerically and in relation to everyday substances.

Table 3.9.3 The pH scale

pH value	Hydrogen ion concentration H+	Hydroxyl ion concentration H-	Nature
0	10^{0}	10^{-14}	Acid
7	10^{-7}	10^{-7}	Neutral
14	10^{-14}	10^{0}	Alkaline

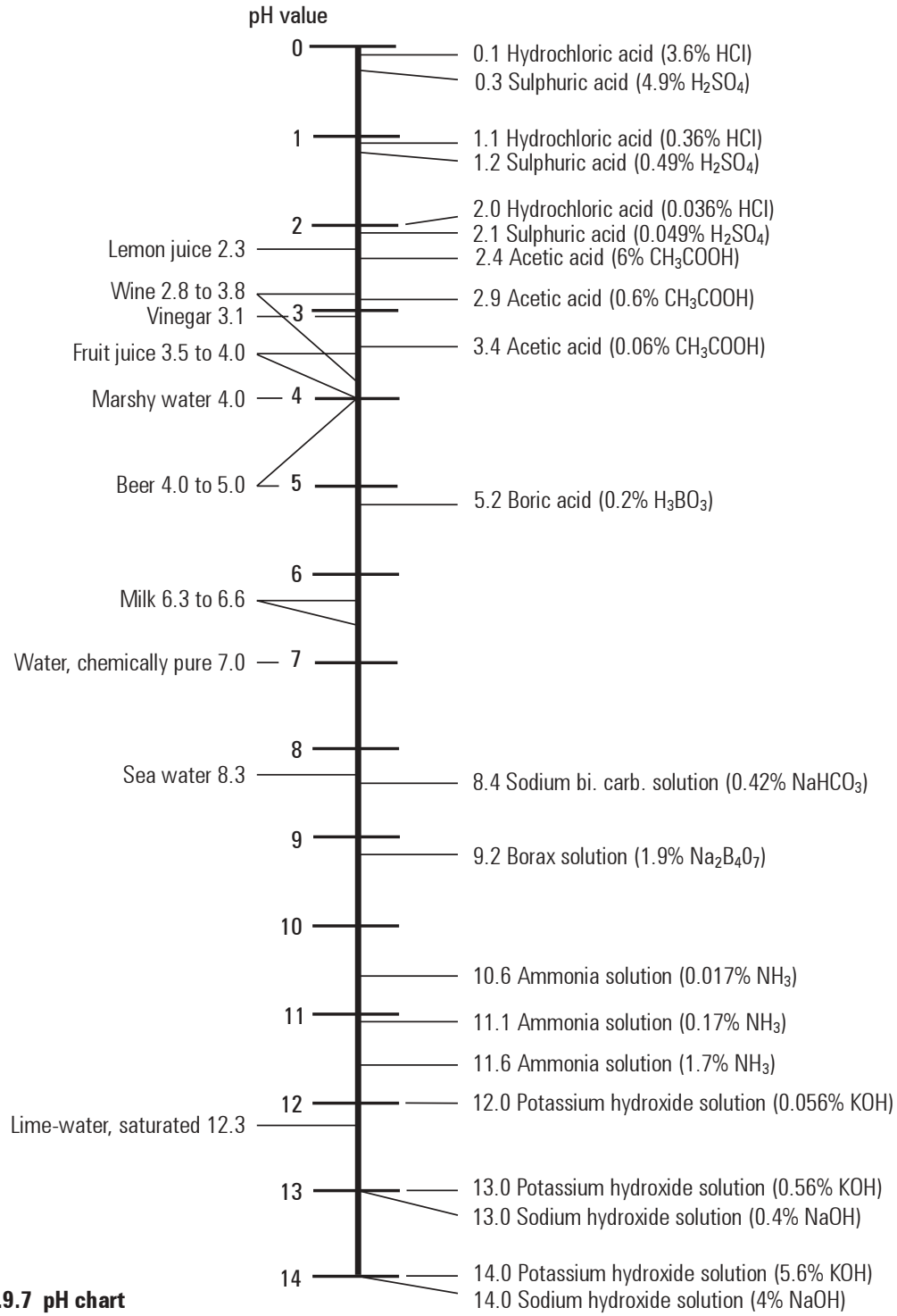


Fig. 3.9.7 pH chart

Questions

1. Temporary hardness salts are reduced by:

a| Raising the water temperature ☐
b| Lowering the water temperature ☐
c| Raising the pH value ☐
d| Letting the water settle ☐

2. What is the effect of CO_2 in a steam system?

a| The formation of scale ☐
b| The formation of sludge ☐
c| Corrosion ☐
d| Acidity ☐

3. Which of the following forms soft scale or sludge?

a| Magnesium sulphate ☐
b| Sodium carbonate ☐
c| Sodium bicarbonate ☐
d| Calcium bicarbonate ☐

4. Which of the following are principal dissolved solids that are scale forming?

a| Carbonates and sulphates of sodium ☐
b| Calcium bicarbonate ☐
c| Carbonates and sulphates of magnesium ☐
d| Bicarbonate of sodium and magnesium ☐

5. What is the effect of temperature on calcium and magnesium sulphates?

a| They separate out as soft scale and sludge ☐
b| They precipitate out of solution and form hard scale ☐
c| Foaming and carryover occurs ☐
d| The TDS is increased ☐

6. What is the treatment for scale forming salts in boiler feedwater?

a| They are chemically treated to modify the pH ☐
b| The feedwater tank is raised to at least 85°C ☐
c| They are chemically treated to produce suspended solids ☐
d| They are removed by filtration means ☐

Answers

1: a, 2: c, 3: d, 4: c, 5: b, 6: c

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Module 3.10

Water for the Boiler

Water for the Boiler

The operating objectives for steam boiler plant include:

- Safe operation.
- Maximum combustion and heat transfer efficiency.
- Minimum maintenance.
- Long working life.

The quality of the water used to produce the steam in the boiler will have a profound effect on meeting these objectives.

There is a need for the boiler to operate under the following criteria:

- **Freedom from scale -** If hardness is present in the feedwater and not controlled chemically, then scaling of the heat transfer surfaces will occur, reducing heat transfer and efficiency - making frequent cleaning of the boiler necessary. In extreme cases, local hot spots can occur, leading to mechanical damage or even tube failure.
- **Freedom from corrosion and chemical attack -** If the water contains dissolved gases, particularly oxygen, corrosion of the boiler surfaces, piping and other equipment is likely to occur.

  If the pH value of the water is too low, the acidic solution will attack metal surfaces. If the pH value is too high, and the water is alkaline, other problems such as foaming may occur.

  Caustic embrittlement or caustic cracking must also be prevented in order to avoid metal failure. Cracking and embrittlement are caused by too high a concentration of sodium hydroxide. Older riveted boilers are more susceptible to this kind of attack; however, care is still necessary on modern welded boilers at the tube ends.

Good quality steam

If the impurities in the boiler feedwater are not dealt with properly, carryover of boiler water into the steam system can occur. This may lead to problems elsewhere in the steam system, such as:

- **Contamination of the surfaces of control valves -** This will affect their operation and reduce their capacity.
- **Contamination of the heat transfer surfaces of process plant -** This will increase thermal resistance, and reduce the effectiveness of heat transfer.
- **Restriction of steam trap orifices -** This will reduce steam trap capacities, and ultimately lead to waterlogging of the plant, and reduced output.

Carryover can be caused by two factors:

1. **Priming -** This is the ejection of boiler water into the steam take-off and is generally due to one or more of the following:
   - Operating the boiler with too high a water level.
   - Operating the boiler below its design pressure; this increases the volume and the velocity of the steam released from the water surface.
   - Excessive steam demand.

2. Foaming - This is the formation of foam in the space between the water surface and the steam off-take. The greater the amount of foaming, the greater the problems which will be experienced.

The following are indications and consequences of foaming:

- Water will trickle down from the steam connection of the gauge glass; this makes it difficult to accurately determine the water level.
- Level probes, floats and differential pressure cells have difficulty in accurately determining water level.
- Alarms may be sounded, and the burner(s) may even 'lockout'. This will require manual resetting of the boiler control panel before supply can be re-established.

These problems may be completely or in part due to foaming in the boiler. However, because foaming is endemic to boiler water, a better understanding of foam itself is required:

- **Surface definition -** Foam on a glass of beer sits on top of the liquid, and the liquid / foam interface is clearly defined. In a boiling liquid, the liquid surface is indistinct, varying from a few small steam bubbles at the bottom of the vessel, to many large steam bubbles at the top.
- **Agitation increases foaming -** The trend is towards smaller boilers for a given steaming rate. Smaller boilers have less water surface area, so the rate at which steam is released per square metre of water area is increased. This means that the agitation at the surface is greater. It follows then that smaller boilers are more prone to foaming.
- **Hardness -** Hard water does not foam. However, boiler water is deliberately softened to prevent scale formation, and this gives it a propensity to foam.
- **Colloidal substances -** Contamination of boiler water with a colloid in suspension, for example. milk, causes violent foaming. Note: Colloidal particles are less than 0.0001 mm in diameter, and can pass through a normal filter.
- **TDS level -** As the boiler water TDS increases, the steam bubbles become more stable, and are more reluctant to burst and separate.

Corrective action against carryover

The following alternatives are open to the Engineering Manager to minimise foaming in the boiler:

- **Operation -** Smooth boiler operation is important. With a boiler operating under constant load and within its design parameters, the amount of entrained moisture carried over with steam may be less than 2%.

  If load changes are rapid and of large magnitude, the pressure in the boiler can drop considerably, initiating extremely turbulent conditions as the contents of the boiler flash to steam. To make matters worse, the reduction in pressure also means that the specific volume of the steam is increased, and the foam bubbles are proportionally larger.

  If the plant conditions are such that substantial changes in load are normal, it may be prudent to consider:

  - Modulating boiler water level controls if on / off are currently fitted.
  - 'Surplussing controls' that will limit the level to which the boiler pressure is allowed to drop.
  - A steam accumulator (see Module 22 of this Block).
  - 'Feed-forward' controls that will bring the boiler up to maximum operating pressure before the load is applied.
  - 'Slow-opening' controls that will bring plant on-line over a pre-determined period.

- **Chemical control -** Anti-foaming agents may be added to the boiler water. These operate by breaking down the foam bubbles. However, these agents are not effective when treating foams caused by suspended solids.
- **Control of TDS -** A balance has to be found between:
  - A high TDS level with its attendant economy of operation.
  - A low TDS level which minimises foaming.
- **Safety -** The dangers of overheating due to scale, and of corrosion due to dissolved gases, are easy to understand. In extreme cases, foaming, scale and sludge formation can lead to the boiler water level controls sensing improper levels, creating a danger to personnel and process alike.

External water treatment

It is generally agreed that where possible on steam boilers, the principal feedwater treatment should be external to the boiler.

A summary of the treated water quality that might be obtained from the various processes, based on a typical hard raw water supply, is shown in Table 3.9.2. This is the water that the external treatment plant has to deal with.

External water treatment processes can be listed as:

- **Reverse osmosis -** A process where pure water is forced through a semi-permeable membrane leaving a concentrated solution of impurities, which is rejected to waste.
- **Lime; lime/soda softening -** With lime softening, hydrated lime (calcium hydroxide) reacts with calcium and magnesium bicarbonates to form a removable sludge. This reduces the alkaline (temporary) hardness. Lime/soda (soda ash) softening reduces non-alkaline (permanent) hardness by chemical reaction.
- **Ion exchange -** Is by far the most widely used method of water treatment for shell boilers producing saturated steam. This module will concentrate on the following processes by which water is treated: Base exchange, Dealkalisation and Demineralisation.

Ion exchange

An ion exchanger is an insoluble material normally made in the form of resin beads of 0.5 to 1.0 mm diameter. The resin beads are usually employed in the form of a packed bed contained in a glass reinforced plastic pressure vessel. The resin beads are porous and hydrophilic - that is, they absorb water. Within the bead structure are fixed ionic groups with which are associated mobile exchangeable ions of opposite charge. These mobile ions can be replaced by similarly charged ions, from the salts dissolved in the water surrounding the beads.

Base exchange softening

This is the simplest form of ion exchange and also the most widely used. The resin bed is initially activated (charged) by passing a 7 - 12% solution of brine (sodium chloride or common salt) through it, which leaves the resin rich in sodium ions. Thereafter, the water to be softened is pumped through the resin bed and ion exchange occurs. Calcium and magnesium ions displace sodium ions from the resin, leaving the flowing water rich in sodium salts. Sodium salts stay in solution at very high concentrations and temperatures and do not form harmful scale in the boiler.

From Figure 3.10.1 it can be seen that the total hardness ions are exchanged for sodium. With sodium base exchange softening there is no reduction in the total dissolved solids level (TDS in parts per million or ppm) and no change in the pH. All that has happened is an exchange of one group of potentially harmful scale forming salts for another type of less harmful, non-scale forming salts. As there is no change in the TDS level, resin bed exhaustion cannot be detected by a rise in conductivity (TDS and conductivity are related). Regeneration is therefore activated on a time or total flow basis.

Softeners are relatively cheap to operate and can produce treated water reliably for many years. They can be used successfully even in high alkaline (temporary) hardness areas provided that at least 50% of condensate is returned. Where there is little or no condensate return, a more sophisticated type of ion exchange is preferable.

Sometimes a lime/soda softening treatment is employed as a pre-treatment before base exchange. This reduces the load on the resins.

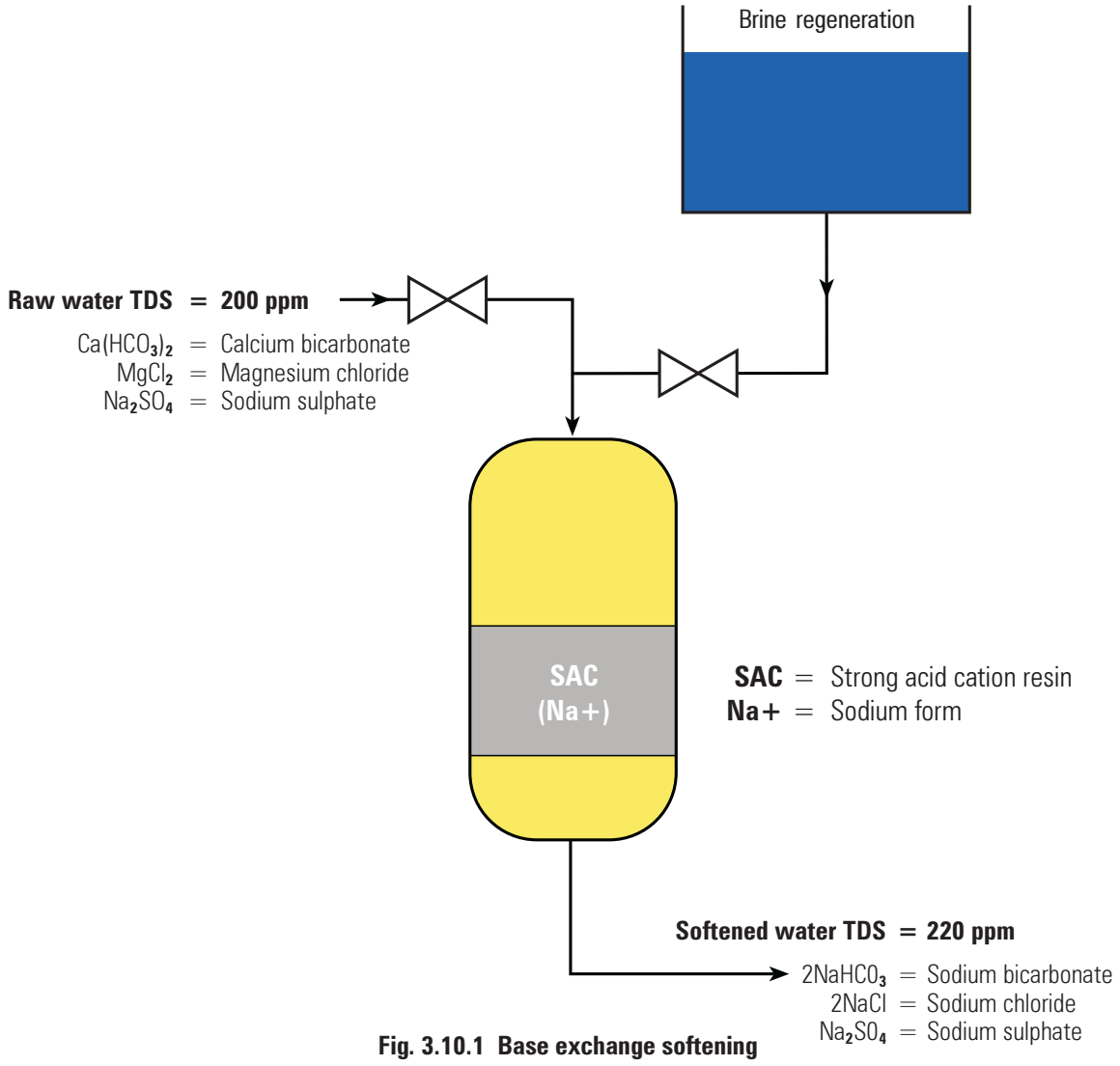
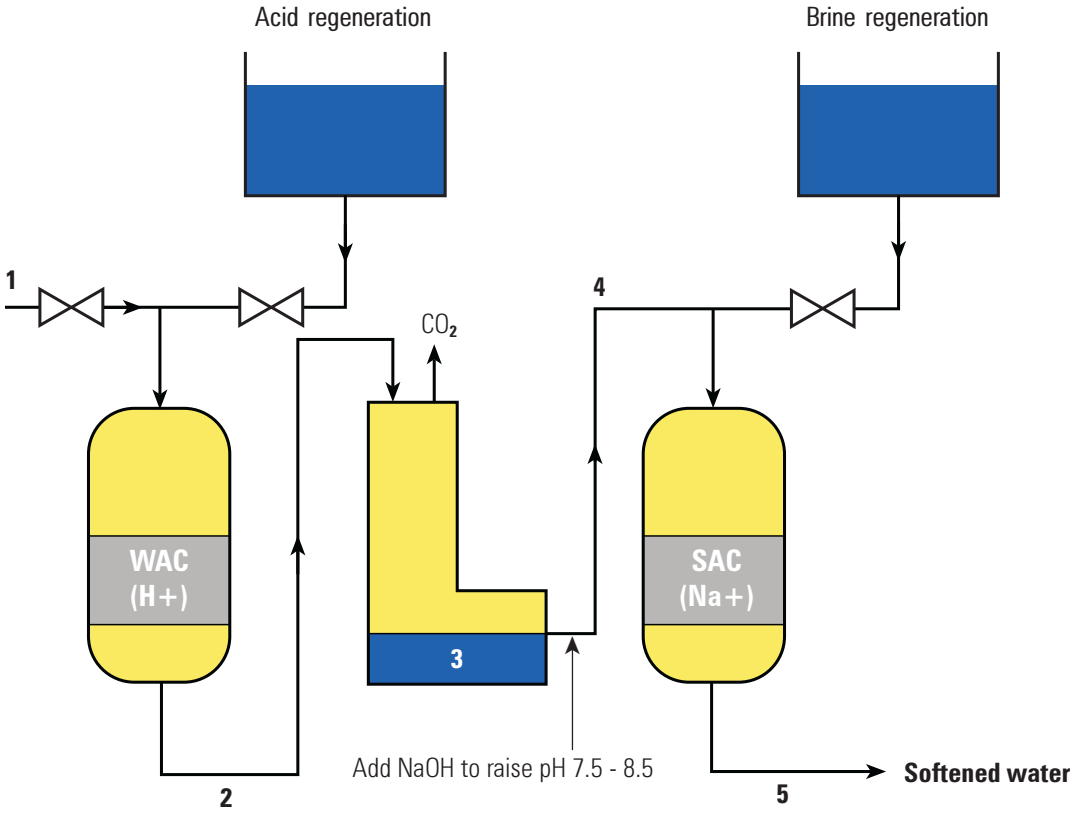


Fig. 3.10.1 Base exchange softening

Dealkalisation

The disadvantage of base exchange softening is that there is no reduction in the TDS and alkalinity. This may be overcome by the prior removal of the alkalinity and this is usually achieved through the use of a dealkaliser.

There are several types of dealkaliser but the most common variety is shown in Figure 3.10.2. It is really a set of three units, a dealkaliser, followed by a degasser and then a base exchange softener.



WAC = Weak acid cation resin
H+ = Hydrogen form

SAC = Strong acid cation resin
Na+ = Sodium form

1	2	3	4	5
$Ca(HCO_3)_2$	$2H_2CO_3$	H_2O	H_2O	H_2O
$MgCl_2$	$MgCl_2$	$MgCl_2$	$MgCl_2$	2NaCl
Na_2SO_4	Na_2SO_4	Na_2SO_4	Na_2SO_4	Na_2SO_4
pH 7.6	pH 4.5 – 5.0	pH 4.5 – 5.0		pH 7.5 – 8.5

Fig. 3.10.2 A dealkalisation plant

Dealkaliser

The system shown in Figure 3.10.3 is sometimes called 'split-stream' softening. A dealkaliser would seldom be used without a base exchange softener, as the solution produced is acidic and would cause corrosion, and any permanent hardness would pass straight into the boiler.

A dealkalisation plant will remove temporary hardness as shown in Figure 3.10.3. This system would generally be employed when a very high percentage of make-up water is to be used.

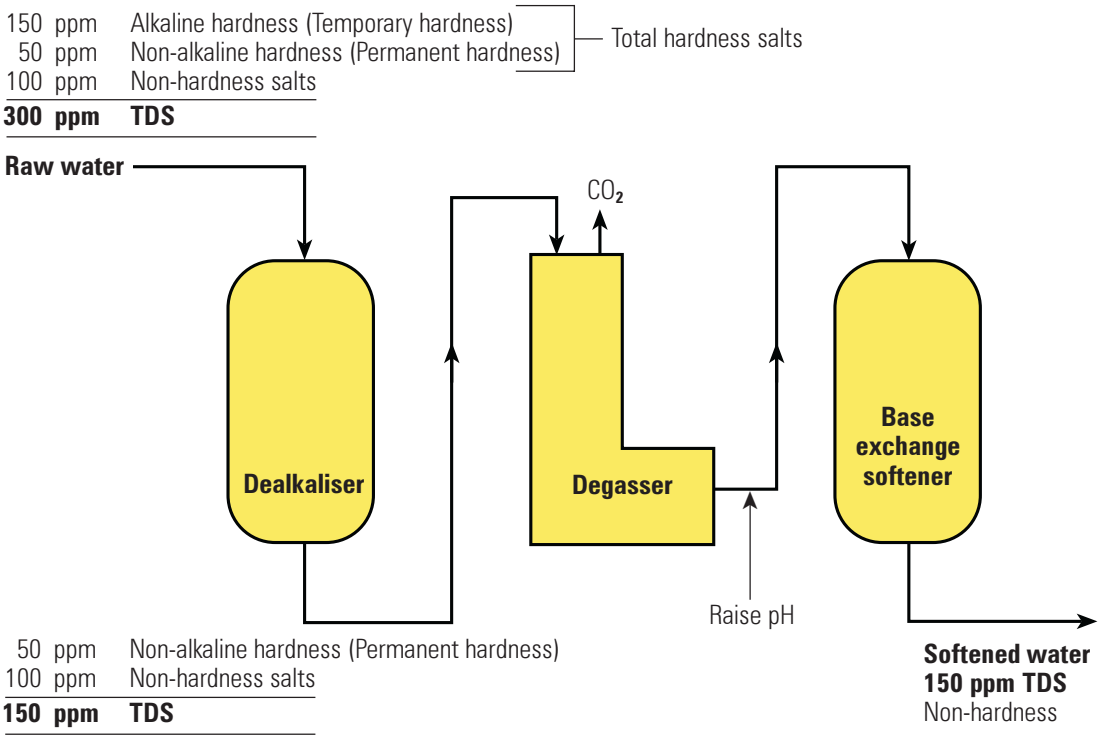


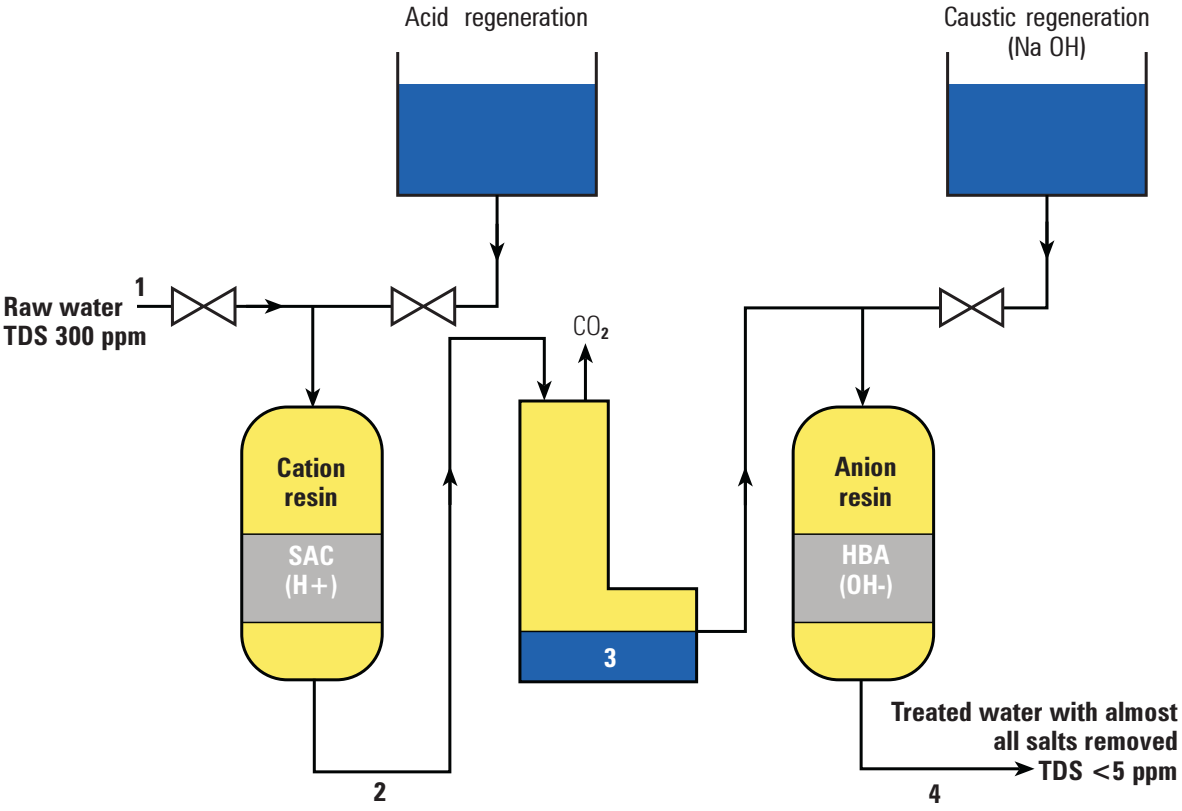
Fig. 3.10.3 The dealkalisation process

Demineralisation

This process will remove virtually all the salts. It involves passing the raw water through both cation and anion exchange resins (Figure 3.10.4). Sometimes the resins may be contained in one vessel and this is termed 'mixed bed' demineralisation.

The process removes virtually all the minerals and produces very high quality water containing almost no dissolved solids. It is used for very high pressure boilers such as those in power stations.

If the raw water has a high amount of suspended solids this will quickly foul the ion exchange material, drastically increasing operating costs. In these cases, some pre-treatment of the raw water such as clarification or filtration may be necessary.



SAC = Strong acid cation resin
H+ = Hydrogen form

HBA = Hydroxyl based anion resin
OH- = Hydroxyl form

1	2	3	4
$Ca(HCO_3)_2$	$2H_2CO_3$	H_2O	H_2O
$MgCl_2$	2HCl	2HCl	H_2O
Na_2SO_4	H_2SO_4	H_2SO_4	H_2O
Na_2SiO_3	H_2SiO_4	H_2SiO_3	H_2O
pH 7.6	pH 2.0 – 2.5	pH 2.0 – 2.5	pH 8.5 – 9.0

Fig. 3.10.4 Demineralisation

Selection of external water treatment plant

Looking at Table 3.10.1, it is tempting to think that a demineralisation plant should always be used. However, each system has a capital cost and a running cost, as Table 3.10.2 illustrates, plus the demands of the individual plant need to be evaluated.

Table 3.10.1 Water quality versus treatment process

Process	Hardness ppm		Non-hardness salts ppm	TDS ppm
	Alkaline	Non-alkaline		
Raw water	200	50	60	310
Lime	30	50	58	138
Lime/soda	30	0	108	138
Lime/base exchange	5	0	133	138
Base exchange	5	0	255	260
Dealkalisation	5	50	60	115
Dealkalisation + base exchange	5	0	110	115
Demineralisation	1	0	2	3
Reverse osmosis	20	5	6	31

Table 3.10.2 Relative costs of water treatment processes

Type of system	Comparative cost scale	
	Capital cost	Running cost
Base exchange	1	1
Dealkalisation + base exchange	4	2
Demineralisation	8	3

Shell boiler plant

Generally, shell boilers are able to tolerate a fairly high TDS level, and the relatively low capital and running costs of base-exchange softening plants (see Table 3.10.2) will usually make them the first choice.

If the raw water supply has a high TDS value, and/or the condensate return rate is low (<40%), there are a few options which may be considered:

- Pre-treatment with lime/soda which will cause the alkaline hardness to precipitate out of solution as calcium carbonate and magnesium hydroxide, and then drain from the reaction vessel.
- A dealkalisation plant to reduce the TDS level of the water supplied to the boiler plant.

Water-tube boiler plant

Water-tube boiler plant is much less tolerant of high TDS levels, and even less so as the pressure increases. This is due to a number of reasons, including:

- Water-tube boilers have a limited water surface area in the steam drum, relative to the evaporation rate.

  This results in very high steam release rates per unit of water area, and turbulence.

- Water-tube boilers tend to be higher rated, perhaps over 1 000 tonnes / h of steam. This means that even a small percentage blowdown can represent a high mass to be blown down.

- Water-tube boilers tend to operate at higher pressures, usually up to 150 bar g. The higher the pressure, the greater the energy contained in the blowdown water.

  Higher pressures also mean higher temperatures. This means that the materials of construction will be subjected to higher thermal stresses, and be operating closer to their metallurgical limitations. Even a small amount of internal contamination hindering the heat transfer from tubes to water may result in the tubes overheating.

- Water-tube boilers often incorporate a superheater.

  The dry saturated steam from the steam drum may be directed to a superheater tubes situated in the highest temperature area of the furnace. Any carryover of contaminated water with the steam would coat the inside of the superheater tubes, and inhibit heat transfer with potentially disastrous results.

The above factors mean that:

- High quality water treatment is essential for the safe operation of this type of plant.
- It may be economically viable to invest in a water treatment plant that will minimise blowdown rates.

In each of these cases, the selection will often be a demineralisation or a reverse osmosis plant.

Summary

The quality of raw water is obviously an important factor when choosing a water treatment plant. Although TDS levels will affect the performance of the boiler operation, other issues, such as total alkalinity or silica content can sometimes be more important and then dominate the selection process for water treatment equipment.

Questions

1. Good quality water is required to:

a| Avoid corrosion ☐

b| Avoid scaling ☐

c| Produce good quality steam ☐

d| All of the above ☐

2. Good quality water will result in:

a| Good surface definition, enabling level controls to operate correctly ☐

b| Better performance of plant equipment due to avoidance of contamination ☐

c| The export of good quality steam ☐

d| All of the above ☐

3. Temporary hardness salts

a| Stay in solution, whatever the water temperature ☐

b| Come out of solution as the water temperature increases ☐

c| Modify the pH of the feedwater ☐

d| Increase as the water temperature increases ☐

4. A base exchange water softener is regenerated using:

a| A sodium sulphate solution ☐

b| A sodium chloride solution ☐

c| A sodium bicarbonate solution ☐

d| A magnesium chloride solution ☐

5. A base exchange water softener regeneration cycle is usually initiated on the basis of:

a| Time or total flow ☐

b| A change in TDS within the vessel ☐

c| Shift change ☐

d| A change in colour of the discharged water ☐

6. A base exchange water softener is generally chosen for shell boiler plant because:

a| It represents a good balance between capital cost, operating cost and effectiveness ☐

b| The operating procedure is similar to shell boilers, so on-site personnel already have the necessary skills ☐

c| It is cheaper in the short term, however, the longer service life of dealkalisation plants means that they represent a saving in the longer term ☐

d| They are more widely available ☐

Answers

1: d, 2: d, 3: b, 4: b, 5: a, 6: a

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Module 3.11

The Feedtank and Feedwater Conditioning

The Feedtank and Feedwater Conditioning

The importance of the boiler feedtank, where boiler feedwater and make-up water are stored and into which condensate is returned, is often underestimated. Most items of plant in the boiler house are duplicated, but it is rare to have two feedtanks and this crucial item is often the last to be considered in the design process.

The feedtank is the major meeting place for cold make-up water and condensate return. It is best if both of these, together with flash steam from the blowdown system, flow through sparge pipes installed well below the water surface in the feedwater tank. The sparge pipes must be made from stainless steel and be adequately supported.

Operating temperature

It is important that the water in the feedtank is kept at a high enough temperature to minimise the content of dissolved oxygen and other gases. The correlation between the water temperature and its oxygen content in a feedtank can be seen in Figure 3.11.1.

If a high proportion of make-up water is used, heating the feedwater can substantially reduce the amount of oxygen scavenging chemicals required.

Example 3.11.1
Cost savings associated with reducing the dissolved oxygen in feedwater by heating.

Basis for calculation:

- The standard dosing rate for sodium sulphite is 8 ppm per 1 ppm of dissolved oxygen.
- It is usual to add an additional 4 ppm to maintain a reserve in the boiler.
- Typical liquid catalysed sodium sulphite contains only 45% sodium sulphite.

For the example:

The average generation rate of the boiler = 10 000 kg/h

The boiler operation per annum = 6 000 h/year

The cost of sodium sulphite = £1 000/1 000 kg = £1/kg

Calculation 1

Feedtank temperature = 60°C

From Figure 3.11.1, the oxygen content of water at 60°C = 4.8 ppm

Amount of sodium sulphite required = (4.8 x 8) + 4 = 42.4 ppm

Amount of sodium sulphite required (45% concentrated) = 42.4 ppm x $\frac{100}{45}$ = 94.2 ppm

Annual amount of sodium sulphite required = 10 000 kg/h x 6 000 h/year x $\frac{94.2 \text{ ppm dissolved } O_2}{1\,000\,000 \text{ ppm to } 1 \text{ kg}}$

Annual amount of sodium sulphite required = 5 653 kg/year

Annual cost of sodium sulphite = 5 653 kg/year x £1/kg

Annual cost of sodium sulphite = £5 653/year

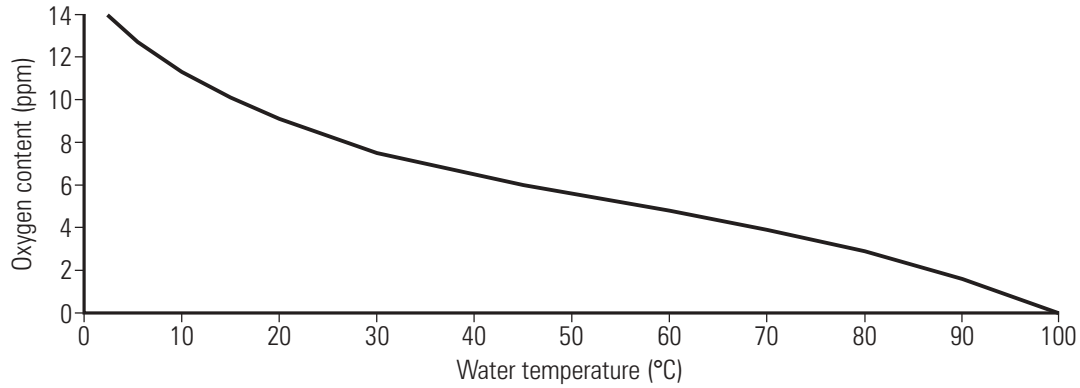


Fig. 3.11.1 Water temperature versus oxygen content

Calculation 2

Feedtank temperature = 85°C

From Figure 3.11.1, the oxygen content of water at 85°C = 2.3 ppm

Amount of sodium sulphite required = (2.3 x 8) + 4 = 22.4

Amount of sodium sulphite required (45% concentrated) = 22.4 ppm x $\frac{100}{45}$ = 49.8 ppm

Annual amount of sodium sulphite required = 10 000 kg/h x 6 000 h/year x $\frac{49.8 \text{ ppm dissolved } O_2}{1\,000\,000 \text{ ppm to } 1 \text{ kg}}$

Annual amount of sodium sulphite required = 2 988 kg/year

Annual cost of sodium sulphite = 2 988 kg/year x £1/kg

Annual cost of sodium sulphite = £2 988/year

Annual cost saving

This is the difference between the two values calculated:

Annual cost saving = £5 653 - £2 988

Annual cost saving = £2 665/year

Percentage of annual cost saving = $\frac{£2\,665}{£5\,653}$ x $\frac{100}{1}$

Percentage of annual cost saving = 47%

Obviously a cost is involved in heating the feedtank, but since the water temperature would be increased by the same amount inside the boiler, this is not additional energy, only the same energy used in a different place.

The only real loss is the extra heat lost from the feedtank itself. Provided the feedtank is properly insulated, this extra heat loss will be almost insignificant.

An important additional saving is reducing the amount of sodium sulphite added to the boiler feedwater. This will reduce the amount of bottom blowdown needed, and this saving will more than compensate for the small additional heat loss from the boiler feedtank.

To avoid damage to the boiler itself

The boiler undergoes thermal shock when cold water is introduced to the hot surfaces of the boiler wall and its tubes. Hotter feedwater means a lower temperature difference and less risk of thermal shock.

To maintain the designed output

The lower the boiler feedwater temperature, the more heat is required in the boiler to produce steam. It is important to maintain the feedtank temperature as high as possible, to maintain the required boiler output.

Cavitation of the boiler feedpump

Caution: very high condensate return rates (typically over 80%) may result in excessive feedwater temperature, and cavitation in the feedpump.

If water close to boiling point enters a pump, it is liable to flash to steam at the low pressure area at the eye of the pump impeller. If this happens, bubbles of steam are formed as the pressure drops below the water vapour. When the pressure rises again, these bubbles will collapse and water flows into the resulting cavity at a very high velocity.

This is known as 'cavitation'; it is noisy and can seriously damage the pump.

To avoid this problem, it is essential to provide the best possible Net Positive Suction Head (NPSH) to the pump so that the static pressure is as high as possible. This is greatly aided by locating the feedtank as high as possible above the boiler, and generously sizing the suction pipework to the feedpump (Figure 3.11.2).

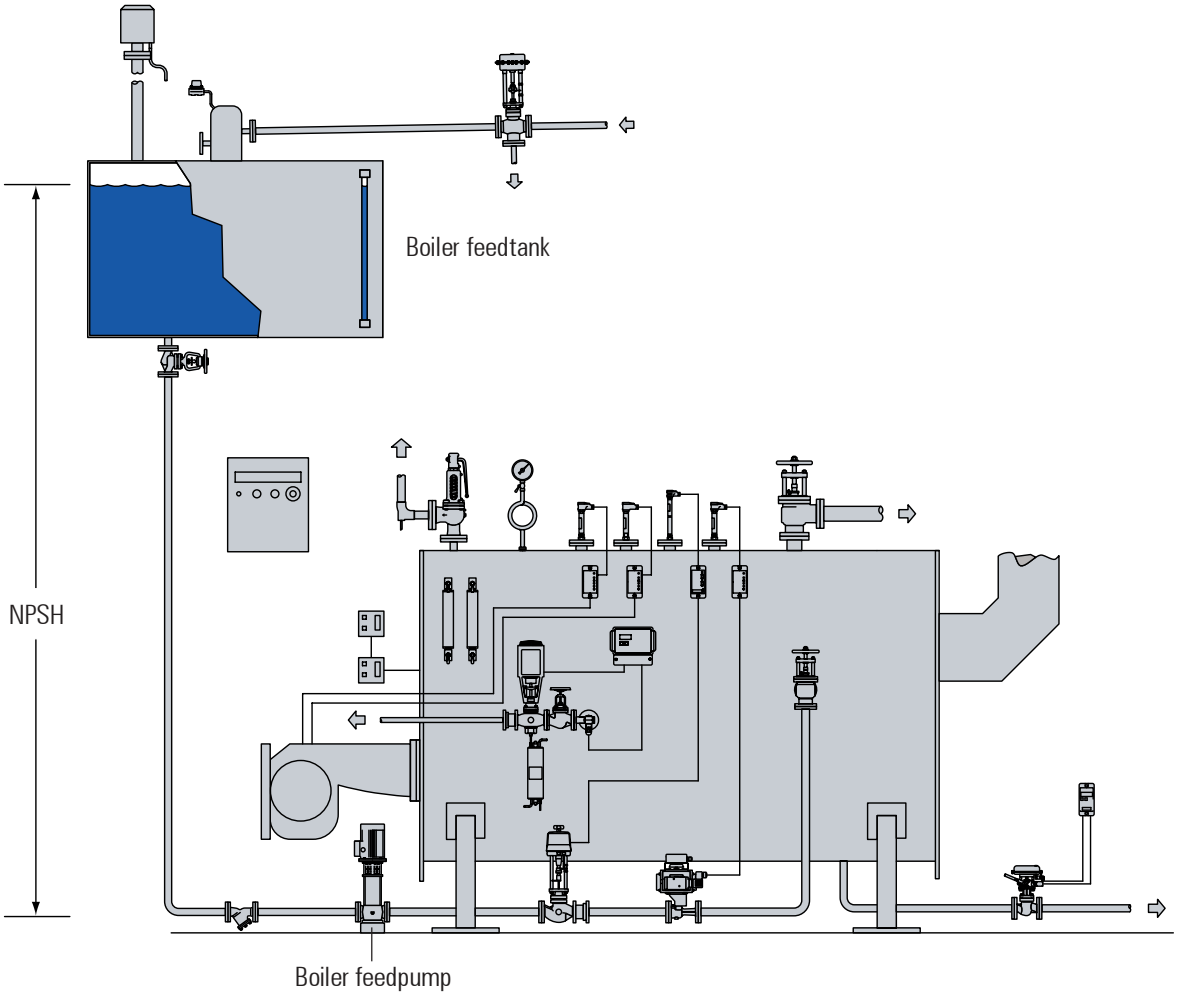


Figure 3.11.2 NPSH above feedpump

Feedtank design

The feedtank (Figure 3.11.3) can influence the way in which the whole boiler house operates in several ways. By careful design of the feedtank and associated systems, substantial savings can be made in energy and water treatment chemicals together with increased reliability of operation.

Whilst cylindrical feedtanks, both vertical and horizontal, are not uncommon in other parts of the world, the rectangular shape is most regularly used in the UK. This normally offers the maximum volume of water storage for the floor area that it occupies.

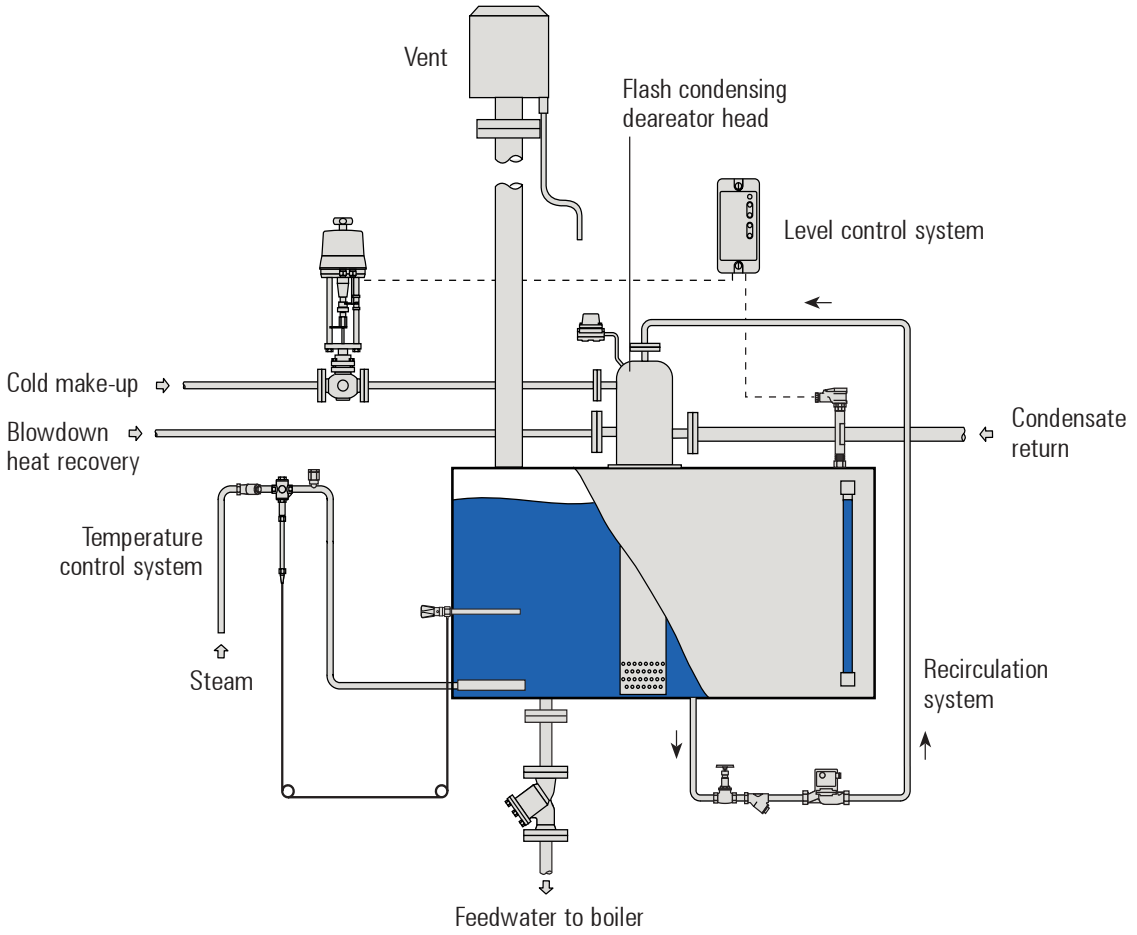


Fig. 3.11.3 Boiler feedtank

Feedtank materials:

- **Cast iron -** Cast iron tanks are usually assembled from rectangular sections:
  - Problems often arise from leaks at the section joints, and they are prone to corrosion.
- **Carbon steel -** Probably the most common construction material for feedtanks:
  - Uncoated, it is a relatively low cost material but it is extremely susceptible to corrosion. This weakness can be improved by applying suitable coatings to the surface, but the cost of this can be more than the cost of the tank, especially as the coating will also need regular maintenance.
- **Plastic -** This material is not usually suitable for feedtanks due to the high cost of materials able to withstand the relatively high temperatures involved. However, plastic is a suitable material for the cold make-up water tank.
- **Austenitic stainless steel -** The enhanced life of a properly made feedtank in this material will invariably justify the higher initial cost. Type 304L is generally selected as the most appropriate grade of stainless steel.

Feedtank capacity

The feedtank provides a reserve of water to cover the interruption of make-up water supply. Traditional practice is to have a feedtank with sufficient capacity to allow one hour of steaming at maximum boiler evaporation. For larger plants this may be impractical and an alternative might be to have a smaller 'hotwell' feedtank with additional cold treated water storage. It should also have sufficient capacity above its normal working level to accommodate any surges in the rate of condensate return. This capacity is referred to as 'ullage'.

A high condensate return rate can occur at start-up when condensate lying in the plant and pipework is suddenly returned to the tank, where it may be lost to drain through the overflow. If this occurs, it may be wise to review the condensate return system, to control the return rate and avoid wastage.

Feedtank construction

The following notes may be useful in designing a feedtank:

- **Stiffening -** The tank should be fully welded and it is very important to use adequate stiffening to strengthen the tank sides and top and to provide adequate support for the base. Failure to do so will result in excessive flexing and premature failure.
- **Piping connections -** All flanged piping connections should stand-off at least 150 mm to facilitate insulation. All screwed connections should stand-off by at least 20 mm.
- **Lifting lugs -** It is essential to fit lifting lugs to allow safe and easy installation.

Feedtank piping

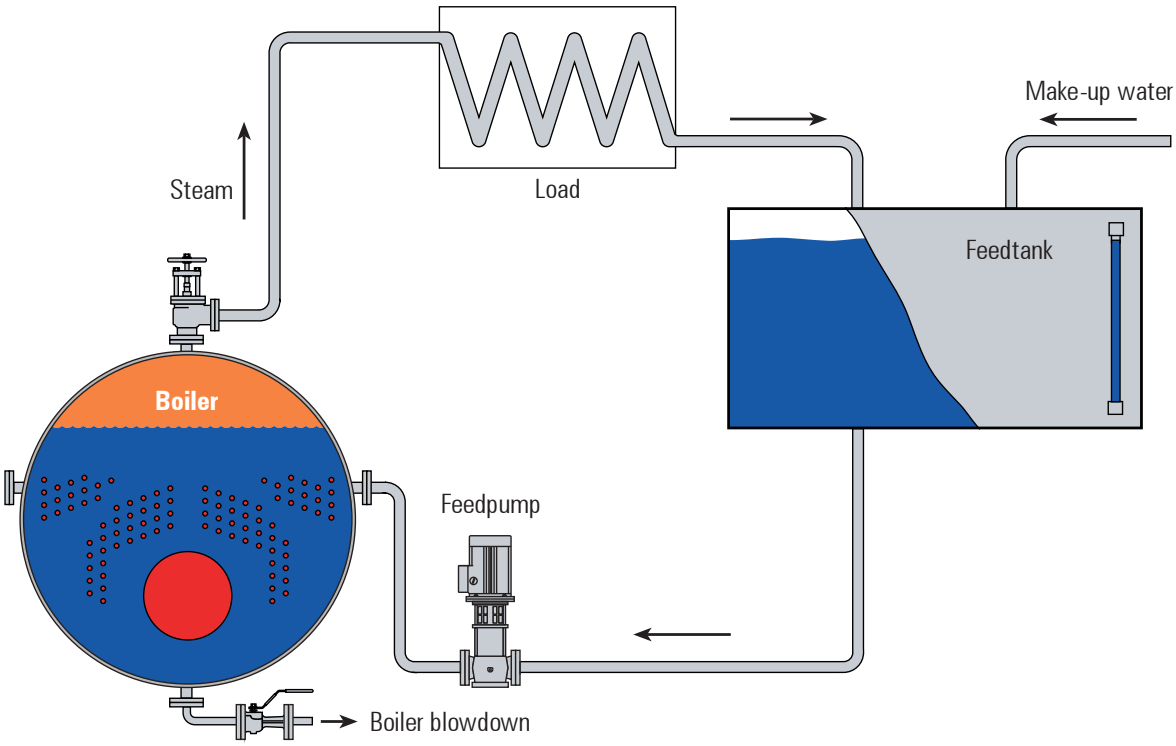


Fig. 3.11.4 The feedtank in relation to the other elements within a steam system

Condensate return

As steam is generated, the water within the boiler evaporates and is replaced by pumping feedwater into the boiler.

As the steam passes around the system to the various items of steam-using plant, it changes state back to condensate, which is, essentially, very good quality hot water.

Unless some contamination is likely (perhaps due to the process), this condensate is ideal boiler feedwater. It makes economic sense, therefore, to return as much as possible for re-use. In reality, it is almost impossible to return all the condensate; some steam may have been injected directly into the process for applications such as humidification and steam injection, and there will usually be water losses from the boiler itself, for instance, via blowdown. Make-up (chemically treated) water will therefore have to be introduced to the system to maintain the correct working levels.

The return of condensate represents huge potential for energy savings in the boiler house. Condensate has a high heat content and approximately 1% less fuel is required for every 6°C temperature rise in the feedtank.

Figure 3.11.5(a) shows the formation of steam at 10 bar g when the boiler is supplied with cold feedwater at 10°C. The portion at the bottom of the diagram represents the enthalpy (42 kJ/kg) available in the feedwater. A further 740 kJ/kg of heat energy has to be added to the water in the boiler before saturation temperature at 10 bar g is reached.

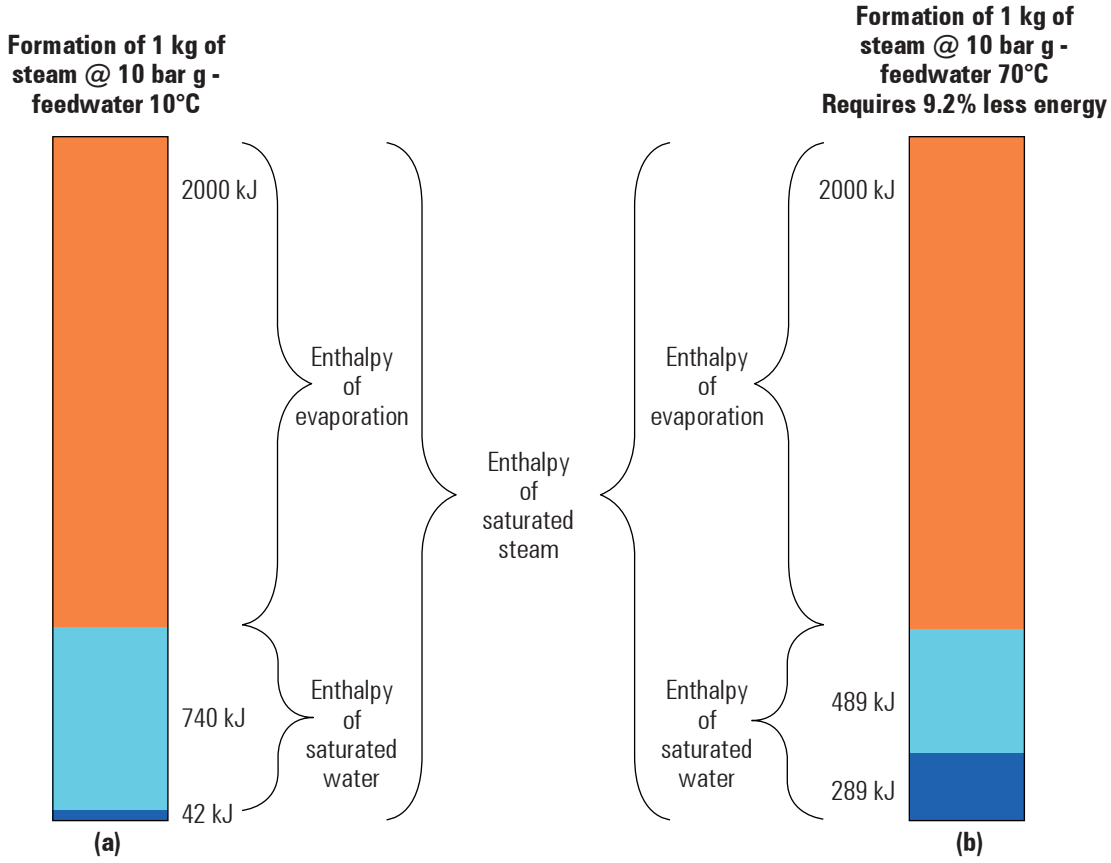


Fig. 3.11.5 Comparison of energy to raise steam at 10 bar g

Figure 3.11.5(b) again shows the formation of steam at 10 bar g, but this time the boiler is fed with feedwater heated to 70°C by returning more condensate.

The increased enthalpy contained in the feedwater means that the boiler now only has to add 489 kJ/kg of heat energy to bring it up to saturation temperature at 10 bar g. This represents a saving of 9.2% in the energy needed to raise steam at this same pressure.

The returned condensate is virtually pure water and this saves not only on water costs but also on water treatment chemicals, which reduces the losses associated with blowdown.

If pressurised condensate is being returned then flash steam will be released in the feedtank. This flash steam needs to be condensed to ensure that both the heat and water content are recovered. The traditional method of doing this has been to introduce it into the feedtank through sparge pipes, but a more modern and effective method is to use a flash condensing deaerator head where cold make-up, condensate return and flash steam are mixed (see Figure 3.11.6).

Flash steam from heat recovery systems
A heat recovery system may, for example, recover flash steam from the boiler blowdown. It is another opportunity to use recovered heat to raise the feedtank temperature and so save fuel.

As with pressurised condensate, the flash steam needs to be condensed. Traditionally, this was achieved using sparge pipes, but a modern and much more effective method is the flash condensing deaerator head.

Make-up water
This is cold water from the water treatment plant that makes up any losses in the system.

Many water treatment plants need a substantial flow through them in order to achieve optimum performance. A 'trickle' flow as a result of a modulating control into the feedtank can, for example, have an adverse effect on the performance of a softener. For this reason a small plastic or galvanised steel cold make-up tank is often fitted. The flow from the softener is controlled 'on/off' into the make-up tank. From there a modulating valve controls its flow into the feedtank.

This type of installation leads to 'smoother' operation of the boiler plant. To avoid the relatively cold make-up water sinking directly to the bottom of the tank (where it will be drawn directly into the boiler feedwater line), and to ensure uniform temperature distribution, it is common practice to sparge the make-up water into the feedtank at a higher level.

Steam injection
As previously mentioned, there are significant advantages to maintaining the feedtank contents at a high temperature. One of the most convenient ways of achieving this higher temperature is by injecting steam into the feedtank.

Vent
The feedtank must be vented to prevent any build-up of pressure. As a guide, this vent will range in size from DN80 on a 2 000 litre tank to DN250 on a 30 000 litre tank. The vent should be fitted with a vent head, which incorporates an internal baffle to separate entrained water from the steam for discharge through a drain connection.

Overflow
This should be fitted with a 'U' tube water seal to prevent flash steam loss.

Feedpump take-off
If the take-off is from the base of the feedtank there should be a 50 mm internal stub to prevent any dirt in the bottom of the tank from entering the pipeline. It should be generously sized so that friction losses are minimised, and the net positive suction head (NPSH) to the feedpump is maximised.

Drain
A drain connection should be fitted in the bottom of the feedtank to facilitate its emptying for inspection.

Insulation
The feedtank should be adequately insulated to prevent heat losses. The advice of a reputable insulation specialist should be sought in selecting the correct material and economic thickness.

Inspection opening
An adequately sized inspection opening should be fitted to enable internal inspection and the fitting of ancillaries, as appropriate.

Water level control

Traditionally, float controls have been used for this application. Modern controls use level probes, which will give an output signal to modulate a control valve. Not only does this type of system require less maintenance but, with the use of an appropriate controller, a single probe may incorporate level alarms and remote indicating devices.

Level probes can be arranged to signal high water level, the normal working (or control) water level, and low water level. The signals from the probe can be linked to a control valve on the cold water make-up supply. The probe is fitted with a protection tube inside the feedtank to protect it from turbulence, which can result in false readings.

Water level indicator

A local level indicator or water level gauge glass on the feedtank is recommended, allowing the viewing of the contents for confirmation purposes, and for commissioning level probes.

Temperature gauge

This can be a local or remote reading device.

Deaerators

Atmospheric deaerator head

The mixing unit of a deaerator head brings together all the incoming flows. It mixes the high oxygen content cold make-up water with flash steam from the condensate and the blowdown heat recovery system. Oxygen and other gases are released from the cold water and can be automatically removed through a vent before the water enters the main feedtank.

The deareator head considerably reduces the amount of steam that would normally be expected to emanate from the tank under working conditions. Because of this, properly designed atmospheric deareator tanks fitted with deareator heads require less venting capacity than an ordinary tank fitted with a vented lid. Typically, vent sizes on an atmospheric deareator tank vary from DN80 on a 2000 L tank, to DN250 on a 30 000 L tank.

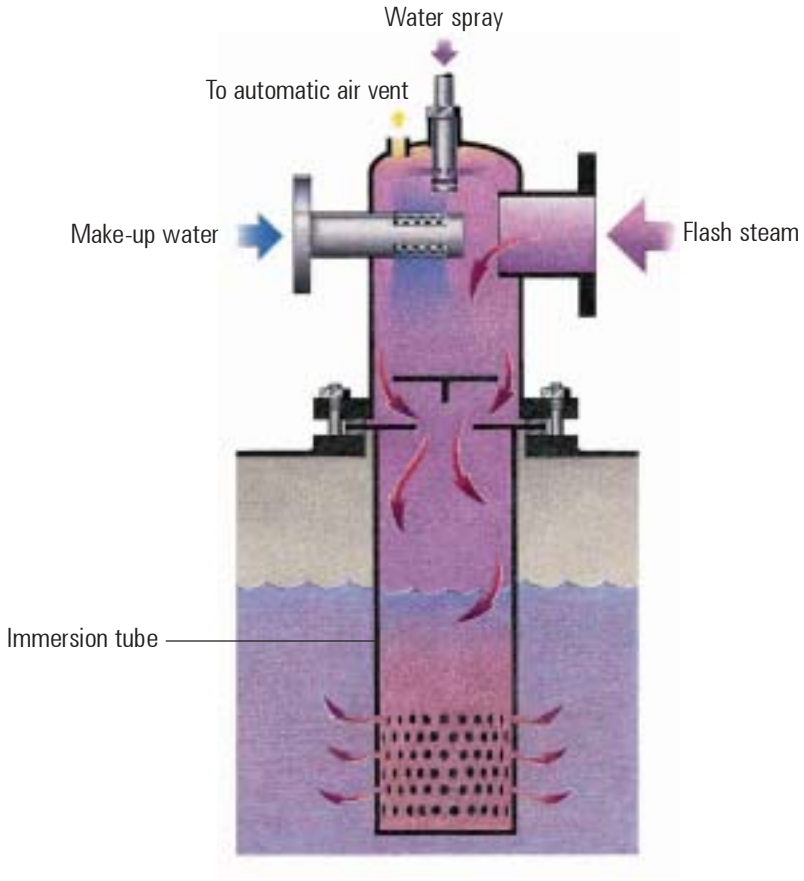


Fig. 3.11.6 Atmospheric deaerator

Pressurised deaerator

On larger boiler plants, pressurised deaerators are sometimes installed and live steam is used to bring the feedwater up to approximately 105°C to drive off the oxygen. Pressurised deaerators are usually thermally efficient and will reduce dissolved oxygen to very low levels.

Pressurised deaerators:

- Must be fitted with controls and safety devices.
- Are classified as pressure vessels, and will require periodic, formal inspection.

This means that pressurised deaerators are expensive, and are only justified in very large boiler houses. If a pressure deaerator is to be considered, its part load performance (or effective turndown) must be investigated.

A detailed review of pressurised deaerators is given in Module 21 of this Block.

Conditioning treatment

This is additional treatment which supplements external treatment, (for example, the base exchange system) and is generally carried out by adding chemicals in metered amounts, into either the feedwater tank or the feedwater pipeline prior to its entry into the boiler.

The chemical treatment required depends on many factors such as:

- The impurities inherent in the make-up water and its hardness.
- The volume of condensate returned for re-use and its quality in terms of pH value, TDS content, and hardness.
- The design of the boiler and its operating conditions.

Deciding on the type of chemical regime and water treatment system is a matter for a skilled water treatment specialist who should always be consulted.

The purpose of the conditioning treatment is to enhance the treatment of the raw water after it has been processed as far as possible by the main water treatment plant. It ensures quality because, inevitably, there will be some impurities that find a way through the main treatment system. The objectives of water treatment are:

- To prevent scale formation from low remaining levels of hardness which may have escaped treatment.

  Sodium phosphate is normally used for this, and causes the hardness to precipitate to the bottom of the boiler where it can be blown down.

- To deal with any other specific impurities present.

  These will be specific substances for specific applications.

- To maintain the correct chemical balance in the boiler water - to prevent corrosion it needs to be somewhat alkaline and not acidic.

  Typically a 1% caustic solution will be used to achieve a target pH of between 9 and 11. British Standards BS 2486 recommends pH 10.5 - 12.0 for shell boilers @ 10 bar, pH 9 could be used in higher pressure boilers only.

- To condition any suspended matter.

  This will be a flocculant or coagulant, which will cause the suspended matter to agglomerate and sink to the bottom of the boiler from where it can be blown down.

- To provide anti-foaming protection.
- To remove traces of dissolved gases.

  These are primarily oxygen and carbon dioxide and the presence of these dissolved gases in the boiler plant and system will cause corrosion. It is, therefore, necessary to remove and/or neutralise them if damage is to be prevented.

Carbon dioxide

Dissolved carbon dioxide is often present in feedwater in the form of carbonic acid and this causes the pH level to fall. Proper pH control will correct this but carbon dioxide is also released in boilers due to heating of carbonates and bicarbonates. These decompose into caustic soda with the release of carbon dioxide. This may need to be dealt with by use of a condensate corrosion inhibitor, to prevent corrosive attack to the condensate system.

Oxygen

The most harmful of the dissolved gases is oxygen, which can cause pitting of metal. Very small amounts of oxygen can cause severe damage. It can be removed both mechanically and chemically. The amount of dissolved oxygen present is dependent on the temperature of the feedwater; the lower the feedwater temperature, the larger the volume of dissolved oxygen present.

Any remaining oxygen is then dealt with by the addition of a chemical oxygen scavenger such as catalysed sodium sulphite.

8 ppm of sodium sulphite is sufficient to deal with 1 ppm of dissolved oxygen. However, it is usual to add an extra (or 'reserve') of 4 ppm of sodium sulphite because:

- There is a significant danger of corrosive damage.
- The chemical dosing system is usually 'open loop' with water samples taken at intervals, and adjustments made to the dosing rate.
- There is a concern about complete dispersion of the chemical, perhaps due to the method of injection, circulation currents, or stratification within the feedtank.

The total dosing rate, therefore, is 8 ppm of sodium sulphite per 1 ppm of dissolved oxygen plus 4 ppm.

Other oxygen scavengers involve organic compounds or hydrazine. The latter, however, is thought to be carcinogenic, and is not generally used in low and medium pressure plants.

Other 'internal treatment' to provide protection for the boiler and the condensate system can include:

- Neutralising amines - These have a neutralising effect on the acid generated by the solution of carbon dioxide in condensate.
- Filming amines - These create an oil attractive, water repellent film on metal surfaces which is resistant to both carbon dioxide and oxygen.

Further detail on this complicated subject is available from water treatment handbooks and water treatment specialists; this is very much a matter for expert advice and professional analysis. There are however, one or two areas which call for further explanation:

- The main boiler water treatment programme is aimed at changing scale-forming salts into soft or mobile sludges. The sludge conditioners used in the chemical dosing prevent these solids from depositing on metal surfaces and keep them in suspension.
- Under high pressures and temperatures, silica can present a real problem because it can combine with the metal heating surfaces to cause hot spots. Special synthetic polymers can prevent this problem.
- Alkalinity levels in the boiler are particularly important and these are controlled by the addition of sodium hydroxide.

  Maintaining a pH level of between 10.5 - 12 will avoid corrosion problems by providing stable conditions for the formation of a film of magnetite (Fe_3O_4) in a thin, dense layer on the metal surfaces, protecting them from corrosive attack.

Chemicals added during the conditioning treatment will increase the TDS level in the boiler water and a higher rate of blowdown will be required.

Questions

1. What is the main purpose of an atmospheric deaerator head?

a| To eliminate sparge pipes in the boiler feedtank ☐
b| To remove air from the boiler feedtank ☐
c| To mix hot and cold incoming flows to the boiler feedtank ☐
d| To vent returning flash steam and prevent overheating of the boiler feedtank ☐

2. What is the main reason for returning condensate to the boiler feedtank?

a| To recover its heat content ☐
b| To reduce the boiler blowdown rate ☐
c| To reduce chemical treatment ☐
d| Deaeration of feedwater ☐

3. The free oxygen content of water is reduced by:

a| Letting the water settle ☐
b| Lowering the water temperature ☐
c| Raising the water temperature ☐
d| Agitation of the water through a sparge pipe ☐

4. Why isn't a boiler feedtank maintained at boiling point?

a| To prevent increased heat loss from the feedtank ☐
b| To prevent dangerous pressurisation of the feedtank ☐
c| Incoming cold water could cause thermal shock ☐
d| It would create the danger of feedpump cavitation ☐

5. Condensate is not returned to a boiler feedtank. Why should the feedtank still be heated to at least 85°C?

a| To reduce the water oxygen content ☐
b| To prevent thermal shock on the boiler ☐
c| To reduce scale formation in the boiler ☐
d| To ensure the feedwater conditioning treatment is effective ☐

6. What would be the approximate % cost saving in sodium sulphite from operating a boiler feedtank at 90°C instead of 70°C? (Oxygen content: 90°C, 1.6%; 70°C, 4%)

a| 42% ☐
b| 76% ☐
c| 51% ☐
d| 24% ☐

Answers

1: c, 2: a, 3: c, 4: d, 5: a, 6: c

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Module 3.12

Controlling TDS in the Boiler Water

Controlling TDS in the Boiler Water

As a boiler generates steam, any impurities which are in the boiler feedwater and which do not boil off with the steam will concentrate in the boiler water.

As the dissolved solids become more and more concentrated, the steam bubbles tend to become more stable, failing to burst as they reach the water surface of the boiler. There comes a point (depending on boiler pressure, size, and steam load) where a substantial part of the steam space in the boiler becomes filled with bubbles and foam is carried over into the steam main.

This is obviously undesirable not only because the steam is excessively wet as it leaves the boiler, but it contains boiler water with a high level of dissolved and perhaps suspended solids. These solids will contaminate control valves, heat exchangers and steam traps.

Whilst foaming can be caused by high levels of suspended solids, high alkalinity or contamination by oils and fats, the most common cause of carryover (provided these other factors are properly controlled) is a high Total Dissolved Solids (TDS) level. Careful control of boiler water TDS level together with attention to these other factors should ensure that the risks of foaming and carryover are minimised.

TDS may be expressed in a number of different units, and Table 3.12.1 gives some approximate conversions from TDS in ppm to other units.

Degrees Baumé and degrees Twaddle (also spelt Twaddell) are alternative hydrometer scales.

Table 3.12.1 Comparison of units used to measure TDS

Total dissolved solids ppm	Conductivity (µS/cm) Neutralised	Conductivity (µS/cm) Unneutralised	Relative density at 15.5°C	Degrees Baumé °Be	Degrees Twaddle °Tw
0	0	0	1.000 00	0.000	0.000
200	286	400	1.000 18	0.026	0.036
400	571	800	1.000 36	0.052	0.073
600	857	1 200	1.000 55	0.079	0.109
800	1 143	1 600	1.000 73	0.105	0.145
1 000	1 429	2 000	1.000 91	0.131	0.182
1 200	1 714	2 400	1.001 09	0.157	0.218
1 400	2 000	2 800	1.001 27	0.184	0.255
1 600	2 286	3 200	1.001 45	0.210	0.291
1 800	2 571	3 600	1.001 64	0.236	0.327
2 000	2 857	4 000	1.001 82	0.262	0.364
2 200	3 143	4 400	1.002 00	0.289	0.400
2 400	3 429	4 800	1.002 18	0.315	0.436
2 600	3 714	5 200	1.002 36	0.341	0.473
2 800	4 000	5 600	1.002 55	0.367	0.509
3 000	4 286	6 000	1.002 73	0.393	0.545
3 200	4 571	6 400	1.002 91	0.420	0.582
3 400	4 857	6 800	1.003 09	0.446	0.618
3 600	5 143	7 200	1.003 27	0.472	0.655
3 800	5 429	7 600	1.003 45	0.498	0.691
4 000	5 714	8 000	1.003 64	0.525	0.727
4 200	6 000	8 400	1.003 82	0.551	0.764
4 400	6 286	8 800	1.004 00	0.577	0.800
4 600	6 571	9 200	1.004 18	0.603	0.836
4 800	6 857	9 600	1.004 36	0.630	0.873
5 000	7 143	10 000	1.004 55	0.656	0.909
5 200	7 429	10 400	1.004 73	0.682	0.945
5 400	7 714	10 800	1.004 91	0.708	0.982
5 600	8 000	11 200	1.005 09	0.735	1.018
5 800	8 286	11 600	1.005 27	0.761	1.055
6 000	8 571	12 000	1.005 45	0.787	1.091
6 200	8 857	12 400	1.005 64	0.813	1.127

Boiler water sampling

The boiler water TDS may be measured either by:

- Taking a sample, and determining the TDS external to the boiler,

or by

- A sensor inside the boiler providing a signal to an external monitor.

Sampling for external analysis

When taking a sample of boiler water it is important to ensure that it is representative. It is not recommended that the sample be taken from level gauge glasses or external control chambers; the water here is relatively pure condensate formed by the continual condensation of steam in the external glass/chamber. Similarly, samples from close to the boiler feedwater inlet connection are likely to give a false reading.

Nowadays, most boilermakers install a connection for TDS blowdown, and it is generally possible to obtain a representative sample from this location.

If water is simply drawn from the boiler, a proportion will violently flash to steam as its pressure is reduced. Not only is this potentially very dangerous to the operator, but any subsequent analysis will also be quite wrong, due to the loss of the flash steam concentrating the sample.

Since a cool sample is required for analysis, a sample cooler will also save considerable time and encourage more frequent testing.

A sample cooler is a small heat exchanger that uses cold mains water to cool the blowdown water sample.

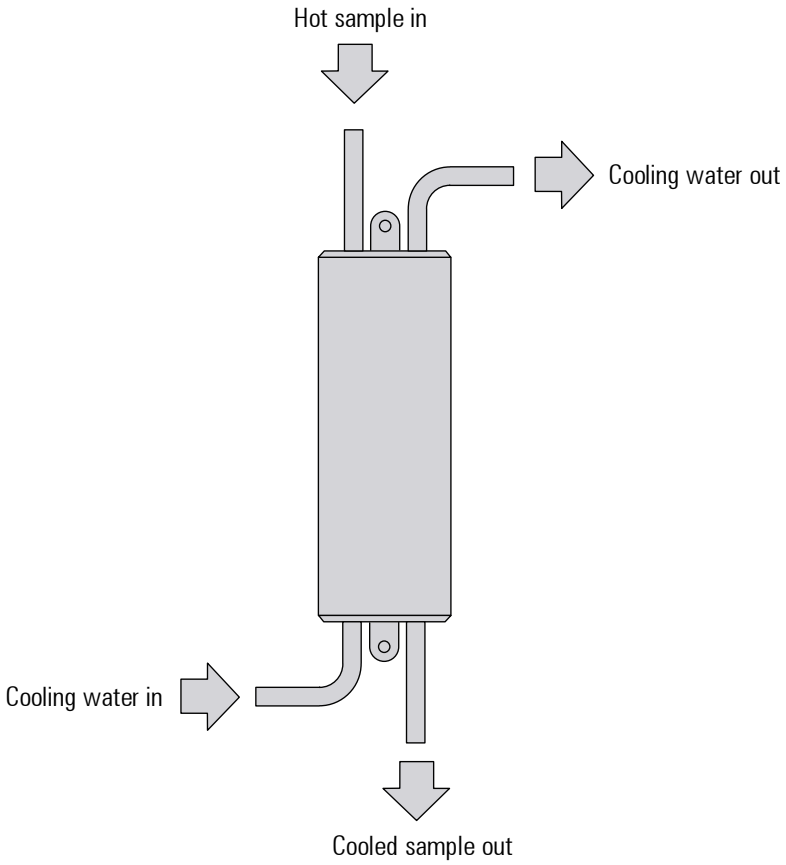


Fig. 3.12.1 A sample cooler

Relative density method

The relative density of water is related to its dissolved solids content. For raw water, feedwater and condensate the relative density is so near to that of pure water that it cannot be measured satisfactorily using a hydrometer. For boiler water, however, a hydrometer can be used to obtain an approximate measurement of the dissolved solids, since for boiler water each increase of 0.000 1 relative density at 15.5°C is approximately equal to 110 ppm. A very sensitive hydrometer is required which needs careful handling and use if a satisfactory measurement of TDS is to be obtained. The procedure is generally as follows:

- Filter the cooled boiler water sample to remove any suspended solids, which would otherwise give a false reading.
- Cool to 15.5°C
- Add a few drops of a wetting agent to help prevent bubbles adhering to the hydrometer.
- Place the hydrometer in the sample and spin gently to remove bubbles.
- Read off the relative density.
- Read off the TDS from a table supplied with the hydrometer **or** calculate the TDS in ppm by using Equation 3.12.1:

$$\text{TDS (ppm)} = (\text{relative density at } 15.5°\text{C} - 1) \times 1.1 \times 10^6 \qquad \textbf{Equation 3.12.1}$$

Example 3.12.1

Relative density at 15.5°C = 1.003 5

TDS = (1.003 5 - 1) x 1.1 x 10^6

TDS = 3 850 ppm

The hydrometer is a delicate instrument, which can easily be damaged. To avoid obtaining false readings it should be regularly checked against distilled water.

Conductivity method

The electrical conductivity of water also depends on the type and amount of dissolved solids contained. Since acidity and alkalinity have a large effect on the electrical conductivity, it is necessary to neutralise the sample of boiler water before measuring its conductivity. The procedure is as follows:

- Add a few drops of phenolphthalein indicator solution to the cooled sample (< 25°C).
- If the sample is alkaline, a strong purple colour is obtained.
- Add acetic acid (typically 5%) drop by drop to neutralise the sample, mixing until the colour disappears.

The TDS in ppm is then approximately as shown in Equation 3.12.2:

$$\text{TDS (ppm)} = (\text{conductivity in } \mu\text{S/cm}) \times 0.7 \qquad \textbf{Equation 3.12.2}$$

Note: This relationship (shown in 3.12.2) is only valid for a neutral sample at 25°C.

Example 3.12.2

Conductivity of a neutralised sample at 25°C = 5 000 µS / cm

TDS = 5 000 µS / cm x 0.7

TDS = 3 500 ppm

Alternatively, the battery powered, temperature compensated conductivity meter shown in Figure 3.12.2 is suitable for use up to a temperature of 45°C.



Fig. 3.12.2 A hand-held conductivity meter

Conductivity measurement in the boiler

It is necessary to measure the conductivity of the boiler water inside the boiler or in the blowdown line. Obviously, the conditions are very different from those of the sample obtained via the sample cooler which will be cooled and subsequently neutralised (pH = 7). The main aspects are the great temperature difference and high pH.

An increase in temperature results in an increase in electrical conductivity. For boiler water, the conductivity increases at the rate of approximately 2% (of the value at 25°C) for every 1°C increase in temperature. This can be written as:

$$\sigma_T = \sigma_{25} \ [1 + \alpha \ (T - 25)] \qquad \textbf{Equation 3.12.3}$$

Where:

σ_T = Conductivity at temperature T (µS/cm)
σ_{25} = Conductivity at 25°C (µS/cm)
α = Temperature coefficient, per °C (Typically 0.02 / °C or 2%°C)
T = Temperature (°C)

Example 3.12.3
A boiler water sample has an unneutralised conductivity of 5 000 µS/cm at 25°C. What is the conductivity of the boiler water at 10 bar g?

At 10 bar g, saturation temperature = 184°C (from steam tables)

$$\sigma_T = 5\,000\,[1 + 0.02\,(184 - 25)]$$

$$\sigma_T = \mathbf{20\,900\ \mu S/cm}$$

This means that the effects of the temperature have to be allowed for in the blowdown controller, either by automatic temperature compensation, or by assuming that the boiler pressure (and hence temperature) is constant. The small variations in boiler pressure during load variations have only a relatively small effect, but if accurate TDS readings are required on boilers which are operated at widely varying pressures then automatic temperature compensation is essential.

Cell constant
A probe used to measure the conductivity of a liquid has a 'cell constant'. The value of this constant depends on the physical layout of the probe and the electrical path through the liquid. The further the probe tip is from any part of the boiler, the higher the cell constant. Any differences in cell constant are taken into consideration when 'calibrating' the controller.

Conductivity and resistance are related by the cell constant, as seen in Equation 3.12.4:

$$R = \frac{K}{\sigma} \qquad \textbf{Equation 3.12.4}$$

Where:
R = Resistance (Ohm)
K = Cell constant (cm^{-1})
σ = Conductivity (S/cm)

Example 3.12.4
From Example 3.12.3 the boiler water conductivity was 20 900 µS/cm. For a cell constant of 0.3, what is the resistance measured by the controller?

$$\text{Resistance} = \frac{0.3}{20\,900 \times 10^{-6}}$$

$$\textbf{Resistance} = \mathbf{14.4\ Ohm}$$

Whilst the boiler water conductivity is converted to a resistance through the probe, it cannot be measured using a simple dc resistance meter. If a dc voltage is applied to the probe, tiny hydrogen or oxygen bubbles are formed on the surface due to electrolysis of the water. This effect, called electrolytic polarisation, causes a much higher resistance to be measured.

It is therefore necessary to use an ac voltage to measure the probe resistance and this is the method always to be preferred in blowdown controllers. A relatively high frequency (for example 1 000 Hz) is necessary to avoid polarisation at the high conductivities of boiler water.

Deciding on the required boiler water TDS

The actual dissolved solids concentration at which foaming may start will vary from boiler to boiler. Conventional shell boilers are normally operated with the TDS in the range of 2 000 ppm for very small boilers, and up to 3 500 ppm for larger boilers, provided the:

- Boiler is operating near to its design pressure.
- Steam load conditions are not too severe.
- Other boiler water conditions are correctly controlled.

Blowing down the boiler to maintain these TDS levels should help to ensure that reasonably clean and dry steam is delivered to the plant.

Table 3.12.2 provides some broad guidelines on the maximum permissible levels of boiler water TDS in certain types of boiler. Above these levels, problems may occur.

Table 3.12.2 Typical maximum TDS for various boiler types

Boiler type	Maximum TDS (ppm)
Lancashire	10 000
Two-pass economic	4 500
Packaged and three-pass economic	3 000 to 3 500
Low pressure water-tube	2 000 to 3 000
Coil boiler and steam generators (TDS in feedwater)	2 000
Medium pressure water-tube	1 500
High pressure water-tube	1 000

Note: The figures in Table 3.12.2 are offered as a broad guide only. The boilermaker should always be consulted for specific recommendations.

Calculating the blowdown rate

The following information is required:

- The required boiler water TDS in parts per million (Table 3.12.1).
- The feedwater TDS in parts per million.

  An average value may be obtained by looking at water treatment records, or a sample of feedwater may be obtained and its conductivity measured

  As with boiler water TDS measurement, conductivity (µS/cm) x 0.7 = TDS in parts per million (at 25°C).

  Note: the sample of feedwater that is required is from the boiler feedline or from the feedtank and is not a sample of the make-up water supplying the feedtank.
- The quantity of steam which the boiler generates, usually measured in kg/h. For selecting a blowdown system, the most important figure is usually the maximum quantity of steam that the boiler can generate at full-load.

When the above information is available the required blowdown rate can be determined using Equation 3.12.5:

$$\text{Blowdown rate} = \frac{F\ S}{B - F} \qquad \textbf{Equation 3.12.5}$$

Where:
F = Feedwater TDS (ppm).
S = Steam generation rate (kg/h).
B = Required boiler water TDS (ppm).

Example 3.12.5
A 10 000 kg/h boiler operates at 10 bar g - Calculate the blowdown rate, given the following conditions:

$$\text{Maximum allowable boiler TDS} = 2\,500 \text{ ppm}$$

$$\text{Boiler feedwater TDS} = 250 \text{ ppm}$$

$$\text{Blowdown rate} = \frac{250 \times 10\,000}{(2\,500 - 250)}$$

$$\textbf{Blowdown rate} = \textbf{1 111 kg / h}$$

Controlling the blowdown rate

There are a number of different ways that the blowdown rate can be controlled. The simplest device is an orifice plate (Figure 3.12.3).

The orifice size can be determined based on:

- **Flowrate -** A means of calculating flowrate is shown above.
- **Pressure drop -** Theoretically this would be from boiler pressure to atmospheric pressure. However, pipeline friction and backpressure is inevitable, so for the purposes of this Module, assume the pressure on the downstream side of the orifice is 0.5 bar g.

There is a problem: an orifice is not adjustable and therefore can only be correct for one specific set of circumstances. If the steaming rate were to:

- **Increase -** The orifice would not pass sufficient water. The boiler TDS level would increase, and priming and carryover would occur.
- **Reduce -** The orifice would pass too much water. The blowdown rate would be too great and energy would be wasted.

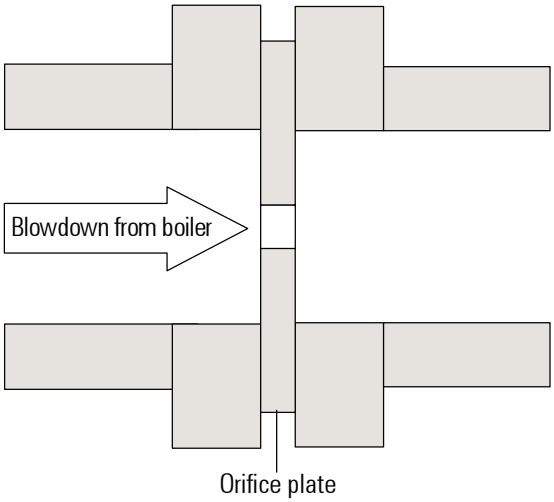


Figure 3.12.3
Controlling the blowdown rate using a fixed orifice

Flashing

The water being drained from the boiler is at saturation temperature, and there is a drop in pressure over the orifice almost equal to the whole boiler pressure. This means that a substantial proportion of the water will flash to steam, increasing its volume by a factor of over 1 000.

This rapid and aggressive change of state and volume over the orifice may result in erosion and wiredrawing of the orifice. This increases both the size and flow characteristic (coefficient of discharge) of the orifice, resulting in a progressively increasing blowdown rate.

The steam, being a gas, can travel much faster than the water (liquid). However, the steam and water do not have the opportunity to separate properly, which results in water droplets travelling at a very high velocity with the steam into the pipework. This leads to further erosion and possibly waterhammer in the pipework and downstream equipment.

The problem of flashing increases with boiler pressure.

It should also be remembered that the water drained from the boiler is dirty and it does not take a great deal of dirt to restrict or even block a small hole.

Blowdown valves

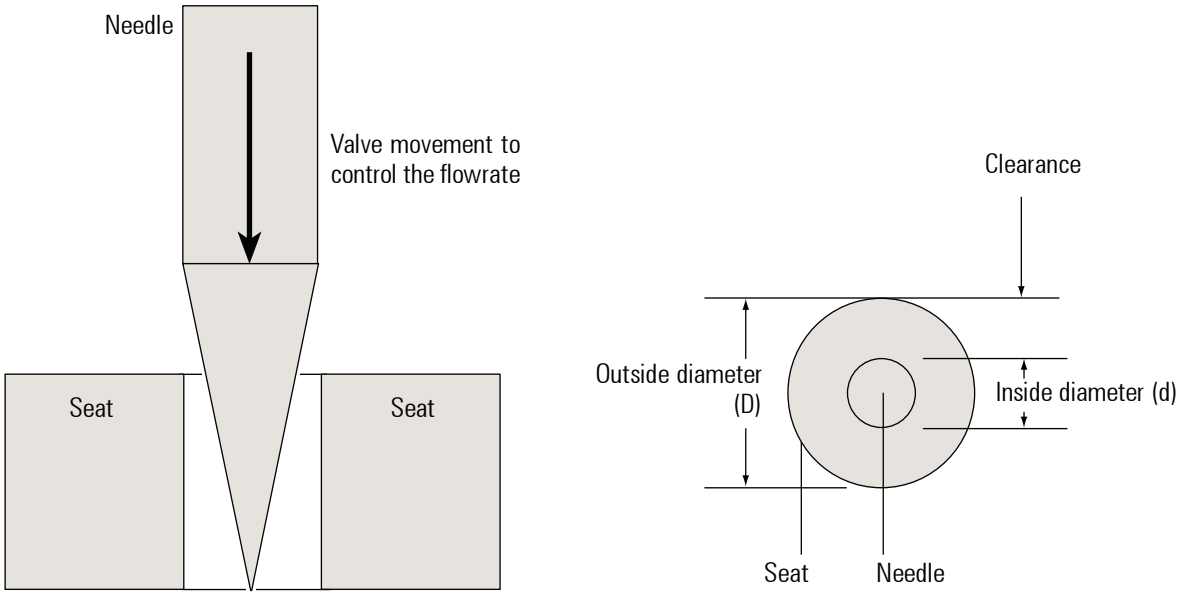


Fig. 3.12.4 A needle valve used to control the blowdown rate

Continuous blowdown valves

In its simplest form, this is a needle valve. In plan view, there is an annulus with the:

- Outer circumference defined by the valve seat.
- Inner circumference defined by the needle.

If an increase in flowrate is required, the needle is adjusted out of the seat and the clearance between the needle and seat is increased.

To ensure a reasonable velocity through the orifice, the size of orifice necessary for the blowdown flowrate of 1 111 kg/h (from Example 3.12.5) would be about 3.6 mm.

Taking the valve seat diameter to be 10 mm, it is possible to calculate the diameter of the needle at the point where it is set to give the required flow of 1 111 kg/h, as follows:

$$d^2 = \sqrt{D_2^2 - D_1^2}$$

Where:
D orifice = D_1 = 3.6 mm
D valve seat = D_2 = 10.0 mm
d needle = d = ?

Therefore: Solving the equation shows that the needle diameter at the correct setting is 9.33 mm.

The clearance is half the difference of the diameters.

$$\text{Clearance} = \frac{10 - 9.33}{2}$$

Clearance = 0.33 mm

This is a fundamental weaknesses of continuous blowdown valves; the clearance is so small that blockage by small particles is difficult to avoid.

In addition, the problem of flashing over the valve seat still has to be addressed. The low clearances mean that a high velocity steam/water mixture is flowing close to the surfaces of the needle and the seat. Erosion (wiredrawing) is inevitable, resulting in damage and subsequent failure to shut off.

Continuous blowdown valves have been developed over many years from simple needle valves, and now incorporate a number of stages, possibly taking the form of three or four progressively larger seats in the valve, and even including helical passageways. The objective is to dissipate the energy gradually in stages rather than all at once.

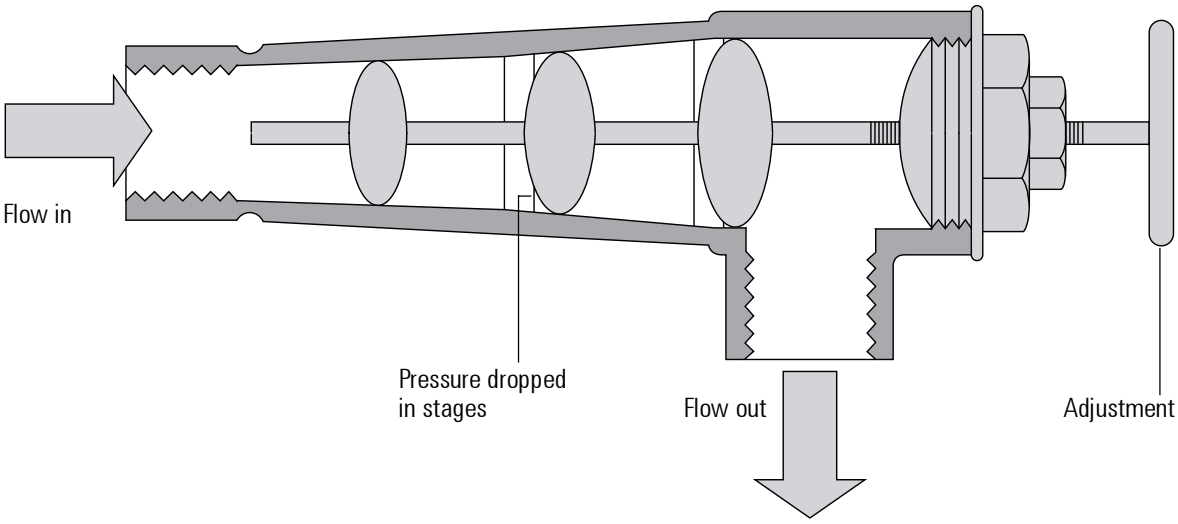


Fig. 3.12.5 Staged blowdown valve

This type of valve was originally designed for manual operation, and was fitted with a scale and pointer attached to the handle. In an operational environment, a boiler water sample was taken, the TDS determined, and an appropriate adjustment made to the valve position.

To keep pace with modern technology and market demands, some of these continuous blowdown valves have been fitted with electric or pneumatic actuators. However, the fundamental problem of small clearances, flashing, and wiredrawing still exist, and damage to the valve seating is inevitable. Despite using a closed loop control system, excessive blowdown will occur.

On/off boiler blowdown valves

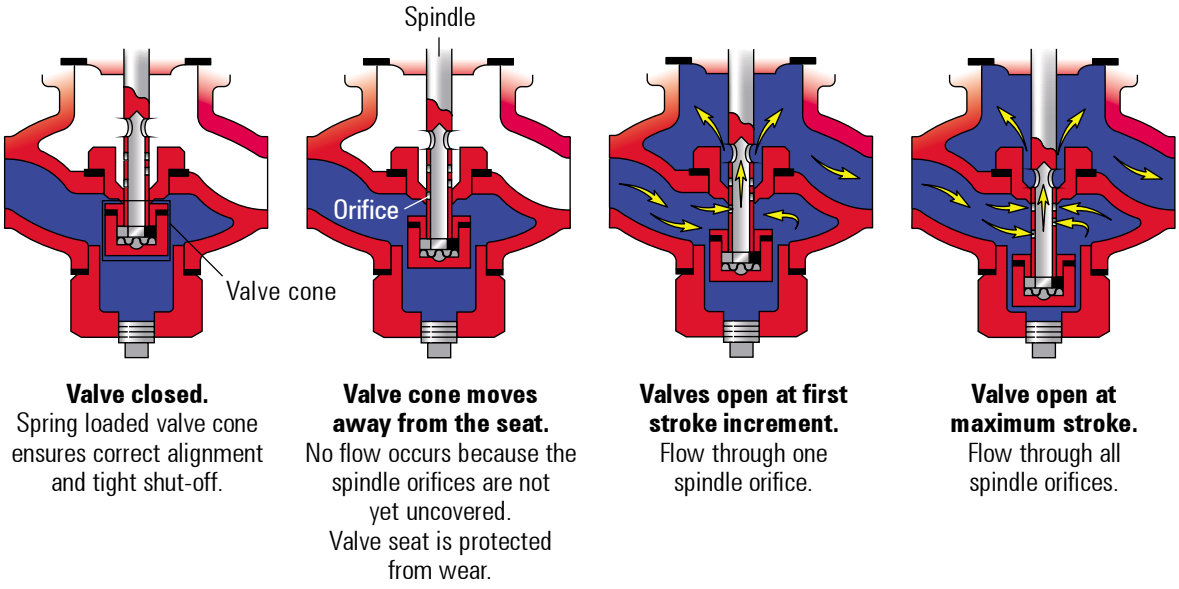
There is an advantage to using a larger control device with larger clearances, but only opening it for some of the time. Clearly, moderation is required if the boiler TDS is to be kept between reasonable values, and DN15 and 20 valves are the most common sizes to be found.

A typical arrangement would be to set the controller to open the valve at, for example, 3 000 ppm, then to close the valve at 3 000 – 10% = 2 700 ppm. This would give a good balance between a reasonable sized valve and accurate control.

The type of valve selected is also important:

- For small boilers with a low blowdown rate and pressures of less than 10 bar g, an appropriately rated solenoid valve will provide a cost-effective solution.
- For larger boilers with higher blowdown rates, and certainly on boilers with operating pressures over 10 bar g, a more sophisticated valve is required to take flashing away from the valve seat in order to protect it from damage.

Valves of this type may also have an adjustable stroke to allow the user the flexibility to select a blowdown rate appropriate to the boiler, and any heat recovery equipment being used.



Valve closed.
Spring loaded valve cone ensures correct alignment and tight shut-off.

Valve cone moves away from the seat.
No flow occurs because the spindle orifices are not yet uncovered.
Valve seat is protected from wear.

Valves open at first stroke increment.
Flow through one spindle orifice.

Valve open at maximum stroke.
Flow through all spindle orifices.

Fig. 3.12.6 Modern blowdown control valve

Closed loop electronic control systems

These systems measure the boiler water conductivity, compare it with a set point, and open a blowdown control valve if the TDS level is too high.

A number of different types are on the market which will measure the conductivity either inside the boiler, or in an external sampling chamber which is purged at regular intervals to obtain a representative sample of boiler water. The actual selection will be dependent upon such factors as boiler type, boiler pressure, and the quantity of water to be blown down.

These systems are designed to measure the boiler water conductivity using a conductivity probe.

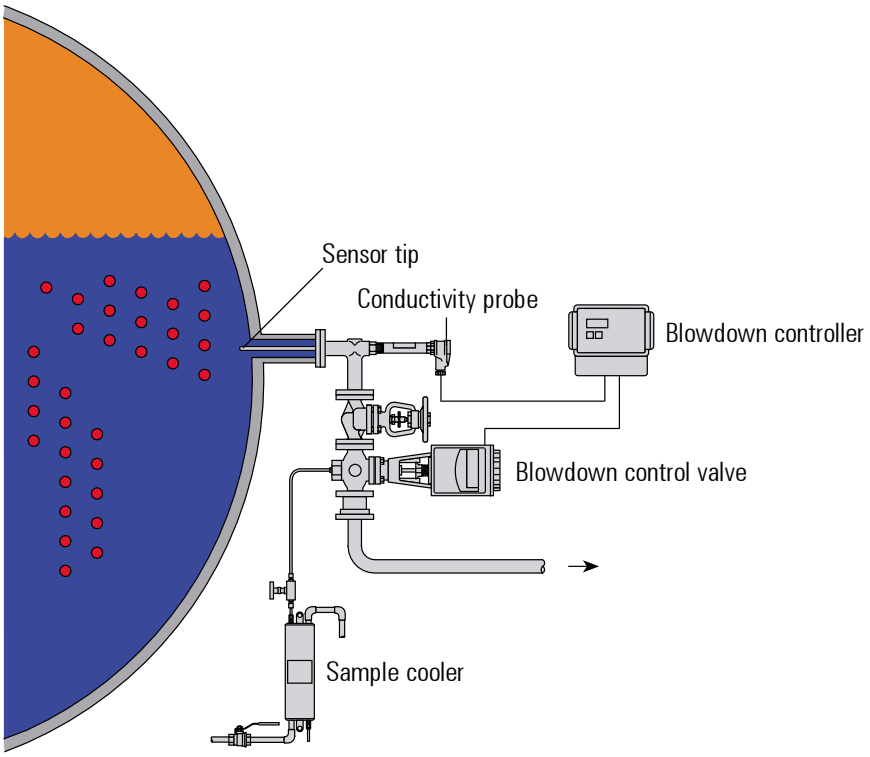


Fig. 3.12.7 A closed loop electronic TDS control system

The measured value is compared to a set point programmed into the controller by the user. If the measured value is greater than the set point, the blowdown control valve is opened until the set point is achieved. Typically, the user can also adjust the 'dead-band'.

As mentioned earlier, an increase in water temperature results in an increase in electrical conductivity. Clearly if a boiler is operating over a wide temperature/pressure range, such as when boilers are on night set-back, or even a boiler with a wide burner control band, then compensation is required, since conductivity is the controlling factor.

The benefits of automatic TDS control:

- The labour-saving advantages of automation.
- Closer control of boiler TDS levels.
- Potential savings from a blowdown heat recovery system (where installed).

The calculation of further savings due to a reduction in the blowdown rate are described in the following text and in Example 3.12.6.

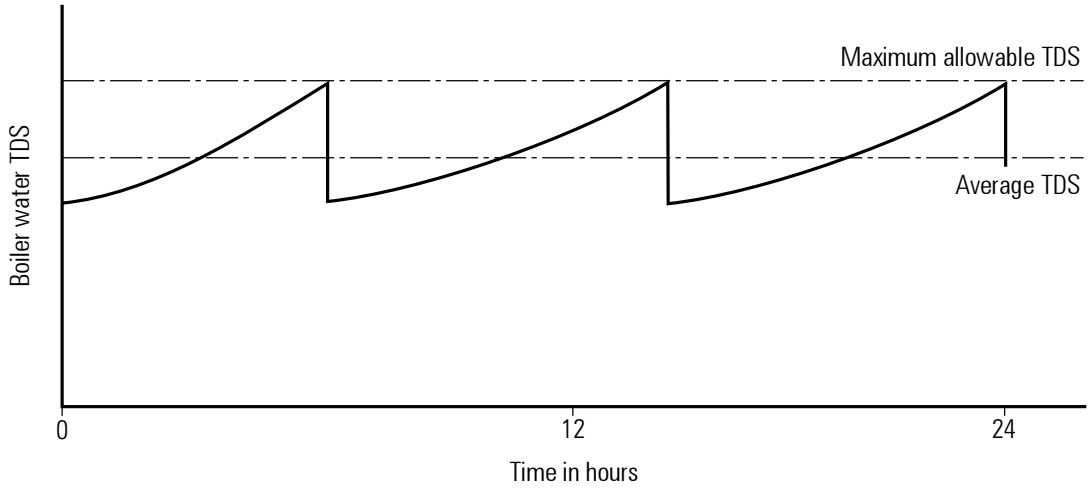


Fig. 3.12.8 Plot of TDS versus time using a manual blowdown 3 times per 24 hours

Where the present method is solely manual blowdown from the bottom of the boiler, it may be possible by looking at past water treatment records, to obtain some idea of how much the boiler TDS varies over a period of weeks. By inspection, an average TDS figure can be established. Where the actual maximum is less than the maximum allowable figure, the average is as shown. Where the actual maximum exceeds the maximum allowable, the average obtained should be scaled down proportionally, since it is desirable that the maximum allowable TDS figure should never be exceeded.

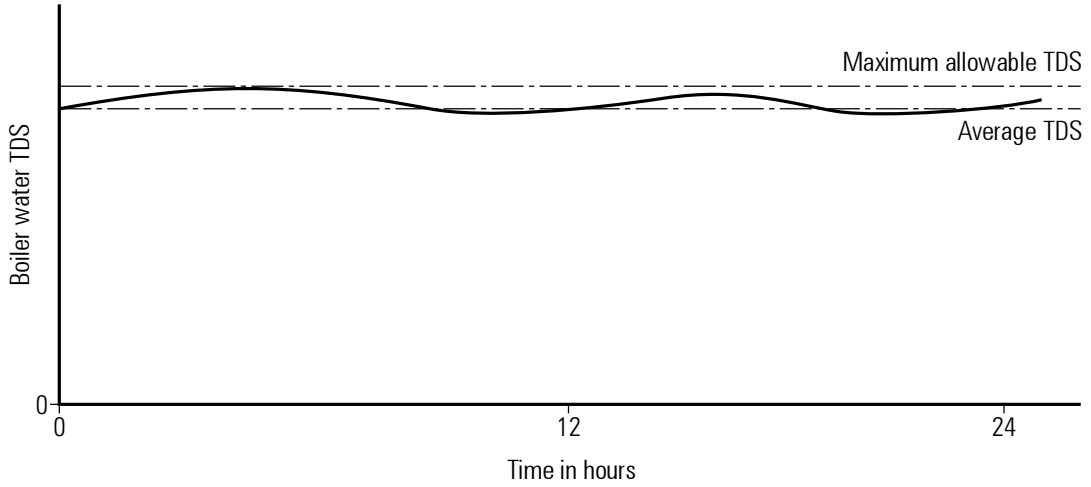


Fig. 3.12.9 Plot of TDS versus time using a closed loop electronic TDS control system

Example 3.12.6
Figure 3.12.8 shows that the average TDS with a well operated manual bottom blowdown is significantly below the maximum allowable. For example the maximum allowable TDS may be 3 500 ppm and the average TDS only 2 000 ppm. This means that the actual blowdown rate is much greater than that required. Based on a feedwater TDS of 200 ppm, the actual blowdown rate is:

$$\frac{\text{200 ppm feedwater TDS}}{\text{2 000 ppm average boiler TDS - 200 ppm feedwater TDS}} \times \frac{100}{1} = 11.1\%$$

By installing an automatic TDS control system the average boiler water TDS can be maintained at a level almost equal to the maximum allowable TDS as shown in Figure 3.12.9;

Evaluating savings by reducing blowdown rate
If a boiler is to supply a given amount of steam, the water blown down must be in addition to this amount. The energy that is lost in blowdown is the energy that is supplied to the additional amount of water that is heated to saturation temperature, and then blown down.

A close approximation can be obtained using steam tables.

Using the figures from Example 3.12.5, if the boiler had been operating at 10 bar g, steaming at 5 000 kg/h and had a feedwater temperature of 80°C (h_f = 335 kJ/kg), the change in energy requirement could be calculated as follows:

Condition 1, manual TDS control: Blowdown rate = 11.1%

To achieve a steaming rate of 5 000 kg/h, the boiler needs to be supplied with:

$$\text{Flowrate of water supplied to the boiler} = \frac{5\,000 \text{ kg/h} \times (100 + 11.1)}{100}$$

Flowrate of water supplied to the boiler = 5 555 kg/h

All of this water will be raised to saturation temperature from feedwater temperature h_f = 782 kJ/kg at 10 bar g saturation temperature; h_f = 335 kJ/kg at 80°C:

$$\text{Energy required} = \frac{5\,555 \text{ kg/h} \times (782 - 335) \text{ kJ/kg}}{3\,600 \text{ second/hour}}$$

Energy required = 690 kW

5 000 kg/h of this is evaporated to steam for export
h_{fg} = 2 000 kJ/kg from steam tables:

$$\text{Energy required} = \frac{5\,000 \text{ kg/h} \times 2\,000 \text{ kJ/kg}}{3\,600 \text{ second/hour}}$$

Energy required = 2 778 kW

Total energy used to generate 5 000 kg/h of steam = 690 kW + 2 778 kW

Total energy used to generate 5 000 kg/h of steam = 3 468 kW

Example 3.12.7

Condition 2, automatic TDS control : Blowdown rate $= \frac{200 \times 100\%}{3\,500 - 200} = 6.1\%$

To achieve a steaming rate of 5 000 kg/h, the boiler needs to supplied with:

$$\text{Flowrate of water supplied to the boiler} = \frac{5\,000 \text{ kg/h} \times (100 + 6.1)}{100}$$

Flowrate of water supplied to the boiler = 5 305 kg/h

All of this water will be raised to saturation temperature from feedwater temperature h_f = 782 kJ/kg at 10 bar g saturation temperature; h_f = 335 kJ/kg at 80°C:

$$\text{Energy required} = \frac{5\,305 \text{ kg/h} \times (782 - 335) \text{ kJ/kg}}{3\,600 \text{ second/hour}}$$

Energy required = 659 kW

5 000 kg/h of this is evaporated to steam for export:

$$\text{Energy required} = \frac{5\,000 \text{ kg/h} \times 2\,000 \text{ kJ/kg}}{3\,600 \text{ second/hour}}$$

Energy required = 2 778 kW

The total energy used to generate 5 000 kg/h of steam = 659 kW + 2 778 kW

The total energy used to generate 5 000 kg/h of steam = 3 437 kW

Since fuel must have supplied the energy used to generate the steam, the reduction in energy used must represent a saving in fuel:

Reduction in energy = 3 468 kW - 3 437 kW

Reduction in energy = 31 kW

This, in turn, can be expressed as a percentage saving in the boiler fuel cost:

$$\text{Reduction in energy cost} = \frac{31 \text{ kW}}{3\,468 \text{ kW}} \times \frac{100}{1}$$

Reduction in energy cost = 0.9% saving in fuel cost

Questions

1. What is the effect of the TDS being too high?

a| Energy used is reduced ☐
b| Water carry over ☐
c| The water level will fall and lockout will occur ☐
d| Waste of energy ☐

2. What is the effect of the TDS being too low?

a| Energy lost through excessive blowdown ☐
b| Energy saved through reduced blowdown ☐
c| Turbulent water conditions and wet steam ☐
d| Waste of feed treatment chemicals ☐

3. A boiler exports steam at the rate of 5 000 kg/h. The required TDS is 2 750 ppm in the boiler. The feedwater has a TDS of 200 ppm. What should be the boiler blowdown rate?

a| 407 kg/h ☐
b| 372 kg/h ☐
c| 358 kg/h ☐
d| 392 kg/h ☐

4. What is the advantage of an automatic blowdown system over a simple manual control?

a| The valve prevents the passage of flash steam ☐
b| It does not need calibration ☐
c| The amount of blowdown is correct for all boiler operating conditions ☐
d| It can be retrofitted to existing manual valves ☐

5. Why is a sample cooler essential for taking a sample of boiler water?

a| Accuracy of reading ☐
b| To neutralise the sample ☐
c| Because TDS readings cannot be taken from inside the boiler ☐
d| It excludes flash steam from the readings ☐

6. For the same steam output, why will a packaged shell boiler tolerate a higher TDS level than a high pressure water-tube boiler?

a| Water treatment is simpler for a packaged boiler and is less able to control TDS formation ☐
b| The larger water surface area in a packaged boiler results in a lower foaming rate per m² ☐
c| There is a greater steam space above the water in a packaged boiler so more foaming is acceptable ☐
d| There is a lower water content in a water-tube boiler and less space for bubbles ☐

Answers

1: b, 2: a, 3: d, 4: c, 5: a, 6: a

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Module 3.13

Heat Recovery from Boiler Blowdown (TDS Control only)

Heat Recovery from Boiler Blowdown (TDS Control Only)

The previous module discussed the water to be blown down from a boiler in order to maintain an acceptable TDS level. This water has a number of characteristics:

- **It is dirty -** This means that:
  - The water is generally unsuitable for other applications.
  - The dirty water may present a disposal problem.
- **It is hot -** This means that:
  - A proportion of the water will flash to steam at atmospheric pressure.
  - The hot water may present a disposal problem. For example, there may be a substantial quantity to dispose of.

A heat recovery system can solve many of these problems.

Energy flowrate in blowdown

Using the data from the blowdown calculation, Example 3.12.5, the amount of energy sent to blowdown can be calculated using the steam tables.

Note: 1 kJ/s = 1 kW

Example 3.13.1

Boiler pressure = 10 bar g

Boiler rating = 10 000 kg/h

Maximum allowable boiler TDS = 2 500 ppm

Feedwater TDS = 250 ppm

Calculated blowdown rate = 1 111 kg/h

To obtain the energy flow in kW:

$$\text{The blowdown rate in kg/s} = \frac{1\ 111\ \text{kg/h}}{3\ 600}$$

The blowdown rate in kg/s = 0.31 kg/s

The amount of energy in each kg from h_f* at 10 bar g = 782 kJ/kg

*h_f is the specific enthalpy of water at the saturation temperature - obtained from steam tables.

Rate of energy blown down = 0.31 kg/s x 782 kJ/kg

Rate of energy blown down = 241 kW

To put the energy flowrate into context, in North West Europe the average domestic central heating system is rated at approximately 13 kW, so the energy flowrate blown down in Example 3.13.1 is sufficient to heat 19 houses.

For clarity the above calculation utilises steam tables where water at 0°C is the datum.

In reality, make-up water to replace the blowdown will be supplied at a temperature greater than this, so the energy blow down will be slightly less. For example, if the make-up water were at 10°C the energy blown down would be 228 kW.

Flash steam

The blowdown water released from the boiler is water at the saturation temperature appropriate to the boiler pressure. In the case of the boiler in Example 3.13.1 - 10 bar g, this temperature is 184°C. Clearly, water cannot exist at 184°C under atmospheric conditions, because there is an excess of enthalpy or energy in the blowdown water.

Assuming the blowdown water is released to a flash steam system operating at 0.5 bar g, steam tables may be used to quantify this energy excess:

Specific enthalpy of water at 10 bar g = 782 kJ/kg (h_f at 10 bar g)

Specific enthalpy of water at 0.5 bar g = 468 kJ/kg (h_f at 0.5 bar g)

Excess energy = 314 kJ/kg

This excess energy evaporates a proportion of the water to steam, and the steam is referred to as flash steam.

The quantity of flash steam is readily determined by calculation or can be read from tables or charts.

Example 3.13.2
The specific enthalpy of evaporation at 0.5 bar g (h_{fg}) from steam tables is 2 226 kJ/kg.

$$\% \text{ Flash steam} = \frac{(h_f \text{ high pressure} - h_f \text{ low pressure})}{h_{fg} \text{ low pressure}} \times 100\%$$

$$\% \text{ Flash steam} = \frac{782 - 468}{2\,226} \times 100\%$$

% Flash steam = 14.1%

Therefore 14.1% of the water blown down from the boiler will change to steam as its pressure drops from 10 to 0.5 bar g across the blowdown valve.

There are two options:

1. Vent this flash steam to atmosphere via the blowdown vessel with the associated waste of energy and potentially good quality water from the condensed steam.

2. Utilise the energy in the flash steam, and recover water by condensing the flash steam. It is useful to quantify the energy flowrate in the flash steam. This can be done using steam tables.

Example 3.13.3

Rate of flash steam generation = 1 111 kg/h x 14.1%

Rate of flash steam generation = 157 kg/h (0.043 5 kg/s)

Total energy per kg of steam = 2 694 kJ/kg (h_g at 0.5 bar g)

Energy flowrate in flash steam = 0.043 5 kg/s x 2 694 kJ/kg

Energy flowrate in flash steam = 117 kW

Compare this to the 241 kW rate of energy blown down from the boiler.

It may be possible to use this flash steam: in this example it represents almost 49% of the energy flowrate in the blowdown, and 14.1% of the water blown down.

Using values from steam tables for the above calculations assumes that feedwater will be supplied at a temperature of 0°C. For greater accuracy, the actual change in feedwater temperature should be used.

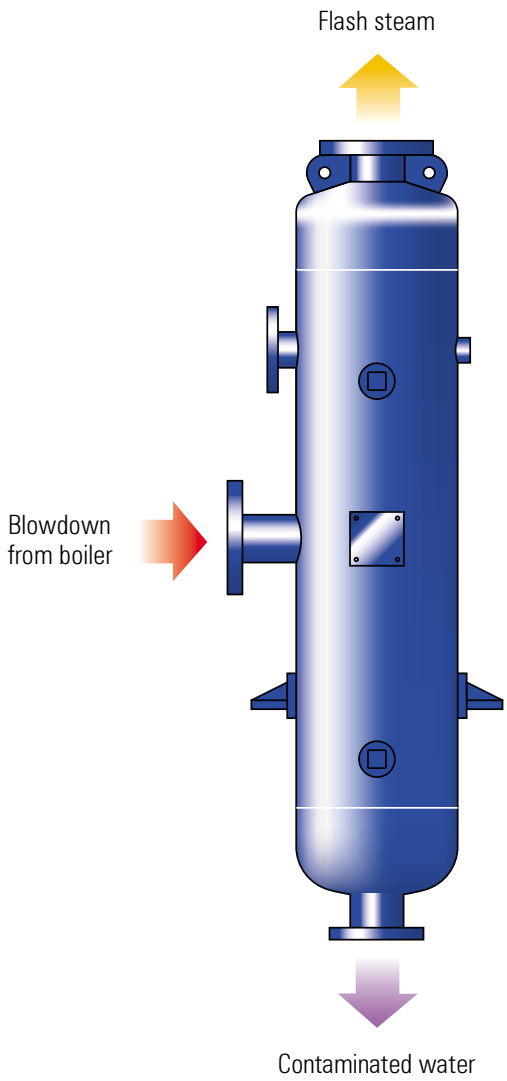


Fig. 3.13.1 Flash vessel

Recovering and using flash steam

The flash steam becomes available for recovery at the flash vessel. In essence, a flash vessel provides a space where the velocity is low enough to allow the hot water and flash steam to separate, and from there to be piped to different parts of the plant.

The design of the flash vessel is important not only from a steam/water separation point of view, but structurally it should be designed and built to a recognised pressure vessel standard, such as PD 5500.

This is not only good engineering practice, the boiler inspector will also insist upon this if the plant is to be insured.

The most obvious place for the flash steam to be used is in the boiler feedtank, which is usually nearby.

The water temperature in the feedtank is important. If it is too low, chemicals will be required to de-oxygenate the water; if it is too high the feedpump may cavitate. Clearly, if heat recovery is likely to result in an excessive high feedtank temperature, it is not practical to discharge flash steam into the tank. Other solutions are possible, such as feedwater heating on the pressure side of the feedpump, or heating the combustion air.

Figure 3.13.2 shows a simple installation, which makes recovery of the 117 kW of energy flow, and 157 kg/h of boiler quality water, extremely cost effective.

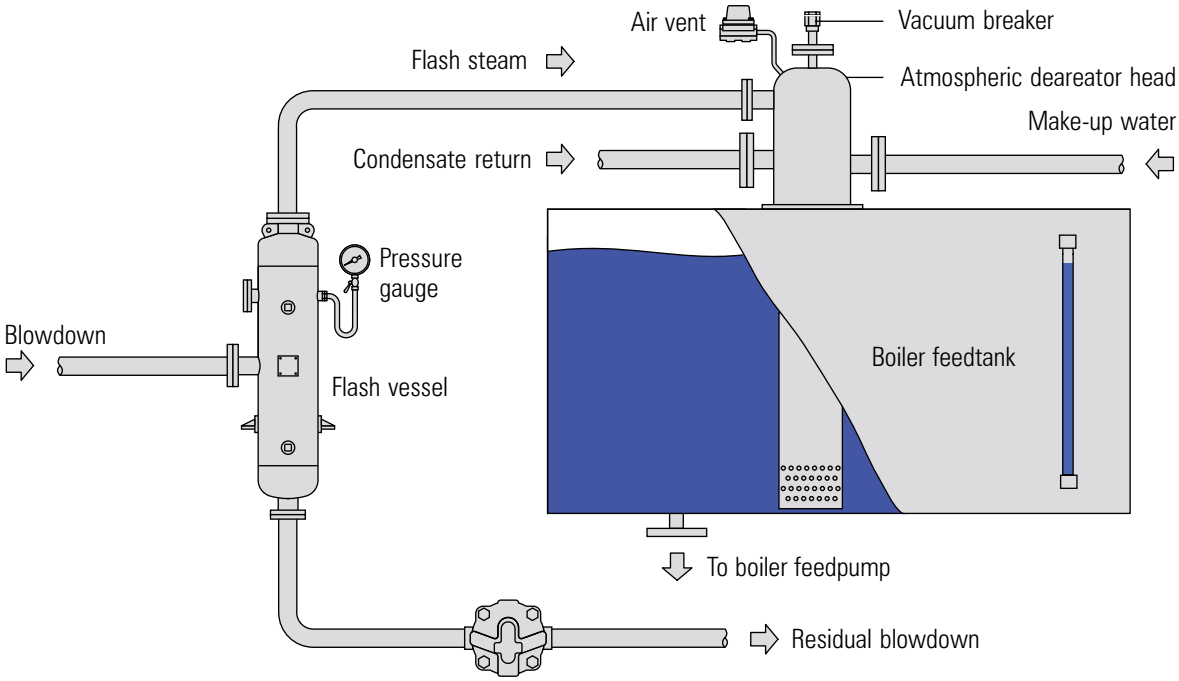


Fig. 3.13.2 Using a flash vessel to return energy to the feedtank

Equipment required

- **Flash vessel -** Manufacturers will have sizing charts for vessels. Note: the steam velocity in the top section of the vessel should not exceed 3 m/s.
- **Steam trap to drain the vessel -** A float trap is ideal for this application as it releases the residual blowdown water as soon as it reaches the trap.

  The flash vessel is working at low pressure so there is virtually no energy to lift the residual blowdown after the steam trap, so this must drain by gravity through the trap and discharge pipework.

  Note: because of the low pressure, the trap will be fairly large. This has the additional advantage that it is unlikely to be blocked by the solids in the residual blowdown water.

  Sometimes strainers are preferred before the steam trap; for this application the strainer cap should be fitted with a blowdown valve to simplify maintenance, and the strainer screen should not be too fine.
- **Vacuum breaker -** There will be occasions when the boiler does not need to blow down. At these times any steam in the flash vessel and associated pipework will condense and a vacuum will be formed. If this vacuum is not released then water will be drawn up from the boiler feedtank into the pipework. When the boiler blows down again this water will be forced along the pipe at high velocity and waterhammer will occur.

  A vacuum breaker fitted to the deareator head will protect against this eventuality.
- **Steam distribution equipment -** Proper distribution of the flash steam in the feedwater tank is clearly important in order to ensure condensation and recovery of the heat and water. The equipment required to do this include, in order of effectiveness:

  1. Atmospheric deaerator.
  2. Steam distributor.
  3. Sparge pipe.

Heat recovery using heat exchangers

Heat recovery from residual blowdown

About 49% of the energy in boiler blowdown can be recovered through the use of a flash vessel and associated equipment; however, there is scope for further heat recovery from the residual blowdown itself.

Continuing on from Example 3.13.3, if the flash vessel operates at a pressure of 0.2 bar g, this means that the residual blowdown passes through the flash vessel float trap at about 105°C. Further useful energy can be recovered from the residual blowdown before passing it to drain. The accepted method is to pass it through a heat exchanger, heating make-up water en route to the feedtank. This approach typically cools the residual blowdown to about 20°C. This system not only recovers the energy in the blowdown effluent, it also cools the water before discharging into the drainage system. (The temperature at which effluent may be discharged is limited to 42°C in the UK; other countries having similar limitations).

Example 3.13.4 (continuing from Example 3.13.3)

The total blowdown = 1 111 kg/h with 157 kg/h flashing to steam

Water flowrate = 1 111 – 157

Water flowrate = 954 kg/h

Energy in the water:

Enthalpy of saturated water (h_f) at 0.2 bar g = 440 kJ/kg

Enthalpy of water at 20°C = 84 kJ/kg

Energy available to heat up the make-up water = 440 – 84

Energy available to heat up the make-up water = 356 kJ/kg

$$\text{Energy recovered} = \frac{954 \text{ kg/h} \times 356 \text{ kJ/kg}}{3\,600}$$

Energy recovered = 94 kW

A typical arrangement for recovering this energy is shown in Figure 3.13.3.

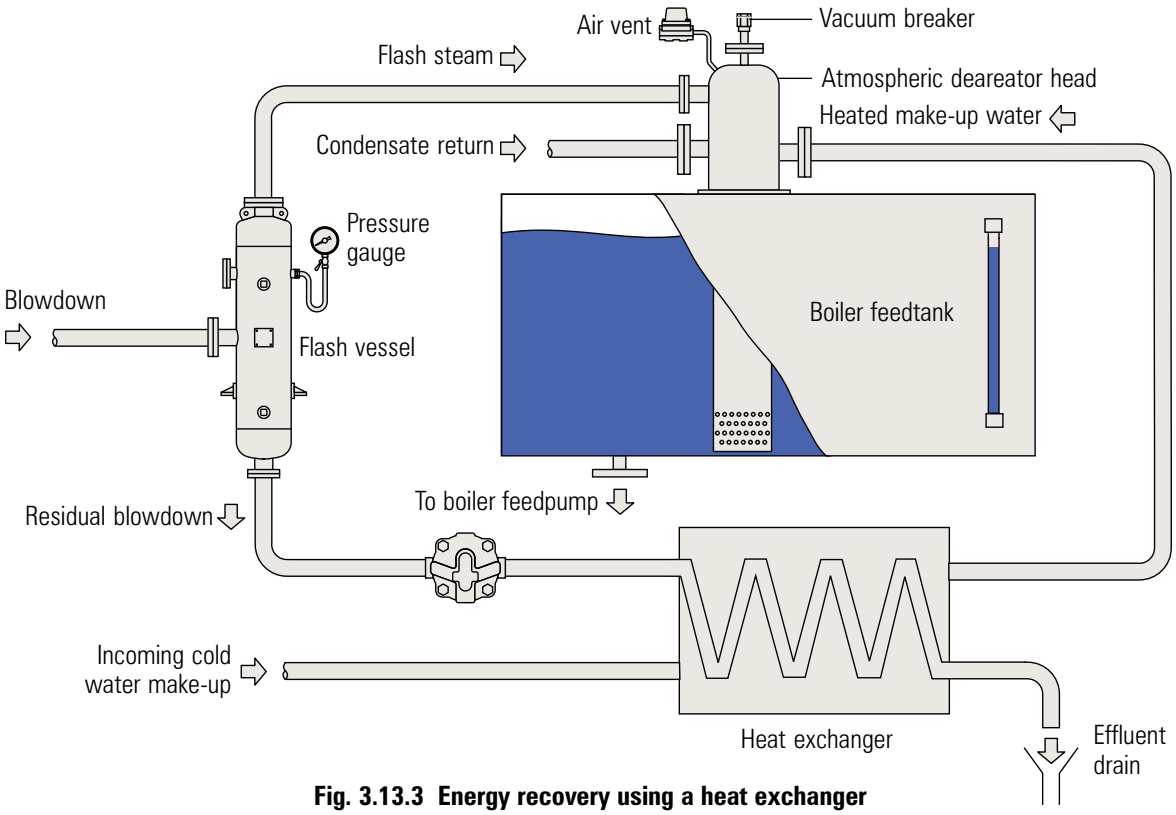


Fig. 3.13.3 Energy recovery using a heat exchanger

Design considerations

A problem with the arrangement shown in Figure 3.13.3 is that the simultaneous flow of incoming cold make-up water and residual blowdown from the flash vessel may not be guaranteed.

One preferred arrangement is shown in Figure 3.13.4, where a cold water break tank is used as a heat sink. A thermostat is used to control a small circulating pump so that when the residual blowdown is at a high enough temperature, water is pumped through the heat exchanger, raising the average tank temperature and saving energy.

If the temperature of the blowdown effluent exiting the heat exchanger can be above 43°C, then it should be directed to the blowdown vessel rather than straight to the effluent drain (See Module 3.14).

Preferred type of heat exchanger

Plate heat exchangers are preferred for this application, as they are very compact and easily maintained.

Experience shows that the higher velocities and turbulence in plate heat exchangers help to keep them clean, and hence dismantling is rarely required. However, should cleaning be required, it is relatively straightforward to open the heat exchanger and clean the plates.

The cleaning of a shell and tube heat exchanger is more complex, and will involve a complete strip down and often the tubes themselves cannot be removed for cleaning.

Total energy saving (from Examples, 3.13.3 and 3.13.4)**:**

From the flash vessel	≈	117 kW
From the heat exchanger	=	94 kW
Total energy recovered	=	117 kW + 94 kW
Total energy recovered	**=**	**211 kW**

When energy is recovered from the flash steam and the condensate, 87% of the total energy contained in the original blowdown has been recovered.

In addition, 14% (by mass) of the water has been recovered, making a further contribution to savings.

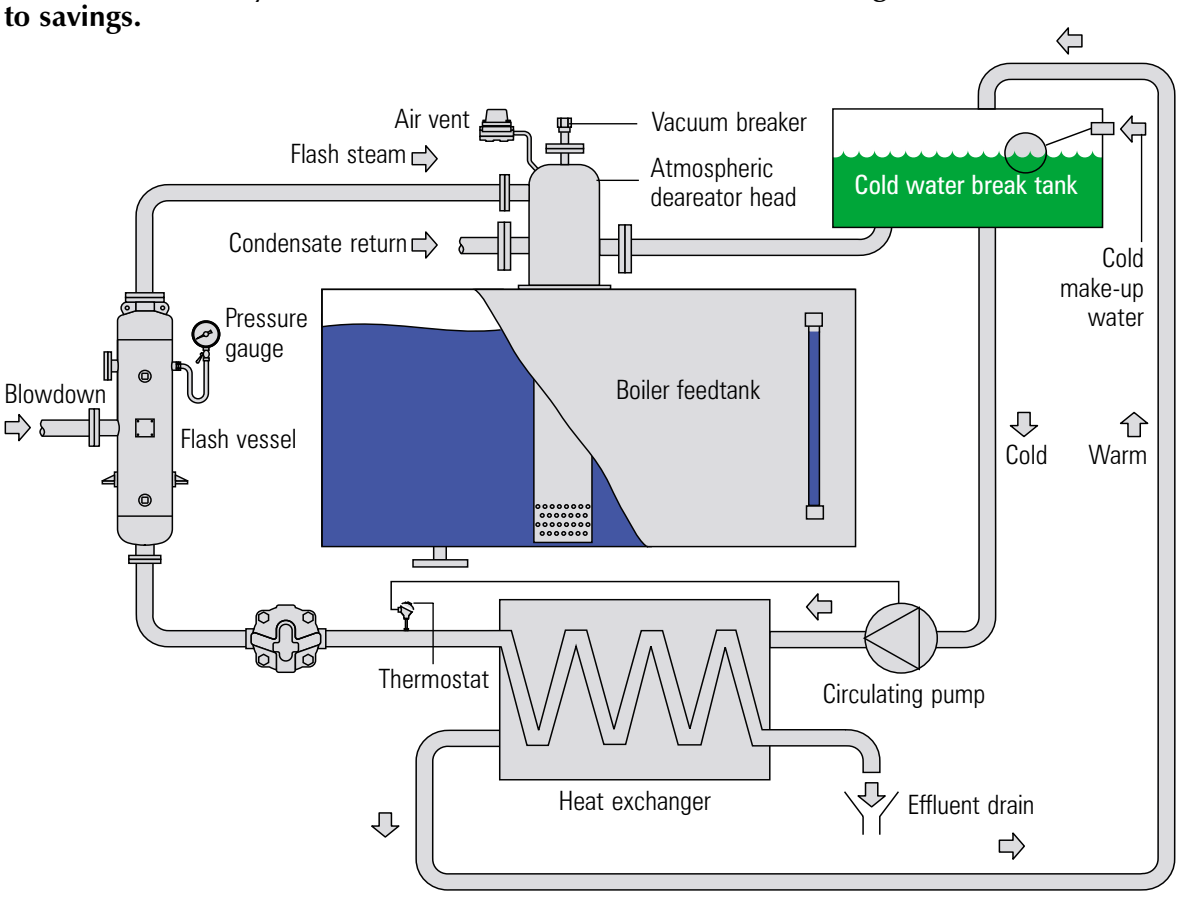


Fig. 3.13.4 Heating make-up water in a cold break tank (level controls have not been shown on the feedtank)

Questions

1. A boiler exports 5 000 kg/h of saturated steam at 14 bar g, If the TDS blowdown rate is 350 kg/h, approximately how much energy is lost to blowdown ?

a| 82 kW ☐
b| 117 kW ☐
c| 189 kW ☐
d| 68 kW ☐

2. Referring to Question 1, the blowdown water passes into a flash vessel operating at 0.7 bar g. How much energy will be released by the flash steam ?

a| 57 kW ☐
b| 51 kW ☐
c| 42 kW ☐
d| 36 kW ☐

3. Why might it be necessary to use flash steam from boiler blowdown in a heat exchanger on the pressure side of the boiler feedpump?

a| The feedtank is positioned too high for the flash steam to reach ☐
b| Flash steam injected into the feedtank will cause contamination ☐
c| When there is no deaerator head in the feedtank ☐
d| When further heating of the feedwater could cause cavitation in the feedpump ☐

4. Referring to Question 2, the residual blowdown from the flash vessel trap passes through a heat exchanger pre-heating cold make up water. If the residual blowdown discharges at 15°C how much energy has been recovered from the residual blowdown ?

a| 22 kW ☐
b| 28 kW ☐
c| 34 kW ☐
d| 47 kW ☐

5. Referring to Question 4, what could happen to the residual blowdown leaving the heat exchanger if it is never higher than 30°C ?

a| Filter the water and take it to drain ☐
b| Take the water to a blowdown vessel ☐
c| Discharge the water into the boiler feedtank ☐
d| Take the water to an effluent drain ☐

6. What controls the pressure of the flash steam taken to the deaerator ?

a| The size of the flash vessel ☐
b| The size of the flash steam pipe ☐
c| The head of water over the steam mixture discharge ☐
d| The blowdown rate ☐

Answers

1: a, 2: c, 3: d, 4: c, 5: d, 6: c

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Module 3.14

Bottom Blowdown

Bottom Blowdown

Suspended solids can be kept in suspension as long as the boiler water is agitated, but as soon as the agitation stops, they will fall to the bottom of the boiler. If they are not removed, they will accumulate and, given time, will inhibit heat transfer from the boiler fire tubes, which will overheat and may even fail.

The recommended method of removing this sludge is via short, sharp blasts using a relatively large valve at the bottom of the boiler. The objective is to allow the sludge time to redistribute itself so that more can be removed on the next blowdown.

For this reason, a single four-second blowdown every eight hours is much more effective than one, twelve-second blowdown in the first eight hour shift period, and then nothing for the rest of the day.

Blowdown water will either pass into a brick-lined blowdown pit encased below ground, or a metal blowdown vessel situated above ground. The size of the vessel is determined by the flowrate of blowdown water and flash steam that enters the vessel when the blowdown valve is opened.

The major influences on blowdown rate are:

- The boiler pressure.
- The size of the blowdown line.
- The length of the blowdown line between the boiler and the blowdown vessel.

In practice, a reasonable minimum length of blowdown line is 7.5 m, and most blowdown vessels are sized on this basis. Blowdown lines will contain bends, check valves and the blowdown valve itself; and these fittings will increase the pressure drop along the blowdown line. They may be thought of in terms of an 'equivalent straight length of pipe', and can be added to the pipe length to give an overall equivalent length. Table 3.14.1 gives equivalent lengths of various valves and fittings.

Table 3.14.1 Equivalent length of blowdown line fittings in metres (m)

Blowdown line size	20 mm	25 mm	32 mm	40 mm	50 mm
Long radius bend	0.4	0.5	0.6	0.7	0.8
Manifold inlet	0.6	1.0	1.4	1.7	2.1
Globe valve	5.9	9.6	12.2	13.9	17.8
Check valve	2.6	3.6	4.2	4.9	6.2
Blowdown valve	0.1	0.2	0.3	0.4	0.5

In the unlikely event that the total equivalent length is less than 7.5 m, the vessel should be sized on a higher flowrate. In these cases, multiply the boiler pressure by 1.15 to calculate the blowdown rate from Figure 3.14.1. Blowdown lines over 7.5 m can be read straight from this graph.

Example 3.14.1: For a boiler pressure of 10 bar g, an equivalent 40 mm blowdown line length is calculated to be 10 m**,** consequently, the blowdown rate is 6.2 kg/s (see Figure 3.14.1).

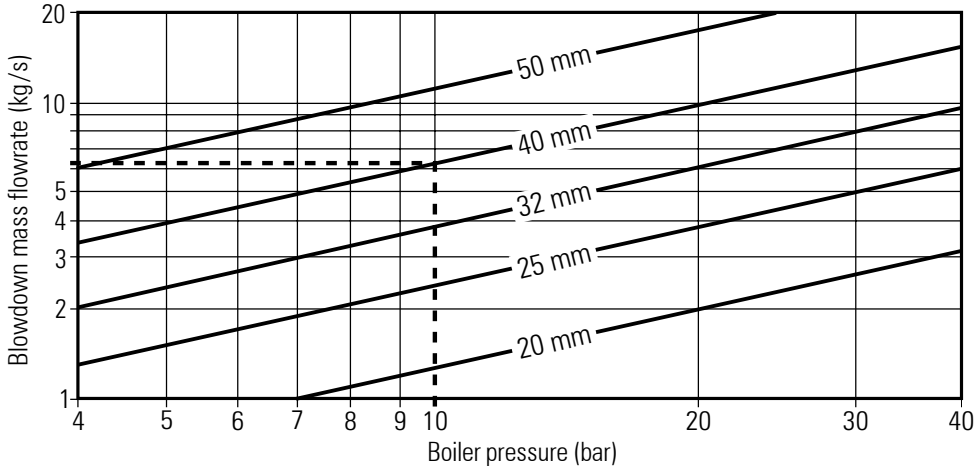


Fig. 3.14.1 Approximate blowdown rate (based on an 8 m equivalent pipe length)

There are two important factors to recognise with bottom blowdown:

- **Energy content of blowdown**
  The energy contained in the water being blown down is the liquid enthalpy of water at saturation temperature at boiler pressure. In Example 3.14.1, the boiler pressure is 10 bar g, and from steam tables, h_f is 782 kJ/kg. So the rate at which energy is being released from the boiler is:

  $$782 \text{ kJ/kg} \times 6.2 \text{ kg/s} = 4.85 \text{ MW}$$

- **Change in volume**

  Over a 3 second blowdown period, the amount of water blown down is:

  $$6.2 \text{ kg/s} \times 3 \text{ seconds} = 18.6 \text{ kg}$$

  The volume of the 18.6 kg of water blown down is:

  $$18.6 \text{ kg} \times 0.001 \text{ m}^3/\text{kg} = 0.018\,6 \text{ m}^3$$

  From flash steam calculations, 16% of water at 10 bar g saturation temperature will flash to steam when the pressure is reduced to atmospheric. Steam at atmospheric pressure has a significantly greater volume than water and each kilogram occupies 1.673 m³ of space.

  The resulting volume of flash steam from the 18.6 kg of boiler water is:

  $$(18.6 \text{ kg} \times 16\%) \times 1.673 \text{ m}^3/\text{kg} = 4.98 \text{ m}^3$$

  For comparison, the volume of water, is reduced to:

  $$\mathbf{(18.6 \text{ kg} \times 84\%) \times 0.001 \text{ m}^3/\text{kg} = 0.015\,6 \text{ m}^3}$$

The very high energy flowrate, and huge change in volume between the upstream and downstream sides of the blowdown valve, mean that substantial reactionary forces are developed, and that boiler blowdown must be handled in a safe manner.

Regulations and guidance notes

In the UK, due to the forces involved, and the potential for injury to personnel and the environment, boiler blowdown is covered in a number of statutes and Guidance Notes from the Health & Safety Executive.

The following are applicable in the UK, and have local equivalents in many other parts of the world:

- Factories Act (1961).
- Health and Safety at Work Act (1974).
- Public Health Act (1936).
- Health and Safety Guidance Notes PM60 and PM5.
- Pressure Systems and Transportable Gas Containers Regulations (1989).
- The European Pressure Equipment Directive (PED), (2002).

Compliance may or may not be mandatory, but an incident on the plant or injury to personnel will certainly involve factory inspectors and possible litigation.

Please note: The illustrations within this Module are schematic and some essential boiler fittings, for example, gauge glasses have been omitted for clarity.

Countries other than the UK should confirm the local equivalents of the above, but in any case should stress the importance of:

- Common sense.
- Good engineering and installation practice.
- Safety.

In all cases, it is important to ensure adequate isolation for maintenance purposes and the prevention of reverse flow. The installation of TDS control equipment on multi-boiler plants should include a non-return valve and an isolation valve to prevent pressure/flow from one boiler being imposed on another. This is particularly important when a boiler is shut down, as the TDS control valve may not be designed to seal against pressure on the downstream side. Good engineering practice will always consider what would happen if the control valve were passing water or steam. At worst, the absence of a non-return valve and isolation valve may endanger personnel working on, or in, the shut down boiler.

Bottom blowdown valve

In the UK, this type of valve is covered in the Factories Act (1961). Section 34 prohibits personnel entering specific boilers unless:

- All inlets through which steam or hot water might enter the boiler (from any other part of the system) are disconnected from that part; **or**
- All valves or taps controlling entry of steam or water are closed and securely locked. Where there is a common blowdown pipe or vessel, the blowdown valve is constructed so that it can only be opened by a key which cannot be removed until the blowdown valve is closed; and that this is the only key in use in the boiler house.

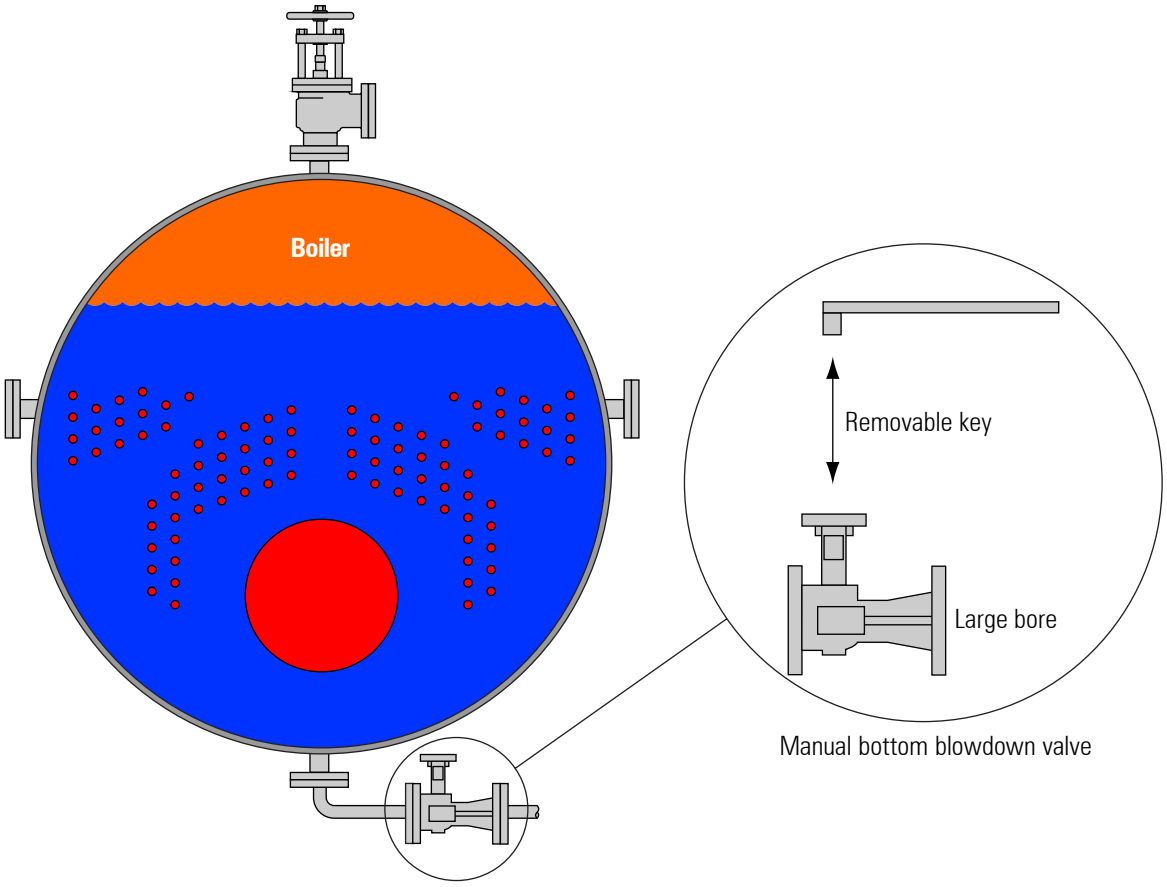


Fig. 3.14.2 Bottom blowdown valve with removable key

Timer controlled automatic bottom blowdown

It is now possible to automate the bottom blowdown valve using a proprietary timer linked to a pneumatically operated ball valve.

The timer should be capable of opening the valve at a specific time, and holding it open for a set number of seconds.

The use of automatic bottom blowdown ensures that this important action is carried out regularly and releases the boiler attendant for other duties.

With multi-boiler installations, it is necessary to interlock the valves so that not more than one can be open at any one time, as this would overload the blowdown vessel. This can be done most simply by staggering the setting times of the individual blowdown timers, or by setting the individual blowdown times in sequence.

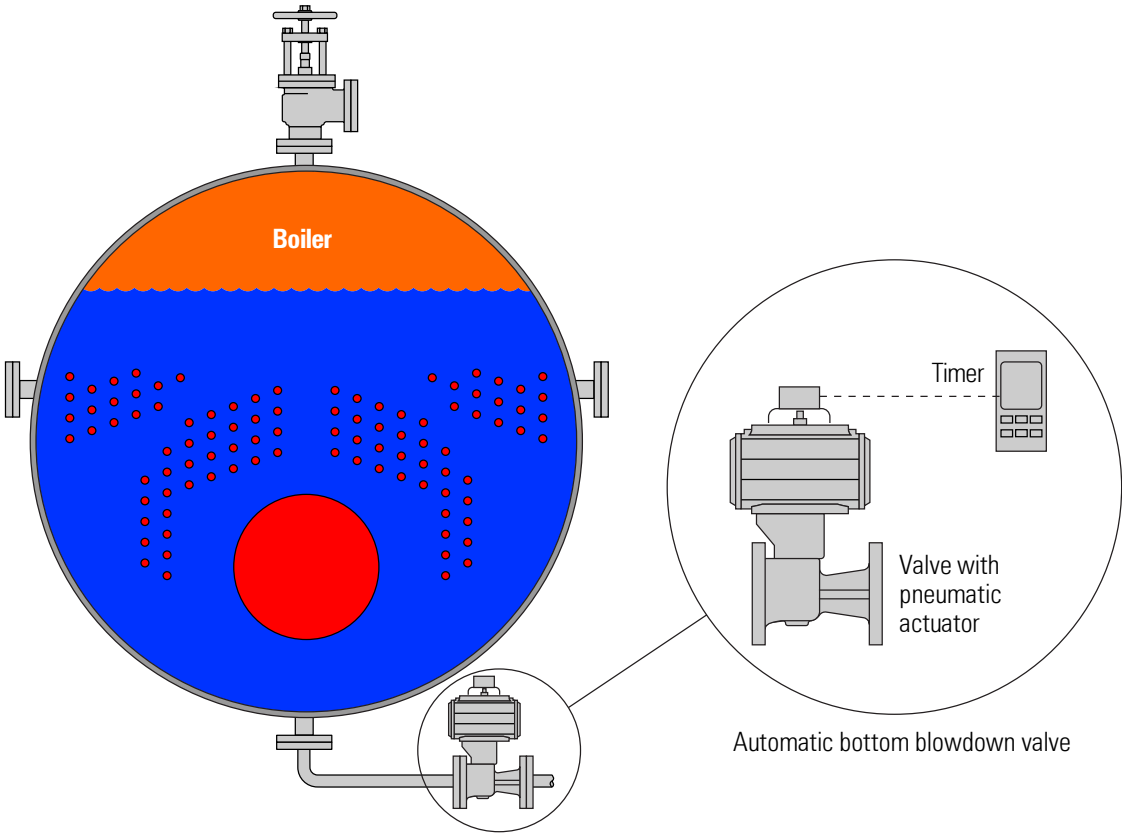


Fig. 3.14.3 Timer controlled automatic bottom blowdown valve

Blowdown vessels, as required by UK standards

Blowdown vessels are a preferred alternative to blowdown pits. The following information is extracted from HSE Guidance Note PM60 and provides information that may be useful in places other than the UK:

Traditionally, blowdown vessels have had tangential inlets. However, this has meant that the vessels have been structurally weak at the point where the inlet enters.

A preferred alternative is to bring the blowdown line in radially, giving a structurally superior vessel, and then fitting a diffuser inside the vessel. This arrangement also reduces the erosion which could occur inside a vessel with a tangential inlet.

Construction standard

The vessel will need to conform to the European Pressure Equipment Directive (2002) for Group 2 gases. This directive instructs the manufacturer to conform to design and manufacturing standards. As this is a pressure vessel specification, the vessel also needs provision for inspection including an access door and a drain.

Design temperature and pressure

The blowdown vessel design pressure should be at least 25% of the boiler maximum working pressure and the design temperature should be greater than or equal to the saturation temperature for the vessel design pressure.

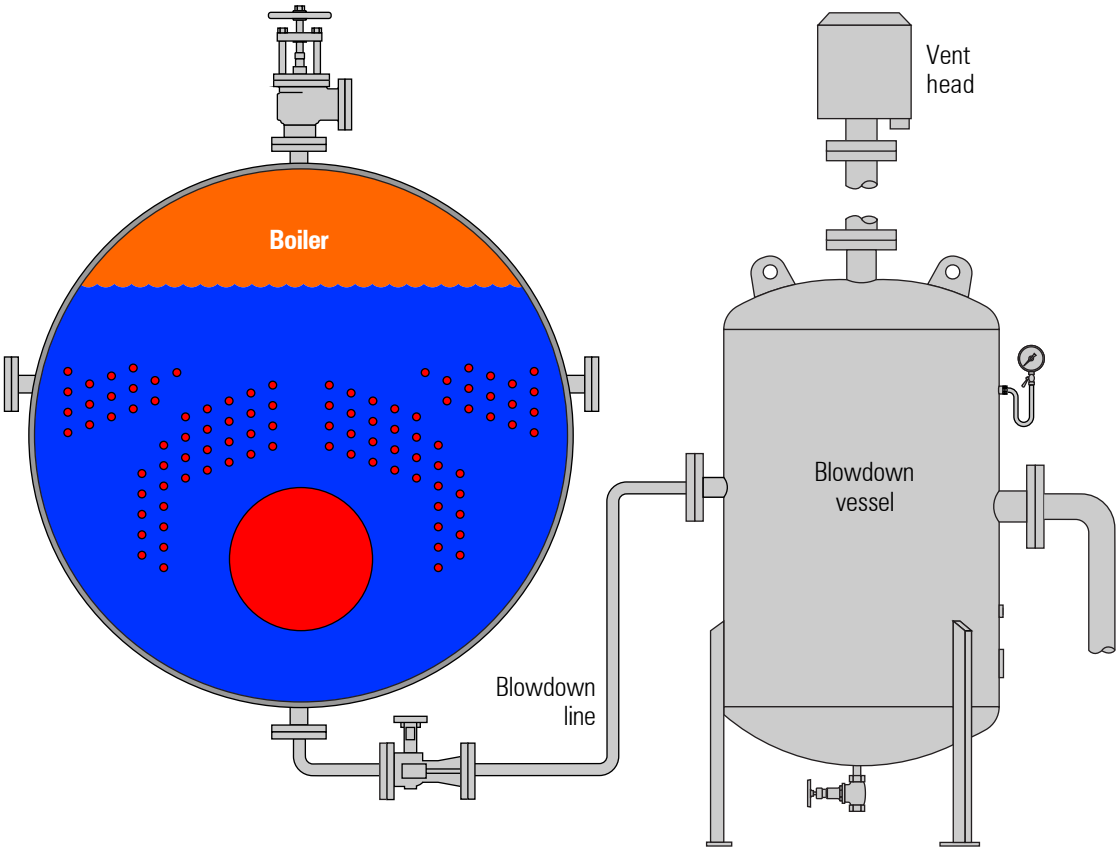


Fig. 3.14.4 A blowdown vessel installation on a single boiler (Not to scale)

Size

This depends on the boiler pressure and blowdown line size, however:

- The vent should be large enough, that pressure within the vessel does not exceed 0.35 bar g.
- The volume of standing water must ensure that the overflowing water temperature does not exceed 43°C.

Operation

The vessel should operate with a quantity of standing water, and the water quantity should be at least twice the quantity of blowdown water. Approximately half of the tank's volume should be occupied by standing water and the remainder as air space.

Vent

The vent should ensure that flash steam is vented safely and there is no significant carryover of water at the exit to the vent pipe. The vent should be as straight as possible and ideally terminated with a vent head.

Tapping for a pressure gauge

The vessel must have a tapping for a pressure gauge, as the vessel is manufactured to a pressure vessel specification and regular testing and inspection are required.

Cooling system

A cooling device should be fitted to the vessel if the hot water temperature causes the outlet temperature at blowdown to exceed the permissible limit. The most cost-effective choice for this application is a self-acting control valve. If the temperature exceeds the set temperature, the valve will open and allow cold mains water into the vessel.

Multi-boiler installations

The piping arrangement for multi-boiler installations is covered in the UK HSE Guidance Note (PM60); the following points are made:

Operation

Only one boiler can be blown down at any one time. In fact, sizing of the blowdown vessel will be based on the highest pressure boiler with the biggest blowdown line size. Reference is also made to the UK Factories Act (1961) which states the same thing.

Piping

Figure 3.14.5 shows the recommended layout for multiple boiler installations where the bottom and TDS blowdown lines are taken back separately to the blowdown vessel. Manifolding should be at the vessel and not at the boiler. Separate connections are required on the vessel for bottom blowdown and for TDS blowdown return lines.

A third connection is also needed on the vessel to comply with UK Guidance Note (PM5) regarding water level control in boilers. This requires a connection for the blowdown from control chambers and level gauge glasses.

Valving

Where blowdown lines connect into an inlet manifold on the vessel, each must be fitted with either a screw down non-return valve or, a non-return valve and an isolating valve. This is to prevent the possibility of steam and pressurised hot water being blown from one working boiler into another (inside which personnel may be working) during maintenance.

The preference is for two separate valves. The check valve will have to work regularly, hence wear on the seat is inevitable.

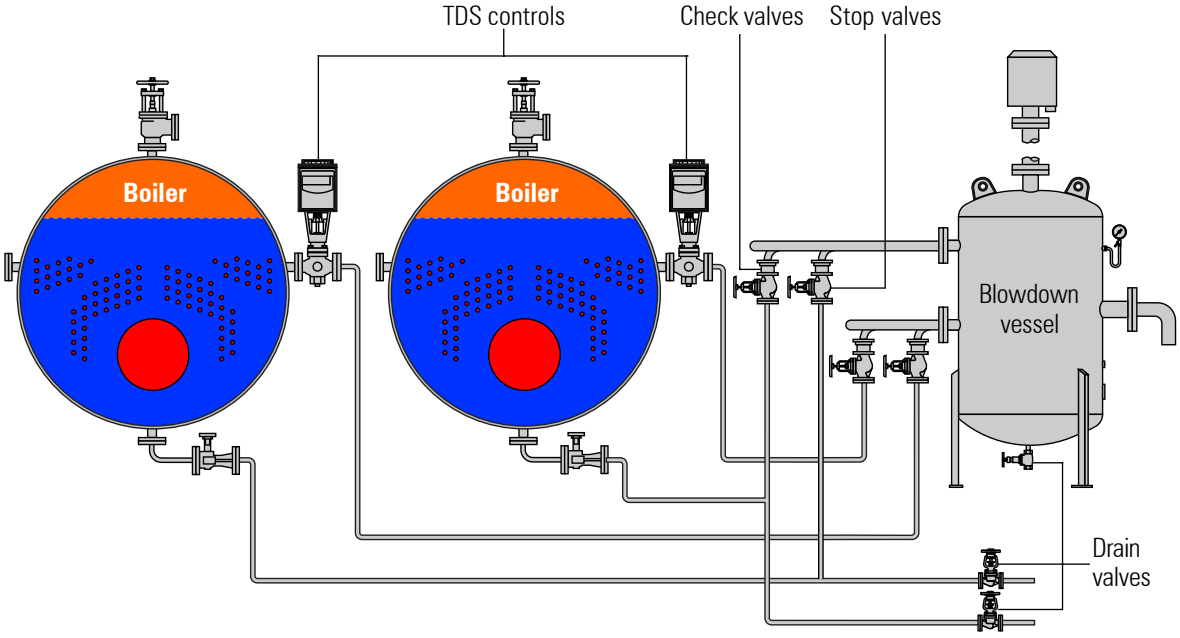


Fig. 3.14.5 A blowdown vessel on a multi-boiler installation

Questions

1. Why is heat recovery from bottom boiler blowdown not carried out ?

a| It is not permitted by boiler regulations ☐

b| The water is too contaminated with solids ☐

c| The blowdown is too intermittent to make it practical ☐

d| The blowdown must go direct to a blowdown vessel ☐

2. A boiler operates at 7 bar g and is fitted with a DN25 bore bottom blowdown valve. The boiler is blown down for 3 seconds every hour. What is the approximate blowdown rate ?

a| 1.9 kg/s ☐

b| 5.7 kg/s ☐

c| 5.0 kg/s ☐

d| 15 kg/s ☐

3. What is the prime purpose of the isolation valve and non-return valve on each blowdown line from a multi-boiler installation ?

a| To act as a reserve for the blowdown valve ☐

b| To assist in maintenance ☐

c| For pressure testing ☐

d| For prevention of reverse flow when a boiler is off-line ☐

4. Why should a blowdown valve be of large bore ?

a| It will give improved purging of sediment ☐

b| There will be a lower pressure drop across it ☐

c| It will be more able to handle the expansion of flash steam ☐

d| It is easier to open for a specified short time ☐

5. What is the purpose of the valve in the base of the blowdown vessel ?

a| For pressure testing ☐

b| To remove sludge from the blowdown vessel ☐

c| For isolating the cooled blowdown line to drain ☐

d| For inspection purposes ☐

6. With reference to the bottom blowdown, TDS blowdown and gauge glass blowdown lines to a blowdown vessel, which of the following statements is correct ?

a| The bottom and TDS blowdown lines can be joined ☐

b| All three lines must have separate connections to the vessel ☐

c| The TDS and gauge glass lines can be joined ☐

d| The bottom and TDS lines can be joined ☐

Answers

1: c, 2: a, 3: d, 4: a, 5: b, 6: b

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Module 3.15

Water Levels in Steam Boilers

Water Levels in Steam Boilers

The task of any steam boiler is to provide the correct amount of high quality steam: safely, efficiently, and at the correct pressure.

Steam is generated by heat from the combustion of fuel in a furnace, or by waste heat from a process. The heat is transferred to water in the boiler shell, which then evaporates to produce steam under pressure.

A certain area of water surface is required in a boiler from which to release the steam. A certain height should also be allowed above the normal working level, to allow the water level to rise with increasing load, but still allowing sufficient area to release the steam without carryover of water taking place.

In horizontal shell boilers, the water level rises with increasing load (due to the presence of more steam being below the water level in the boiler). As it does so, the water surface area (steam release area) will decrease because, as the water level is above the centre line of the boiler, the sides of the containing shell converge.

The boilermaker will have designed the boiler to ensure that the area of the normal water level (NWL) is such that steam will be released at an acceptable velocity. The design will also allow a specific minimum height of the steam off-take above the NWL.

Clearly, as steam is generated, the water in the boiler evaporates, and the boiler must receive a supply of water to maintain the level. Because of the factors outlined above, water must be maintained at the correct level.

Safety is also of paramount importance. If the boiler operates with insufficient water, severe damage could occur and there is ultimately the risk of explosion.

For this reason, controls are required which will:

- Monitor and control the water level.
- Detect if a low water level point is reached, and take appropriate action.

  This action may include:
  Sounding an alarm, shutting down the feedwater supply and shutting down the burner(s).

It is also essential to provide an external indication of the water level.



Photograph courtesy of BIB Cochrane

Fig. 3.15.1 Typical packaged steam boiler

The following Sections within this Module give basic information on the automatic level controls and alarms as applied to shell and tube boilers. This information is also generally applicable to the steam drum of water-tube boilers.

For the purpose of continuity, much of the information in this Module is based upon UK legislation. Other national regulations must be consulted where relevant.

Water level indication and boiler water levels

Water level indication applies to steam boilers where the water level can be detected. It includes most steam boilers, the exception being those of the 'once through' or coil type, where there is no steam drum. In such cases, steam outlet temperatures exceeding a pre-set value are taken to indicate insufficient water input.

In most cases, the simple gauge glass on the steam/water drum or boiler shell is used as the indicator. Many standards stipulate the provision of two gauge glasses. Arrangements are usually required to prevent a breakage from causing a hazard to the operator. The most common form of protection is a toughened glass screen to the front and sides of the water gauge glass.

Water gauge glass constructed from flat or prismatic glass may be required for high-pressure boilers.

The gauge glass device, which has stood the test of time, is used on the vast majority of boilers and is usually arranged to give a visible range of water level above and below the normal water level.

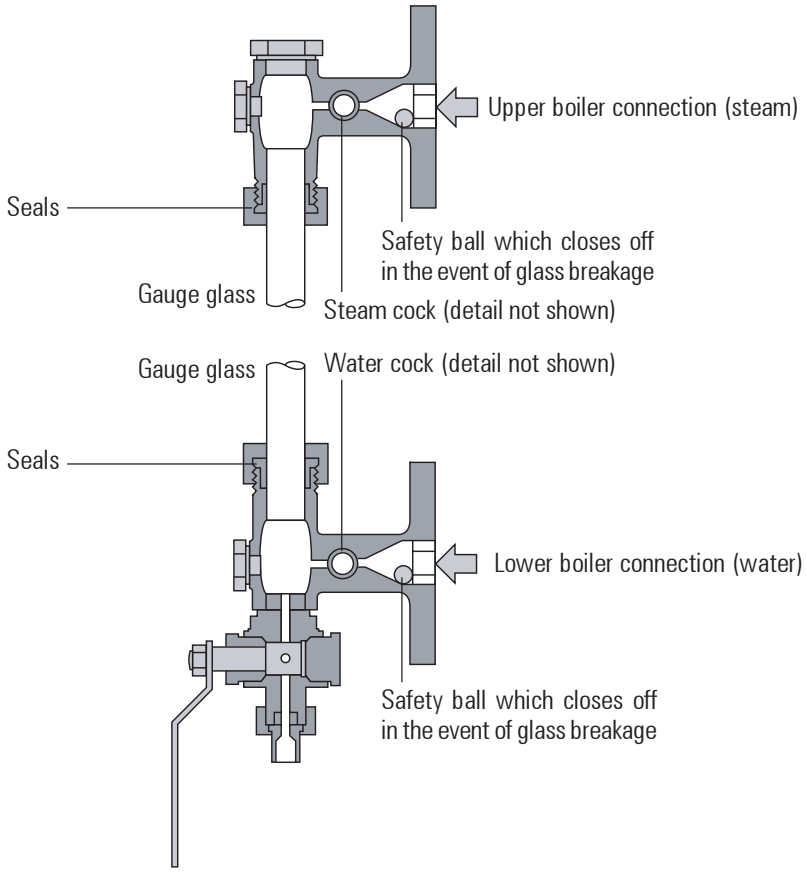


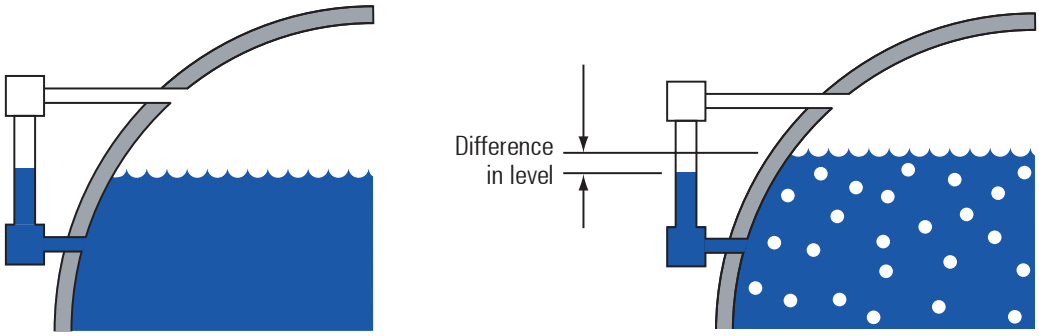
Fig. 3.15.2 Water gauge glass and mountings

It is essential to understand what is seen in a boiler gauge glass. The following Section explains some of the factors which will influence the level of water indicated in the gauge glass.

It is not possible to define the exact water level in a steaming boiler, because the water surface is made up of a mass of bubbles with a strong horizontal circulation. There are therefore, level variations both across and along the boiler shell. Conversely, the gauge glass contains water which:

- Is not subject to current and agitation.
- Does not contain steam bubbles.
- Is cooler than the water in the boiler.

This means that the water in the gauge glass (and other external fittings) is denser than the water within the boiler shell. This in turn, means that the level gauge glass will show a lower level than the average water surface level in the boiler shell.



Boiler off
No steam bubbles, and the level gauge glass shows the true water level in the shell

Boiler at high load
Many steam bubbles and a lower indicated level in the gauge glass

Fig. 3.15.3 Water level difference in the gauge glass

The difference between the level in the gauge glass and the level in the boiler shell at high steaming rates, depends on such factors as:

- The boiler steam generation rating.
- The height of the gauge glass water connection into the boiler.
- The TDS and chemical analysis of the boiler water.
- The size of the boiler shell.

Level changes due to boiler circulation

With a boiler on high load, the strong circulation of the boiler water will cause the water level to vary along the length of the boiler. These circulation currents are normally considered to be upwards along the front and back of the boiler, and upwards along the centreline over the furnace.

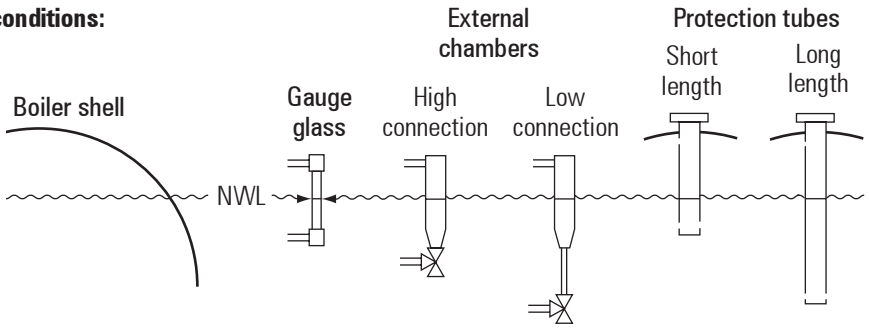
The downward circulation must therefore be at the sides, in the centre section of the boiler. There could also be a 'drawing' effect from the steam off-take connection which will tend to raise the water locally.

During sudden load changes there is also the possibility of waves developing in the boiler, which can often be seen in the level gauge glass, but should ideally be ignored by the water level controls.

A summary of the level changes to be expected under various boiler conditions is illustrated in Figure 3.15.4.

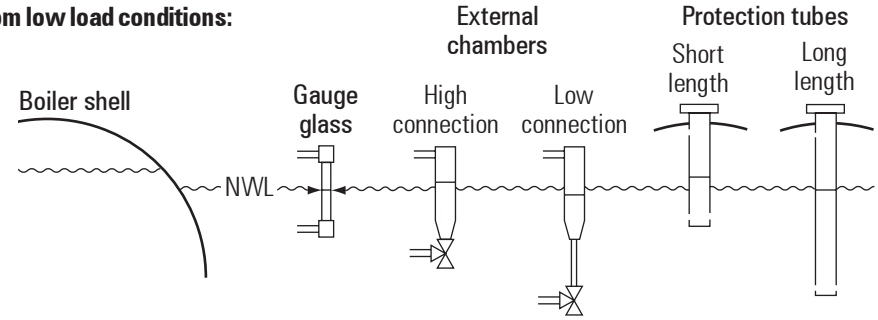
Boiler off-load or at low load conditions:

- All levels are the same.



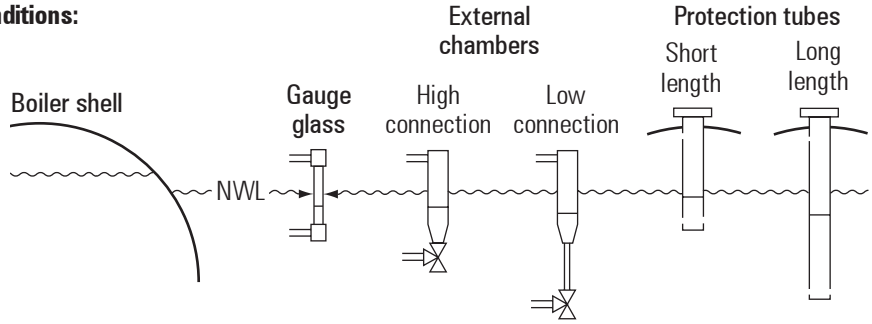
Boiler on sudden high load from low load conditions:

- Water quantity in the boiler is initially the same as at low load.
- If control is in a short length protection tube, feed supply will be cut off and high alarm may sound.



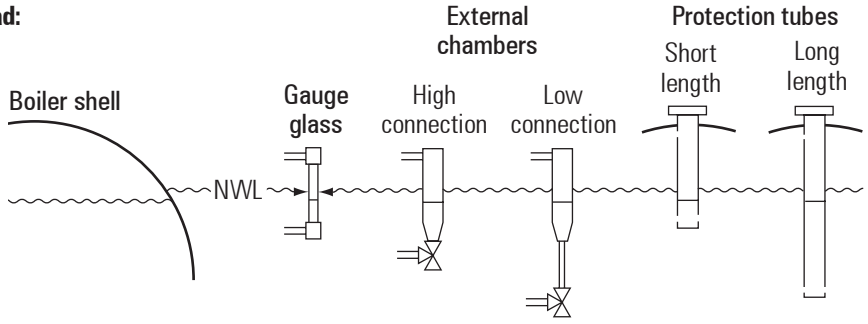
Boiler on high load, steady conditions:

- Control in short length protection tube.
- Level drops in boiler and gauge glass.



Boiler load drops from high load:

- Boiling rate reduces.
- Far fewer steam bubbles are formed so water level drops rapidly in the boiler shell.
- Low alarm may sound.



Boiler at high steady load:

- Control in external chamber with low connection or in long length protection tube.
- Level in boiler is high, but control is stable.
- Levels at the control point hardly change with load or feedwater flowrate.
- Boiler at high steady load.

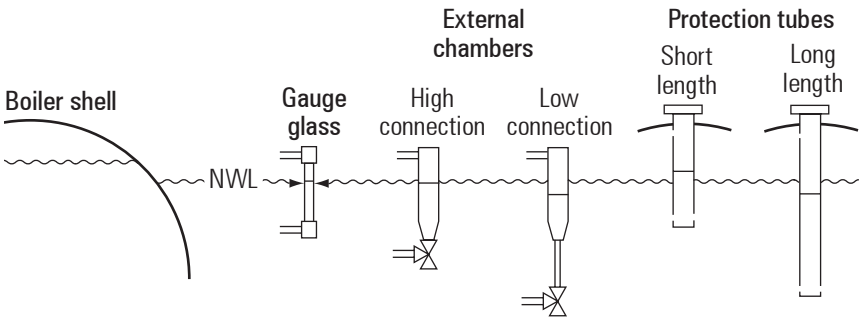


Fig. 3.15.4 Summary of level changes under various boiler conditions

Questions

1. The surface area of water in a steam boiler...

a| Remains constant during operation ☐

b| Increases during operation ☐

c| Reduces during operation ☐

d| Varies during operation ☐

2. It is important to maintain a water level over the furnace tubes to ensure:

a| Steam is generated at the appropriate pressure ☐

b| That sufficient steam is available for export from the boiler ☐

c| The tubes are maintained at a safe operating temperature ☐

d| That the water treatment plant operates at peak efficiency ☐

3. The functions of the level monitoring equipment in the boiler include:

a| Monitoring and controlling TDS ☐

b| Monitoring and controlling water level ☐

c| Monitoring and controlling the burner flame ☐

d| Monitoring and controlling air quality ☐

4. The true water level in all steam boilers may be observed through the gauge glass:

a| True ☐

b| False ☐

5. The circulation currents in a shell boiler are generally considered to be:

a| Upwards at the ends and downwards in the middle when viewed from the side ☐

b| Downwards at the ends and upwards in the middle when viewed from the side ☐

c| Clockwise when viewed from the burner ☐

d| Anticlockwise when viewed from the burner ☐

6. The placing of level monitoring equipment is crucial if accurate monitoring and control is to be assured:

a| Untrue, because the electronic equipment can be calibrated to take the effects of the currents into consideration ☐

b| Untrue because the position of the 'highs' and 'lows' are impossible to determine ☐

c| True, because the position of the 'highs' and 'lows' are known, and the level monitoring equipment must be placed so they read 'average' positions ☐

d| True, because the position of the 'highs' and 'lows' are known, and the level monitoring equipment must be placed so they read 'minimum' positions ☐

Answers

1: d, 2: c, 3: b, 4: b, 5: a, 6: c

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Module 3.16

Methods of Detecting Water Level in Steam Boilers

Methods of Detecting Water Level in Steam Boilers

On a steam raising boiler there are three clear applications for level monitoring devices:

- **Level control -** To ensure that the right amount of water is added to the boiler at the right time.
- **Low water alarm -** For safe boiler operation, the low water alarm ensures that the combustion of fuel does not continue if the water level in the boiler has dropped to, or below a predetermined level. For automatically controlled steam boilers, national standards usually call for two independent low level alarms, to ensure safety. In the UK, the lower of the two alarms will 'lockout' the burner, and manual resetting is required to bring the boiler back on line.
- **High water alarm -** The alarm operates if the water level rises too high, informing the boiler operator to shut off the feedwater supply. Although not usually mandatory, the use of high level alarms is sensible as they reduce the chance of water carryover and waterhammer in the steam distribution system.

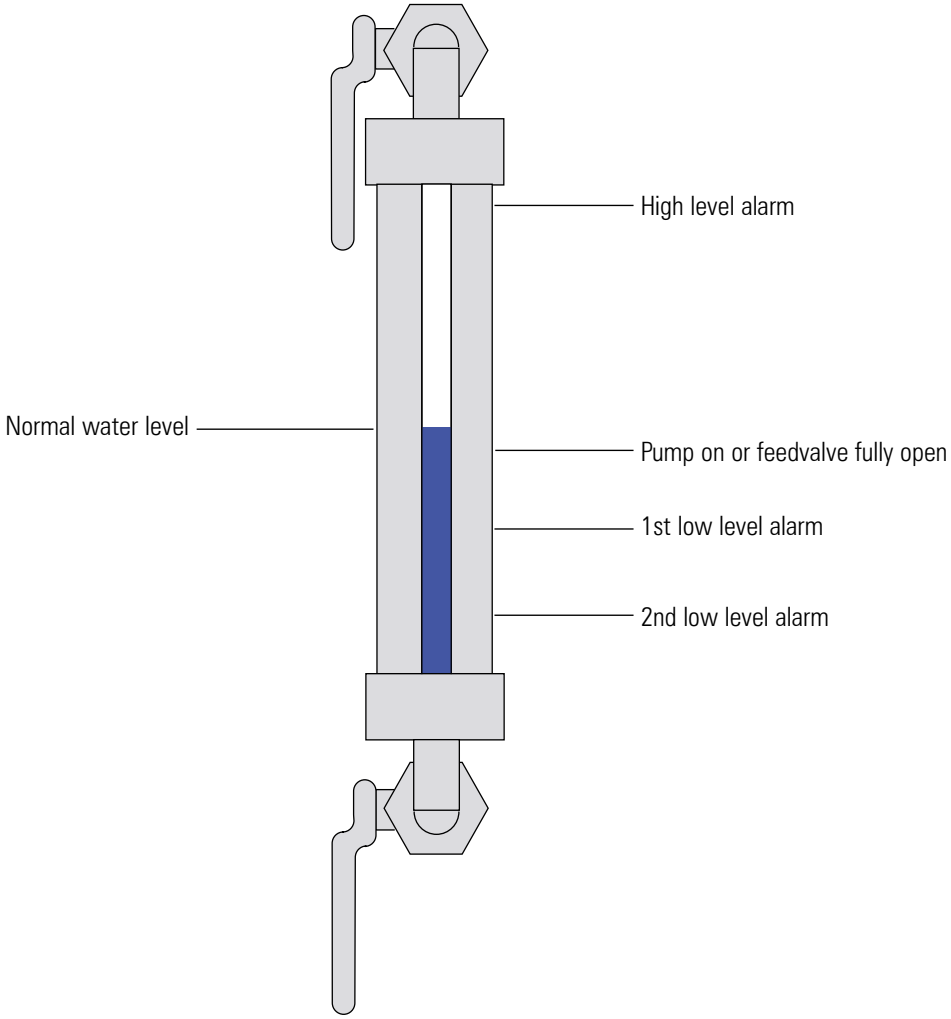


Fig. 3.16.1 Operating levels for water controls and alarms

Methods of automatic level detection

The following Sections within this Module discuss the principal types of level detection device which are appropriate to steam boilers.

Basic electric theory

The way in which electricity flows can be compared with a liquid. Liquid flows through a pipe in a similar way that electricity flows through a conductor (see Figure 3.16.2).

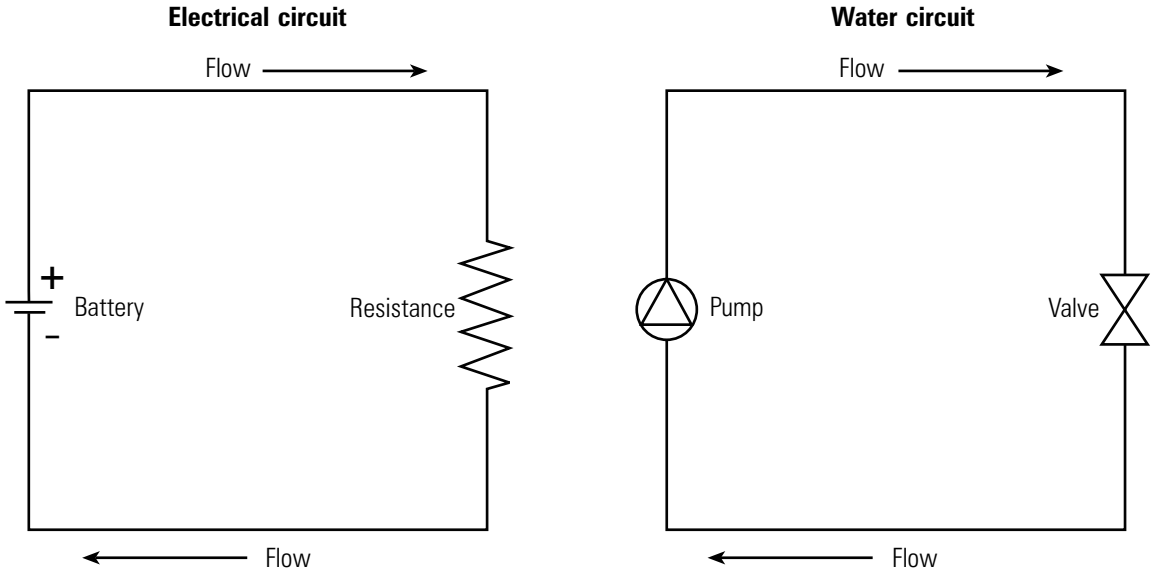


Fig. 3.16.2 Analogy of an electrical circuit with a water circuit

A conductor is a material, such as metal wire, which allows the free flow of electrical current. (The opposite of a conductor is an insulator which resists the flow of electricity, such as glass or plastic). An electric current is a flow of electric 'charge', carried by tiny particles called electrons or ions. Charge is measured in coulombs. 6.24×10^{18} electrons together have a charge of one coulomb, which in terms of SI base units is equivalent to 1 ampere second.

When electrons or ions are caused to move, the flow of electricity is measured in Coulombs per second rather than electrons or ions per second. However, the term 'ampere' (or A) is given to the unit in which electric current is measured.

- 1 A = A flow of 6.24×10^{18} electrons per second.
- 1 A = 1 coulomb per second.

The force causing current to flow is known as the electromotive force or EMF. A battery, a bicycle dynamo or a power station generator (among other examples) may provide it.

A battery has a positive terminal and a negative terminal. If a wire is connected between the terminals, a current will flow. The battery acts as a pressure source similar to the pump in a water system. The potential difference between the terminals of an EMF source is measured in volts and the higher the voltage (pressure) the greater the current (flow). The circuit through which the current flows presents a resistance (similar to the resistance presented by pipes and valves in a water system).

The unit of resistance is the ohm (given the symbol Ω) and Ohm's law relates current, voltage and resistance, see Equation 3.16.1:

$$I = \frac{V}{R}$$ **Equation 3.16.1**

Where:
I = Current (amperes)
V = Voltage (volts)
R = Resistance (ohms)

Another important electrical concept is 'capacitance'. It measures the capacity of the charge between two conductors (roughly analogous to the volume of a container) in terms of the charge required to raise its potential by an amount of one volt.

A pair of conductors has a large capacitance if they need a large amount of charge to raise the voltage between them by one volt, just as a large vessel needs a large quantity of gas to fill it to a certain pressure.

The unit of capacitance is one coulomb per volt, which is termed one farad.

Conductivity probes

Consider an open tank with some water in it. A probe (metal rod) is suspended in the tank (see Figure 3.16.3). If an electrical voltage is applied and the circuit includes an ammeter, the latter will show that:

- With the probe immersed in the water, current will flow through the circuit.
- If the probe is lifted out of the water, current will not flow through the circuit.

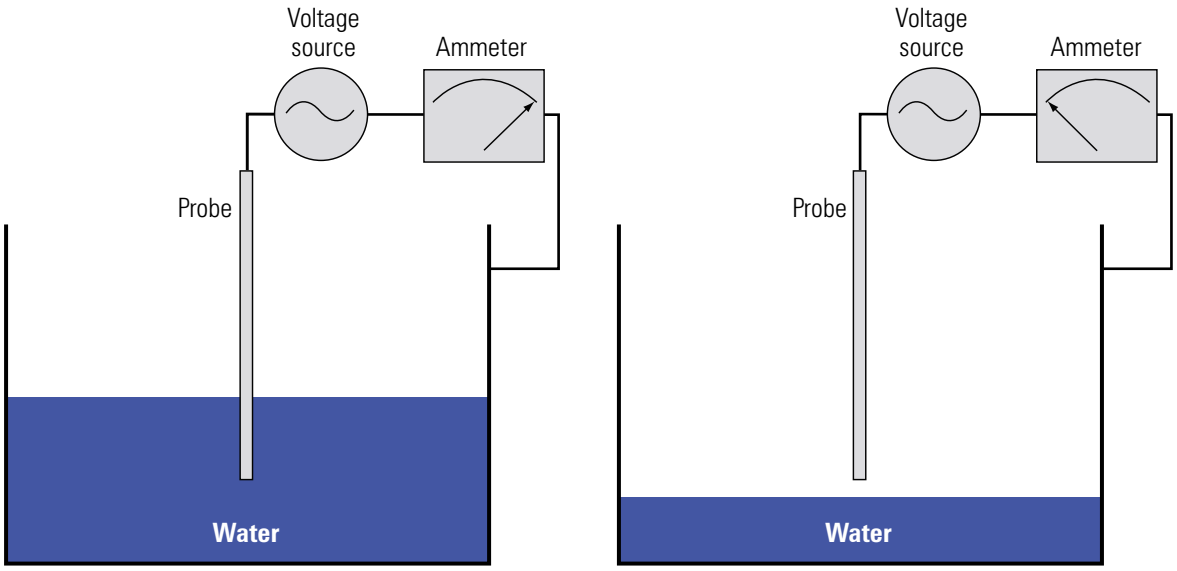


Fig. 3.16.3 Operating principle of conductivity probes - single tip

This is the basis of the conductivity probe. The principle of conductivity is used to give a point measurement. When the water level touches the probe tip, it triggers an action through an associated controller.

This action may be to:

- Start or stop a pump.
- Open or close a valve.
- Sound an alarm.
- Open or close a relay.

But a single tip can only provide a single or point action. Thus, two tips are required with a conductivity probe in order to switch a pump on and off at predetermined levels, (Figure 3.16.4). When the water level falls and exposes the tip at point **A**, the pump will begin to run. The water level rises until it touches the second tip at point **B**, and the pump will be switched off.

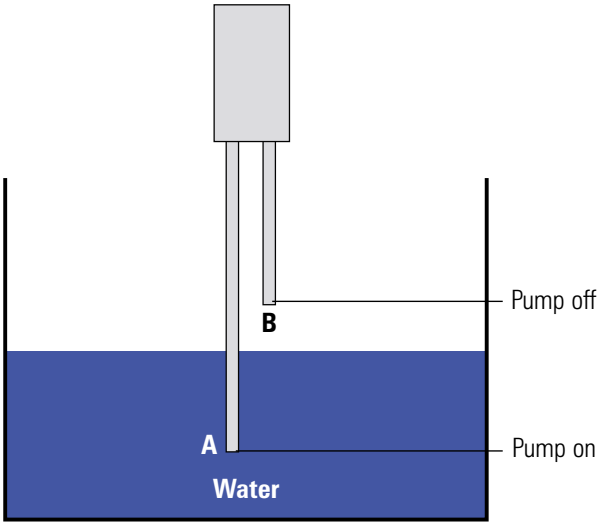


Fig. 3.16.4 Conductivity probes arranged to switch a feedpump on and off - two tip

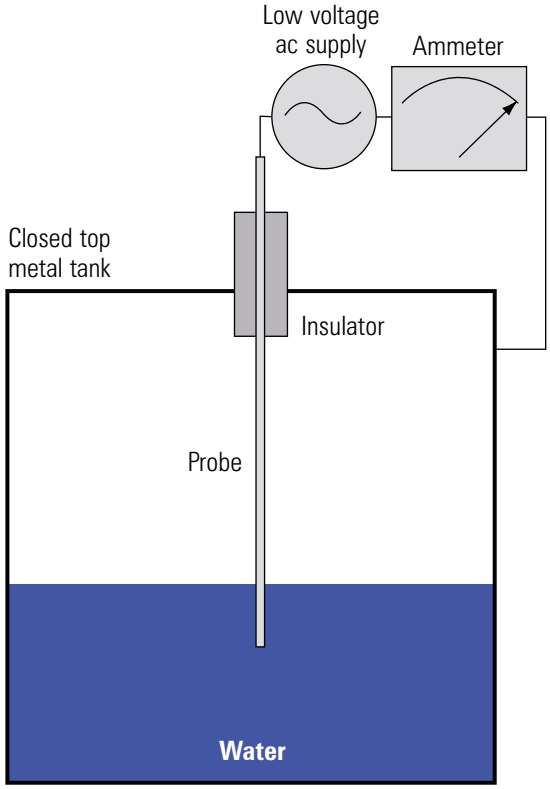


Fig. 3.16.5 Conductivity probe in a closed top tank

Probes can be installed into closed vessels, for example a boiler. Figure 3.16.5 shows a closed top metal tank - Note; an insulator is required where the probe passes through the tank top.

Again:

- With the probe immersed, current will flow.
- With the probe out of the water, the flow of current ceases.

Note: An alternating current is used to avoid polarisation and electrolysis (the splitting of water into hydrogen and oxygen) at the probe. A standard conductivity probe must be used to provide low water alarm in a boiler.

Under UK regulations, this must be tested daily.

For a simple probe there is a potential problem - If dirt were to build up on the insulator, a conductive path would be created between the probe and the metal tank and current would continue to flow even if the tip of the probe were out of the water. This may be overcome by designing and manufacturing the conductivity probe so that the insulator is long, and sheathed for most of its length with a smooth insulating material such as PTFE/Teflon®. This will minimise the risk of dirt build-up around the insulator, see Figure 3.16.6.

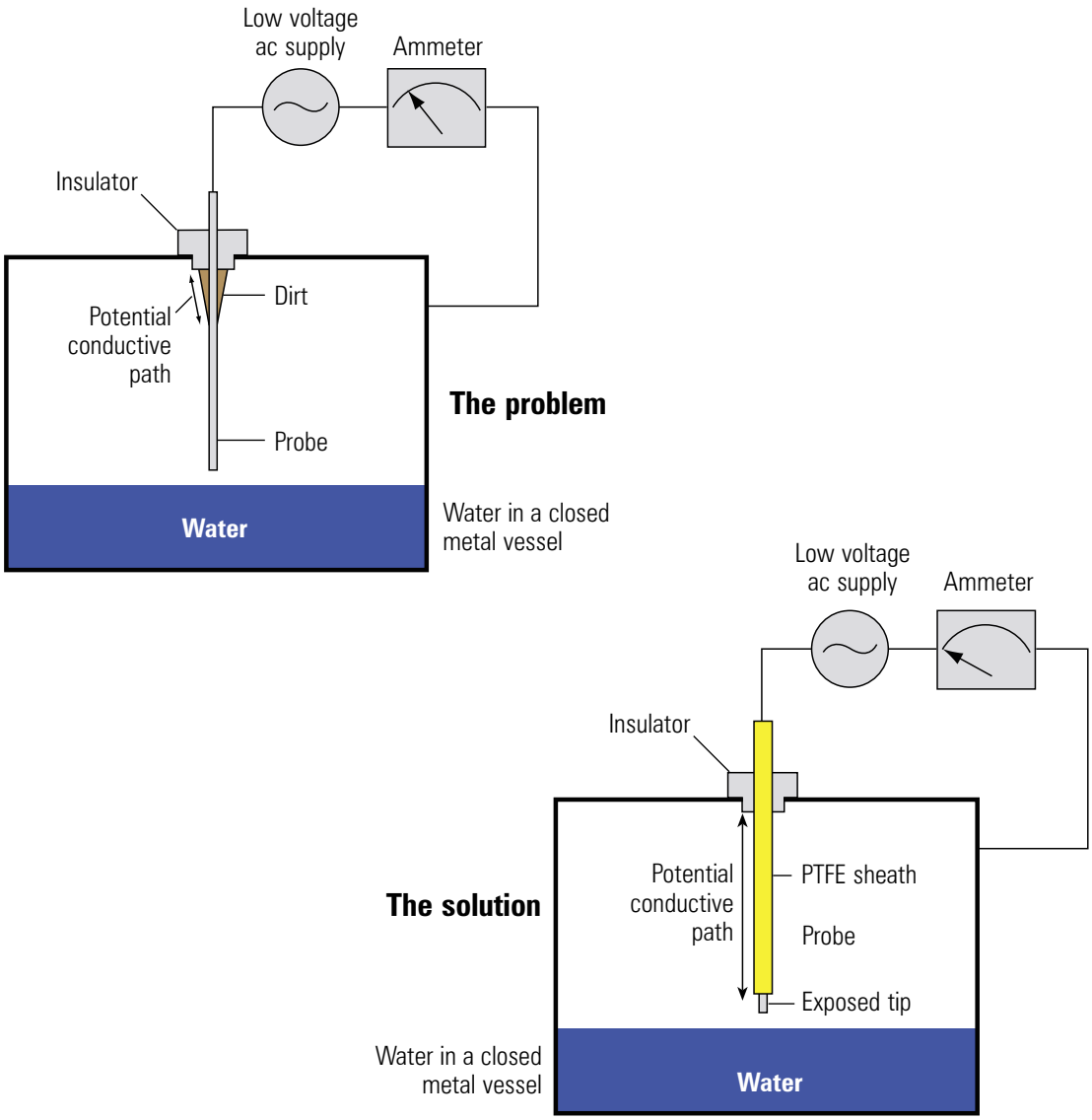


Fig. 3.16.6 Dirt on the insulator: the problem and the solution

The problem has been solved by:

- Using an insulator in the steam space.
- Using a long smooth PTFE sheath as an insulator virtually along the whole length of the metal probe.
- Adjustable sensitivity at the controller.

Special conductivity probes are available for low level alarms, and are referred to as 'self-monitoring'. Several self-checking features are incorporated, including:

- A comparator tip which continuously measures and compares the resistance to earth through the insulation and through the probe tip.
- Checking for current leakage between the probe and the insulation.
- Other self-test routines.

Under UK regulations, use of these special systems allows a weekly test rather than a daily one. This is due to the inherently higher levels of safety in their design.

The tip of a conductivity probe must be cut to the correct length so that it accurately represents the desired switching point.

Conductivity probes summary

Conductivity probes are:

- Normally vertically mounted.
- Used where on/off level control is suitable.
- Often supplied mounted in groups of three or four in a single housing, although other configurations are available.
- Cut to length on installation.

  Since the probes use electrical conductivity to operate, applications using very pure water (conductivity less than 5 µ Siemens/cm) are not suitable.



Fig. 3.16.7 A typical conductivity probe (shown with four tips) and associated controller

Capacitance probes

A simple capacitor can be made by inserting dielectric material (a substance which has little or no electrical conductivity, for example air or PTFE), between two parallel plates of conducting material (Figure 3.16.8).

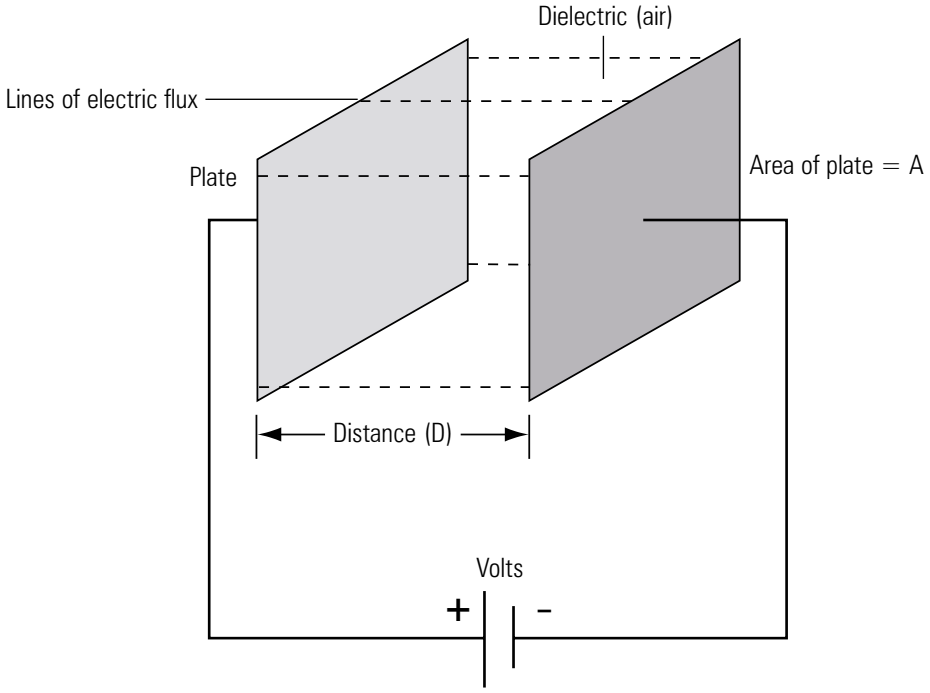


Fig. 3.16.8 A capacitor

The basic equation for a capacitor, such as the one illustrated in Figure 3.16.8, is shown in Equation 3.16.2:

$$C = K \frac{A}{D}$$ **Equation 3.16.2**

Where:
C = Capacitance (farad)
K = Dielectric constant (a function of the dielectric between the plates)
A = Area of plate (m^2)
D = Distance between plates (m)

Consequently:

- The larger the area of the plates, the higher the capacitance.
- The closer the plates, the higher the capacitance.
- The higher the dielectric constant, the higher the capacitance.

Therefore if A, D or K is altered then the capacitance will vary!

A basic capacitor can be constructed by dipping two parallel conductive plates into a dielectric liquid (Figure 3.16.9). If the capacitance is measured as the plates are gradually immersed, it will be seen that the capacitance changes in proportion to the depth by which the plates are immersed into the dielectric liquid.

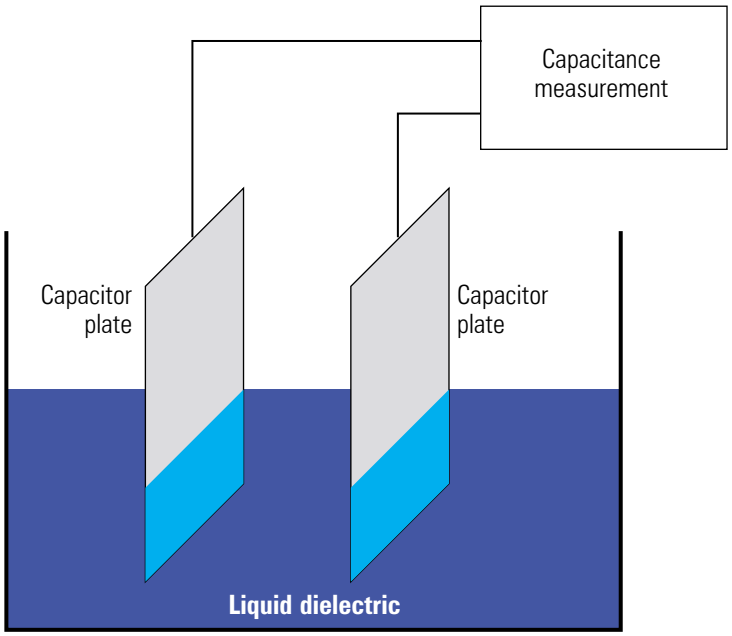


Fig. 3.16.9 A basic capacitor in a liquid

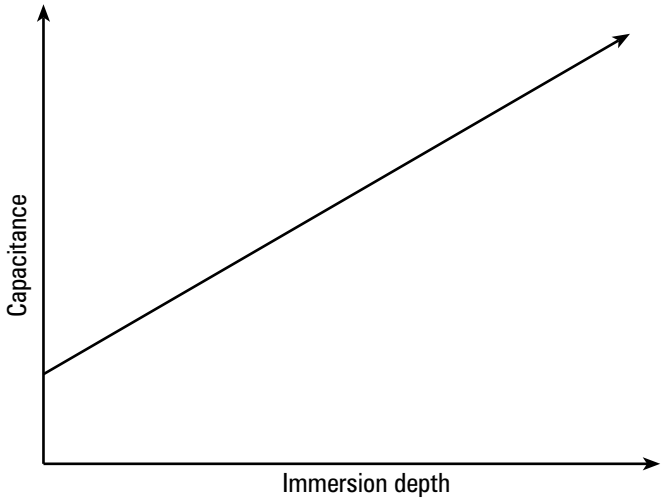


Fig. 3.16.10 Output from a capacitor in a liquid

The capacitance increases as more of the plate area is immersed in the liquid (Figure 3.16.10). A simple capacitor can be made by inserting dielectric material (a substance which has little or no electrical conductivity, for example air), between two parallel plates of conducting material (Figure 3.16.8).

The situation is somewhat different in the case of plates immersed in a conductive liquid, such as boiler water, as the liquid no longer acts as a dielectric, but rather an extension of the plates.

The capacitance level probe therefore consists of a conducting, cylindrical probe, which acts as the first capacitor plate. This probe is covered by a suitable dielectric material, typically PTFE. The second capacitor plate is formed by the chamber wall (in the case of a boiler, the boiler shell) together with the water contained in the chamber. Therefore, by changing the water level, the area of the second capacitor plate changes, which affects the overall capacitance of the system (see Equation 3.16.2).

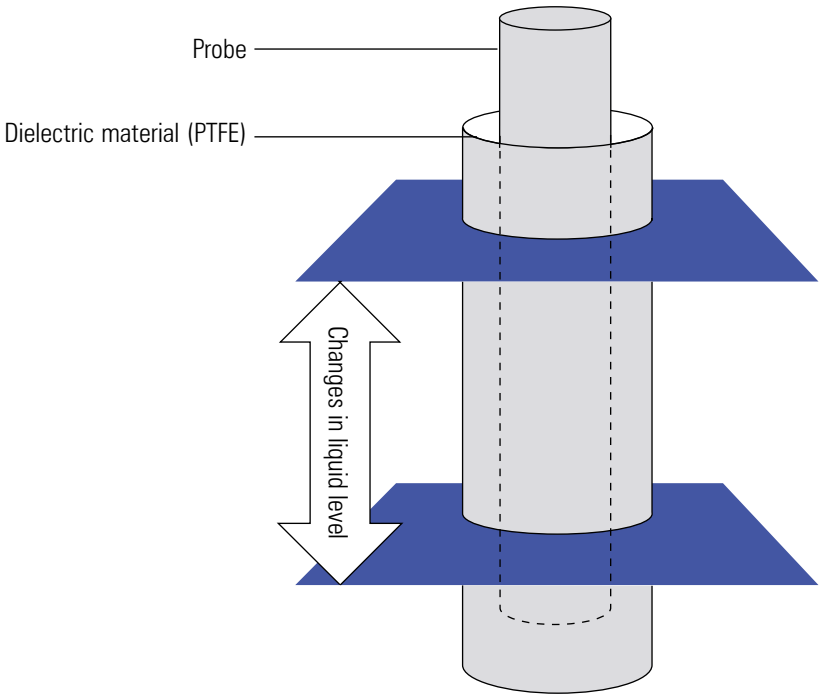


Fig. 3.16.11 Capacitance in water

The total capacitance of the system therefore has two components (illustrated in Figure 3.16.12):

- **C_A, the capacitance above the liquid surface -** The capacitance develops between the chamber wall and the probe. The dielectric consists of both the air between the probe and the chamber wall, and the PTFE cover.
- **C_B, the capacitance below the liquid surface -** The capacitance develops between the water surface in contact with the probe and the only dielectric is the PTFE cover.

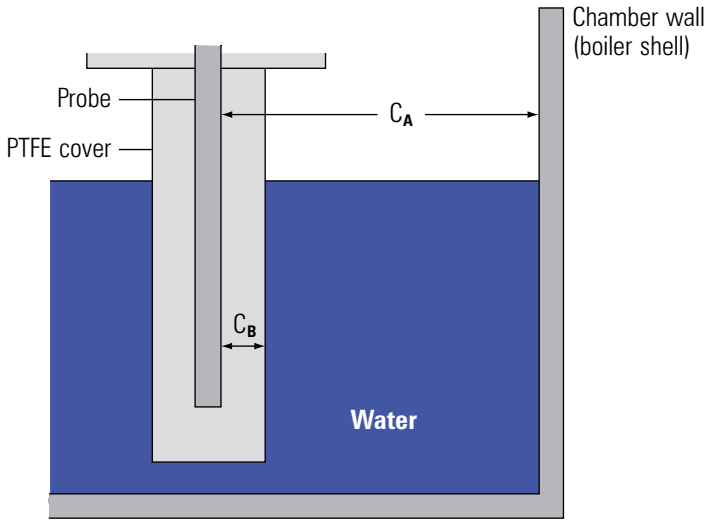


Fig. 3.16.12 Components of a capacitor signal (not to scale)

Since the distance between the two capacitance plates above the water surface (the chamber wall and the probe) is large, so the capacitance C_A is small (see Equation 3.16.2). Conversely, the distance between the plates below the water surface (the probe and the water itself) is small and therefore, the capacitance C_B will be large compared with C_A. The net result is that any rise in the water level will cause an increase in capacitance that can be measured by an appropriate device.

The change in capacitance is, however, small (typically measured in pico farads, for example, 10^{-12} farads) so the probe is used in conjunction with an amplifier circuit. The amplified change in capacitance is then signalled to a suitable controller.

Where the capacitance probe is used in, for example, a feedtank, (Figure 3.16.13) liquid levels can be monitored continuously with a capacitance probe. The associated controller can be set up to modulate a control valve, and / or to provide point functions such as a high level alarm point or a low level alarm.

The controller can also be set up to provide on / off control. Here, the 'on' and 'off' switching points are contained within a single probe and are set via the controller, removing any need to cut the probe. Since a capacitance probe must be wholly encased in insulating material, it must not be cut to length.

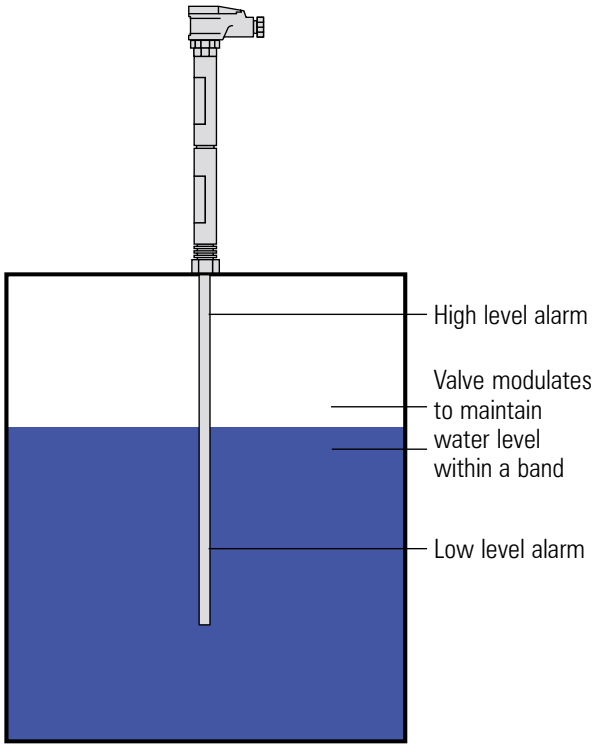


Fig. 3.16.13
Typical control using a capacitance probe in a feedtank (not to scale)

NOTE:
CAPACITANCE PROBES MUST NOT BE CUT TO LENGTH

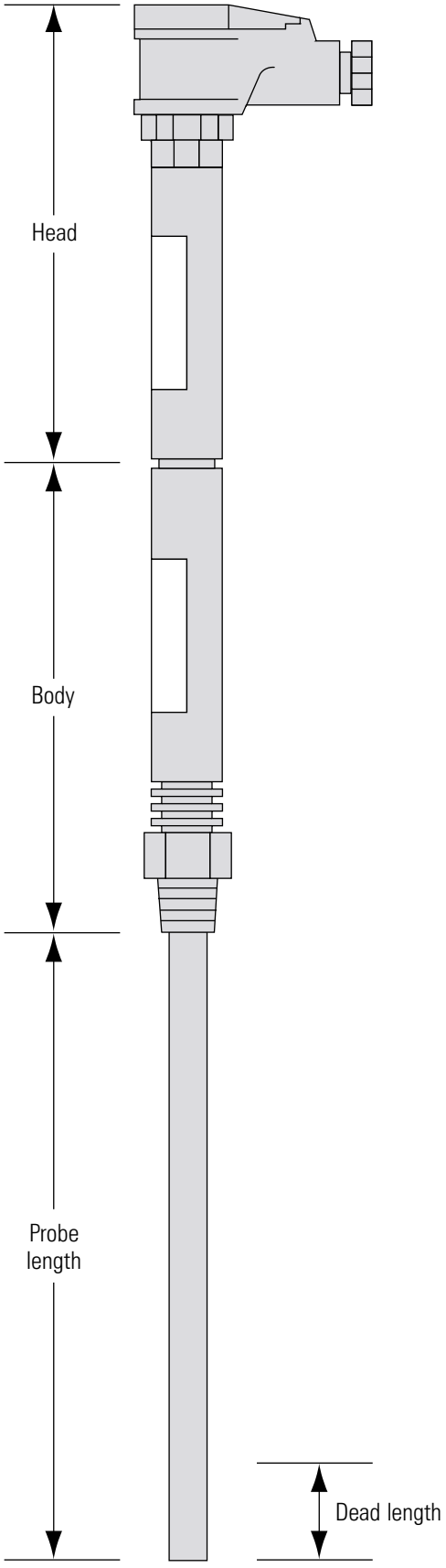


Fig. 3.16.14
Typical capacitance probe (shown with head)

Float control

This is a simple form of level measurement. An everyday example of level control with a float is the cistern in a lavatory. When the lavatory is flushed, the water level drops in the cistern, the float follows the water level down and opens the inlet water valve. Eventually the cistern shuts and as fresh water runs in, the water level increases, the float rises and progressively closes the inlet water valve until the required level is reached.

The system used in steam boilers is very similar. A float is mounted in the boiler. This may be in an external chamber, or directly within the boiler shell. The float will move up and down as the water level changes in the boiler. The next stage is to monitor this movement and to use it to control either:

- A feedpump (an on/off level control system)

  or

- A feedwater control valve (a modulating level control system)

Because of its buoyancy, the float follows the water level up and down.

- At the opposite end of the float rod is a magnet, which moves inside a stainless steel cap. Because the cap is stainless steel, it is (virtually) non-magnetic, and allows the lines of magnetism to pass through it.

In its simplest form, the magnetic force operates the magnetic switches as follows:

- The bottom switch will switch the feedpump on.
- The top switch will switch the feedpump off.

However, in practice a single switch will often provide on/off pump control, leaving the second switch for an alarm.

This same arrangement can be used to provide level alarms.

A more sophisticated system to provide modulating control will use a coil wrapped around a yoke inside the cap. As the magnet moves up and down, the inductance of the coil will alter, and this is used to provide an analogue signal to a controller and then to the feedwater level control valve.

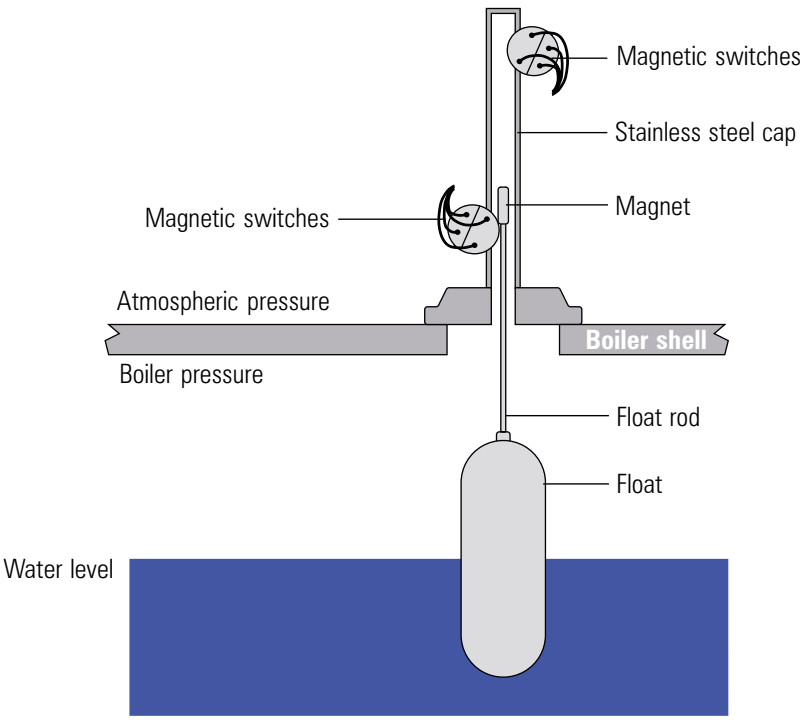


Fig. 3.16.15 Float control

Float control application

Vertically or horizontally mounted, the level signal output is usually via a magnetically operated switch (mercury type or 'air-break' type); or as a modulating signal from an inductive coil due to the movement of a magnet attached to the float. In both cases the magnet acts through a non-magnetic stainless steel tube.

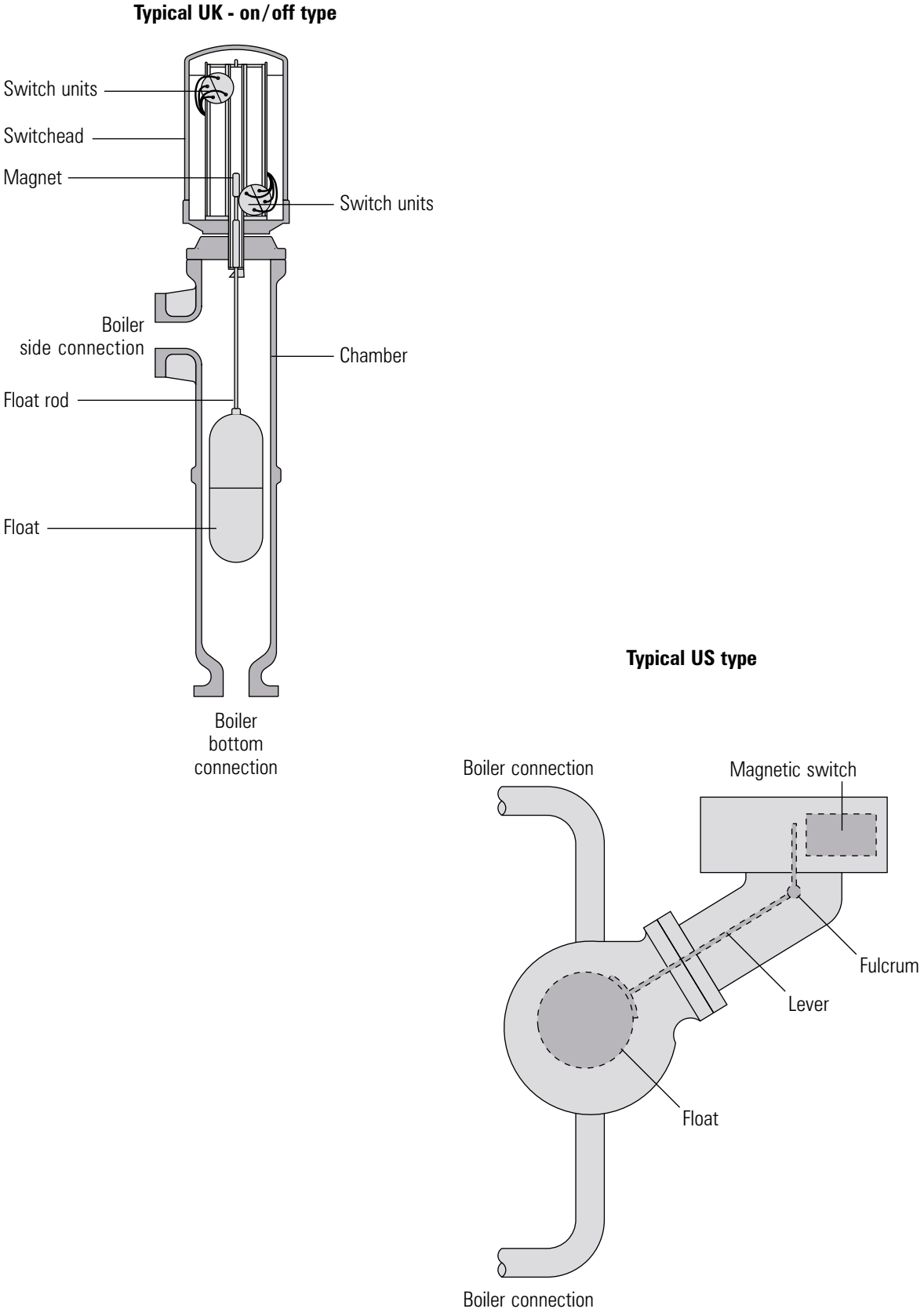


Fig. 3.16.16 Magnetic level controller in a chamber

Differential pressure cells

The differential pressure cell is installed with a constant head of water on one side. The other side is arranged to have a head which varies with the boiler water level.

Variable capacitance, strain gauge or inductive techniques are used to measure the deflection of a diaphragm, and from this measurement, an electronic level signal is produced.

Use of differential pressure cells is common in the following applications:

- High-pressure water-tube boilers where high quality demineralised water is used.
- Where very pure water is used, perhaps in a pharmaceutical process.

In these applications, the conductivity of the water is very low, and it can mean that conductivity and capacitance probes will not operate reliably.

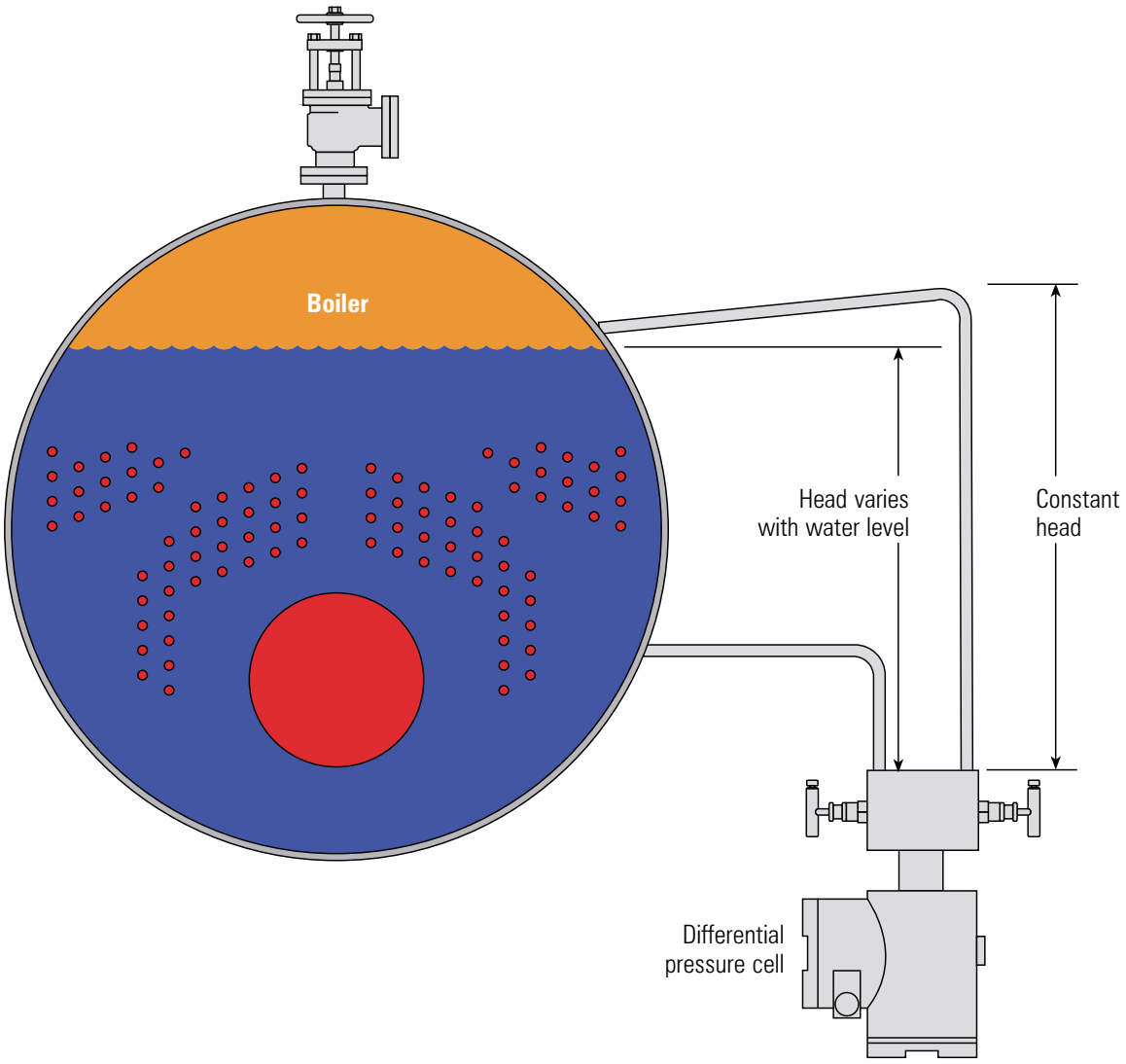


Fig. 3.16.17 Level control using a differential pressure cell (not to scale)

Other types of modulating control systems may occasionally be encountered. However, in order to comply with (UK) Health and Safety Executive (HSE) or insurance company demands, most boilers use one or other of the systems described above.

Questions

1. With regard to high water level conditions, which of the following statements is incorrect ?

a| Water carryover can occur in the distribution system ☐
b| It is usually mandatory to fit shell boilers with high level alarms ☐
c| Waterhammer can occur in the distribution system ☐
d| High water levels result in a lower steam release area ☐

2. Why are conductivity level probes often fitted in groups of three or four ?

a| As a safety back-up ☐
b| They monitor each other for example, for current leakage ☐
c| Because they incorporate separate self-monitoring probes for low level alarms ☐
d| Each probe serves a different function for example, pump on, pump off ☐

3. Which of the following is true of self-monitoring low level probes over standard conductivity probes ?

a| Self-monitoring level probes do not need cutting to length ☐
b| They are less susceptible to dirt collection ☐
c| They do not need testing at regular intervals ☐
d| Self-monitoring probes need testing once a day ☐

4. What is the advantage of a capacitance probe over a conductivity probe ?

a| Only one probe is required ☐
b| A capacitance probe is more accurate ☐
c| A capacitance probe can provide modulating alarms ☐
d| There is only one probe to cut ☐

5. Which of the following statements is true of a float control compared with a capacitance level probe ?

a| A float control can be used to operate a modulating control system ☐
b| A float must be fitted in an external chamber ☐
c| One float can provide level control and all necessary alarms ☐
d| A float can puncture and become inoperative ☐

6. Probe type level controls have failed to function in a clean steam application. The likely cause is:

a| The probes are defective ☐
b| The installation is incorrect ☐
c| The conductivity of the water is too low ☐
d| The insulation has broken down ☐

Answers

1: b, 2: c, 3: b, 4: c, 5: d, 6: c

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Module 3.17

Automatic Level Control Systems

Automatic Level Control Systems

On / off control

All the methods of level detection described so far can be used to produce an on / off signal for level control. The most common method of level control is simply to start the feedpump at a low level and allow it to run until a higher water level is reached within the boiler.

- With a float level control, a magnetic switch with a built-in hysteresis or dead-band will be used.
- With conductivity probes, two probes are necessary, (pump on and pump off) which will give fixed switching levels.
- A capacitance probe can be used to give adjustable on / off switching levels.

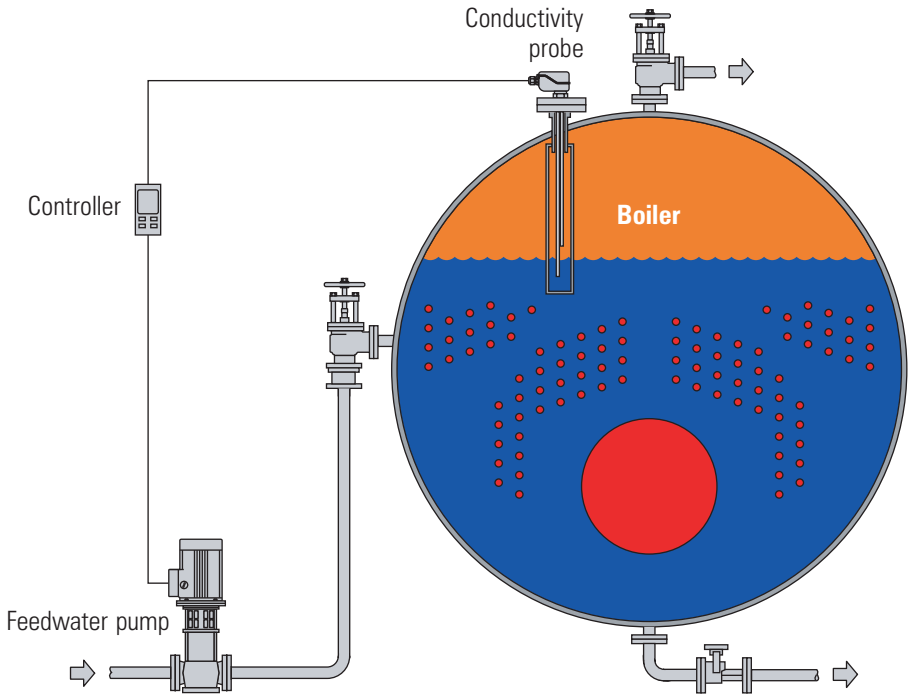


Fig. 3.17.1 On/off control

In the UK, on / off type control is almost universal on boilers below about 5 000 kg / h steam generation rate because it is the least expensive option. (In Australia and New Zealand, standards state that for boilers exceeding 3 MW (typically 5 000 kg / h), modulating control must be fitted).

It can be argued, however, that this type of on / off control is not ideal for boiler control, because the relatively high flowrate of 'cold' feedwater when the pump is on reduces the boiler pressure.

This causes the burner firing rate to continuously vary as the pump switches on and off.

Taking a typical example, it can be shown by calculation that even with feedwater at 80°C, the burner firing rate may have to be 40% higher with the feedpump on, than with the feedpump off.

This continuous variation causes:

- Wear on the burner controls.
- Temperature cycling of the boiler.
- Reduced efficiency.
- A 'saw-tooth' type steam flowrate as depicted by the chart recorder shown in Figure 3.17.2.

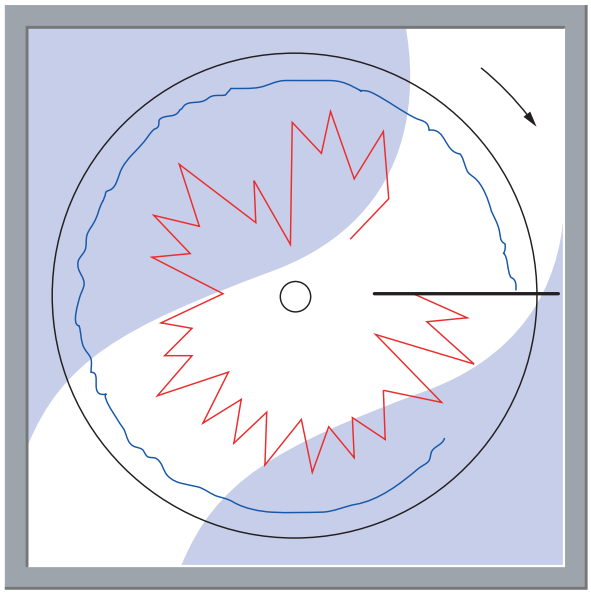


Fig. 3.17.2 Saw tooth trace on a chart recorder

If steam loads are high, the variable steam flowrate will tend to increase water carryover with the steam, and will tend to make water levels increasingly unstable with the associated danger of low water level lockout, particularly on multi-boiler installations.

However, the fact remains that on/off control is very widely used on boilers of small to medium output, as defined above, and that many problems associated with steam boilers operating with large swings in load are due in part to on/off level control systems.

Summary of on/off level control

Advantages:

- Simple.
- Inexpensive.
- Good for boilers on stand-by.

Disadvantages:

- Each boiler requires its own feedpump.
- More wear and tear on the feedpump and control gear.
- Variable steam pressure and flowrate.
- More boiler water carryover.
- Higher chance of daily operating problems under large load swings.

Modulating control

In this type of system the feedpump runs continuously, and an automatic valve (between the feedpump and the boiler) controls the feedwater flowrate to match the steam demand.

When operating correctly, modulating control can dramatically smooth the steam flowrate chart and ensure greater water level stability inside the boiler.

For modulating level control, the following methods can be used to sense the water level:

- Floats with a continuous signal output.
- Capacitance probes.
- Differential pressure cells.

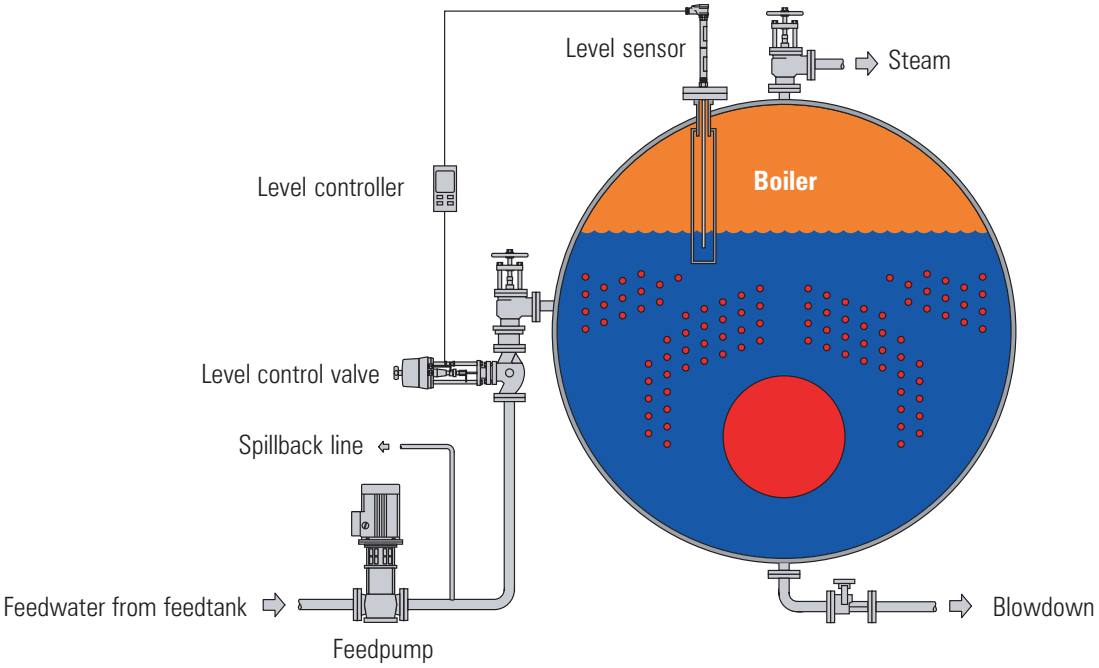


Fig. 3.17.3 Modulating control

Recirculation

To protect the feedpump from overheating when pumping against a closed modulating valve, a recirculation or spill-back line is provided to ensure a minimum flowrate through the pump.

This recirculation may be controlled by a valve or with an orifice plate. The amount of water to be recirculated is not great, and guidance is usually available from the pump manufacturer. As an indication, the orifice size will usually be between 5 mm and 7 mm for a typical boiler.

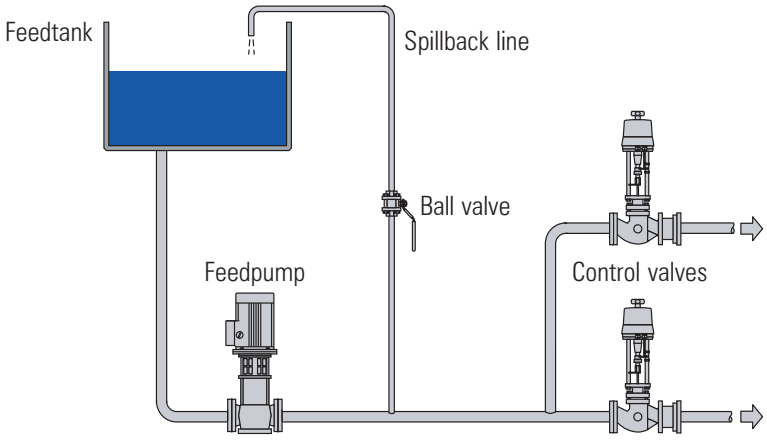


Fig. 3.17.4 Recirculation of feedwater

Modulating level control by varying the speed of the boiler feedwater pump

In this type of system, a modulating signal representing boiler water level (for example, from a capacitance probe) is directed to an electrical frequency controller. This controller in turn varies the frequency of the ac voltage to the boiler feedwater pump motor, and hence varies its speed.

- If a lot of water is required, the pump runs at high speed.
- If less water is required, the pump speed is reduced.

In this way the speed of the pump is modulated to provide a feedwater flowrate which matches the boiler's demand for feedwater.

There are two ways that variable speed drive technology is generally applied:

- **With recirculation -** When demand is satisfied and the motor speed is reduced to its minimum, and some recirculation of feedwater to the feedtank is still required to avoid the pump overheating (see Figure 3.17.5).
- **Without recirculation -** In this case the motor controller stops the feedpump at very low boiler loads, so recirculation is not required.

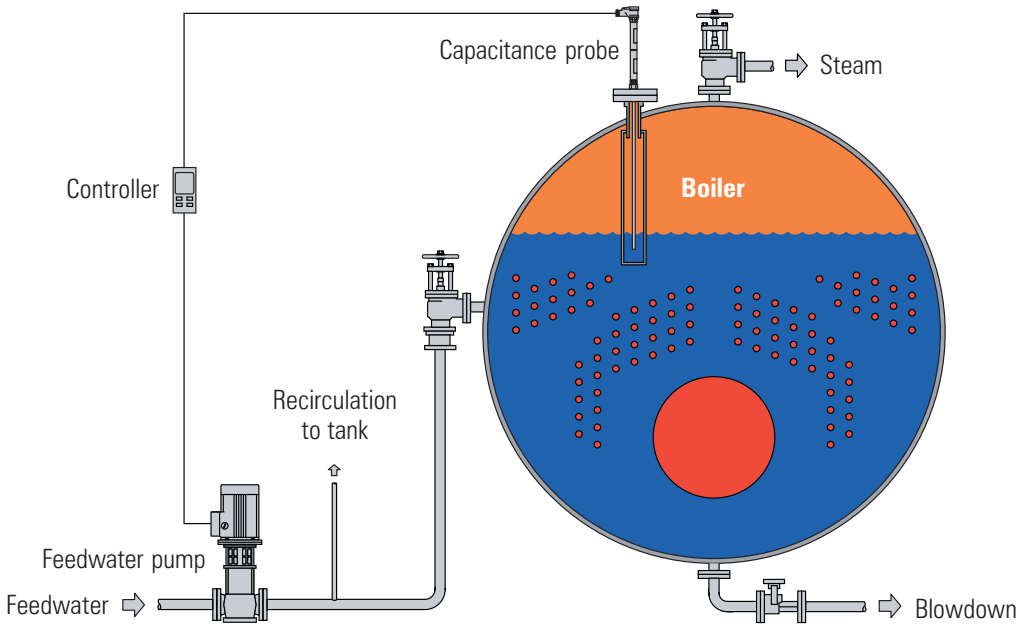


Fig. 3.17.5 Variable speed drive of a boiler water feedpump, with spill-back

Two important factors related to stopping and starting of the pump are:

- The pump must not be started and stopped within a given period of time more than is recommended by the manufacturer.
- When starting, the frequency controller should be ramped up from low speed, to minimise wear on the pump.

The principle advantage of variable speed drives is that as the speed of the pump varies, so does its power consumption, and, of course, reduced power consumption means reduced running costs.

However, the cost savings from using variable speed drives must be related to the higher cost of the control equipment. This is usually only viable for large boilers with wide variations in load or which operate in a lead/lag manner.

Single element water level control

The standard single element boiler water level control system, with proportional control, gives excellent control on the majority of boiler installations.

However, with single element proportional control, the water level must fall for the feedwater control valve to open. This means that the water level must be higher at low steaming rates and lower at high steaming rates: a falling level control characteristic.

However, where there are very sudden load changes, on some types of water-tube boiler, single element control has its limitations.

Consider the situation when a boiler is operating within its rated capacity:

- The boiler 'water' will actually contain a mixture of water and steam bubbles, which will be less dense than water alone.
- If the demand for steam increases, the pressure in the boiler initially falls, and the control system will increase the burner firing rate. The rate of evaporation will increase to meet the increased demand.
- The increased rate of evaporation means that the boiler water will contain more steam bubbles and become even less dense.

If a sudden load is now applied to the boiler:

- The pressure inside the boiler is further reduced, and a proportion of the boiler water will flash to steam. The flashing of the boiler water, plus the increased heat input as the burners turn up to maximum, means that the boiler 'water' will contain even more steam bubbles, and its density will be further reduced.
- As the pressure falls, the specific volume of the steam increases, and the resulting higher velocity at which the steam is drawn off the boiler can create a 'swell' of the steam bubble/water mixture, resulting in an apparent rise in water level.
- The level controls will detect this apparent rise in water level, and start to close the feedwater control valve, when in fact more water is required. The situation now, is that there is a high steam demand, and no water is being added to the boiler to maintain the level.
- A point is reached where the 'swell' in the water will collapse, possibly to a level below the low level alarms, and the boiler can suddenly 'lockout', bringing the plant off-line.

Two element water level control

Two element control reverses the falling level control characteristic to ensure that the water level is made to rise at high steaming rates. This strives to ensure that the quantity of water in the boiler stays constant at all loads, and that during periods of increased, sudden steam demand, the feedwater control valve opens.

The system works by using the signal from a steam flowmeter installed in the steam discharge pipework to increase the level controller set point at high steam loads.

The two elements of the signal are:

- **First element -** Level signal from the water within the boiler.
- **Second element -** Flow signal from the steam flowmeter in the boiler steam off-take.

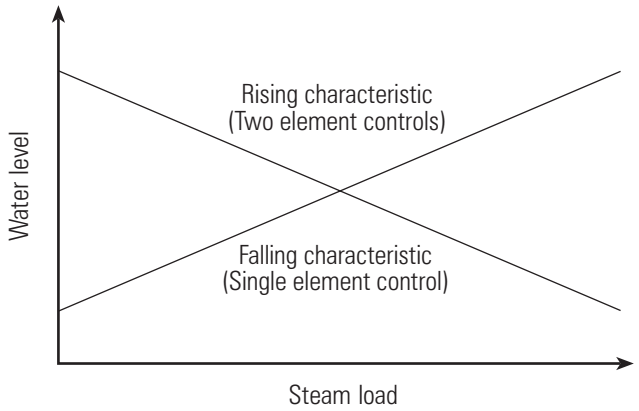


Fig. 3.17.6 Level control characteristics

Summary of two element water level control

Any boiler installation which experiences frequent, sudden changes in load may work better with a two element feedwater control system.

Where process load changes are severe (breweries are a common application) two element control should be considered and would appear to be necessary where there are sudden load changes of more than 25%, on a boiler.

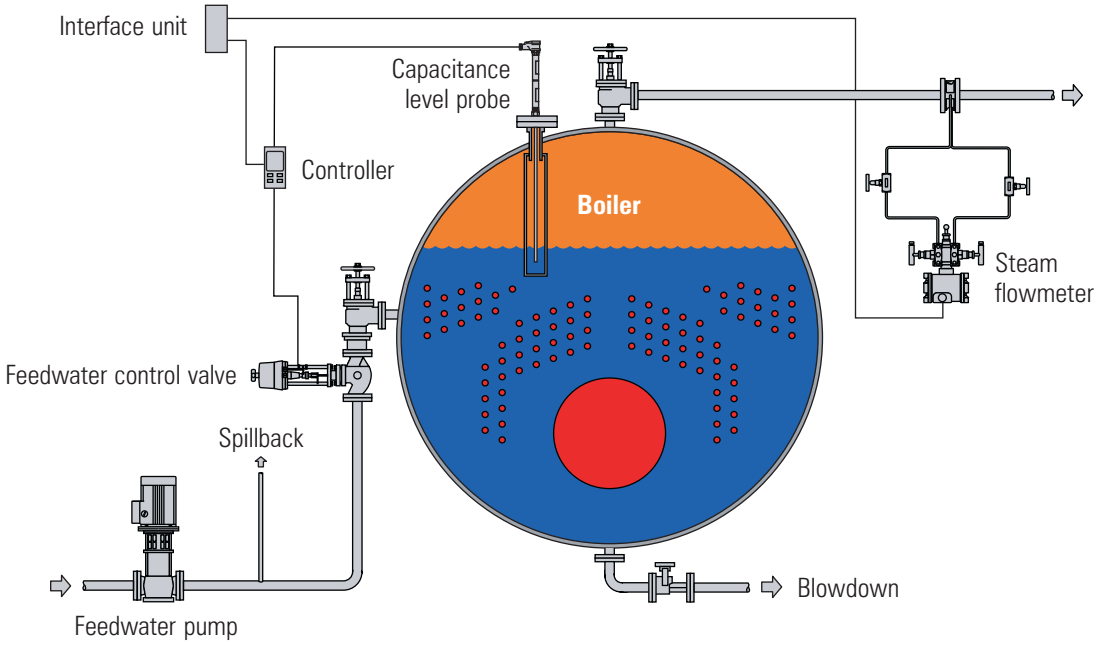


Fig. 3.17.7 Two element boiler water level control

Three element water level control

Three element control as shown in Figure 3.17.8, involves the two signal elements as previously mentioned, plus a third element, which is the actual measured flowrate of feedwater into the boiler. Three element control is more often seen in boiler houses where a number of boilers are supplied with feedwater from a common, pressurised ring main.

Under these circumstances the pressure in the feedwater ring main can vary depending on how much water is being drawn off by each of the boilers.

Because the pressure in the ring main varies, the amount of water which the feedwater control valve will pass will also vary for any particular valve opening. The input from the third element modifies the signal to the feedwater control valve, to take this variation in pressure into consideration.

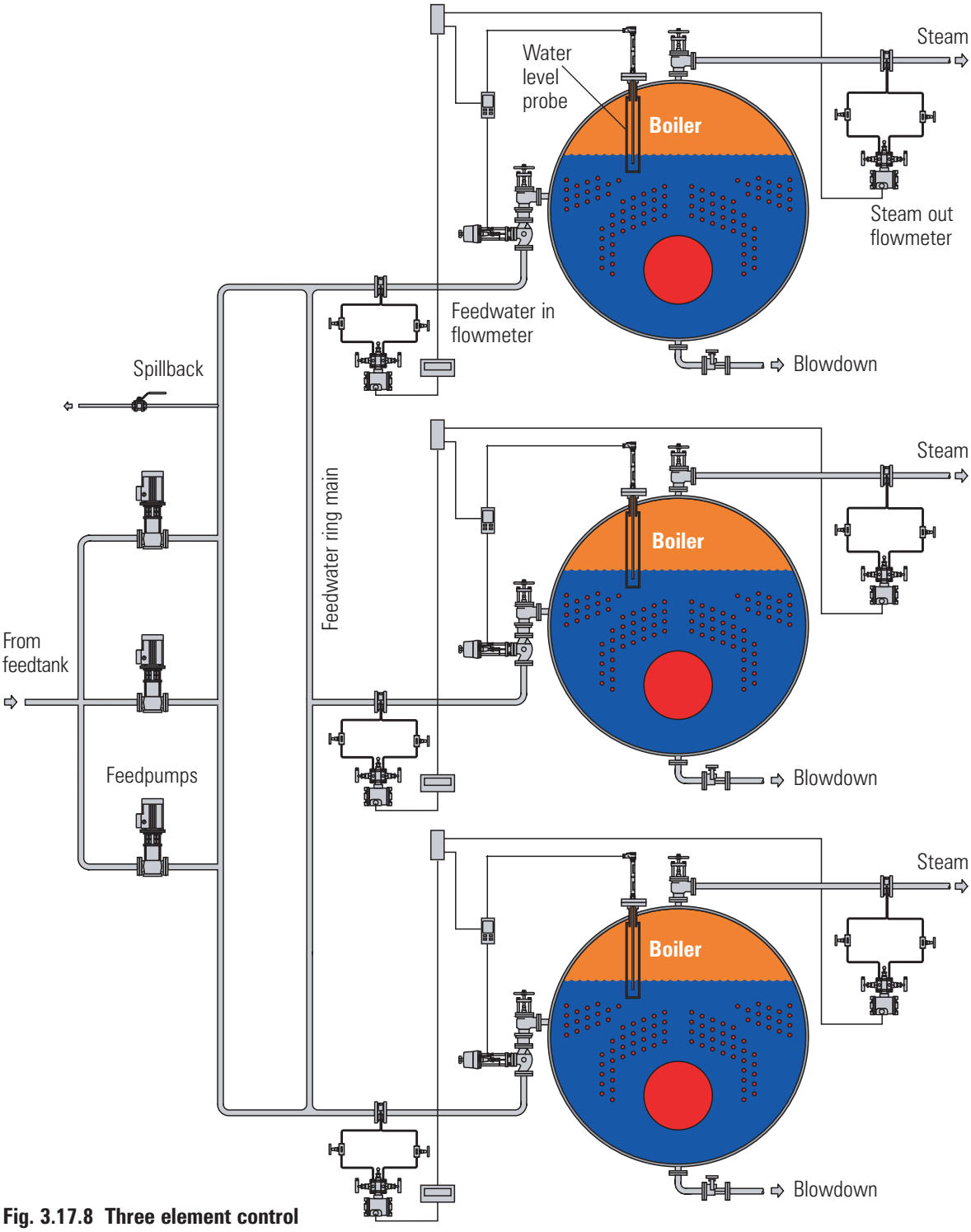


Fig. 3.17.8 Three element control

Summary of modulating level control

Advantages:

- Steady steam pressure and flowrate within the boiler's thermal capacity.
- More efficient burner operation.
- Less thermal stress on the boiler shell.
- Less boiler water carryover.
- Can use a central feedpump station.
- Less wear and tear on the feedpump and burner.

Disadvantages:

- More expensive.
- Feedpump must run continually.
- Less suitable for 'stand-by' operation.
- Possibly greater electricity consumption.

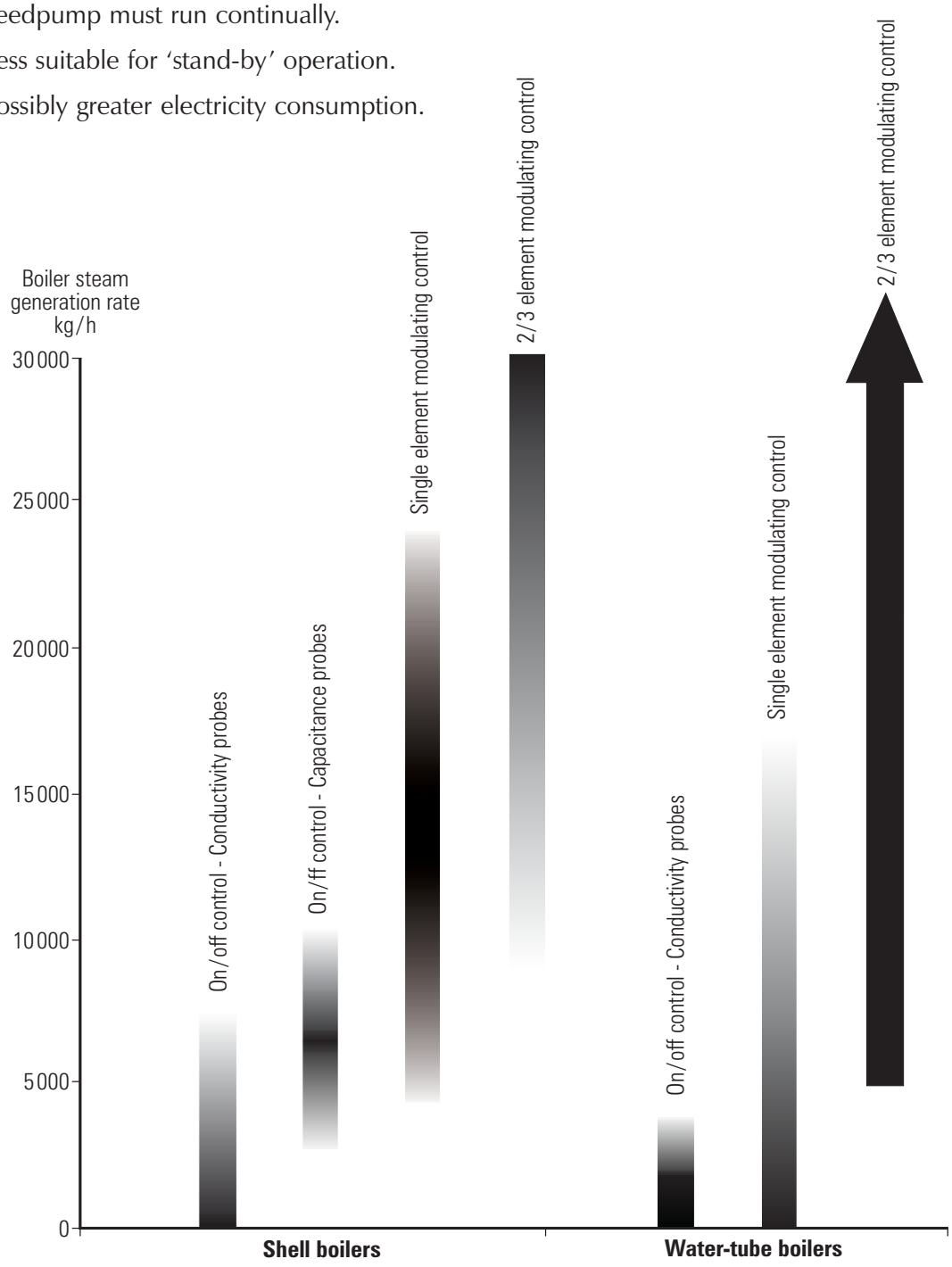


Fig. 3.17.9 The application of level controls

Questions

1. Which one of the following statements is true of on/off control of a steam boiler ?

a| Two capacitance level probes are required ☐

b| Boiler pressure is reduced when the pump operates ☐

c| An electrical frequency controller is required on the pump ☐

d| It is unable to operate effectively against varying boiler pressure conditions ☐

2. What is the purpose of the control valve after a feedpump with a modulating boiler level control system ?

a| For isolation purposes ☐

b| To maintain the pump outlet pressure ☐

c| To modulate the water flow ☐

d| To modulate the water pressure ☐

3. What is the purpose of the water recirculation line on the outlet of a pump on a modulating boiler control arrangement ?

a| To protect against the pump overheating ☐

b| To prevent pump cavitation ☐

c| To regulate water flow from the pump ☐

d| As an indication that the pump is delivering ☐

4. Which of the following is a disadvantage of a single element level control ?

a| It requires a pump with a variable speed drive for accurate level control ☐

b| A low steam demand can result in low water level lockout ☐

c| A high steam demand can result in high water level lockout ☐

d| The water level must fall for the feedwater valve to open ☐

5. Which of the following is an advantage of two element control over single element control ?

a| A variable speed drive pump is not required ☐

b| Steam demand has little effect on water level ☐

c| Only one conductivity level probe is required ☐

d| Steam flowrate can be adjusted in accordance with the prevailing water level ☐

6. A three element water level control:

a| Is controlled by the water level probe and a steam flowmeter ☐

b| Controls the number of pumps operating at any one time in a multi-boiler installation ☐

c| Makes it unnecessary to re-circulate water after the feedpump ☐

d| Caters for changes in feedwater pressure ☐

Answers

1: b, 2: c, 3: a, 4: d, 5: b, 6: d

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Module 3.18

Water Level Alarms

Water Level Alarms

Where boilers are operated without constant supervision (which includes the majority of industrial boilers) low water level alarms are required to shut down the boiler in the event of a lack of water in the boiler. Low level may be caused by:

- A feedwater shortage in the feedtank.
- Failure of a feedpump.
- Accidental isolation of the feedwater line.
- Failure of the level control system.

The regulations covering boilers have built up over the years in response to boiler explosions, damage and loss of life. Whilst boiler explosions are now very rare, damage to boilers which is attributable to low water level still occurs.

The effect of low water level in a boiler is that the heated tubes or the furnace tube(s) become uncovered and are no longer cooled by the boiler water. The metal temperature rapidly increases, its strength is reduced and collapse or rupture follows.

Low water alarm

The action of the low water level alarms under UK regulations is as follows:

- **1st low level alarm** - Shuts down the burner at the alarm level, but allows it to re-fire if the level recovers.
- **2nd low level alarm (often called lockout)** - Also shuts down the burner at the alarm level, but the burner controls remain 'locked out' even if the water level recovers and any faults have been rectified. The lockout has to be manually reset to allow the burner to re-fire.

The rules and regulations covering boiler operation, and the controls required, will vary from country to country, although demands for higher levels of safety, plus a desire to run steam boilers without the permanent presence of a boiler attendant, are tending to drive the regulations in the same direction.

The action of low water alarms outlined above, relates to the regulations governing unattended steam boiler plant in the UK. However, they are similar to the rules which are applied in many European countries and further afield.

High water alarm

With the exception of one or two operating standards, the risks from a water level too high are treated very lightly, if not ignored altogether.

The dangers of an excessively high water level in a steam boiler include:

- Increased carryover of water into the steam will result in poor operation and/or malfunction of the steam system components, due to dirt.
- Wet and dirty steam can contaminate or spoil the product where it is used directly. Wet steam can increase the water film thickness of the heat transfer surface, lower processing temperatures, perhaps interfering with proper sterilisation of food products or processing of pharmaceuticals, and causing wastage. At best, lower process and production efficiency will increase process time and unit costs.
- Overfilling the boiler can lead to waterhammer in the steam system, risking damage to plant and even injury to personnel.

All of these, taken together, can result in:

- Spoilt product.
- Lower production rates.
- Poor product quality.
- Increased plant and component maintenance.
- Damage to the steam system.
- Risk to personnel.

As can be seen, the dangers of an excessively high water level are too serious to ignore, and deserve equal consideration to that given to low water level conditions.

A high water condition could:

- Simply sound an alarm if the boiler house is manned.
- Shut-down the feedpump.
- Lockout the burner.
- Close the feedwater valve.

The action to be taken largely depends on the individual plant requirements.

Questions

1. Which of the following occurs when the 2nd low water level alarm sounds ?

a| The water feedpump is shut off ☐

b| The burner is shut off ☐

c| The water feedpump is started ☐

d| The burner is shut off and locked out ☐

2. After the 1st low water alarm has sounded, what action is required to reactivate the burner ?

a| The pump should be restarted ☐

b| The level probes should be recalibrated ☐

c| Nothing ☐

d| The reset button should be pressed ☐

3. With regard to a high water level condition which of the following statements is not true ?

a| The boiler might be flooded ☐

b| Water carryover into the steam space can occur ☐

c| The boiler will be damaged ☐

d| There is a risk to personnel ☐

4. What as a minimum should a high water level alarm do ?

a| Shut off the water feedpump ☐

b| Isolate the steam off-take ☐

c| Switch the burner to low fire ☐

d| Open the bottom blowdown valve ☐

Answers

1: d, 2: c, 3: c, 4: a

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Module 3.19

Installation of Level Controls

Installation of Level Controls

It has already been acknowledged that the water level in a steam boiler varies considerably as a result of:

- The load.
- The rate of load change.
- Water circulation within the boiler.

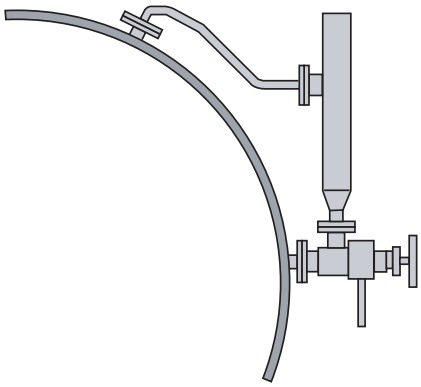
These circumstances combine to make it very difficult to monitor and control the boiler water level to any accuracy. What is required is a calm area of water which is representative of the actual boiler water level.

With float and probe type level controls, this is achieved in two ways:

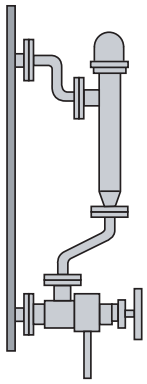
- External chambers.
- Internal protection tubes.

External chambers

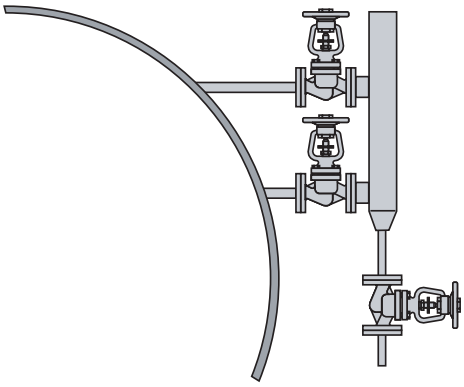
These are externally mounted chambers which have pipe connections to the boiler. They are usually, but not always, fitted with float controls. Some typical arrangements are shown in Figure 3.19.1.



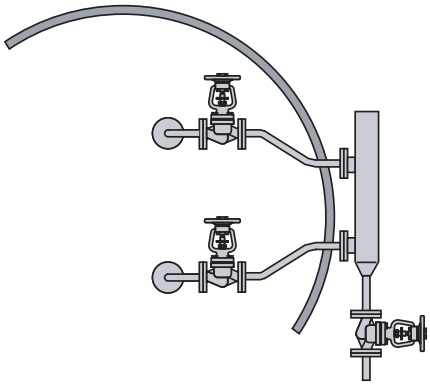
Side and bottom entry chamber with sequencing valve on a horizontal boiler



Side and bottom entry chamber with sequencing valve on a vertical boiler

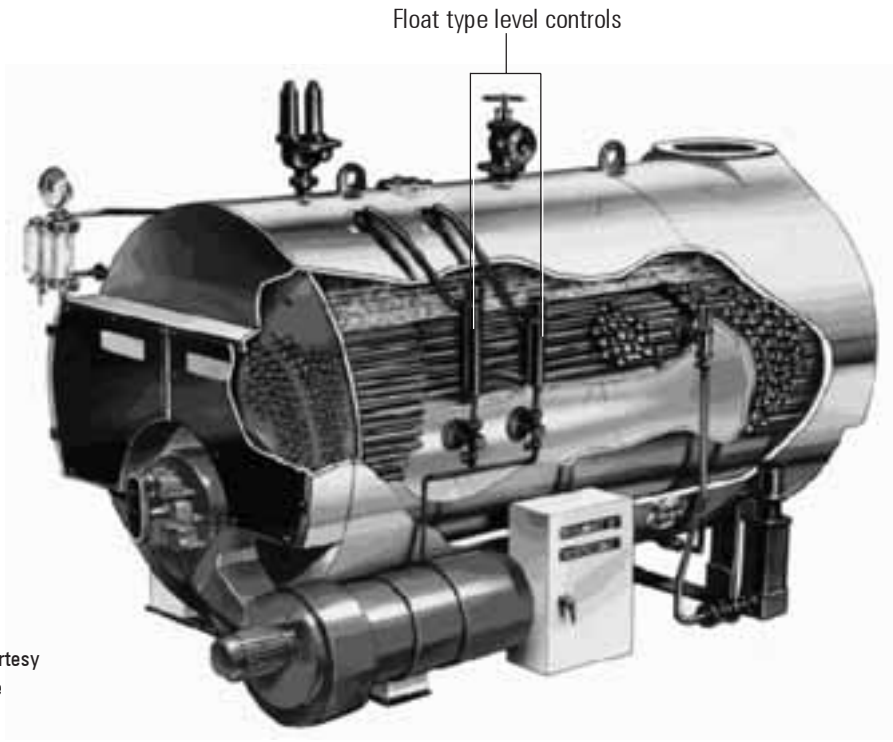


Side and side entry chamber on a horizontal boiler



Side and side entry chamber on the steam drum of a water tube boiler

Fig. 3.19.1 Alternative external chamber mounting methods for float or probe type level controls



Photograph courtesy of BIB Cochrane

Fig 3.19.2 External float level controls fitted in two independent chambers

Two external chambers are required:

- One chamber houses the level control plus the first low level alarm.
- The other houses the second low level alarm plus the high level alarm (if fitted).

This ensures that the two low alarms are in independent chambers.

The external chambers would be fitted with 'sequencing purge valves' and (optionally) with steam isolating valves.

Note: If isolating valves are fitted, UK regulations demand that they are locked open.

Traditionally float controls have been installed into external chambers, although probes work equally well, and have the advantage of no moving parts to wear out.

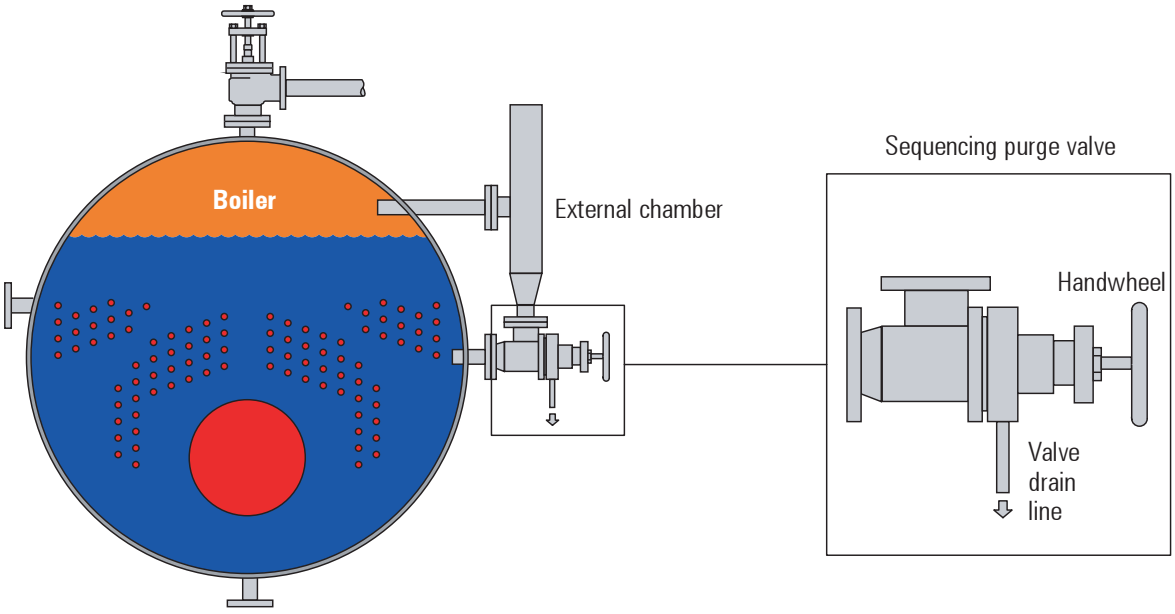


Fig. 3.19.3 Sequencing valve
(For the operation of sequencing valves see Figure 3.20.1)

Internal protection tubes (direct mounted level controls)

These are sometimes referred to as direct mounted level controls, and they require protection tubes to be installed inside the boiler shell as shown in Figure 3.19.4.

The first and second low level devices must be mounted in separate protection tubes, so that they are completely independent of each other.

The protection tubes themselves are not standard items, and will be uniquely manufactured for each individual boiler. However, because the design of the protection tubes can have such a major effect on the successful operation of the level controls, the following provide some guidance for their design and installation:

- **Diameter:**
  An 80 mm nominal bore protection tube will ensure steady conditions and provide sufficient clearance for probe centering.

  Where two probes (for example, level control / high alarm probe plus self-monitoring low alarm probe) are to be installed in a single protection tube, 100 mm nominal bore is usually required.

- **Length:**
  The protection tube should go as far down between the boiler tubes as physically possible

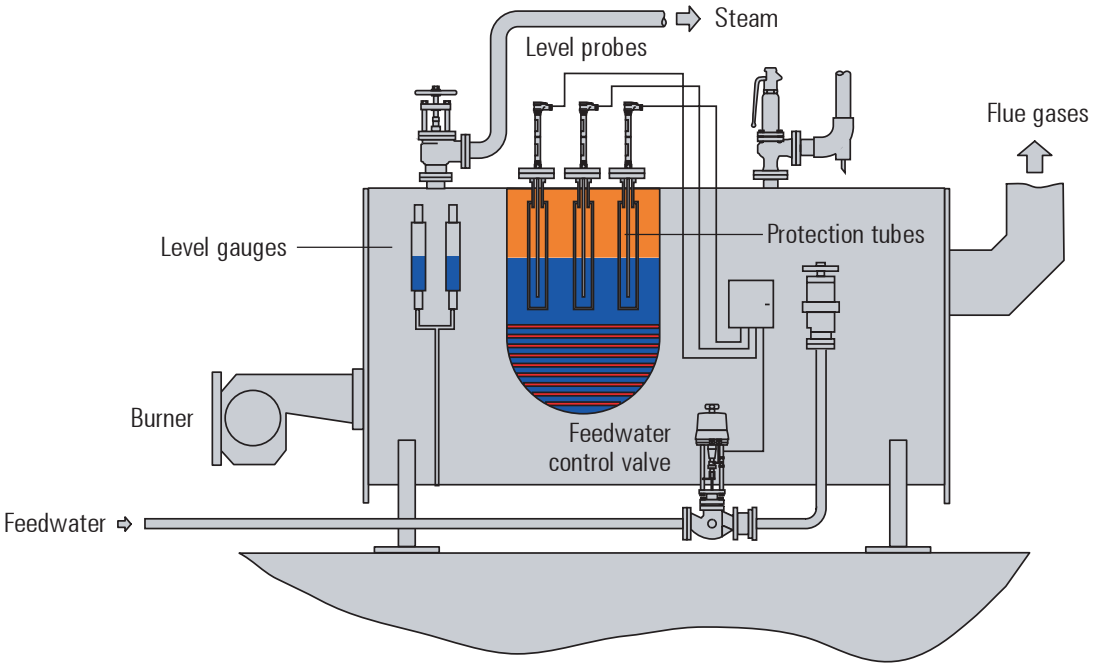


Fig. 3.19.4 Shell boiler with direct mounted level probes

Location

Where there is a choice of probe installation positions, the general recommendations are as follows:

- As far away as possible from the steam off-take and safety valve connection (minimum 1 m), but not too near the boiler end plates.
- As close to the level gauge as possible. Connections across the boiler shell, near the front are often convenient.
- Installation in protection tubes with top and bottom holes for steam and water entry, with a blanked bottom to prevent steam bubbles entering and without a full length slot along the protection tube.

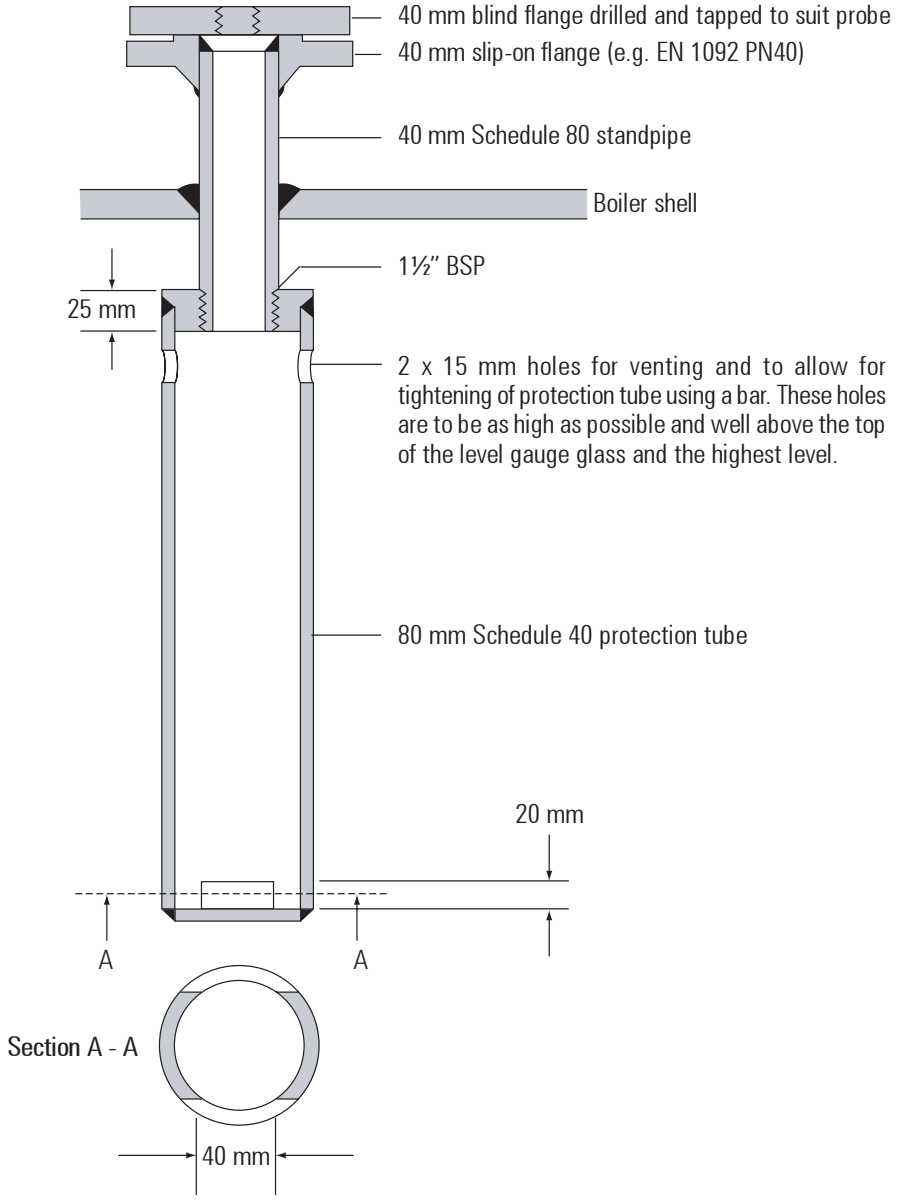


Fig. 3.19.5 Protection tubes

There are a number of significant advantages to using direct mounted controls in internal protection tubes:

- It is often a cheaper alternative with a new boiler as the cost of two or three protection tubes is usually less than two external control chambers and the associated sequencing purge valves.
- Full advantage can be taken of the advances in electronics provided by modern technology.

Float controls

Although the trend is towards using probe-type direct mounted controls, it is still common to see direct mounted float controls, where the float is situated inside the boiler shell using a flange and protection tube assembly.

Standard models

Direct mounted float controls employ the same principles of operation and piece parts as their chamber mounted equivalents, except that the chamber is exchanged for a large round flange and protection tube assembly for mounting the control directly onto the boiler shell connection. The protection tube may be fixed or removable, and will ensure that the float rod is not damaged and the correct vertical movement is achieved.

Direct mounted float controls incorporating test facilities

To comply with the UK HSE Guidance Note for unmanned boiler houses, direct mounted float controls may incorporate a facility for testing the operation of the mechanism without lowering the level of water in the boiler. Testing can be manual, or initiated/controlled by a timer. The test is achieved by lowering the float to the low water alarm level.

Hydraulic cup test facility

The test is achieved by lowering the float to the low water alarm level, by the following means: The float rod includes a cup above the float, which is fed for approximately 24 seconds with water from the boiler feedpump, via small bore pipework and valves, through the control mounting flange (see Figure 3.19.6).

The additional weight overcomes the buoyancy of the float, causing it to sink. This stops the burner from firing and operates the alarm system. After closing the test valve in the supply from the feedpump to the control, a small hole in the bottom of the cup drains off the water, permitting the float to rise to the normal operating position. Control of the water supply to the cup can also be achieved by means of a solenoid valve, which can be initiated by a timer or a manually operated push button.

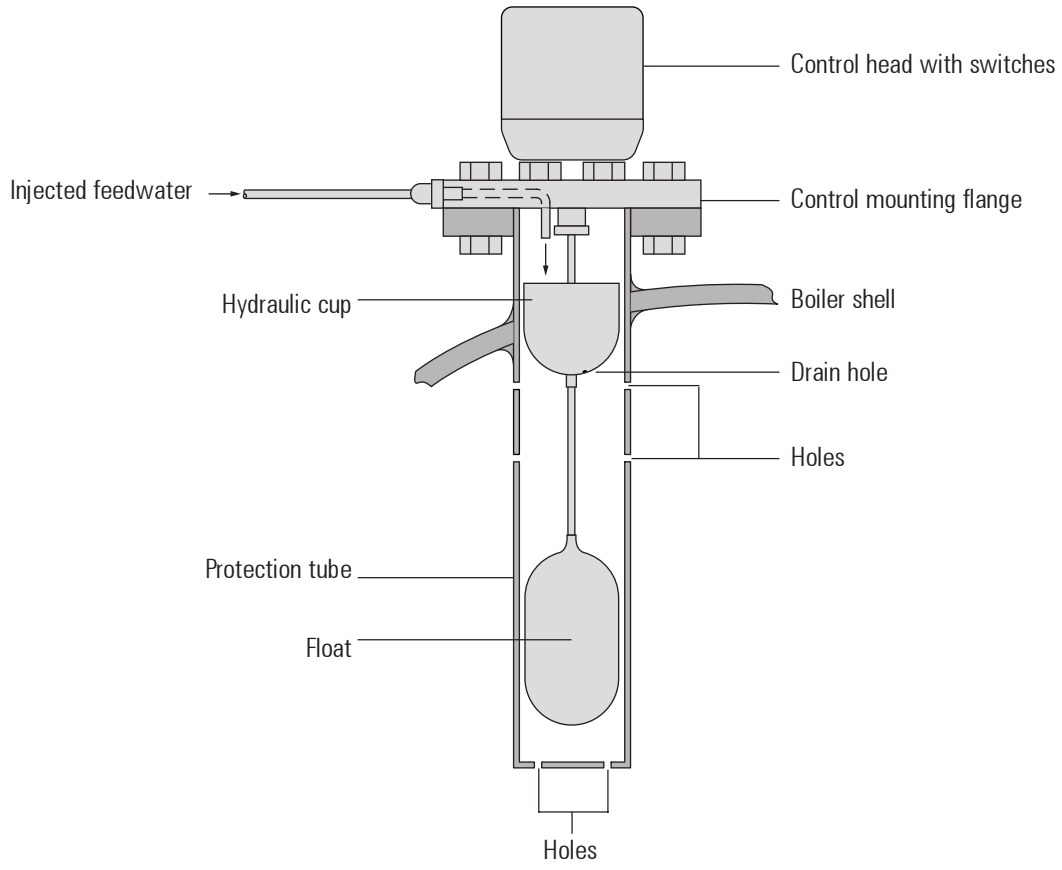


Fig. 3.19.6 Direct mounted float control with hydraulic cup

Electromagnetic test facility

The switch head includes a solenoid coil below the single switch sub-assembly. This surrounds an armature, which is located inside the stainless steel centre tube and fixed to the float rod.

To initiate the test cycle, the coil can be energised by a timer or a manually operated push button, and the float will be thrust downwards, to stop the burner firing and thus operate the alarm system. When the coil is de-energised the float rises to its normal level.

Probe controls

Single channel (non self-monitoring high integrity probes) may be installed in protection tubes, and, because they have no moving parts, they will often last longer than an equivalent float control system.

The use of internal protection tubes in conjunction with high integrity, self-monitoring probes and controllers, brings significant advantages in terms of testing requirements and the level of supervision demanded by authorities such as the UK Health and Safety Executive. This is discussed further in the next Module.

Questions

1. When external level sensing chambers are used on a boiler why are two fitted?

a| One is a stand-by and used to check against the other ☐

b| One to house 1st and 2nd low level alarm and one to house the level control ☐

c| One to house 1st low alarm and one to house 2nd low alarm ☐

d| To average out differences in the level of water sensed in the boiler ☐

2. On an external level sensing chamber what is the purpose of the two connections into the boiler?

a| In the event of one becoming blocked ☐

b| If there was only one connection the chamber could not fill properly and empty ☐

c| To enable water swell in the boiler to be sensed ☐

d| The top connection is used for sensing a high water level condition ☐

3. What is the purpose of the sequencing purge valve on an external level sensing chamber?

a| To check that the connections to the chamber are clear ☐

b| To take water samples ☐

c| For isolating the water connection to the chamber ☐

d| To check that the connections to the chamber are clear and to drain them down to check the alarms ☐

4. When using direct mounted level controls why are the 1st and 2nd low level devices mounted in separate protection tubes?

a| So that the alarms positions are independent of each other ☐

b| Because it is not physically practical to fit both alarms in one tube ☐

c| To average out the level sensed in the boiler ☐

d| To assist in maintaining each alarm separately ☐

5. As well as providing protection against physical damage of the level sensing devices what is the purpose of the protection tubes?

a| To give more protection against corrosion of the level sensors ☐

b| To afford some protection of the sensors against heat from the fire tubes ☐

c| As pockets so that the level controls can be withdrawn without shutting down the boiler ☐

d| To allow the level controls to sense more steady water conditions than that surrounding them ☐

6. How are direct mounted float controls tested?

a| By blowing down the water in the boiler ☐

b| By lowering the float ☐

c| By an electromagnetic test ☐

d| All of the above ☐

Answers

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Module 3.20

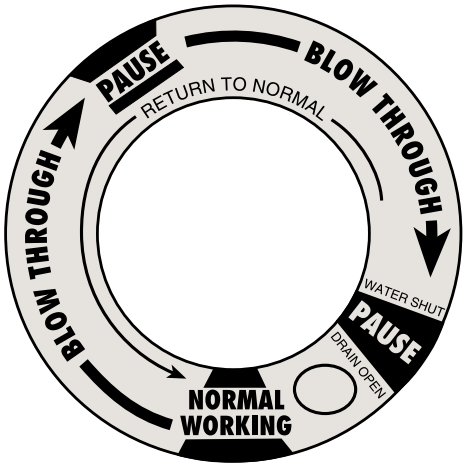
Testing Requirements in the Boiler House

Testing Requirements in the Boiler House

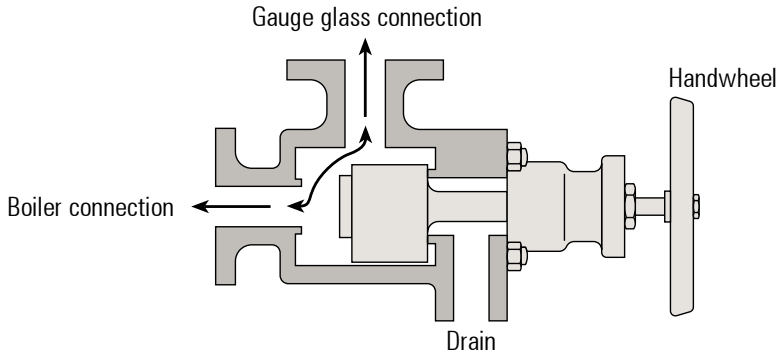
The following test routines are required by the UK HSE (Health and Safety Executive) for a manned boiler house.

External chambers (float or probe type controls)

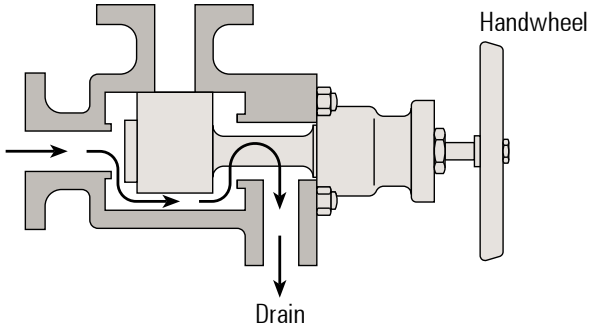
- **Daily:**
  1. Blow through of the chambers is required, using the sequencing purge valves to remove any accumulated sludge.
  2. Separately, the first and second low alarms are tested.
- **Weekly:**
  1. Lower the actual boiler water level to the 1st low (by evaporation), and then blow down to the 2nd low.
     The main reason for this weekly test is to ensure that the alarm is given, and at the correct level, when the level drops slowly in the boiler (because floats could stick).
  2. A high alarm is usually tested weekly.



Position 1 - Normal working



Position 2 - Blow through water



Position 3 - Blow through chamber

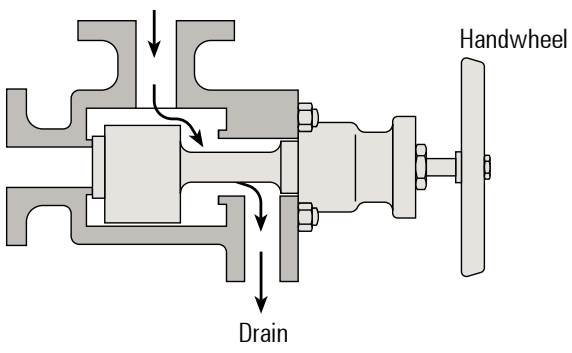


Fig. 3.20.1 Operation of sequencing valves

Direct mounted level controls with internal protection tubes

A daily test is still required, but this means dropping the actual level, unless test facilities are incorporated. The time involved and the loss of heat, water and treatment chemicals means that this is only really practical in smaller boilers.

The UK regulations for supervision state that, for 'standard' (for example, non-self-monitoring, high integrity) controls there must be a trained boiler attendant on site at all times that the boiler is operating.

Testing requirements in the unmanned boiler house

In many countries and in all types of industries, there is a need or desire to run steam boiler plant unattended. This has led to the development of special, high integrity 'self-monitoring' level alarms, and controls for increased safety in the event of low water conditions.

For externally mounted float controls, automatic sequencing valves are required, plus a control system which will then carry out automatic sequenced blowdown of the external chambers and electrical testing of the externally mounted boiler level controls (Figure 3.20.2).

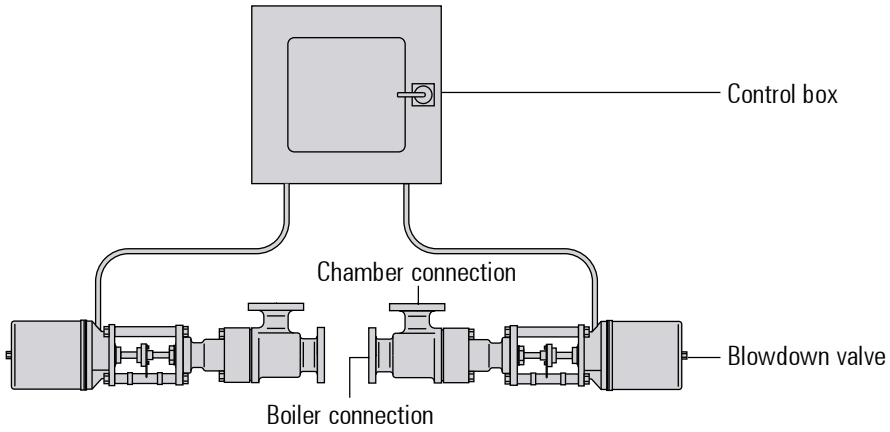


Fig. 3.20.2 Automatic sequencing valves and control systems for externally mounted float type level controls

Direct mounted float type level controls must be fitted with a test device, plus a control system which will then automatically and electrically test the direct mounted level controls (Figure 3.20.3).

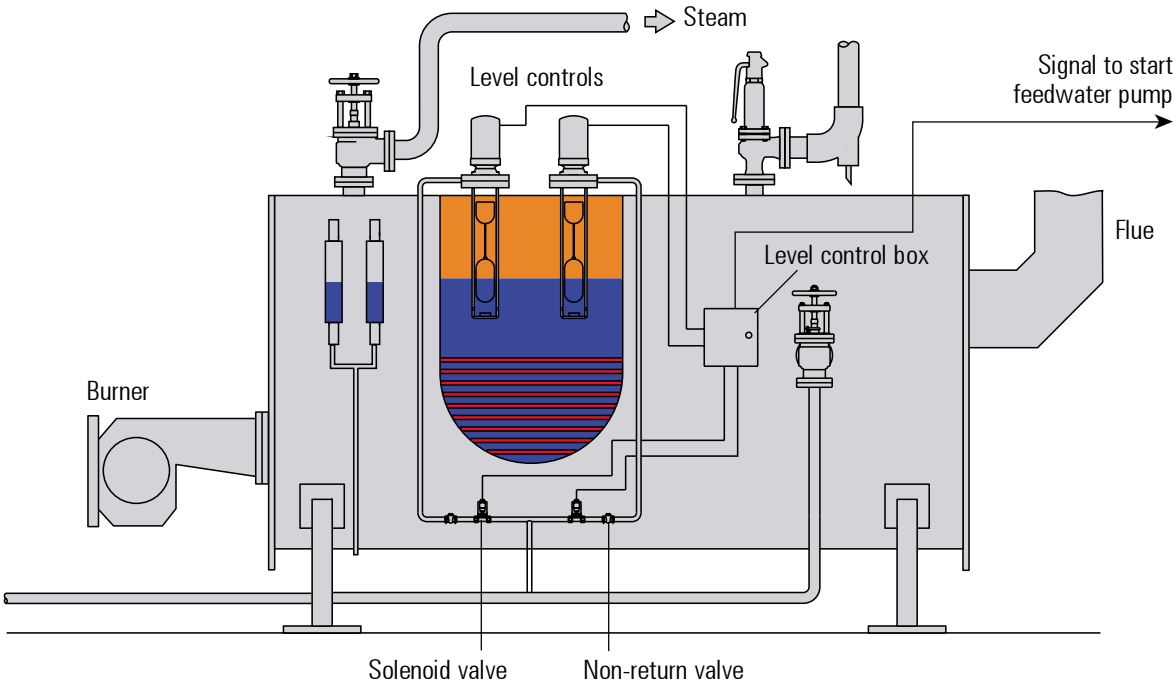


Fig. 3.20.3 Direct mounted float controls in a shell boiler

Automatic test system for direct mounted float type level controls

With probe type, high integrity, self-monitoring level controls, the 'self-checking' facility is carried out via the probe and its associated controller, so a further, special control system is not required.

The latest conductivity systems which incorporate a high integrity self-monitoring feature, will check for faults continuously, and electronically.

Faults can include the build-up of scale or dirt on the probe and also any moisture leakage into the probe. If such a fault is detected, the control system will initiate an alarm and cause the boiler to safely shut down.

The main user advantage of these special low water level alarms is not only increased safety but also that daily testing is not necessary. This means that there is little point in fitting high integrity probe controls in external chambers, where it would still be necessary to blow through the chambers, on a daily basis, to remove any sludge.

Probe type, high integrity, self-monitoring low water level alarms are therefore fitted in internal protection tubes.

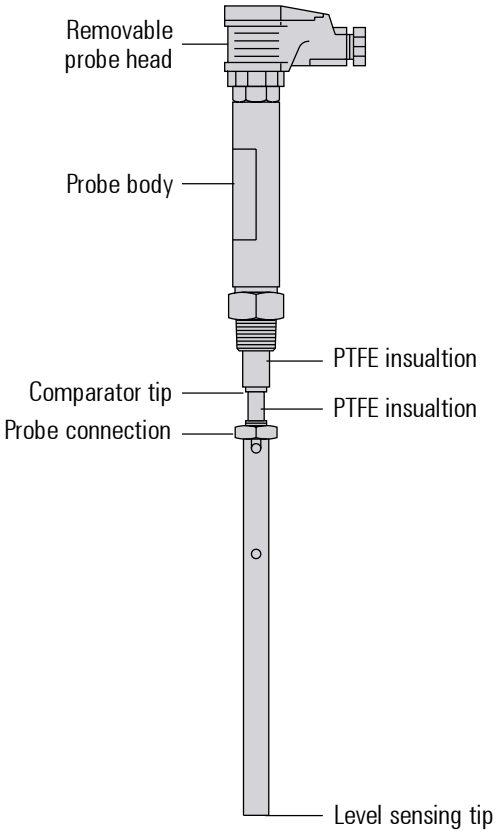


Fig. 3.20.4 Typical high integrity self-monitoring conductivity probe

The manual weekly test must still be carried out under UK regulations. In Germany, where approved probe-type high integrity self-monitoring low water alarms are fitted, the interval between manual tests is 6 months.

Under the UK regulations, if high integrity self-monitoring systems are fitted, supervision requirements are reduced to the need to have someone available to respond to any alarm and call for further assistance. An adequately trained security guard or porter could be considered suitable.

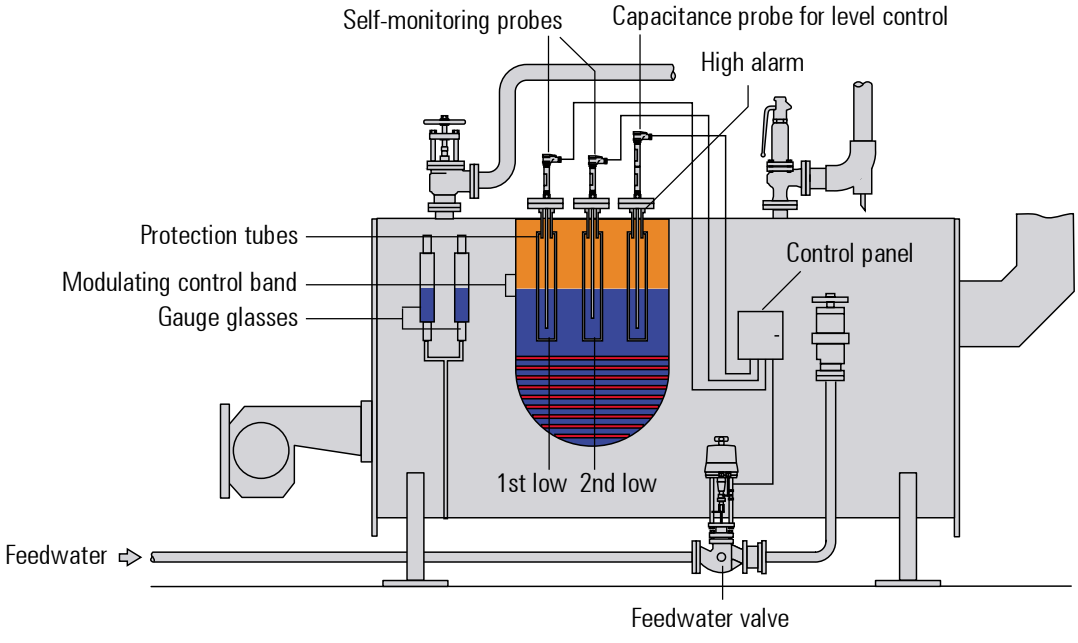


Fig. 3.20.5 High integrity, self-monitoring, modulating control system

Summary

When the low water level alarm systems are housed in external chambers they will require manually blowing down and testing, and this must be carried out at least once per day. In these cases a trained boiler attendant must be on site whenever the boiler is operating including during 'silent hours' (nights and weekends).

The trained boiler attendant need not be permanently situated in the boiler house but must be able to respond immediately to the level alarms.

When high integrity self-monitoring low level alarms are mounted in the boiler shell, since they are automatically self-testing, they only require a full operational test by a trained boiler attendant once per week.

When standard low level alarms (floats or probes) are fitted in external chambers, automatic sequencing valves have to be fitted in order for the alarm system to be deemed self-monitoring. A trained boiler attendant need not be on site at all times and another person (watchman or porter) can be put in charge of the boiler instead, as part of his duties during the silent hours.

This person should always be ready to respond correctly to the boiler alarms, shutting down the boiler if necessary. Thus, depending on the type of installation there are two possible types of supervision: A trained boiler attendant (or technician), who must be fully conversant with the operation of the boiler and its controls; or an individual such as a watchman who, although not a fully trained boiler attendant, must be familiar with the alarm protocol and know the procedure for shutting down the boiler.

Table 3.20.1 Testing required by UK HSE (Health and Safety Executive)

	Standard controls	High integrity, self-monitoring controls
In external chambers	Daily test (plus true test weekly)	-
In shell	Daily (true) test	Weekly (true) test

Testing steam boiler control systems

Any boiler regulations will emphasise that regular testing of any boiler control system, particularly with respect to the water level, is an important requirement. All testing should be carried out with the water in the visible region of the water level gauge.

All testing should be carried out by a trained boiler attendant. In the case of level devices mounted in chambers with manual sequencing valves, testing involves operating the sequencing valves at least once per day to lower the water in each chamber and to test the operation of the water level control, and the controls / alarms at first and second low levels. Similarly for traditional (non-self-monitoring) low water level alarms mounted directly in the boiler, the trained boiler attendant must lower the actual boiler water level every day in order to test these alarms.

However, for high integrity self-monitoring controls mounted directly in the boiler, there is no need for daily testing.

For all types of level control system there is a weekly test to be carried out, and this involves isolating the feedwater supply, lowering the water by evaporation to first low level and blowing down to second low level. This weekly test is a full functional test of the system's ability to cope with actual boiler water level change.

It is recommended that all tests be properly logged in a boiler house log book, for which the Engineering Manager is responsible.

Footnote: These basic notes are based on UK boiler house practice, rules, and regulations. These regulations vary around the world, some examples follow:

Australia	Dynamic (float type) low water alarm systems are currently called for by boiler regulations.
Canada	A boiler plant engineer must always be present during boiler operation.
Italy	Boiler regulations state that the second low water alarm has to be of a mechanical type.

Questions

1. Which weekly tests of level controls should be conducted on a boiler with external level chambers?

a| Evaporate to 1st low and then blowdown to 2nd low ☐
b| Evaporate to 2nd low ☐
c| Blowdown to 1st and 2nd low ☐
d| Weekly tests are not necessary ☐

2. By which means are level controls in external level chambers tested?

a| Hydraulic test ☐
b| Electromagnetic test ☐
c| Draining the chambers through an isolation valve ☐
d| By a sequencing purge valve ☐

3. Which tests are required on high integrity self-monitoring direct mounted level controls?

a| None ☐
b| Daily functional test by evaporating to 1st low and then blowing down to 2nd low ☐
c| Weekly conductivity functional test ☐
d| Weekly functional test by evaporating to 1st low and then blowing down to 2nd low ☐

4. Why would high integrity self-monitoring low-level controls not normally be used in external level chambers?

a| They are only suitable for direct mounting in a boiler ☐
b| Three chambers would be required, one for 1st alarm, one for 2nd alarm and one for the level control ☐
c| There would be no advantage because the chambers would still need to be blown down daily ☐
d| Cost; automatic sequencing purge valves would have to be fitted ☐

5. Which of the following is a possible financial disadvantage of level controls fitted in external chambers?

a| The sensed level is never the same as the level in the boiler ☐
b| Water in the boiler must be lowered each day ☐
c| The alarms must be tested weekly by evaporating to 1st low and then blowing down to 2nd low ☐
d| A trained boiler attendant must be on site whenever the boiler is in operation ☐

6. If boiler plant fitted with float level controls in external chambers is to be left unattended –

a| The plant cannot be left unattended unless self-monitoring probe controls are fitted ☐
b| There will be no need for daily lowering of the water in each chamber ☐
c| Automatic sequenced blowdown of the chambers and electrical tests facility must be fitted ☐
d| In addition to 'c', a high integrity self-monitoring probe must be fitted in the boiler ☐

Answers

1: a, 2: d, 3: d, 4: c, 5: d, 6: d

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Module 3.21

Pressurised Deaerators

Pressurised Deaerators

Why gases need to be removed from boiler feedwater

Oxygen is the main cause of corrosion in hotwell tanks, feedlines, feedpumps and boilers. If carbon dioxide is also present then the pH will be low, the water will tend to be acidic, and the rate of corrosion will be increased. Typically the corrosion is of the pitting type where, although the metal loss may not be great, deep penetration and perforation can occur in a short period.

Elimination of the dissolved oxygen may be achieved by chemical or physical methods, but more usually by a combination of both.

The essential requirements to reduce corrosion are to maintain the feedwater at a pH of not less than 8.5 to 9, the lowest level at which carbon dioxide is absent, and to remove all traces of oxygen. The return of condensate from the plant will have a significant impact on boiler feedwater treatment - condensate is hot and already chemically treated, consequently as more condensate is returned, less feedwater treatment is required.

Water exposed to air can become saturated with oxygen, and the concentration will vary with temperature: the higher the temperature, the lower the oxygen content.

The first step in feedwater treatment is to heat the water to drive off the oxygen. Typically a boiler feedtank should be operated at 85°C to 90°C. This leaves an oxygen content of around 2 mg/litre (ppm). Operation at higher temperatures than this at atmospheric pressure can be difficult due to the close proximity of saturation temperature and the probability of cavitation in the feedpump, unless the feedtank is installed at a very high level above the boiler feedpump.

The addition of an oxygen scavenging chemical (sodium sulphite, hydrazine or tannin) will remove the remaining oxygen and prevent corrosion.

This is the normal treatment for industrial boiler plant in the UK. However, plants exist which, due to their size, special application or local standards, will need to either reduce or increase the amount of chemicals used. For plants that need to reduce the amount of chemical treatment, it is common practice to use a pressurised deaerator.

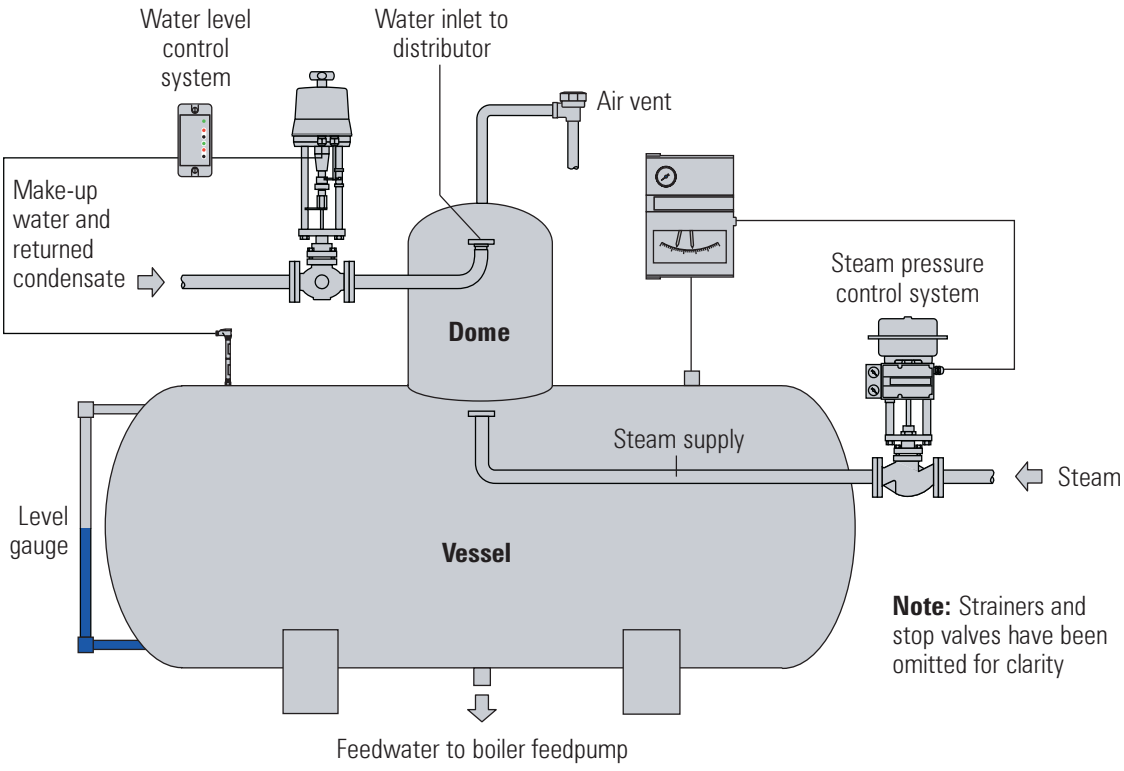


Fig. 3.21.1 General arrangement of a pressure deaerator

Operating principles of a pressurised deaerator

If a liquid is at its saturation temperature, the solubility of a gas in it is zero, although the liquid must be strongly agitated or boiled to ensure it is completely deaerated.

This is achieved in the head section of a deaerator by breaking the water into as many small drops as possible, and surrounding these drops with an atmosphere of steam. This gives a high surface area to mass ratio and allows rapid heat transfer from the steam to the water, which quickly attains steam saturation temperature. This releases the dissolved gases, which are then carried with the excess steam to be vented to atmosphere. (This mixture of gases and steam is at a lower than saturation temperature and the vent will operate thermostatically). The deaerated water then falls to the storage section of the vessel.

A blanket of steam is maintained above the stored water to ensure that gases are not re-absorbed.

Water distribution

The incoming water must be broken down into small drops to maximise the water surface area to mass ratio. This is essential to raising the water temperature, and releasing the gases during the very short residence period in the deaerator dome (or head).

Breaking the water up into small drops can be achieved using one of the methods employed inside the dome's steam environment.

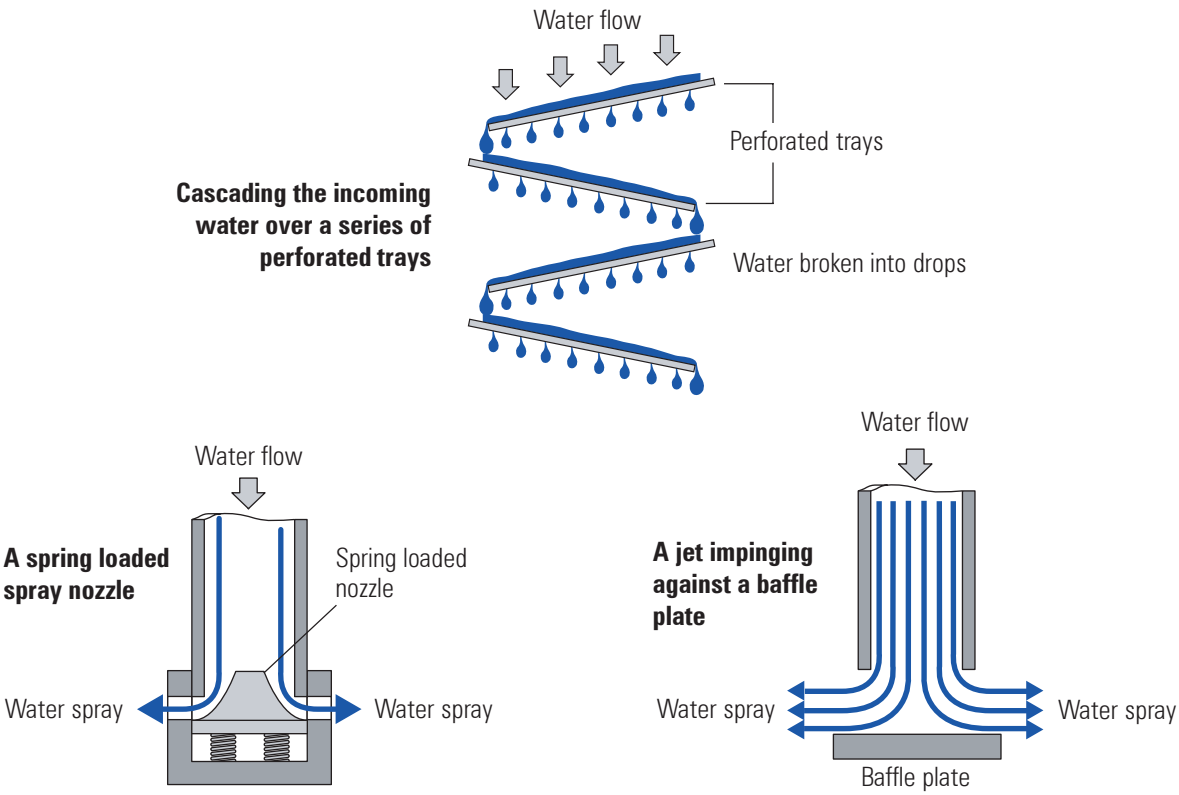


Fig. 3.21.2 Deaerator water inlet options

There are of course advantages and disadvantages associated with each type of water distribution, plus cost implications. Table 3.21.1 compares and summarises some of the most important factors:

Table 3.21.1 Comparison of tray and spray type deaerators

	Tray type	Spray type
Life expectancy (years)	40	20
Turndown (maximum/minimum)	Very high	5
Cost factor	1	0.75
Typical application	Power plant	Process plant

Control systems

Water control

A modulating control valve is used to maintain the water level in the storage section of the vessel. Modulating control is required to give stable operating conditions, as the sudden inrush of relatively cool water with an on/off control water control system could have a profound impact on the pressure control, also the ability of the deaerator to respond quickly to changes in demand.

Since modulating control is required, a capacitance type level probe can provide the required analogue signal of water level.

Steam control

A modulating control valve regulates the steam supply. This valve is modulated via a pressure controller to maintain a pressure within the vessel. Accurate pressure control is very important since it is the basis for the temperature control in the deaerator, therefore a fast acting, pneumatically actuated control valve will be used. Note: A pilot operated pressure control valve may be used on smaller applications, and a self-acting diaphragm actuated control valve may be used when the load is guaranteed to be fairly constant.

The steam injection may occur at the base of the head, and flow in the opposite direction to the water (counter flow), or from the sides, crossing the water flow (cross flow). Whichever direction the steam comes from, the objective is to provide maximum agitation and contact between the steam and water flows to raise the water to the required temperature.

The steam is injected via a diffuser to provide good distribution of steam within the deaerator dome.

The incoming steam also provides:

- A means of transporting the gases to the air vent.
- A blanket of steam required above the stored deaerated water.

Deaerator air venting capacity

In previous Modules, typical feedwater temperatures have been quoted at around 85°C, which is a practical maximum value for a vented boiler feedtank operating at atmospheric pressure. It is also known that water at 85°C contains around 3.5 grams of oxygen per 1 000 kg of water, and that it is the oxygen that causes the major damage in steam systems for two main reasons. First, it attaches itself to the inside of pipes and apparatus, forming oxides, rust, and scale; second, it combines with carbon dioxide to produce carbonic acid, which has a natural affinity to generally corrode metal and dissolve iron. Because of this, it is useful to remove oxygen from boiler feedwater before it enters the boiler. Low-pressure and medium-pressure plant supplied with saturated steam from a shell type boiler will operate quite happily with a carefully designed feedtank incorporating an atmospheric deaerator (referred to as a semi-deaerator). Any remaining traces of oxygen are removed by chemical means, and this is usually economic for this type of steam plant. However, for high-pressure water-tube boilers and steam plant handling superheated steam, it is vital that the oxygen level in the boiler water is kept much lower (typically less than seven parts per billion - 7 ppb), because the rate of attack due to dissolved gases increases rapidly with higher temperatures. To achieve such low oxygen levels, pressurised deaerators can be used.

If feedwater were heated to the saturation temperature of 100°C in an atmospheric feedtank, the amount of oxygen held in the water would theoretically be zero; although in practice, it is likely that small amounts of oxygen will remain. It is also the case that the loss of steam from a vented feedtank would be quite high and economically unacceptable, and this is the main reason why pressurised deaerators are preferred for higher pressure plant operating typically above 20 bar g.

A pressurised deaerator is often designed to operate at 0.2 bar g, equivalent to a saturation temperature of 105°C, and, although a certain amount of steam will still be lost to atmosphere via a throttled vent, the loss will be far less than that from a vented feedtank.

It is not just oxygen that needs to be vented; other non-condensable gases will be rejected at the same time. The deaerator will therefore vent other constituents of air, predominantly nitrogen, along with a certain amount of steam. It therefore follows that the rejection rate of air from the water has to be somewhat higher than 3.5 grams of oxygen per 1 000 kg of water. In fact, the amount air in water at 80°C under atmospheric conditions is 5.9 grams per 1 000 kg of water. Therefore, a rejection of 5.9 grams of air per 1 000 kg of water is needed to ensure that the required amount of 3.5 grams of oxygen is being released. As this air mixes with the steam in the space above the water surface, the only way it can be rejected from the deaerator is by the simultaneous release of steam.

The amount of steam/air mixture that needs to be released can be estimated by considering the effects of Dalton's Law of partial pressures and Henry's Law.

Consider the feasibility of installing a deaerator. Prior to installation, the boiler plant is fed by feedwater from a vented feedtank operating at 80°C. This essentially means that each 1 000 kg of feedwater contains 5.9 gram of air. The proposed deaerator will operate at a pressure of 0.2 bar g, which corresponds to a saturation temperature of 105°C. Assume, therefore, that all the air will be driven from the water in the deaerator. It follows that the vent must reject 5.9 gram of air per 1 000 kg of feedwater capacity.

Consider that the air being released from the water mixes with the steam above the water surface. Although the deaerator operating pressure is 0.2 bar g (1.2 bar a), the temperature of the steam/air mixture might only be 100°C.

$$\text{Total pressure in the deaerator} = 1.2 \text{ bar a}$$

$$\text{Temperature of the vapour in the deaerator} = 100°C$$

$$\text{100°C corresponds to a saturation pressure of 1 atm} = 1.013\,25 \text{ bar a}$$

Therefore, from Dalton's Law:-

If the vapour space in the deaerator were filled with pure steam, the vapour pressure would be 1.2 bar a. As the vapour space has an actual temperature of 100°C, the partial pressure caused by the steam is only 1.013 25 bar a.

The partial pressure caused by the non-condensable gases (air) is therefore the difference between these two figures = 1.2 – 1.013 25 = 0.186 75 bar a.

$$\text{The proportion by volume of air to steam in the mixture} = \frac{0.186\,75}{1.013\,25}$$

$$= 18.43\%$$

$$\text{Therefore every litre of released air is accompanied by :-} \frac{100 - 18.43}{18.43} \text{ litres of steam}$$

$$= 4.42 \text{ litres of steam}$$

The density of air at 100°C is approximately 0.946 grams/L

The density of steam at 100°C is approximately 0.6 grams/L

$$\text{Therefore, 0.946 g of air is released with } 0.6 \times 4.42 = 2.65 \text{ g of steam}$$

$$\text{and, 5.9 g of air is released with: } \frac{2.65 \times 5.9}{0.946} \approx 16.5 \text{ g of steam}$$

Therefore, **the total mixture of air and steam released per 5.9 g of oxygen** can be calculated:

$$5.9 \text{ g} + 16.5 \text{ g} = \textbf{22.4 gram of air/steam mixture}$$

However:

- Because there is no easy way to accurately measure the discharge temperature;
- Because there is only a small pressure differential between the deaerator and atmospheric pressure;
- Because the vent rates are so small,

. . . an automatic venting mechanism is rarely encountered on deaerator vent pipes, the task usually being accomplished by a manually adjusted ball valve, needle valve, or orifice plate.

It is also important to remember that the prime objective of the deaerator is to remove gases. It is vital, therefore, that once separated out, these gases are purged as quickly as possible, and before there is any chance of re-entrainment.

Based on practical experience, deaerator manufacturers will aim to vent 22.4 kg/h of steam/air mixture per 1 000 kg/h of deaerator capacity.

A typical way of controlling the vent rate is to use a DN20 steam duty ball valve of a suitable pressure rating, which can be secured in a part-open condition.

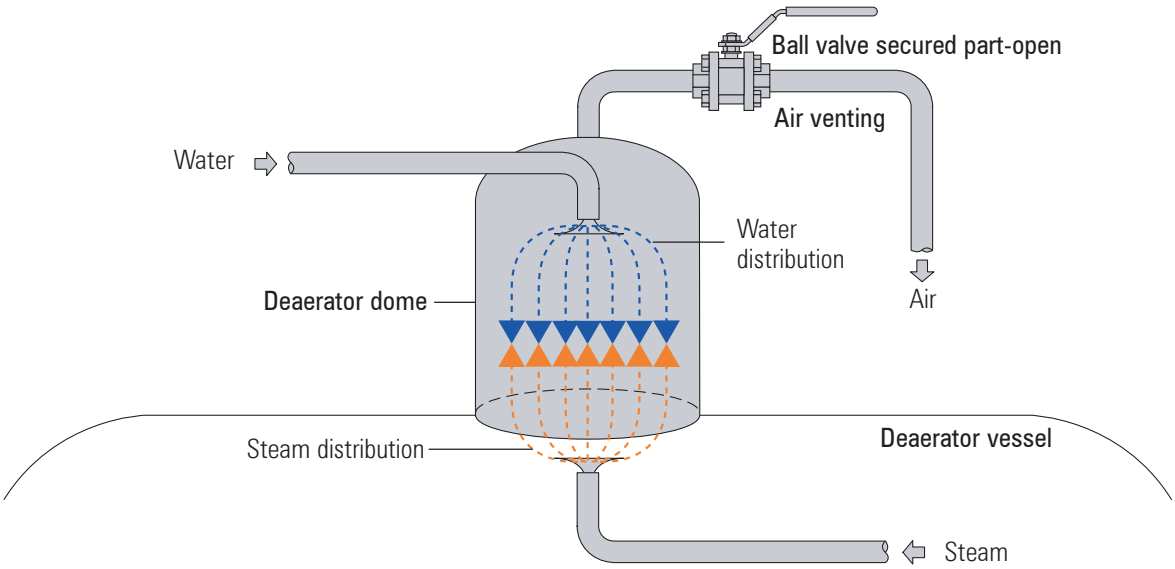


Fig. 3.21.3 Inside a deaerator dome

Typical operating parameters for a pressurised deaerator

The following information is typical and any actual installation may vary from the following in a number of ways to suit the individual requirements of that plant:

- The operating pressure will usually be approximately 0.2 bar (3 psi), which gives a saturation temperature of 105°C (221°F).
- The vessel will contain between 10 and 20 minutes water storage for the boiler on full-load.
- The water supply pressure to the deaerator should be at least 2 bar to ensure good distribution at the nozzle.

This implies either a backpressure on the steam traps in the plant or the need for pumped condensate return.

- Steam supply pressure to the pressure control valve will be in the range 5 to 10 bar.
- Maximum turndown on the deaerator will be approximately 5:1.
- At flowrates below this from the process, there may be insufficient pressure to give good atomisation with nozzle or spray type water distributors.
- This can be overcome by having more than one dome on the unit. The total capacity of the domes would be equal to the boiler rating, but one or more of the domes may be shut down at times of low demand.
- Heating may be required in the storage area of the vessel for start-up conditions; this may be by coil or direct injection
- However, the type of plant most likely to be fitted with a pressurised deaerator will be in continuous operation and the operator may consider the low performance during the occasional cold start to be acceptable.

The vessel design, materials, manufacture, construction, and certification will be in compliance with a recognised standard, for example: in the UK the standard is PD 5500.

The heat balance on the deaerator will typically (but not always) have been calculated on a 20°C increase in the incoming water temperature.

It is normal for water at 85°C to be supplied to the deaerator. If the incoming water temperature is significantly higher than this, then the amount of steam required to achieve the set pressure will be less. This, in turn, means that the steam valve will throttle down and the steam flowrate may be too low to ensure proper dispersal at the steam nozzle.

This may suggest that, with a very high percentage of condensate being returned, some alternative action may be required for proper deaeration to occur.

In this instance, the deaerator heat balance may be calculated using different parameters, or the deaerator may operate at a higher pressure.

Cost and justification

Cost

There is no additional energy cost associated with operating a deaerator, and the maximum amount of steam exported to the plant is the same with, or without the deaerator, because the steam used to increase the feedwater temperature comes from the higher boiler output

However:

- There will be some heat loss from the deaerator (This will be minimised by proper insulation).
- There is the additional cost of running the transfer pump between the feedtank and the deaerator.
- Some steam is lost with the vented non-condensable gases.

Justification

The principle reasons for selecting a pressurised deaerator are:

- To reduce oxygen levels to a minimum (< 20 parts per billion) without the use of chemicals. This will eliminate corrosion in the boiler feed system.
- A cost saving can be achieved with respect to chemicals - this argument becomes increasingly valid on large water-tube type boilers where flowrates are high, and low TDS levels (< 1 000 ppm) need to be maintained in the boiler feedwater.
- Chemicals added to control the oxygen content of the boiler water will themselves require blowing down. Therefore by reducing/eliminating the addition of chemicals, the blowdown rate will be reduced with associated cost savings.
- To prevent contamination where the steam is in direct contact with the product, for example: foodstuffs or for sterilisation purposes.

Deaerator heat balance

To enable correct system design and to size the steam supply valve, it is important to know how much steam is needed to heat the deaerator. This steam is used to heat the feedwater from the usual temperature experienced prior to the installation of the deaerator to the temperature needed to reduce the dissolved oxygen to the required level.

The required steam flowrate is calculated by means of a mass/heat balance. The mass/heat balance works on the principle that the initial amount of heat in the feedwater, plus the heat added by the mass of injected steam must equal the final amount of heat in the feedwater plus the mass of steam that has condensed during the process.

Equation 2.11.3 is the mass/heat balance equation used for this purpose.

$$\dot{m}\, h_1 + \dot{m}_s\, h_g = (\dot{m} + \dot{m}_s)\, h_2 \qquad \textbf{Equation 2.11.3}$$

Where:

$\dot{m}$ = Maximum boiler output at the initial feedwater temperature (kg/h) - This is the boiler 'From and At' figure x the boiler evaporation factor.

$\dot{m}_s$ = Mass of steam to be injected (kg/h)

h_1 = Enthalpy of water at the initial temperature (kJ/kg)

h_2 = Enthalpy of water at the required temperature (kJ/kg)

h_g = Enthalpy of steam supplying the control valve (kJ/kg) - Note: if the supply steam is superheated, this value is the total heat in the superheated steam (h).

To calculate the required steam flowrate, Equation 2.11.4 is transposed to solve for $\dot{m}_s$, and becomes Equation 3.21.1.

$$\dot{m}_s = \dot{m}\, \frac{(h_2 - h_1)}{(h_g - h_2)} \qquad \textbf{Equation 3.21.1}$$

Example 3.21.1 Determine the amount of steam needed to heat a deaerator

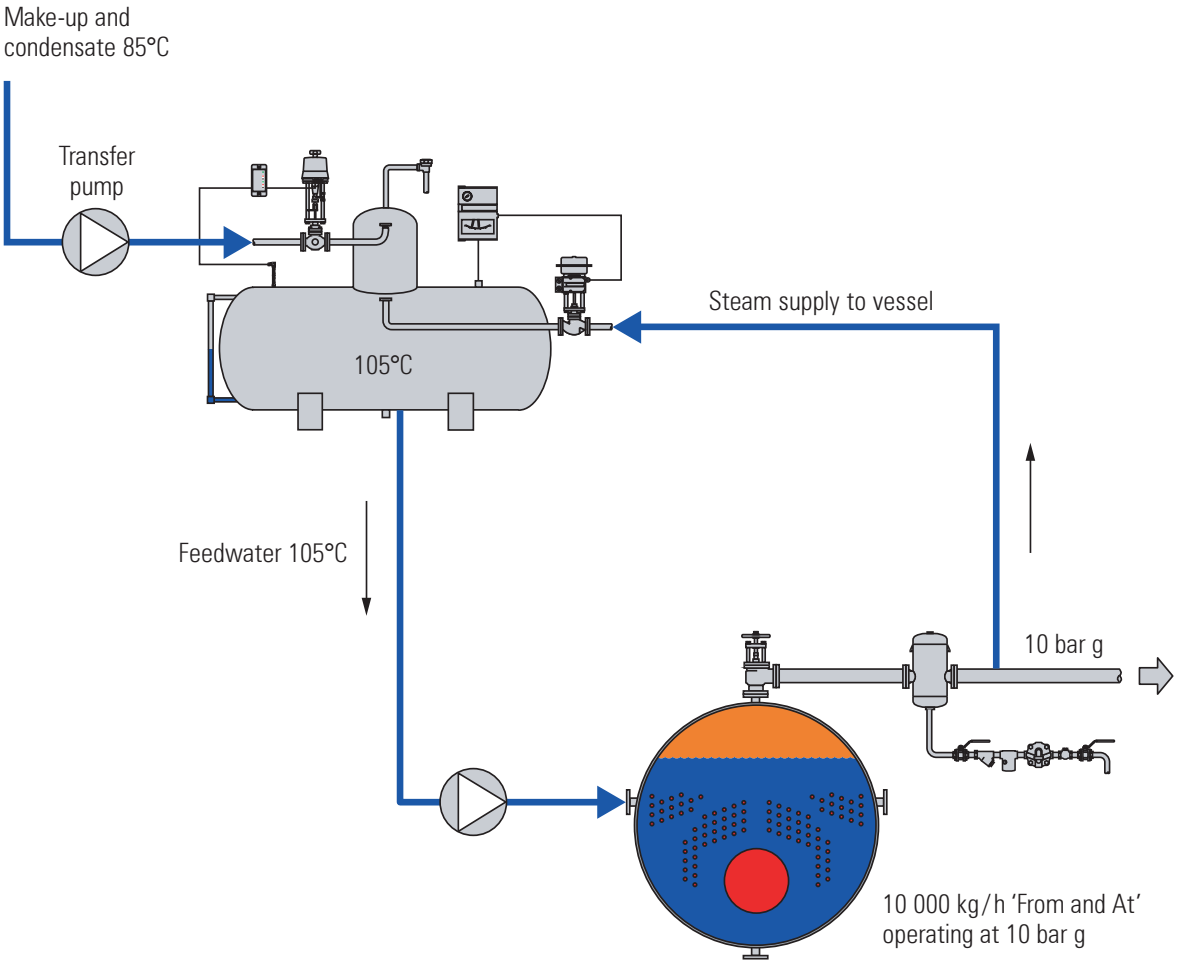


Fig. 3.21.4 Typical pressurised deaerator installation

An existing boiler plant is fed with feedwater at a temperature of 85°C. Due to the rising cost of chemical treatment, it is proposed that a pressurised deaerator be installed, operating at 0.2 bar g to raise the feedwater temperature to 105°C, reducing the solubility of oxygen to quantities typically measured in parts per billion. Steam, produced in the boiler at 10 bar g, is to be used as the heating agent. If the 'From and At' rating of the boiler plant is 10 tonne/h, determine the flowrate of steam required to heat the deaerator.

Where:

Boiler 'From and At' rating = 10 000 kg/h
Initial feedwater temperature = 85°C
Initial feedwater enthalpy at 85°C (h_1) = 356 kJ/kg (from steam tables)
Boiler pressure = 10 bar g
Enthalpy of saturated steam at 10 bar g (h_g) = 2 781 kJ/kg

Before any calculations can be made to estimate the size of the deaerator, it is important to know the maximum likely feedwater requirement. This is determined by calculating the boiler(s') maximum useful steaming rate, which in turn, depends on the initial feedwater temperature. The maximum steaming rate is found by determining the Boiler Evaporation Factor.

From Equation 3.5.1

$$\text{Evaporation factor} = \frac{A}{B - C} \qquad \textbf{Equation 3.5.1}$$

Where:

A = Specific enthalpy of evaporation at atmospheric pressure is 2 258 kJ/kg
B = Specific enthalpy of saturated steam at boiler pressure (h_g) in (kJ/kg)
C = Specific enthalpy of the feedwater (h_1) in (kJ/kg)

$$\text{Evaporation factor} = \frac{2\,258}{2\,781 - 356}$$

$$= 0.931\,1$$

The maximum possible boiler output = 'From and At' rating x evaporation factor

$$= 10\,000 \times 0.931\,1$$

$$= 9\,311 \text{ kg/h}$$

Equation 3.21.1 is used to find the required amount of steam to heat the deaerator.

From steam tables;

Enthalpy of feedwater at the required temperature of 105°C (h_2) = 440 kJ/kg
Enthalpy of steam supplying the control valve @ 10 bar g (h_g) = 2 781 kJ/kg

From above;

Enthalpy of the feedwater at 85°C (h_1) = 356 kJ/kg
Mass flowrate of water make-up to deaerator ($\dot{m}$) = 9 311 kg/h

$$\dot{m}_s = \dot{m} \frac{(h_2 - h_1)}{(h_g - h_2)} \qquad \textbf{Equation 3.21.1}$$

$$\dot{m}_s = \frac{9\,311\,(440 - 356)}{(2\,781 - 440)}$$

$$\dot{m}_s = 334 \text{ kg/h}$$

Therefore, the control valve has to be able to supply 334 kg/h of steam with a supply pressure of 10 bar g, and with a downstream pressure of 0.2 bar g.

Example 3.21.2 Sizing and selecting a control system for a pressurised deaerator

The selections in this example are not the only solutions, and the designer will need to consider the demands of an individual site with respect to the availability of electric and pneumatic services.

The objective of this Section is the selection of control valves and systems. Pipeline ancillaries such as strainers and stop valves have been omitted for clarity, they are, nevertheless, vitally important to the smooth running and operation of a pressurised deaerator.

Data

As shown in Figure 3.21.4 plus the actual output shown below:

Boiler:

- Operating pressure (P_1) = 10 bar g
- 'From and At' rating = 10 000 kg/h
- Actual output = 9 311 kg/h with a feedwater temperature of 85°C

Deaerator:

- Operating pressure (P_2) = 0.2 bar g (Saturation temperature 105°C)

The steam control valve

Sizing a control valve for saturated steam service can be determined using Equation 3.21.2:

$$\dot{m}_s = 12\, K_v\, P_1 \sqrt{1 - 5.67\,(0.42 - \varkappa)^2} \qquad \textbf{Equation 3.21.2}$$

Where:

$\dot{m}_s$ = Steam mass flowrate (kg/h)
K_v = Valve coefficient required
P_1 = Pressure upstream of the control valve (bar a)
P_2 = Pressure downstream of the control valve (bar a)
$\varkappa$ = Pressure drop ratio $\left(\frac{P_1 - P_2}{P_1}\right)$

However, since P_2 (1.2 bar a) is less than 58% of P_1 (11 bar a) the steam flow is subjected to critical pressure drop, so K_v can be calculated from the simpler equation (Equation 6.4.3) used for critical flow conditions.

$$\dot{m}_s = 12\, K_v\, P_1 \qquad \textbf{Equation 6.4.3}$$

From Equation 6.4.3 $$K_v = \frac{334}{12 \times 11} = 2.53$$

The selected control valve should have a K_{vs} larger than 2.53, and would normally be provided by a DN15 valve with a standard K_{vs} of 4, and an equal percentage trim.

Steam control equipment selection

This control will need to respond quickly to changes in pressure in the deaerator, and to accurately maintain pressure; a valve with a pneumatic actuator would operate in the required manner. The pressure sensing and control functions may be provided either by pneumatic or electronic equipment and the control signal output (0.2 to 1 bar or 4 - 20 mA) should go to an appropriate positioner.

Equipment required:

- A DN15 two port valve with standard equal percentage trim (K_{vs} = 4).
- A pneumatic actuator able to close a DN15 valve against a pressure of 10 bar.
- A pneumatic-pneumatic positioner with mounting kit (alternatively an electropneumatic positioner with mounting kit).
- A pneumatic controller with a range of 0 - 7 bar (alternatively an electronic controller and sensor with an appropriate range).

As mentioned earlier, a pilot operated self-acting pressure control may be acceptable. A direct acting diaphragm actuated self-acting pressure control, however, should be avoided if the deaerator load changes considerably, as the wide P-band associated with such valves may not give accurate enough pressure control over the load range.

Control for the water system (level control)

Water supply:

- Transfer pump discharge pressure = 2 bar g
- Feedtank temperature = 85°C
- Steam flowrate to the deaerator ($\dot{m}_s$) has already been calculated at 334 kg/h.

In this example the maximum water flowrate (the 'actual' capacity of the boiler) to the deaerator is 9 311 kg/h. Water valves are sized on volume flowrates, so it is necessary to convert the mass flow of 9 311 kg/h to volumetric flow in m³/h.

The pump discharge pressure onto the control valve is 2 bar g. From steam tables, the specific volume of water at 2 bar g and 85°C is 0.001 032 m³/kg.

It is important to determine the pressure required behind the water distribution nozzle to give proper distribution; the control valve selection must take this into consideration. For this example, it is assumed that a pressure of 1.8 bar is required at the inlet to the distributor nozzle.

The sizing parameters for the water control valve are:

$\dot{V}$ = 9 311 kg/h x 0.001 032 m³/kg = 9.6 m³/h

P_1 = 2 bar g

P_2 = 1.8 bar g

Sizing a control valve for liquid service can be determined by calculating the K_v, see Equation 3.21.3:

$$\dot{V} = K_v \sqrt{\frac{\Delta P}{G}} \qquad \textbf{Equation 3.21.3}$$

Where:

$\dot{V}$ = Volumetric flowrate (m³/h)
K_v = Valve coefficient required
ΔP = Pressure drop across the valve (bar)
G = Relative density of fluid (water = 1)

For water, as G = 1,

$$\dot{V} = K_v \sqrt{\Delta P}$$

$$K_v = \frac{\dot{V}}{\sqrt{\Delta P}}$$

$$K_v = \frac{9.6}{\sqrt{2 - 1.8}}$$

$$K_v = \frac{9.6}{\sqrt{0.2}}$$

$$K_v = 21.5$$

The selected control valve should have a K_{vs} larger than 21.5

Water control equipment selection

Because of the relatively large mass of water held in the deaerator, the speed of control signal response is not normally an issue, and an electrically actuated control may provide an adequate solution.

However, a pneumatically actuated control will provide equally as good a solution.

Equipment required:

- A DN40 two port valve with standard trim (K_{vs} = 25).
- An electric actuator that will close a DN40 valve against the maximum transfer pump pressure.
- A feedback potentiometer will be needed with the actuator.
- A capacitance level probe of appropriate length with a preamplifier.
- A level controller to accept the signal from the capacitance probe, and then pass a modulating signal to the valve actuator.

Note that this only gives water level control plus either a high or low alarm. Should additional low or high alarms be required, the options are either:

1. A capacitance level probe with level controller, which can provide two additional level alarms.
2. A four-tip conductivity level probe, with a level controller, which can provide up to four level alarms.

   or

3. A single tip high integrity, self-monitoring level probe and associated level controller which will provide either a high or low level alarm.

Table 3.21.2 identifies the major difficulties that may be encountered with a pressurised deaerator, and their possible causes.

Table 3.21.2 Diagnosing deaerator malfunctions

Deaerator malfunction	Possible cause
High level of oxygen in feedwater	Leakage of air into the deaerator.
	Insufficient residence time.
	Water/steam mixing equipment not designed / installed / operating correctly.
	Flowrate outside design specification.
Pressure fluctuations	Control valves incorrectly sized.
	Wide temperature variation in the incoming water supply.
Low outlet temperature	Insufficient steam.
	Water/steam mixing equipment not designed / installed / operating correctly.
High level of carbon dioxide in feedwater	Feedwater pH is too high.

Acknowledgement

Spirax Sarco gratefully acknowledge the help and information provided by:
Satec Ltd., Regency Court, 36 High Street, Crewe, Cheshire, UK CW2 7BN

Questions

1. What is the advantage of a pressurised deaerator over an atmospheric deaerator?

a| A boiler feedtank is no longer required ☐

b| Less overall energy will be required to produce the steam ☐

c| It can be fitted at ground level ☐

d| It removes more oxygen ☐

2. At which typical pressure will a pressurised deaerator supplying a shell boiler normally operate?

a| 0.2 bar g ☐

b| 1.2 bar g ☐

c| 5 bar g ☐

d| Boiler pressure ☐

3. How is the released oxygen in a pressurised deaerator prevented from being reabsorbed by the water?

a| By an air vent ☐

b| There is insufficient water surface for the air to be reabsorbed ☐

c| By a steam blanket over the water ☐

d| By the incoming steam against the incoming water ☐

4. What might be the likely affect of supplying water to a pressurised deaerator at, for example 95°C instead of 80°C?

a| The incoming water might be overheated ☐

b| No effect ☐

c| There might be insufficient steam flow to provide efficient heating of the water ☐

d| There might be insufficient residence time for effective oxygen removal ☐

5. How could the installation of a pressurised deaerator be justified?

a| Savings in chemical treatment ☐

b| Savings in energy required to produce the steam ☐

c| Removal of the boiler feedtank will be possible ☐

d| Savings in boiler bottom blowdown ☐

6. How is the water in a pressurised deaerator heated to the required temperature?

a| By a blanket of steam above the water ☐

b| By direct steam injection into the water ☐

c| By a spray of steam as it enters the deaerator dome ☐

d| It is not heated further but just held at a higher pressure ☐

Answers

1: d, 2: a, 3: c, 4: c, 5: a, 6: c

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Module 3.22

Steam Accumulators

Steam Accumulators

The purpose of a steam accumulator is to release steam when the demand is greater than the boiler's ability to supply at that time, and to accept steam when demand is low.

Steam accumulators are sometimes thought of as relics of the 'steam age' with little application in modern industry. The following Sections within this Module will:

- Illustrate how a steam accumulator can improve the operation of a modern plant.
- Discuss the factors which make steam accumulators even more necessary now, than in the past.
- Provide guidance on the sizing and selection of appropriate ancillary equipment.

Boiler design

Contemporary boilers are significantly smaller than their counterparts of only 30 years ago. This reduction in boiler size has been brought about by users, who demand that boilers be:

- More efficient in terms of fuel input to steam output.
- More responsive to changes in demand.
- Smaller, and so take up less floor space.
- Cheaper to buy and install.

These targets have been met in part by today's more sophisticated controls/burners which respond faster and more accurately to changes in demand than those of bygone years. However, a boiler's response to changes in demand is also affected by the laws of nature, for example: how much water is to be heated and the heat transfer area available to transfer that heat from the burner flame to the water.

Response times have been improved by physically reducing the external dimensions of the boiler for any given output, and by cramming the insides full of tubes to increase the heat transfer area. This means that the modern boiler holds less water, and the heat transfer area per kg of water is greater. Consider the situation of today:

1. Steam demand from the plant is increased, and the pressure in the boiler falls to the burner control set point.
2. The burner control purges the combustion chamber, and the burner is ignited.
3. The large heat transfer area and the lower mass of water combine to rapidly evaporate the water in the boiler to satisfy the demand for steam.

As covered in Module 3.7, 'Boiler Fittings and Mountings', the energy stored in a boiler is contained in the water which is held at saturation temperature. The greater the amount of water inside a boiler, the greater the amount of stored energy to cope with changes in demand/load.

Table 3.22.1 compares an old Lancashire boiler of the 1950s with a modern packaged boiler. Note that the modern packaged boiler contains only 20% of the water held in a similarly rated Lancashire boiler. It follows from this that the reserve of energy held in the modern packaged boiler is only 20% of the Lancashire boiler. This suggests that the modern packaged boiler cannot cope with peak demands in the way an old Lancashire boiler could.

Also note from Table 3.22.1, that the 'steam release rate' from the surface of the water inside the modern packaged boiler has increased by a factor of 2.7. This means that the steam has only 1/2.7 (40%) of the time available in a Lancashire boiler to separate itself from the water. At times of peak demand this may mean that wet steam is being exported from the modern packaged boiler, and possibly at a lower pressure than that which it was designed to operate - Covered in Module 3.12 'Controlling TDS in the Boiler Water'.

Water which is carried over with the steam will be dirty (approximately 3 000 ppm TDS), and will contaminate control valves and heat transfer surfaces. It may even block some of the smaller orifices in pressure sensing devices, steam traps and so on.

Table 3.22.1 Comparison of Lancashire and modern packaged boilers

Boiler type	Length x diameter	'From and At' rating	Water content	Surface area of water in the boiler	Steam release rate from surface
Lancashire	9.1 m x 2.7 m (30' x 9')	4 540 kg/h (10 000 lb/h)	45 400 kg (10 000 gal.)	18.6 m² (200 ft²)	244 kg/m² h (50 lb/ft² h)
Modern packaged	3.9 m x 2.5 m (13' x 8')	4 540 kg/h (10 000 lb/h)	9 100 kg (2 000 gal.)	7m² (75 ft²)	649 kg/m² h (133 lb/ft² h)

Note: The information to create Table 3.22.1 was supplied by Thermsave. Imperial units are also shown in the Table to provide an insight into the factors applied in the designing of boilers in the past.

Peak demands

Steam demands on any process plant are rarely steady, but the size and type of the fluctuations depend on the application and the industry. Peaks may occur once a week or even once a day during start-up.

The biggest problems caused by peak demands are usually associated with batch processing industries:

- Brewing.
- Textiles.
- Dry-cleaning.
- Canning.
- Lightweight concrete block manufacturers.
- Specialised areas of the steel making industry.
- Rubber industries with large autoclaves.

For these processes the peaks may be heavy and long-term, and measured in fractions of an hour.

Alternatively, load cycles can consist of short-term frequent peaks of short duration but very high instantaneous flowrate:

- Hosiery finishing.
- Rubber.
- Plastic and polystyrene moulding.
- Steam peeling.
- Hospital and industrial sterilisation.

Figure 3.22.1, shows that in each case the demands are almost instantaneous and the peaks are well above the average load. The result of a sudden demand on boiler plant is a pressure drop in the boiler, because the boiler and its associated combustion equipment are unable to generate steam at the rate at which it is being drawn off.

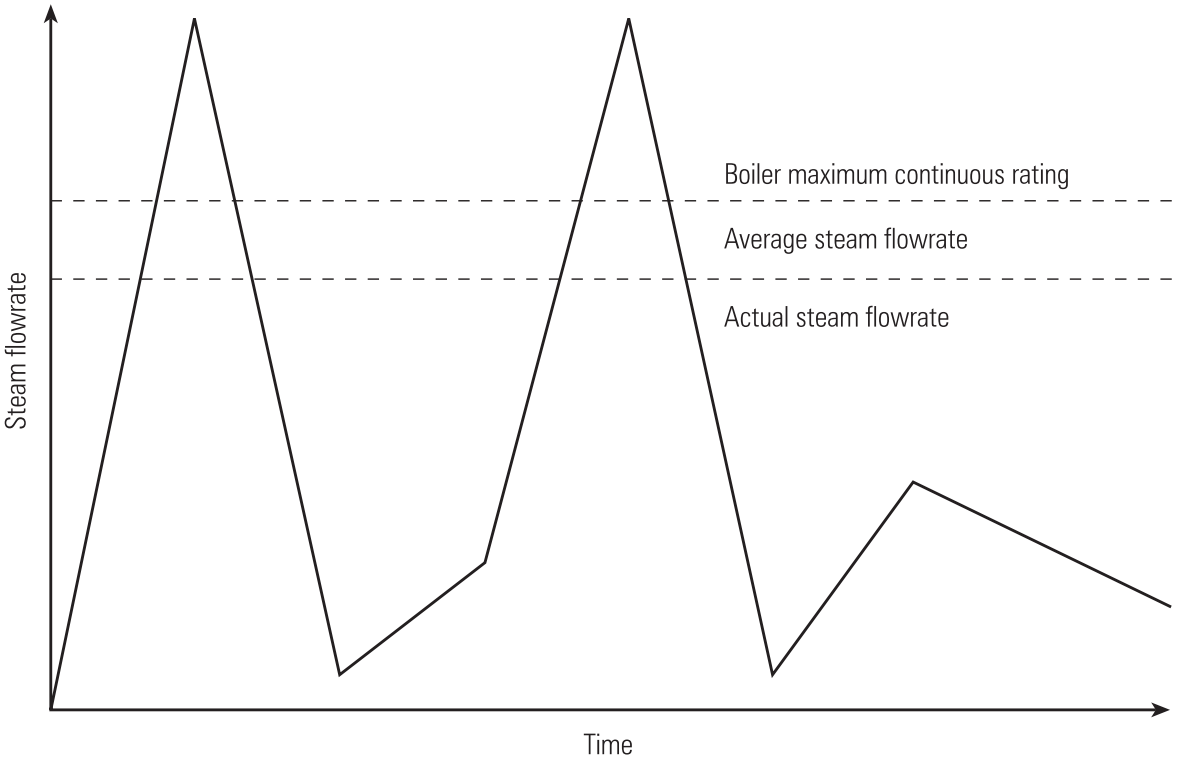


Fig. 3.22.1 Typical steam flow chart for a batch process plant

Peak demands and subsequent pressure drops may have quite serious consequences on factory production.

At worst, the result is a boiler 'lockout', due to the elevation of water level caused by rapid boiling, followed by its collapse. This is seen as a low water level alarm by the level controls.

At best, the steam produced is wet and contaminated. This, coupled with a reduction in pressure, can lead to:

- Increased process times.
- A reduction in product quality or even damage or loss of the product.
- Waterhammer in the steam mains causing distress to pipework and fittings, and possible danger to personnel.

For the boiler plant, peak demands are responsible for:

- A higher level of maintenance.
- Reduced boiler life.
- Reduced fuel efficiency.

This is because the combustion equipment is continually cycling from low to high fire, and even shutting off during periods of very low demand, only to fire again a few minutes later, with all the pre and post-purge chilling effects.

Multiple or oversized boilers may be used in an effort to cope with peak demands (and the subsequent dips in demand) which inevitably result in low efficiencies.

To illustrate this point, it can be assumed that:

- For an average steam boiler, less than 1% of the losses are due to heat radiated from the boiler shell (for example: 1% of the Maximum Continuous Rating (MCR) of the boiler).
- If a boiler is then producing 50% of its MCR, the losses due to radiation are 2% relative to its production rate.
- If a boiler is producing 25% of its MCR the losses are 4% of its production rate.

And so on, until a boiler is simply maintained at a pressure without exporting any steam to the factory. At this point, 1% of its MCR is a 100% loss relative to its steam production rate.

If boiler plant is sized for peak loads, problems arise due to oversizing relative to the average demand. In practice, a boiler may shut off during a period of low demand. If this is then followed by a sudden surge of demand and the boiler is not firing, an alarm situation may arise.

Alarms will ring, the boiler may lockout and steam recovery will be slow and arduous.

In short, peaks are responsible for:

- Loss of production.
- Reduced product quality.
- Increased production times.
- Poor quality steam from the boiler.
- Low fuel efficiency.
- High maintenance costs.
- Reduced boiler life.

Load levelling techniques

Modern boilers are very efficient when properly loaded and respond quickly to load increases, provided that the boiler is firing. However, conventional shell boilers are generally unable to meet large peak demands in a satisfactory way and should be protected from large fluctuating loads.

Various methods are used in an attempt to create a stable load pattern to protect the boiler plant from the effects of large fluctuating loads.

Engineering methods:

- Pressure maintaining valves (also called surplussing valves) can be used as load shedding devices by isolating non-essential parts of the plant and thereby giving priority to essential plant, a typical arrangement is shown in Figure 3.22.2. The success of this method again depends on the severity of the peaks and the assumption that the boiler is firing when the peak develops.

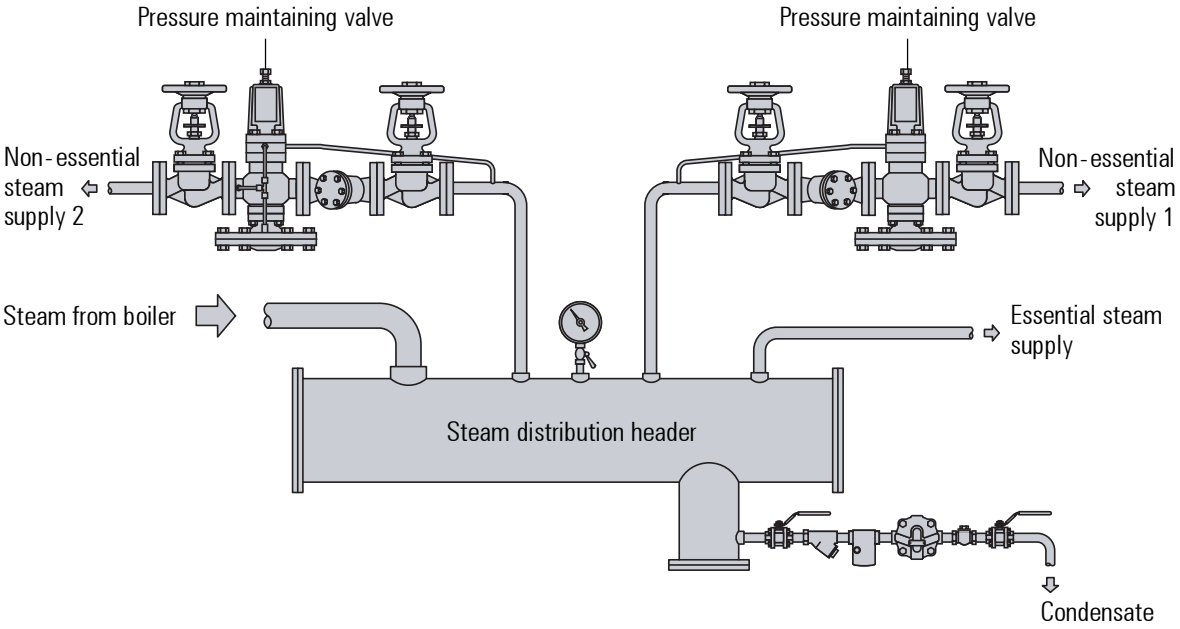


Fig. 3.22.2 Surplussing valves used as load shedding devices

Surplussing valves can also be fitted directly to the boiler or on the steam main to the factory, as shown in Figure 3.22.3.

The set pressure should be:

- Less than the 'high fire' control pressure, to prevent any interference of the surplussing control with the burner controls.
- High enough to maintain the pressure in the boiler at a safe level.

In terms of sizing the surplussing valve, the requirement is for minimum pressure drop. As a general indication, a line size valve should be considered.

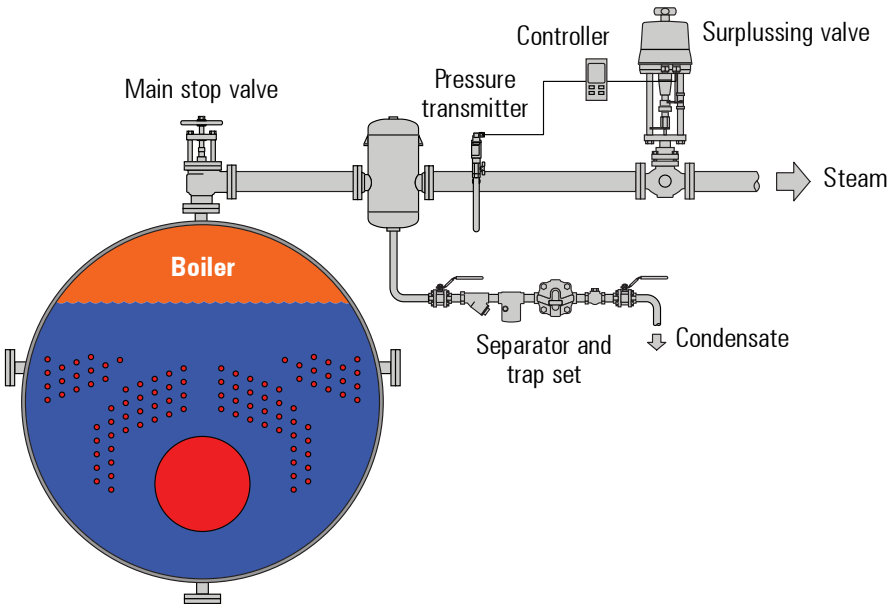


Fig. 3.22.3 Surplussing valve on a boiler main

- Two-element or three-element water level control. These can be successful as long as the peaks are not violent and the boiler is firing when the peak develops; the boiler must also have sufficient capacity.

  Two-element control uses inputs from the boiler water level controls and the steam flowrate to position the feedwater control valve.

  Three-element control uses the above two elements plus an input from a feedwater flow measuring device to control the incoming feedwater flowrate, rather than just the position of the feedwater control valve. (This third element is only appropriate on boilers which use modulating level control in boiler houses with a feedwater ring main.)

Example 3.22.1

A boiler is rated at 5 000 kg/h 'From and At'

The high/low fire pressure settings are 11.3/12.0 bar g respectively (12.3/13.0 bar a).

The surplussing valve setting is 11.0 bar g (12.0 bar a).

1. Based on a velocity of approximately 25 m/s, a 100 mm steam main would be selected.
2. K_{VS} of a standard DN100 surplussing control valve is 160 m³/h
3. Using the following mass flow equation for saturated steam the pressure downstream of the surplussing valve (P_2) can be calculated:

$$\dot{m}_s = 12 K_V P_1 \sqrt{1 - 5.67 (0.42 - \chi)^2} \qquad \textbf{Equation 3.21.1}$$

Where:
$\dot{m}_s$ = Steam mass flowrate (kg/h)
K_v = Valve flow coefficient
P_1 = Pressure upstream of the control valve (bar a)
P_2 = Pressure downstream of the control valve (bar a)
χ = Pressure drop ratio $\left(\frac{P_1 - P_2}{P_1}\right)$

In this example, at low fire, the boiler pressure is given as 12 bar g (13 bar a).

It can be calculated from Equation 3.21.1 that the pressure after the fully open surplussing valve is 11.89 bar g (12.89 bar a).

Consequently, the pressure drop is small (0.11 bar) and would not be significant in normal operation. However, if the pressure should fall to 11.0 bar g, the surplussing valve will start to close in order to maintain upstream pressure.

The proportional band on the controller should be set as narrow as possible without making the valve 'hunt' about the set point.

Both methods of applying pressure-maintaining valves may provide protection to the boiler plant, but they will not overcome the fundamental requirement of more steam for the process.

Management methods

These include, for example, staggered starts on processes to keep peak loads as low as possible. This method of smoothing out peaks can be beneficial to the boiler plant but may be detrimental and restrictive to production, having much the same effect as the pressure-maintaining valve.

It is, however, impossible to smooth out short-term peaks using only management methods. In a factory where there are many individual processes imposing such peaks it is possible for this to have a levelling effect on the load, but equally so, it is also possible for the many individual processes to peak simultaneously, with disastrous effects.

If the above methods do not provide the required stability of demand, it may be time to consider a means of storing steam.

The steam accumulator

The most appropriate means of providing clean dry steam instantaneously, to meet a peak demand is to use a method of storing steam so that it can be 'released' when required. Storing steam as a gas under pressure is not practical due to the enormous storage volume required at normal boiler pressures.

This is best explained in an example:

In the example used later in this Module, a vessel with a volume of 52.4 m³ is used.

- Charging pressure is 10 bar g (specific volume = 0.177 m³/kg).
- Discharge pressure is 5 bar g (specific volume = 0.315 m³/kg).

Based on these parameters, the resultant energy stored and ready for instant release to the plant is contained in 130 kg of steam. This amounts to only 5.2% of the energy stored and ready for use, compared to a water filled accumulator.

In practice there are two ways of generating steam:

- By adding heat to boiling water, indirectly via a combustion tube and burner, as in a conventional boiler.
- By reducing the pressure on water stored at its saturation temperature. This results in an excess of energy in the water, which causes a proportion of the water to change into steam.

  This phenomenon is known as 'flashing', and the equipment used to store the pressurised water is called a steam accumulator. There are, in principle, two types of systems available for steam storage; the pressure-drop accumulator and the constant pressure accumulator. This module only considers the former type.

A steam accumulator is, essentially, an extension of the energy storage capacity of the boiler(s). When steam demand from the plant is low, and the boiler is capable of generating more steam than is required, the surplus steam is injected into a mass of water stored under pressure. Over a period of time the stored water content will increase in temperature and pressure until it finally achieves the saturation temperature for the pressure at which the boiler is operating.

Demand will exceed the capability of the boiler when:

- A load is applied faster than the boiler's ability to respond - for example, the burner(s) may be extinguished and a purging cycle must be completed before the burner can be safely ignited. This may take up to 5 minutes, and rather than adding heat to the boiler, the purging cycle will actually have a slight cooling effect on the water in the boiler. Add to this the fact that the flashing of the boiler water will cause a drop in water level, and the boiler level control system will automatically compensate for this by bringing feedwater in at, for example, 90°C. This will have a quenching effect on the water already at saturation temperature, and will aggravate the situation.
- A heavy demand occurs over a longer then normal period.

In either case, the result is a drop in pressure inside the steam accumulator, and as a result of this some of the hot water will flash to steam. The rate at which the water flashes to steam is a function of the storage pressure, and the rate at which steam is required by the system being supplied.

Charging

The pressure-drop steam accumulator consists of a cylindrical pressure vessel partially filled with water, at a point between 50% and 90% full depending on the application. Steam is charged beneath the surface of the water by a distribution manifold, which is fitted with a series of steam injectors, until the entire water content is at the required pressure and temperature.

It is natural that the water level will rise and fall during charging and discharging.

If the steam accumulator is charged using saturated (or wet) steam, there may be a small gain in water due to the radiation losses from the vessel. Normally, a slightly greater mass of steam is discharged than is admitted.

A steam trap (ball float type) is fitted at the working level and acts as a level-limiter, discharging the small amount of surplus water to the condensate return system.

However, if the steam accumulator were charged using superheated steam, or if the radiation losses are very small, there would be a gradual loss of water due to evaporation, and a feedvalve or pump, under the control of level probes, would be required to make up the deficit.

Discharging

As a pressure drop occurs in a steam accumulator with the stored water at saturation temperature, flash steam will be generated at the rate demanded by any load above the boiler capacity; consequently the overload condition will be satisfied. When the overload is followed by a demand below the boiler capacity the steam accumulator is charged using surplus steam from the boiler. This charge and discharge cycle explains the name 'steam accumulator' and continually allows the boiler to fire up to its maximum continuous rating.

The charging / discharging cycle

The accumulator needs to be fully charged at the beginning of its discharge period, for it to operate correctly. To allow this, two main events must be satisfied:

1. Enough time must be available from the end of one overload period to the beginning of the next, to recharge the water stored in the accumulator.
2. The average off-load steam demand must be lower than the boiler capacity (the maximum continuous rating or MCR), such that sufficient surplus boiler capacity is available to recharge the water stored in the accumulator during off-peak times.

Other criteria are also important to ensure the accumulator has enough capacity, and these must be satisfied by the design:

1. Enough water must be stored to provide the required amount of flash steam during the discharge period. This can be satisfied by ensuring the accumulator volume is large enough.
2. Higher steam release rates will produce wet steam. The velocity and flowrate at which the flash steam is released from the water surface must be below a predetermined value. This can be satisfied by ensuring the water surface area is large enough which, in turn, depends on the accumulator size.
3. The evaporation capacity must be sufficient. This depends on the pressure at which the water is stored when fully charged (the boiler pressure) and the minimum pressure at which the accumulator will operate at the end of the discharge period (the accumulator design pressure). The larger the differential between these two pressures, the more flash steam will be produced.
4. The accumulator design pressure must be higher than the downstream distribution pressure. This is necessary to create a pressure differential across the downstream pressure reducing valve (PRV), to allow the required flow from the accumulator to the plant. The closer the accumulator pressure to the distribution pressure, the smaller the accumulator but this also gives a smaller differential across the PRV. This requires a larger PRV; large enough to pass the highest overload demand when the accumulator is at its design pressure (the minimum pressure in the accumulator at the end of the discharging period).

Sizing a steam accumulator

A steam accumulator in the steam system gives increased storage capacity. Proper design of the steam accumulator ensures that any flowrate can be catered for. There are no theoretical limits to the size of a steam accumulator, but of course practical considerations will impose restrictions.

In practice the steam accumulator volume is based on the storage required to meet a peak demand, with an allowable pressure drop, whilst still supplying clean dry steam at a suitable steam release velocity from the water surface. Example 3.22.2 below, is used to calculate the potential of steam capacity in a horizontal steam accumulator.

Example 3.22.2

Boiler:

Maximum continuous rating = 5 000 kg/h

Normal working pressure = 10 bar g (h_f = 782 kJ/kg, from steam tables)

Burner switching differential = 1 bar (0.5 bar either side of 10 bar g)

Plant requirements:

Maximum instantaneous overload = 12 000 kg/h

Distribution pressure = 5 bar g

Although the maximum instantaneous overload is 12 000 kg/h, the mean value of the overload should be used to size the accumulator.

This prevents unnecessary oversizing of the accumulator. Equally, it is necessary to determine and use the mean 'off-peak' load in the sizing calculation. Off-peak load is any load below the boiler MCR.

Finding the mean value of the overload and off-peak load

There are three possible methods to establish the mean loads for existing boiler plant:

1. To guestimate, based on experience.
2. To interrogate the existing boiler steam output charts to establish the mean loads and the time periods over which they occur.
3. To program a steam meter's computer to integrate the steam load over both the overload and off-peak load periods.

Method 1 could prove to be rather reckless, if an expensive accumulator ended up too small.

However, if the boiler plant is still at the design stage, an educated guess will be the only option. From the designer's knowledge of the installation, it should be possible to give a reasonable estimate of the maximum plant load, the load diversity, and the times over which they occur.

Method 2 is quite easy to expedite, and should give a reasonably accurate result.

Method 3 would provide the most accurate results, and the cost of the steam meter is small relative to the overall cost of an accumulator project.

The following procedure shows how to determine the mean steam load from an existing chart recording the load pattern. The procedure is built up from Figure 3.22.4, which shows the flow pattern for Example 3.22.2.

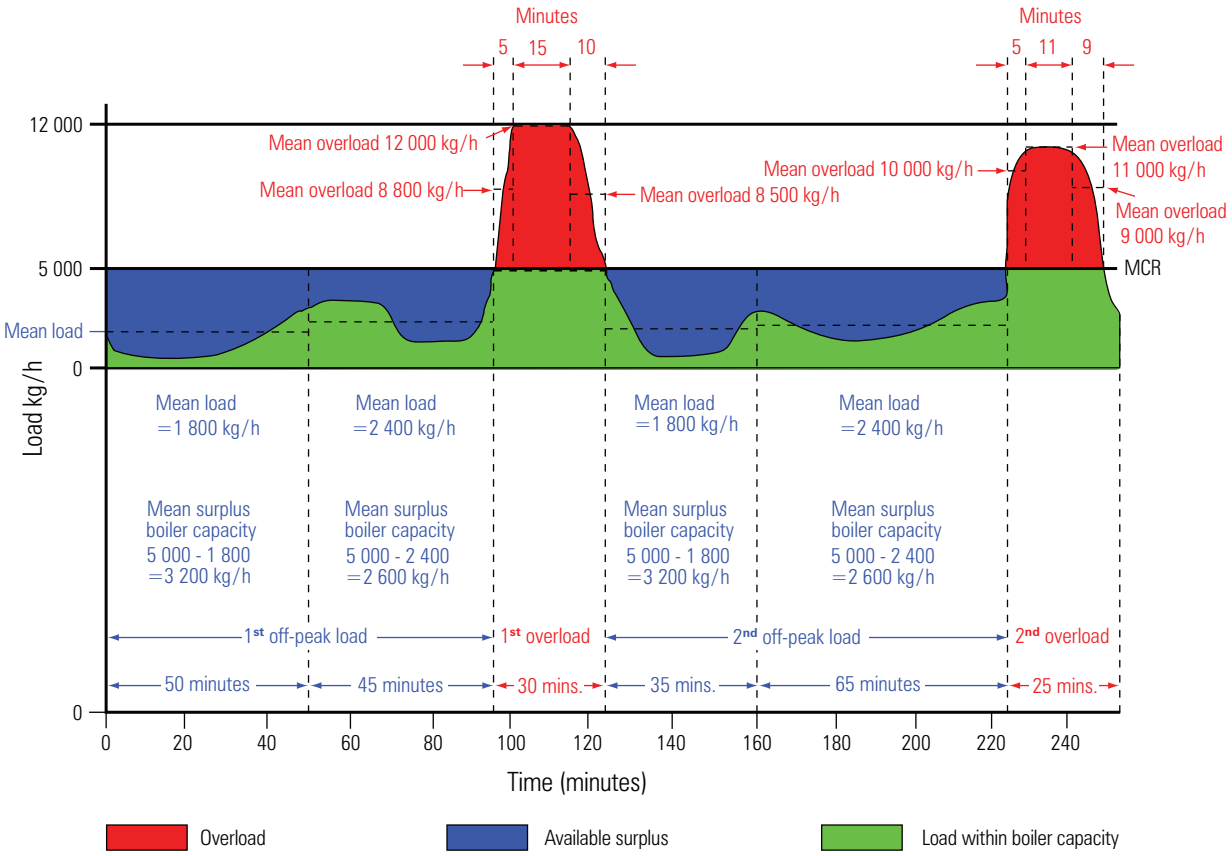


Fig. 3.22.4 Shows the boiler MCR, and allows the mean load periods to be defined

From Figure 3.22.4, it can be seen that the off-peak loads have been divided up into the following mean loads and time periods. From this data, the average load for each time off-peak period can be determined.

The mean load is calculated in the following way:

$$\frac{\Sigma \text{ (Mean loads x times)}}{\text{Total time period}}$$

1st off-peak load

Mean load kg/h	3 200	2 600
Each period time (minutes)	50	45
Total period time (minutes)	95	95

$$\text{Mean load} = \frac{(3\,200 \times 50) + (2\,600 \times 45)}{95}$$

$$\text{Mean load} = \frac{(160\,000) + (117\,000)}{95}$$

$$\text{Mean load} = \frac{277\,000}{95}$$

$$\text{Mean load} = 2916 \text{ kg/h}$$

2nd off-peak load

Mean load kg/h	3 200	2 600
Each period time (minutes)	35	65
Total period time (minutes)	97	97

$$\text{Mean load} = \frac{(3\,200 \times 35) + (2\,600 \times 65)}{100}$$

$$\text{Mean load} = \frac{(112\,000) + (169\,000)}{100}$$

$$\text{Mean load} = \frac{281\,000}{100}$$

$$\text{Mean load} = 2\,810 \text{ kg/h}$$

A similar exercise is carried out for the overload periods from Figure 3.22.9.

1st overload

Mean load kg/h	8 800	12 000	8 500
Each period time (minutes)	5	15	10
Total period time (minutes)	30	30	30

$$\text{Mean load} = \frac{(8\,800 \times 5) + (12\,000 \times 15) + (8\,500 \times 10)}{30}$$

$$\text{Mean load} = \frac{(44\,000) + (180\,000) + (85\,000)}{30}$$

$$\text{Mean load} = \frac{309\,000}{30}$$

$$\text{Mean load} = 10\,300 \text{ kg/h}$$

2nd overload

Mean load kg/h	10 000	12 000	9 000
Each period time (minutes)	5	11	9
Total period time (minutes)	25	25	25

$$\text{Mean load} = \frac{(10\,000 \times 5) + (11\,000 \times 11) + (9\,000 \times 9)}{25}$$

$$\text{Mean load} = \frac{(50\,000) + (121\,000) + (81\,000)}{25}$$

$$\text{Mean load} = \frac{252\,000}{25}$$

$$\text{Mean load} = 10\,080 \text{ kg/h}$$

From the above data, it can be seen that:

- The boiler maximum continuous rating = 5 000 kg/h
- The maximum instantaneous overload = 12 000 kg/h
- The largest mean off peak load = 2 916 kg/h
- The largest mean overload = 10 300 kg/h
- The minimum time between overloads = 95 minutes
- The distribution pressure = 5 bar g

The accumulator design pressure needs to be chosen, and it is usual to choose a pressure 1 bar higher than the distribution pressure. This gives a reasonable flash steam capacity, without unduly oversizing the downstream PRV.

In this example the distribution pressure is 5 bar g, so the accumulator design pressure can initially be considered at 6 bar g.

From this information, an accumulator may now be sized.

Steam accumulator:

Design pressure = 6 bar g

h_f = 698 kJ/kg

Length = 7 m

Diameter = 4 m

Water capacity = 78 909 litres
(Typically 90% of the volume
of the steam accumulator vessel) = 87 976 Litres
= 87.976 m³

At 10 bar g, density of water = 0.882 kg/litre

Water mass = 78 909 L x 0.882 kg/litre

Water mass = 69 598 kg at 90% full

The potential steam capacity in a steam accumulator can be calculated using Equation 3.22.1:

$$\text{Steam storage capacity (kg)} = \frac{\text{Difference in enthalpy of water (kJ/kg) x mass of water (kg)}}{\text{Enthalpy of evaporation at the lower pressure (kJ/kg)}}$$ **Equation 3.22.1**

$$\text{Steam storage capacity} = \frac{(781 \text{ kJ/kg} - 698 \text{ kJ/kg}) \times 69\,598 \text{ kg}}{2\,065 \text{ kJ/kg}}$$

Steam storage capacity = 2 797 kg

Note that this 2 797 kg of flash steam will be released in the time taken for the pressure to drop. If this has been an hour, the steaming rate is 2 797 kg/h; if it were over 30 minutes, then the steaming rate would be:

$$\frac{2\,797 \text{ kg/h} \times 60 \text{ minutes}}{30 \text{ minutes}} = 5\,594 \text{ kg/h}$$

If the steam accumulator is connected to a boiler rated at 5 000 kg/h, and supplying an average demand within its capacity, the combined boiler and accumulator outputs could meet average overload conditions of 5 594 + 5 000 = 10 594 kg/h for 30 minutes. The alternative is an additional combination of boilers capable of generating 10 594 kg/h for 30 minutes with the limitations previously noted.

It is now possible to check the accumulator size.

The figures as used in Example 3.22.2 are used below to facilitate checking.

Boiler

Maximum continuous rating = 5 000 kg/h

Normal working pressure = 10 bar g

Plant requirements

Largest mean overload = 10 300 kg/h for 30 minutes every 95 minutes

Pressure = 5 bar g

Required steam storage = 10 300 kg/h - 5 000 kg/h steam supplied by the boiler

Required steam storage = 5 300 kg/h

However, steam is only required for 30 minutes every hour, so the steam storage required must be:

$$\text{Steam storage required} = \frac{5\,300 \text{ kg/h} \times 30 \text{ minutes/cycle}}{60 \text{ minutes/h}}$$

Steam storage required = 2 650 kg/cycle

The amount of water required to release 2 650 kg of steam is a function of the proportion of flash steam released due to the drop in pressure.

This satifies the criterion of having enough water to produce the required amount of flash steam. It can be seen that the storage capacity of 2 797 kg is greater than the storage required of 2 650 kg of steam.

If the steam accumulator will be charged at 10 bar g by the boiler, and discharged at 6 bar g to the plant, the proportion of flash steam can be calculated as follows:

$$\text{Proportion of flash steam} = \frac{(h_f \text{ at } P_1) - (h_f \text{ at } P_2)}{h_{fg} \text{ at } P_2} \qquad \textbf{Equation 2.2.5}$$

$$\text{Proportion of flash steam} = \frac{(781 - 698)}{2\,065}$$

Proportion of flash steam = 0.040 2 kg/kg water

To produce 2 650 kg of flash steam:

$$\text{The amount of water required at saturation temperature} = \frac{2\,650 \text{ kg of flash steam}}{0.040\,2 \text{ kg/kg water}}$$

The amount of water required at saturation temperature = 65 920 kg

The water content will typically account for only 90% of the volume of the steam accumulator:

$$\text{Water mass} = \frac{65\,920 \text{ kg}}{90\%} = 73\,245 \text{ kg}$$

$$\text{As the density of water at 10 bar g} = 882 \text{ kg/m}^3$$

$$\text{The total vessel volume} = \frac{73245 \text{ kg}}{882 \text{ kg/m}^3}$$

The total vessel volume = 83 m³

The vessel capacity is 87.9 m³, so the vessel satisfies this criterion.

Using the vessel dimensions given earlier, the water surface area is approximately 20.53 m² when fully charged, at a volume of 90% of the vessel capacity.

The maximum steaming rate from the accumulator is given as 5 300 kg/h, therefore:

$$\text{Maximum steam release rate} = \frac{5\,300 \text{ kg/h}}{20.53 \text{ m}^2}$$

Maximum steam release rate = 258 kg/m² h

Empirical test work shows that the rate at which dry steam can be released from the surface of water is a function of pressure. A working approximation suggests:

Maximum release rate without steam entrainment (kg/m² h) = 220 x pressure (bar a)

The steam accumulator in Example 3.22.2 is operating at 6 bar g (7 bar a). The maximum release rate without steam entrainment will be:

$$220 \times 7 \text{ bar a} = 1\,540 \text{ kg/m}^2 \text{ h}$$

This is shown graphically in Figure 3.22.10

The example at 258 kg/m² h is well below the maximum value, and dry steam can be expected. Had the steam release rate been too high, different diameters and lengths giving the same vessel volume would need to be considered.

It must be emphasised that this is only an indication, and design details should always be delegated to specialist manufacturers.

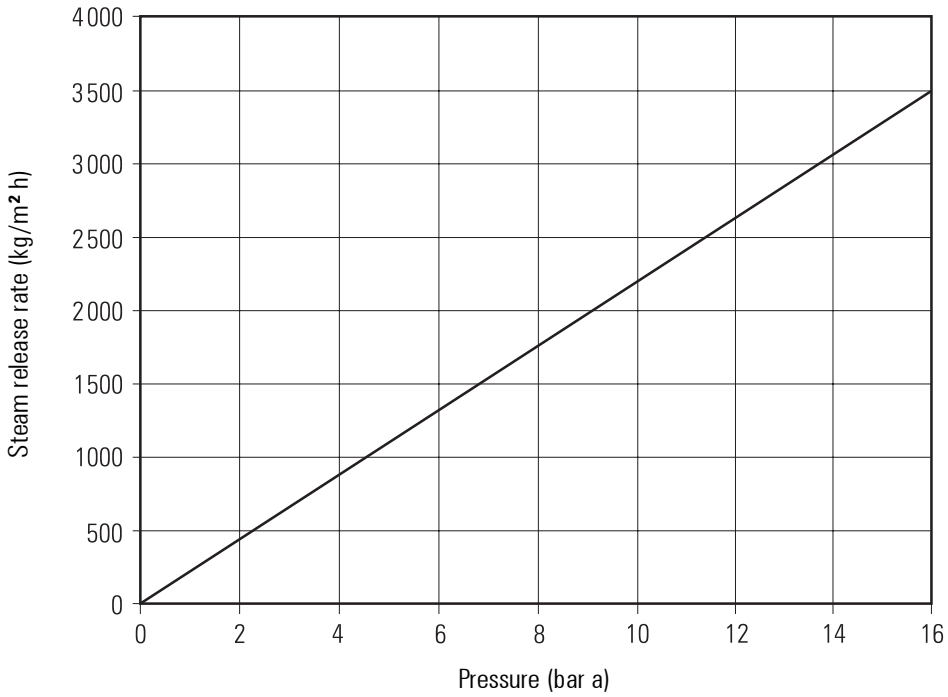


Fig. 3.22.5 Steam release rate without steam entrainment

Steam accumulator controls and fittings

The following is a review of the equipment required for a steam accumulator installation, together with some guidance on sizing and selection of appropriate equipment.

Using figures from Example 3.22.2:

Boiler:

Maximum continuous rating = 5 000 kg/h

Normal working pressure = 10 bar g

Accumulator:

Mass of water = 69 598 kg - fully charged and 90% of vessel volume

P_1 (boiler pressure) = 10 bar g - fully charged

P_2 (discharge pressure) = 6 bar g - fully charged

Plant requirements:

Pressure = 5 bar g

Largest mean overload = 10 300 kg/h for 30 minutes every 95 minutes, of which, 5 000 kg/h is supplied by the boiler.

From these figures it can be deduced that 69 598 kg of water must be heated from saturation temperature at 6 bar g to saturation temperature at 10 bar g in 95 minutes.

Pipework

The pipework between the boiler and the steam accumulator should be sized, as per normal practice, on a steam velocity of 25 to 30 m/s and the maximum output of the boiler.

In the case of Example 3.22.2, this would require a DN100 pipeline from the boiler to the accumulator.

The pipework from the accumulator to the downstream PRV should be sized on the maximum instantaneous overload and a velocity of no more than 20 m/s. This would require a DN250 nominal bore pipe for this example, with an accumulator design pressure of 6 bar g.

Stop valve

A line-size stop valve is required in addition to the boiler crown valve. A suitably rated stop valve, preferably in cast steel, would be appropriate.

Check or non-return valve

A line-size check valve is required to prevent reverse flow of the steam back to the boiler in the event of the boiler being deliberately shut down, or perhaps, the boiler locking-out.

A disc check valve would be an appropriate choice.

Surplussing valve

The surplussing valve is essential to ensure that the rate at which steam is flowing from the boiler to the accumulator is within the capability of the boiler. Example 3.22.1, shows how the valve would be sized.

Pilot operated, self-acting surplussing valves may be used in smaller installations, provided the narrow (and non-adjustable) proportional band is acceptable. A pneumatic controller and control valve is more appropriate to larger installations, and offers the advantage of an adjustable proportional band.

For this application a DN100 pneumatically operated control valve with appropriate operating and shut-off capability, would be selected.

Steam injection equipment

A properly sized steam inlet pipe must feed to well below the water surface level and into a steam distribution header/manifold system such as shown in Figure 3.22.11.

The steam is injected into the water.

It is important to remember that the injector capacity will reduce as the pressure in the vessel increases, as the differential pressure between the injected steam and the vessel pressure is reduced.

At very low flowrates the steam will tend to issue from the injectors closest to the steam inlet pipe(s).

The design of the inlet pipe(s) and the manifold system, together with the placement of the injectors, must provide even injection of steam throughout the length of the accumulator regardless of actual steam flowrate.

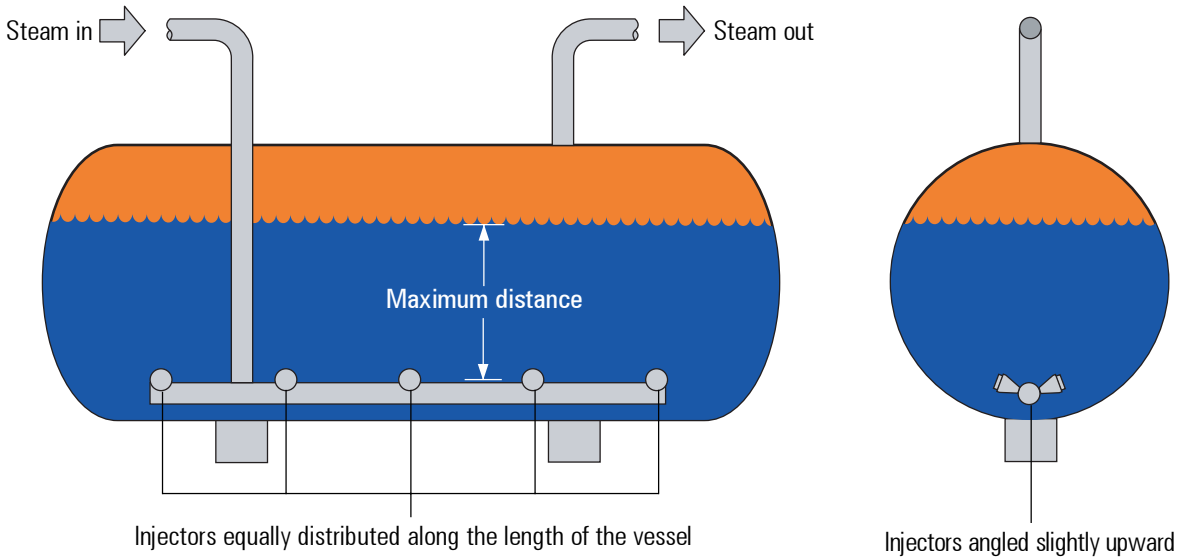


Fig. 3.22.6 Installation of injectors in a steam accumulator

The discharge from the injectors will be very hot water and steam, possibly with some condensing steam bubbles, at very high velocity, promoting turbulence and mixing in the water mass. They should not discharge directly against, or close to, the walls of the vessel. Angled installation may therefore be advisable. Ideally, they should also be angled in different directions to assist with more even distribution.

A nominal arrangement is shown in Figure 3.22.7.

In very long vessels, more regular distribution may be achieved if two or more inlet pipes are used. In such cases, it is very important that the inlet pipes are carefully manifolded together from the supply main.

All the injectors should be installed as low down in the accumulator as possible to ensure the maximum possible liquid head above them. It may also be appropriate to install the injectors at a slight angle to avoid erosion of the vessel.



Fig. 3.22.7 A steam injector

Returning to Example 3.22.2:

Boiler pressure (P_1) = 10 bar g

Minimum accumulator pressure (P_2) = 6 bar g

ΔP(maximum) = 10 - 6 = 4 bar

Flowrate = Boiler maximum continuous rating (5 000 kg/h on example)

Manufacturers' sizing tables will give the K_{VS} value of the nozzles (see Table 3.22.2)

Table 3.22.2 Steam injector capacity index values

Injector Size	IN15	IN25M	IN40M
K_{VS}	1.4	5.8	15.3

Using the data from Table 3.22.2 and referring to Figure 3.22.8, an extract from the saturated steam sizing chart Figure 3.22.9:

1. Draw a line horizontally to the right across from the 'x' axis at 11 bar a (10 bar g) until it intersects the critical pressure drop line, point (**A**).
2. Draw a line vertically down the chart from point (**A**) until it intersects the K_{VS} value of the injector, point (**B**), (For example K_{VS} 5.8 for an IM25M injector).
3. Draw a line horizontally to the left, until it intersects the 'y' axis, point (**C**). The value shown will be the capacity of the injector. (Approximately 760 kg/h for this example).

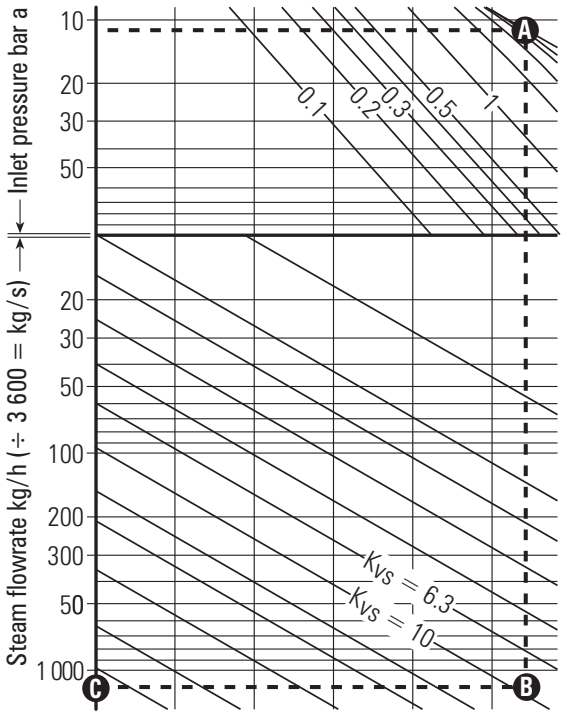


Fig. 3.22.8 Extract from saturated steam sizing chart

The flowrate may also be calculated using Equation 3.21.1:

$$\dot{m}_s = 12 K_v P_1 \sqrt{1 - 5.67 (0.42 - \chi)^2}$$ **Equation 3.21.1**

Where:
$\dot{m}_s$ = Steam flow (kg/h)
K_V = Capacity index of injector
P_1 = Boiler pressure bar a
χ = Pressure drop ratio $\Delta P/P_1$

Sizing and quantifying the injectors

The above exercise gives a capacity of 760 kg/h for one injector; but this only relates to the start of the charging period, when the vessel pressure is at its lowest, and the injector capacity is at its highest.

It must be remembered that, as more steam is injected into the vessel, the vessel pressure will rise, effectively reducing the injectors' capacities, until the vessel pressure may eventually equalise with the boiler pressure, and no flow can take place.

Because of this, it is not practical to use the one (highest) flowrate, 760 kg/h in this example.

Instead, it is necessary to find the mean injection rate over the charging period.

This can be done by using Equation 3.21.1 to calculate the flow at different vessel pressures. (The Spirax Sarco Engineering Support Centre has a valve sizing utility, which can be used to calculate the injector capacities easily from the injector Kv values – see http://www.spiraxsarco.com/esc)

In this example, the vessel pressure will vary between 6 bar g and 10 bar g. The greater the number of pressures taken, the greater the accuracy but, in general, taking ranges at 10% of the difference will give a reliable mean value. Table 3.22.3 refers, using an IN25 injector (1") with a K_V of 5.8.

Table 3.22.3

Boiler pressure bar g	Accumulator pressure bar g	Injector K_V	Injector capacity kg/h
10	6.0	5.8	759
10	6.4	5.8	747
10	6.8	5.8	729
10	7.2	5.8	704
10	7.6	5.8	671
10	8.0	5.8	631
10	8.4	5.8	579
10	8.8	5.8	515
10	9.2	5.8	431
10	9.6	5.8	311
10	10.0	5.8	0
			TOTAL = 6 076

The total flow of 6 076 kg/h is divided by the number of entries. it must be remembered to include the zero entry as well; hence there are eleven entries to consider.

$$\text{The mean injector flowrate over the charging period} = \frac{6\,076 \text{ kg/h}}{11}$$

$$\textbf{The mean injector flowrate over the charging period} = \textbf{553 kg/h}$$

It can be seen that the mean flowrate of 553 kg/h is somewhat less than the maximum capacity of 759. If the maximum capacity were used to quantify the number of injectors, then not enough injectors would be chosen.

The number of injectors required can be determined by dividing the steam flow by the amount a single injector can supply.

For example: $$\frac{\text{Capacity required}}{\text{Injector capacity}} = \frac{5\,000 \text{ kg/h}}{553 \text{ kg/h}} = (9.1)$$ Use 10 injectors.

Note: A number of smaller injectors would be preferable to one large injector to ensure proper mixing within the steam accumulator.

This sizing chart is empirical and should not be used for critical applications

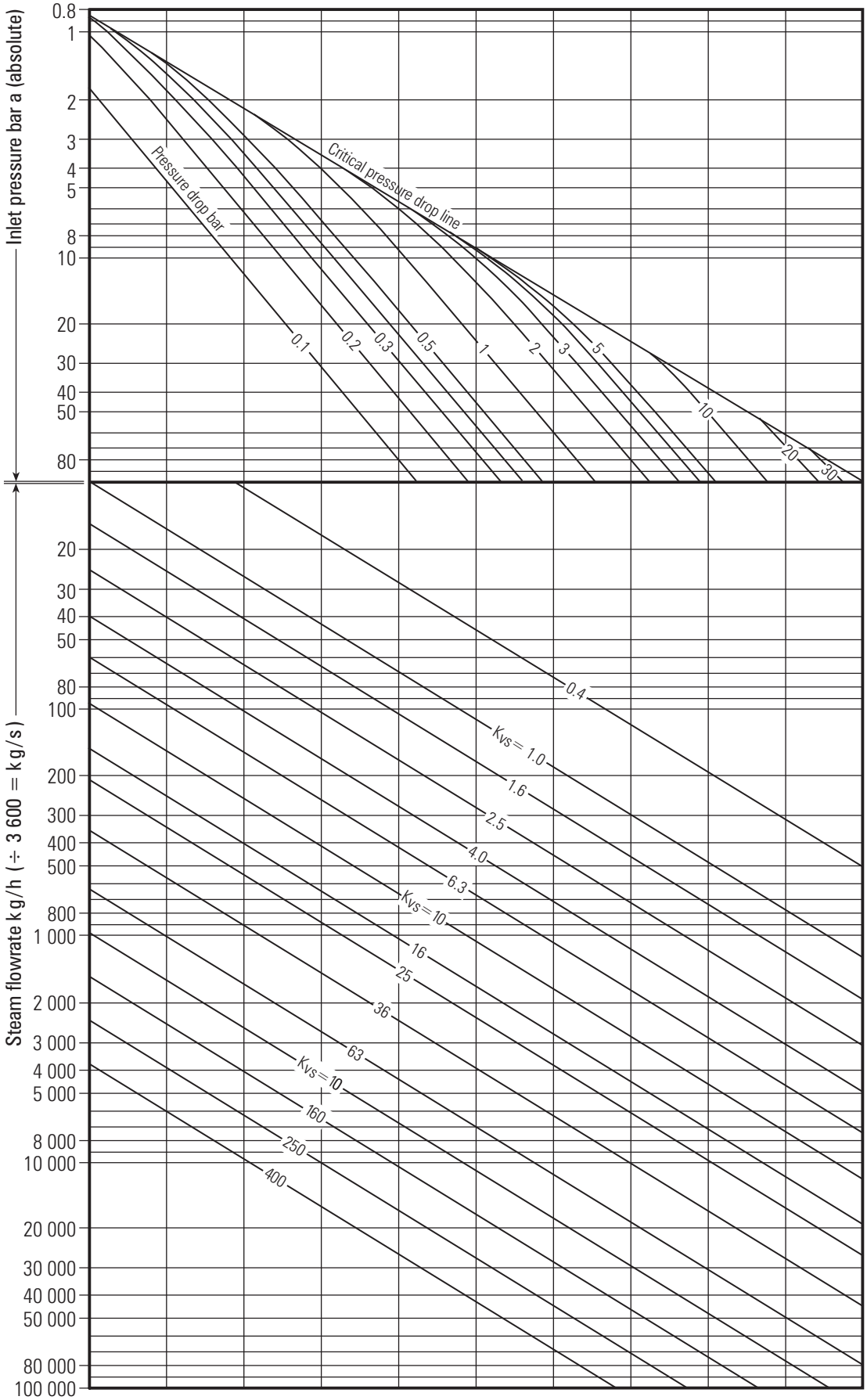


Fig. 3.22.9 Saturated steam sizing chart

Calculating the time required to recharge the vessel

From the load patterns shown in Figure 3.22.4, it has been shown that the minimum time between charge cycles is 95 minutes. It is now necessary to check that the vessel can be recharged in less time than this.

It has been shown that the quantity of steam used during the discharge period is 2 650 kg.

The surplus amount of steam available during the recharging period is actually the boiler capacity minus the mean off-peak demand.

In this example,

Steam available for recharging = 5 000 kg - 2 953 kg

Steam available for recharging = 2 047 kg

The time required for recharging is proportional to the ratio of the steam used to the steam available:

$$\text{Required recharging time} = \frac{2\,650 \text{ kg}}{2\,047 \text{ kg}} \times 60 \frac{\text{minutes}}{\text{hour}}$$

Required recharging time = 78 minutes

As the required recharging time is less than the time between the shortest overload cycle of 95 minutes, the balance between the overload time and the recharging time can be satisfied by the accumulator.

Therefore, the accumulator size of 7 metres long by 4 metres diameter provides sufficient capacity for this particular example.

Pressure gauge

A suitably ranged pressure gauge is required to show the pressure within the steam accumulator.

Ideally it should be marked to show:

- Minimum pressure (plant steam pressure).
- Maximum pressure (boiler steam pressure).
- Vessel maximum working pressure.

Safety valve

If the maximum working pressure of the accumulator is equal to, or greater than that of the boiler, then a safety valve(s) may not be required.

However, the user may be concerned about other less obvious scenarios. For example, in the event of a plant fire, if the accumulator were fully charged and all the inlets and outlets were closed, the pressure in the accumulator could rise. A discussion with the insurance inspector would be essential before a decision is made.

As with all safety valve installations, the discharge should be to a safe area through an adequately sized vent pipe, which is properly drained.

Air vent and vacuum breaker

When the steam accumulator starts from cold, the steam space is full of air. This air has no heat value, in fact it will adversely affect the steam plant performance (as demonstrated in Dalton's Law) and also have the effect of blanketing heat exchange surfaces. The air will also give rise to corrosion in the condensate system.

The air may be purged using a simple cock, normally left open until the steam accumulator is pressurised to about 0.5 bar. An alternative to the cock is a balanced pressure air vent, which not only relieves the boiler plant operator of the task of manually purging air (and hence ensuring that it is actually done), but is also more dependable in purging any other gases which accumulate in the vessel during use.

Conversely, when the steam accumulator is taken off line, the steam in the steam space condenses and leaves a vacuum. This vacuum causes pressure to be exerted on the vessel from the outside, and can result in air leaking in through the inspection doors. A vacuum breaker will avoid this situation.

Drain cock

This valve would be used to drain the vessel for maintenance and inspection work.

A DN40 valve would be suitable for the size of the accumulator in Example 3.22.2.

Overflow

A ball float trap with integral thermostatic air vent must be fitted as in Figure 3.22.10. When installed as shown, the water level inside the accumulator will not rise above this point because the trap will operate as an automatic overflow valve. When the water level drops, that is, when steam is drawn off at a faster rate than it is replaced, the trap will automatically close to prevent the escape of steam.

The use of a float trap with an integral thermostatic capsule as a level limiting device, offers the additional advantage of air venting.

The trap should be installed near to the gauge glass. The discharge from the trap should be directed back to the boiler feedtank, taking care to avoid excessive backpressure or lift.

The size of float/thermostatic trap will vary according to the size of the accumulator, and would typically be size DN32 or DN40 for Example 3.22.2.

Water level gauge

The variation in level within the steam accumulator will not be great because only 5% (approximately) of the mass of water will flash to steam, however, some means of viewing the water level is essential. Clearly the gauge should be rated to operate at the steam accumulator maximum working pressure. However, from a stock holding and plant standardisation point of view, there is some merit in using a gauge the same as the boiler.

Only a single gauge glass is required.

Pressure reducing station

A pressure reducing station is fitted to the discharge. As the pressure reducing valve opens to maintain the downstream pressure, a reduction in pressure occurs in the steam accumulator causing some of the water to flash to steam.

The pressure reducing valve should be sized on the following data:

P_1 = Accumulator pressure (6 bar g on example)
P_2 = Plant pressure (5 bar g on example)
ΔP = 6 - 5 = 1 bar
Flowrate = Maximum overload flowrate (12 000 kg/h on example)

An appropriate valve can now be selected either from the manufacturer's sizing charts or using the saturated steam sizing chart shown in Figure 3.22.9.

For sizes up to DN80, a pilot operated self-acting valve would be suitable, whilst a pneumatically actuated control valve is appropriate on larger sizes.

Pipework

It is appropriate at this point to check that the pipework between the steam accumulator pressure reducing station and the plant is adequately sized. This pipe should be sized as per normal practice on a steam velocity of 25 to 30 m/s, but using the peak flowrate from the steam accumulator at the plant pressure, in this instance 5 bar g.

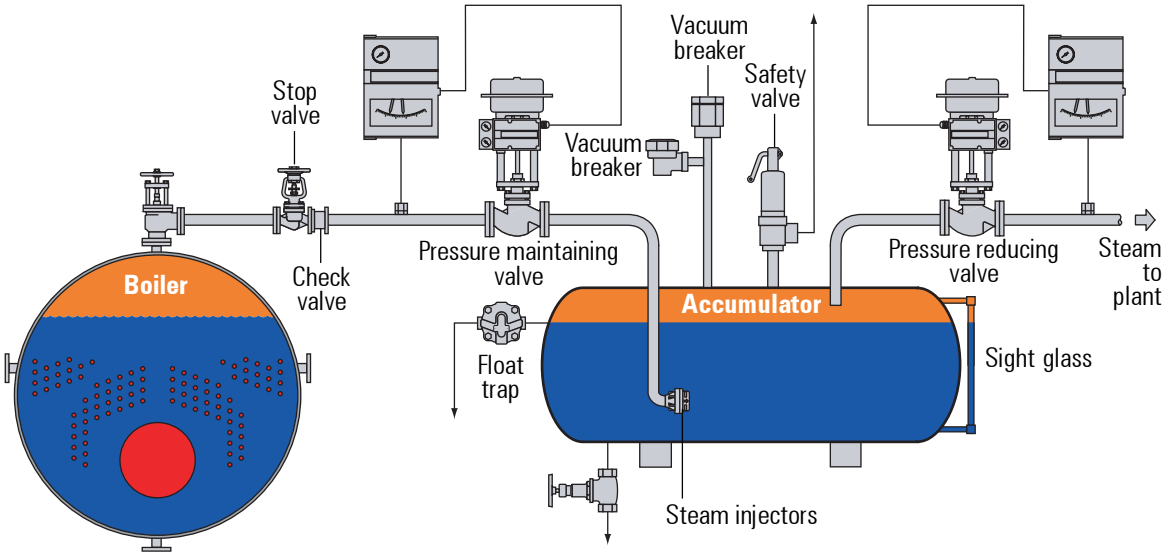


Fig. 3.22.10 A steam accumulator with fittings

Typical arrangements of steam accumulators:

Figure 3.22.11 shows all the steam generated by the boiler plant passing through the steam accumulator. This is the more modern generally preferred arrangement.

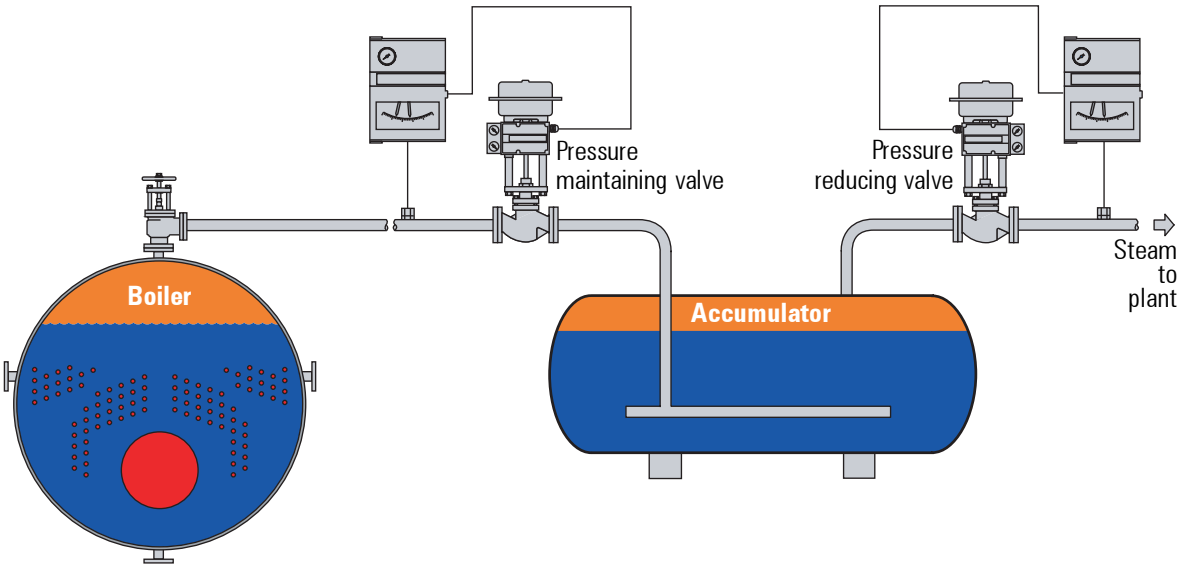


Fig. 3.22.11 Steam accumulator adjacent to the boiler

The arrangement shown in Figure 3.22.12 was more commonly used in the past and is still useful when the steam accumulator must be sited some distance from the steam main. However, the check valves should be checked regularly, as a combination of 'sticking' and 'passing' valves can result in steam being charged to the steam accumulator above the steam surface, which brings no benefit.

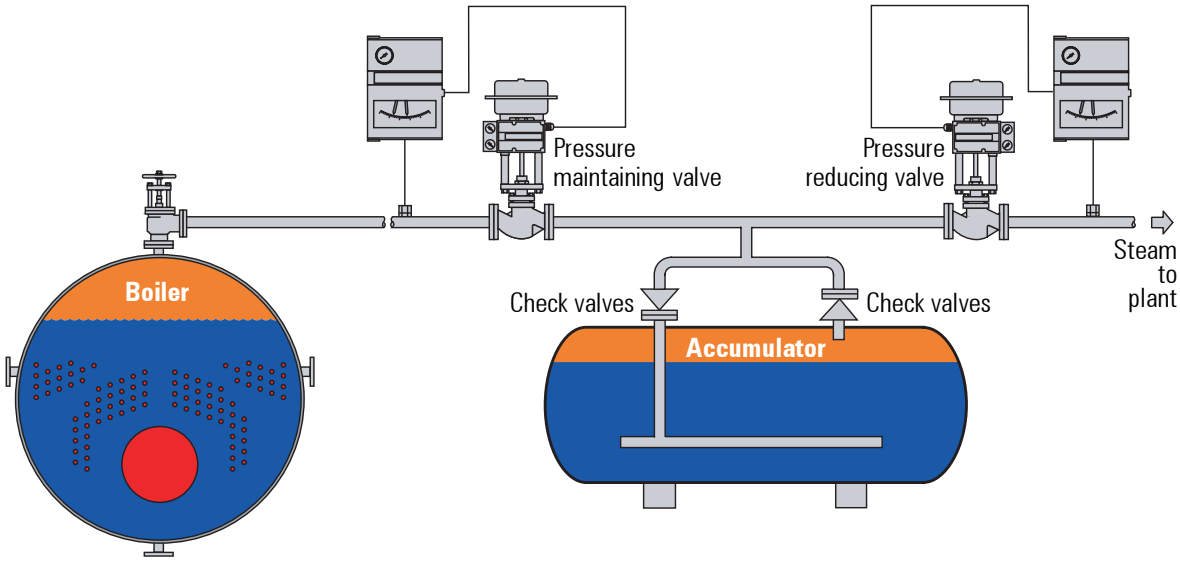


Fig. 3.22.12 Steam accumulator remote from the boiler

Figure 3.22.13 shows an arrangement where steam at boiler pressure is required as well as steam at a lower pressure.

Some process applications cannot tolerate low pressure steam, and steam at boiler pressure may be required at all times (typically for a drying process). If a peak load is caused by the high pressure users, the pressure maintaining valve in Figure 3.22.13 would sense a pressure drop, and modulate towards its seat, thereby reserving high pressure steam for the high pressure users, thus leaving the steam accumulator to supply the low pressure demand during this period. In this way the system supplies a low pressure fluctuating load via the steam accumulator and the maximum possible flowrate for the high pressure load is ensured by the action of the pressure maintaing valve.

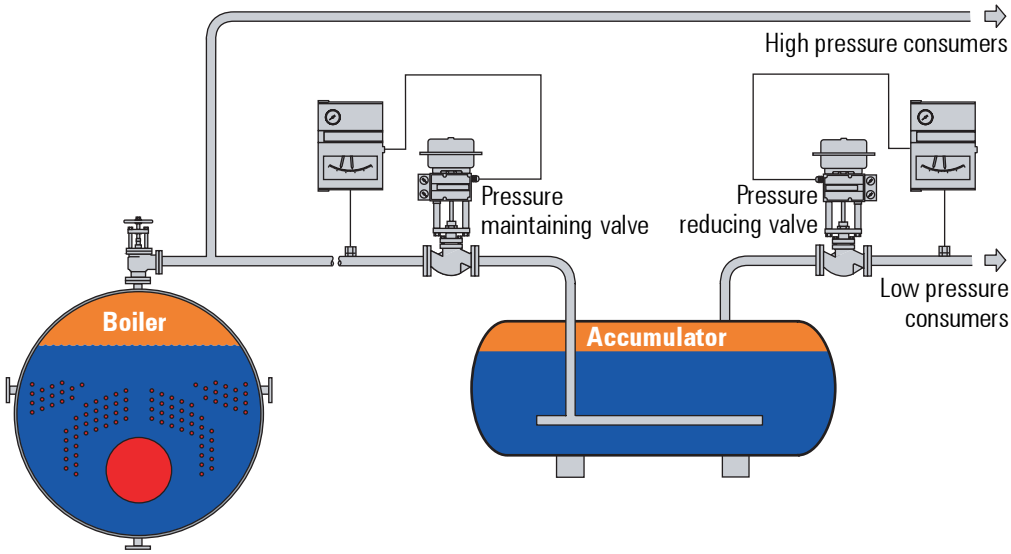


Fig. 3.22.13 Steam required at boiler pressure as well as at lower pressure

In Figure 3.22.14, the boiler is steaming at its normal design pressure, for example 10 bar, and the steam passes to variable loads which require not more than, for example 5 bar. Pressure reducing valve **A** is reducing pressure between the boiler header and the distribution main in the plant, responding to the pressure sensed in the 5 bar line.

If the steam demand should exceed the capacity of this supply from the boiler, and the pressure in the low pressure main falls below, for example 4.8 bar, valve **B** will begin to open and supplement the supply. This draws steam from the steam accumulator, and over a sustained period the steam accumulator pressure will fall. Valve **B** is responding to the downstream pressure in the distribution main, thus acting as a pressure reducing valve also. Its capacity should match the discharge rate permitted for the steam accumulator, and it will be smaller than pressure reducing valve **A**.

Valve **C** is a pressure-maintaining valve, responding to the boiler pressure. If the pressure rises because of reduced demand from the plant, pressure-maintaining valve **C** opens. Steam is then admitted to the steam accumulator that is recharged towards its maximum pressure, a little below boiler pressure. Pressure reducing valve **B** will be closed at this time because the plant is receiving sufficient steam through the (partially closed) pressure reducing valve **A**.

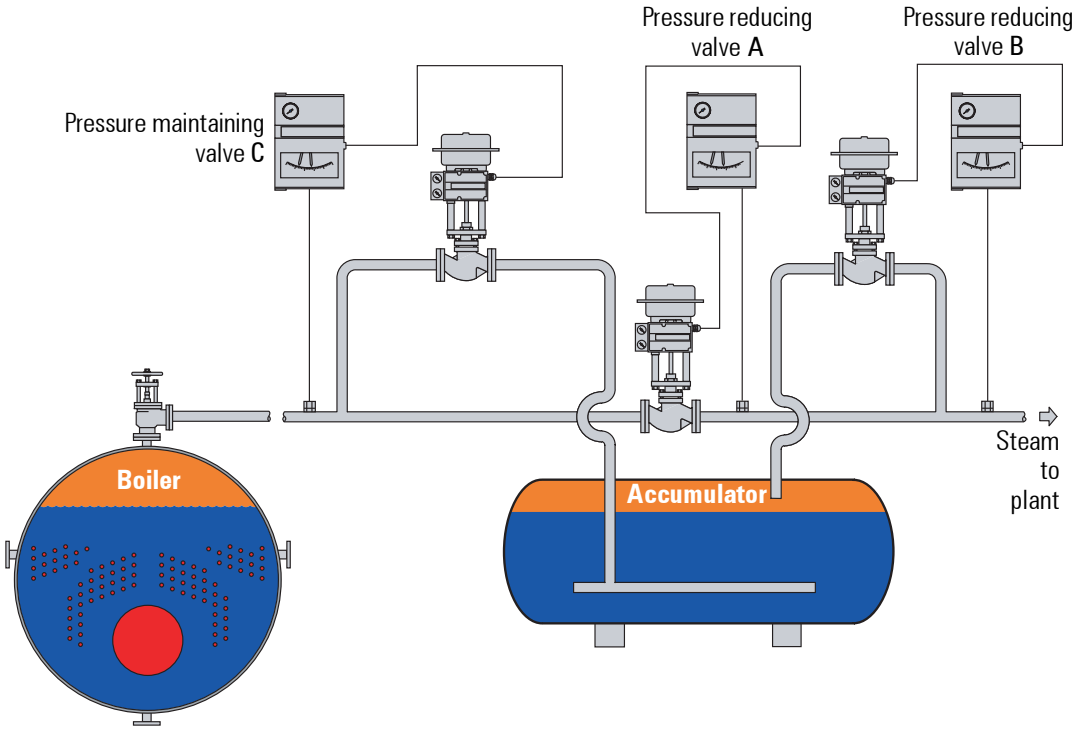


Fig. 3.22.14 Alternative standard arrangement

Practical considerations for steam accumulators

Bypasses

In any plant, the engineering manager must endeavour to provide at least a minimum service in the event that the steam accumulator and its associated equipment either requires maintenance or breaks down.

This will include the provision of adequate and safe isolation of the accumulator with valves, and perhaps some means of protecting the boiler from overload if large changes in demand cannot be avoided. The most obvious solution here is a stand-by pressure-maintaining valve.

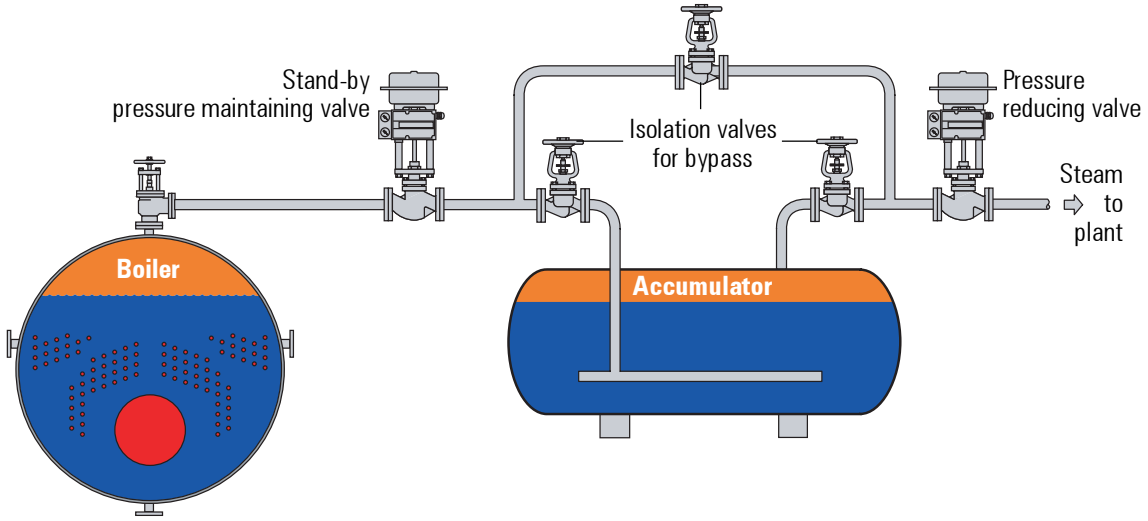


Fig. 3.22.15 Accumulator bypass arrangement (valve controls not shown)

Effects on the boiler firing rate

The steam accumulator and pressure maintaining valve together protect the boiler from overload conditions and allow the boiler to operate properly up to its design rating. This is important to achieve good efficiencies and at the same time to supply clean, dry, saturated steam. Figures 3.22.16 and 3.22.17 illustrate respectively the firing rate without a steam accumulator and the firing rate with a steam accumulator.

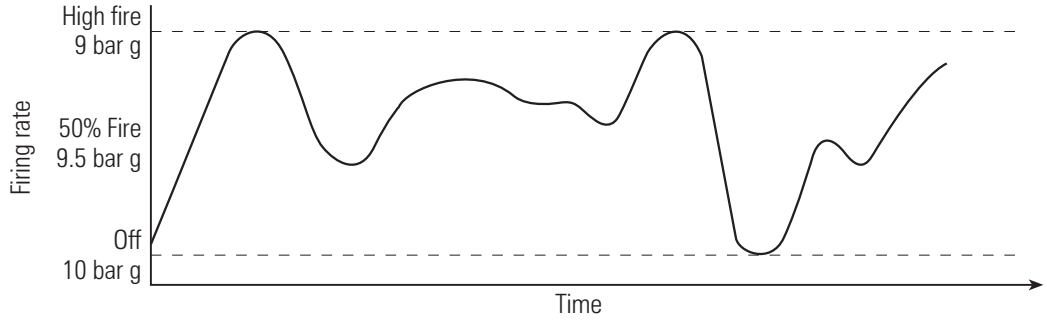


Fig. 3.22.16 Boiler without a steam accumulator

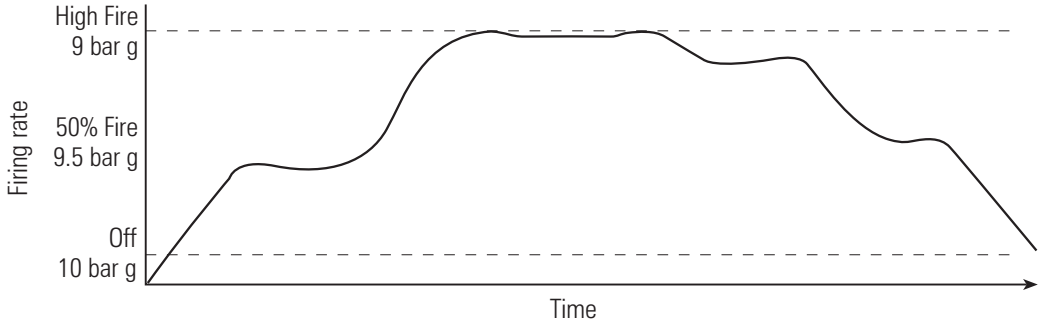


Fig. 3.22.17 Boiler with a steam accumulator and surplussing regulator

Steam quality

When correctly designed and operated, steam from a steam accumulator is always clean, and has a dryness fraction quite close to 1. The steam accumulator is designed with a large water surface and sufficient steam space in order to produce high quality steam almost instantaneously during periods of peak demand. In the case of some vertical steam accumulators the steam space is enlarged to compensate for the smaller water surface.

Water

Water in the steam accumulator is steam that has condensed and is therefore clean and pure, with a typical TDS level of 20 - 100 ppm (compared with a shell boiler TDS of seldom less than 2 000 ppm) which promotes a clean and comparatively stable water surface. Steam accumulators are sometimes used to ensure clean steam is provided where steam is in direct contact with the product; as in hospital and industrial sterilisers, textile finishing and certain applications within the food and drinks industry.

Once the accumulator has been filled with water, and at normal running conditions, water additions and overflow rates are very small indeed.

- If superheated steam is used, the amount of water to be added would be related to the amount of superheat, but since the specific heat of superheated steam is lower than water, it will have a smaller effect on changes in water level.
- If saturated steam is used, the increase in water level is simply a function of heat loss from the vessel. With proper insulation, heat loss is minimal, so the increase in water level, and hence overflow through the steam trap (used as a level limiting device) is also minimal.

Steam accumulator designs

The steam accumulators described and illustrated in this Module have been large and of a horizontal configuration. Steam accumulators are always designed and manufactured to suit the application, and vessels of only 1 m diameter are not uncommon. It is also usual for the smaller steam accumulators to be of a vertical configuration (although large vertical steam accumulators exist). Either configuration can maintain the same values of storage and discharge rate, and it may be easier to find space for a vertical unit.

The storage vessel

This is usually the most expensive part of a steam accumulator system, and will be individually designed for each application. It must be designed to hold the water/steam at the temperatures that are required for the plant. For industrial plant this typically means between 5 and 30 bar, although power station units may be rated up to 150 bar.

Typically the ratio of diameter to total length is between 1.4 to 1.6, but this can vary substantially depending on site conditions.

Steam accumulators are generally cylindrical in form with elliptical ends, as this is structurally the most effective shape. They will be manufactured from boiler plate. In Europe the design and construction will comply with the European Pressure Equipment Directive 97/23/EC.

The greater the acceptable pressure differential between the boiler pressure and the plant pressure, the greater the proportion of flash steam, and hence the lower the live steam capacity required.

In addition to the live storage capacity, the vessel must have:

- Sufficient water in the bottom of the vessel, under minimum conditions, to accommodate and cover the steam injectors
- Sufficient clearance above the water under fully charged conditions to give a reasonable surface area for steam release. This is important because the instantaneous steam release velocity alone could be the final criteria if the peak loads are heavy and abrupt

Justifying the cost of an accumulator

There are several ways in which the capital cost of an accumulator installation can be justified, and they will often pay back in a short period of time. The following points should be considered during an initial analysis.

- Compare the capital cost of a boiler-only installation to meet the peak demand, with that of a smaller boiler used with an accumulator
- Estimate the fuel savings as a result of a smaller boiler operating closer to its maximum output and on a steadier load. In a recent case study, a brewery calculated a 10% fuel saving and a payback period of approximately 18 months
- As a result of levelling out the peaks and troughs of steam generation, determine if the unit cost of the fuel will be less. It may then be possible to contract for a lower maximum supply rate
- Estimate the financial advantage of reduced maintenance on boiler plant, steam control valves, and the steam using equipment. These benefits will result from a steadier boiler load and better quality steam

Conclusion

Steam accumulators are not old fashioned relics from the past. Indeed, far from it. Steam accumulators have been installed throughout modern industry including bio-technology, hospital and industrial sterilisation, product testing rigs, printing and food manufacturing, as well as more traditional industries such as breweries and dyehouses.

Modern boilers have become smaller and there is also an increase in the use of small water-tube boilers, coil boilers and annular boilers, all of which are efficient, but which reduce the thermal capacity of the system, and make it vulnerable to peak load problems.

There are many further applications for steam accumulators. For long term peaks which the boiler plant must ultimately handle, a steam accumulator can be used to store, for example, 5 minutes of the peak flowrate, allowing time for the boiler plant to reach the appropriate output safely. Steam accumulators can also be used with electrode or immersion heater boilers so that steam can be generated off peak, stored, and used during peak times. The possibilities are endless.

In summary, the steam accumulator is an efficient tool, as it may well provide the most cost effective way of supplying steam to a batch process.

Acknowledgement

Spirax Sarco acknowledges the help and information provided by:
Thermsave Engineering (UK) Ltd., Dinnington, South Yorkshire. S25 3QX

Questions

1. After filling, how is the level of water maintained in a steam accumulator?

a| With a conductivity or capacitance level probe ☐

b| With an overflow pipe ☐

c| With a steam trap and overflow pipe ☐

d| With a float level control ☐

2. How is the water that flashes off in an accumulator replaced?

a| By the condensed steam heating the incoming water ☐

b| By make-up water from the boiler feedpump ☐

c| By a connection to the boiler and a self-adjusting level ☐

d| The water level does not change ☐

3. What would be the effect of a peak demand for steam above a boiler's maximum continuous rating?

a| A drop in pressure and a rise in water level ☐

b| Foaming within the boiler and carryover ☐

c| A rise in pressure as the water level rises, and burner shut-off ☐

d| A drop in pressure and water carryover ☐

4. What is the purpose of a surplussing valve on the steam main leaving a boiler?

a| To remove any carryover of water ☐

b| To open further to meet any peak demand for steam ☐

c| To reduce the steam pressure and overloading of the boiler ☐

d| To maintain pressure in the boiler ☐

5. A steam accumulator provides steam to meet peak demand

a| From the water flashing off ☐

b| By steam generated from the steam injected into the water ☐

c| From steam stored in the vessel ☐

d| From steam flowing into and out of the vessel from the boiler ☐

6. An accumulator stores 30 tonnes of water and operates at 6 bar g. The boiler operates at 12 bar g. What will be the steaming rate from the accumulator over a 15 minute period?

a| 1 707 kg/h ☐

b| 6 854 kg/h ☐

c| 5 928 kg/h ☐

d| 7 830 kg/h ☐

Answers

1: c, 2: a, 3: d, 4: d, 5: a, 6: b

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Module 4.1

Fluids and Flow

Introduction

'When you can measure what you are speaking about and express it in numbers, you know something about it; but when you cannot measure it, when you cannot express it in numbers, your knowledge is of a meagre and unsatisfactory kind'.
William Thomson (Lord Kelvin) 1824 - 1907

Many industrial and commercial businesses have now recognised the value of:

- Energy cost accounting.
- Energy conservation.
- Monitoring and targeting techniques.

These tools enable greater energy efficiency.

Steam is not the easiest media to measure. The objective of this Block is to achieve a greater understanding of the requirements to enable the accurate and reliable measurement of steam flowrate.

Most flowmeters currently available to measure the flow of steam have been designed for measuring the flow of various liquids and gases. Very few have been developed specifically for measuring the flow of steam.

Spirax Sarco wishes to thank the EEBPP (Energy Efficiency Best Practice Programme) of ETSU for contributing to some parts of this Block.

Fundamentals and basic data of Fluid and Flow

Why measure steam?

Steam flowmeters cannot be evaluated in the same way as other items of energy saving equipment or energy saving schemes. The steam flowmeter is an essential tool for good steam housekeeping. It provides the knowledge of steam usage and cost which is vital to an efficiently operated plant or building. The main benefits for using steam flowmetering include:

- Plant efficiency.
- Energy efficiency.
- Process control.
- Costing and custody.

Plant efficiency

A good steam flowmeter will indicate the flowrate of steam to a plant item over the full range of its operation, i.e. from when machinery is switched off to when plant is loaded to capacity. By analysing the relationship between steam flow and production, optimum working practices can be determined.

The flowmeter will also show the deterioration of plant over time, allowing optimum plant cleaning or replacement to be carried out.

The flowmeter may also be used to:

- Track steam demand and changing trends.
- Establish peak steam usage times.
- Identify sections or items of plant that are major steam users.

This may lead to changes in production methods to ensure economical steam usage. It can also reduce problems associated with peak loads on the boiler plant.

Energy efficiency
Steam flowmeters can be used to monitor the results of energy saving schemes and to compare the efficiency of one piece of plant with another.

Process control
The output signal from a proper steam flowmetering system can be used to control the quantity of steam being supplied to a process, and indicate that it is at the correct temperature and pressure. Also, by monitoring the rate of increase of flow at start-up, a steam flowmeter can be used in conjunction with a control valve to provide a slow warm-up function.

Costing and custody
Steam flowmeters can measure steam usage (and thus steam cost) either centrally or at individual user points. Steam can be costed as a raw material at various stages of the production process thus allowing the true cost of individual product lines to be calculated.

To understand flowmetering, it might be useful to delve into some basic theory on fluid mechanics, the characteristics of the fluid to be metered, and the way in which it travels through pipework systems.

Fluid characteristics

Every fluid has a unique set of characteristics, including:

- Density.
- Dynamic viscosity.
- Kinematic viscosity.

Density
This has already been discussed in Block 2, Steam Engineering Principles and Heat Transfer, however, because of its importance, relevant points are repeated here.

Density (ρ) defines the mass (m) per unit volume (V) of a substance (see Equation 2.1.2).

$$\text{Density } (\rho) = \frac{\text{Mass (m) kg}}{\text{Volume (V) m}^3} = \frac{1}{\text{Specific volume } (v_g)} \qquad \textbf{Equation 2.1.2}$$

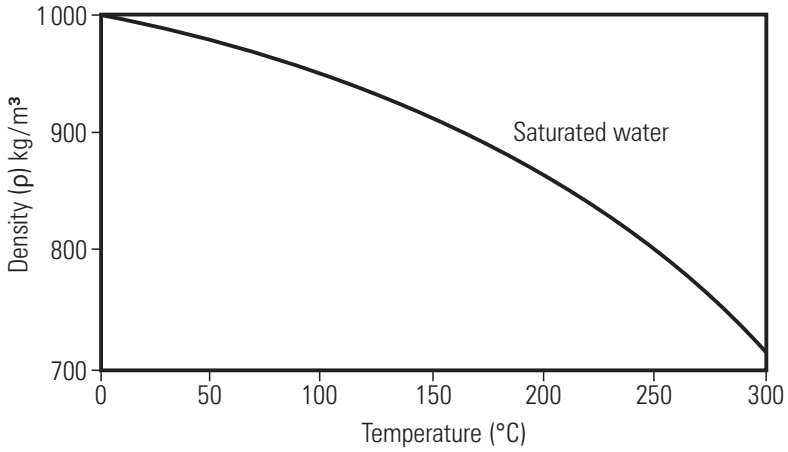
Steam tables will usually provide the specific volume (v_g) of steam at various pressures/temperatures, and is defined as the volume per unit mass:

$$\text{Specific volume } (v_g) = \frac{\text{Volume (V)}}{\text{Mass (m)}} = \text{m}^3/\text{kg}$$

From this it can be seen that density (ρ) is the inverse of specific volume (v_g):

$$\text{Density } (\rho) = \frac{1}{\text{Specific volume } (v_g)} = \text{kg}/\text{m}^3$$

The density of both saturated water and saturated steam vary with temperature. This is illustrated in Figure 4.1.1.



Note: The density of saturated steam increases with temperature (it is a gas, and is compressible) whilst the density of saturated water decreases with temperature (it is a liquid which expands).

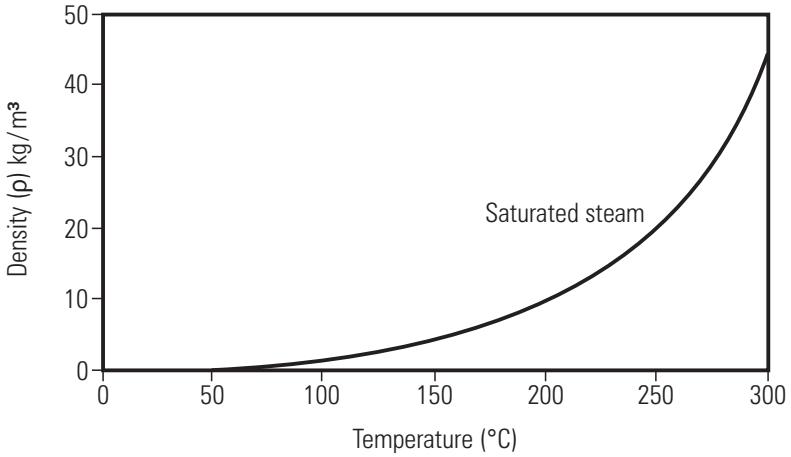


Fig. 4.1.1 The density (ρ) of saturated water (ρ_f) and saturated steam (ρ_g) at various temperatures

Dynamic viscosity

This is the internal property that a fluid possesses which resists flow. If a fluid has a high viscosity (e.g. heavy oil) it strongly resists flow. Also, a highly viscous fluid will require more energy to push it through a pipe than a fluid with a low viscosity.

There are a number of ways of measuring viscosity, including attaching a torque wrench to a paddle and twisting it in the fluid, or measuring how quickly a fluid pours through an orifice.

A simple school laboratory experiment clearly demonstrates viscosity and the units used:

A sphere is allowed to fall through a fluid under the influence of gravity. The measurement of the distance (d) through which the sphere falls, and the time (t) taken to fall, are used to determine the velocity (u).

The following equation is then used to determine the dynamic viscosity:

$$\text{Dynamic viscosity } (\mu) = \frac{2\,\Delta\rho\,g\,r^2}{9\,u} \qquad \textbf{Equation 4.1.1}$$

Where:

μ = Absolute (or dynamic) viscosity (Pa s)
$\Delta\rho$ = Difference in density between the sphere and the liquid (kg/m³)
g = Acceleration due to gravity (9.81 m/s²)
r = Radius of sphere (m)

$$u = \text{Velocity} \left(\frac{\text{d - Distance sphere falls (m)}}{\text{t - Time taken to fall (seconds)}} \right)$$

There are three important notes to make:

1. The result of Equation 4.1.1 is termed the absolute or dynamic viscosity of the fluid and is measured in pascal seconds. Dynamic viscosity is also expressed as 'viscous force'.
2. The physical elements of the equation give a resultant in kg/m, however, the constants (2 and 9) take into account both experimental data and the conversion of units to pascal seconds (Pa s).
3. Some publications give values for absolute viscosity or dynamic viscosity in centipoise (cP), e.g.: 1 cP = 10^{-3} Pa s

Example 4.1.1
It takes 0.7 seconds for a 20 mm diameter steel (density 7 800 kg/m³) ball to fall 1 metre through oil at 20°C (density = 920 kg/m³).

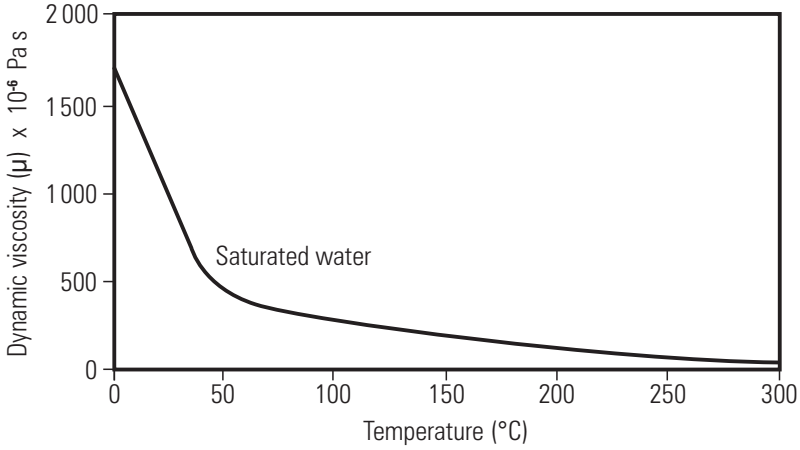
Determine the viscosity where:
$\Delta\rho$ = Difference in density between the sphere (7 800) and the liquid (920) = 6 880 kg/m³
g = Acceleration due to gravity = 9.81 m/s²
r = Radius of sphere = 0.01 m

u = Velocity $\left(\frac{d}{t} = \frac{1}{0.7}\right)$ = 1.43 m/s

$$\text{Dynamic viscosity } (\mu) = \frac{2\ \Delta\rho\ g\ r^2}{9\ u}$$

$$\text{Dynamic viscosity } (\mu) = \frac{2\ \text{x}\ 6\,880\ \text{x}\ 9.81\ \text{x}\ 0.01^2}{9\ \text{x}\ 1.43} = 1.05\ \textbf{Pa s}$$

Values for the dynamic viscosity of saturated steam and water at various temperatures are given in steam tables, and can be seen plotted in Figure 4.1.2.



Note: The values for saturated water decrease with temperature, whilst those for saturated steam increase with temperature.

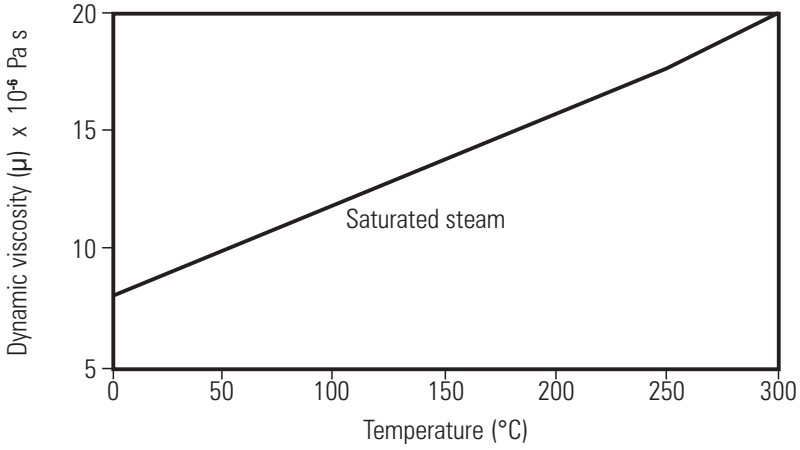


Fig. 4.1.2 The dynamic viscosity of saturated water (μ_f) and saturated steam (μ_g) at various temperatures

Kinematic viscosity
This expresses the relationship between absolute (or dynamic) viscosity and the density of the fluid (see Equation 4.1.2).

$$\text{Kinematic viscosity } (\nu) = \frac{\text{Dynamic viscosity } (\mu) \times 10^3}{\text{Density } (\rho)} \qquad \textbf{Equation 4.1.2}$$

Where:
Kinematic viscosity is in centistokes
Dynamic viscosity is in Pa s
Density is in kg/m³

Example 4.1.2
In Example 4.1.1, the density of the oil is given to be 920 kg/m³ - Now determine the kinematic viscosity:

$$\text{Kinematic viscosity } (\nu) = \frac{1.05 \times 10^3}{920} = 1.14 \text{ centistokes (cSt)}$$

Reynolds number (R_e)
The factors introduced above all have an effect on fluid flow in pipes. They are all drawn together in one dimensionless quantity to express the characteristics of flow, i.e. the Reynolds number (R_e).

$$\text{Reynolds number } (R_e) = \frac{\rho \, u \, D}{\mu} \qquad \textbf{Equation 4.1.3}$$

Where:
ρ = Density (kg/m³)
u = Mean velocity in the pipe (m/s)
D = Internal pipe diameter (m)
μ = Dynamic viscosity (Pa s)

Analysis of the equation will show that all the units cancel, and Reynolds number (R_e) is therefore dimensionless.

Evaluating the Reynolds relationship:

- For a particular fluid, if the velocity is low, the resultant Reynolds number is low.
- If another fluid with a similar density, but with a higher dynamic viscosity is transported through the same pipe at the same velocity, the Reynolds number is reduced.
- For a given system where the pipe size, the dynamic viscosity (and by implication, temperature) remain constant, the Reynolds number is directly proportional to velocity.

Example 4.1.3
The fluid used in Examples 4.1.1 and 4.1.2 is pumped at 20 m/s through a 100 mm bore pipe.
Determine the Reynolds number (R_e) by using Equation 4.1.3 where: ρ = 920 kg/m³
μ = 1.05 Pa s

$$\text{Reynolds number } (R_e) = \frac{\rho \, u \, D}{\mu} \qquad \textbf{Equation 4.1.3}$$

$$\textbf{Reynolds number } (R_e) = \frac{920 \times 20 \times 0.1}{1.05} = \mathbf{1\,752}$$

From looking at the above Reynolds number it can be seen that the flow is in the laminar region (see Figure 4.1.7).

Flow regimes

If the effects of viscosity and pipe friction are ignored, a fluid would travel through a pipe in a uniform velocity across the diameter of the pipe. The 'velocity profile' would appear as shown in Figure 4.1.3:

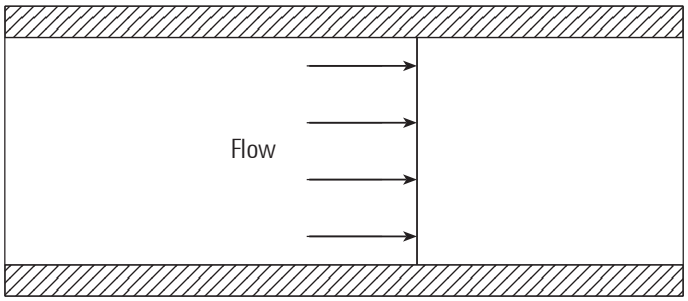


Fig. 4.1.3 Velocity profile ignoring viscosity and friction

However, this is very much an ideal case and, in practice, viscosity affects the flowrate of the fluid and works together with the pipe friction to further decrease the flowrate of the fluid near the pipe wall. This is clearly illustrated in Figure 4.1.4:

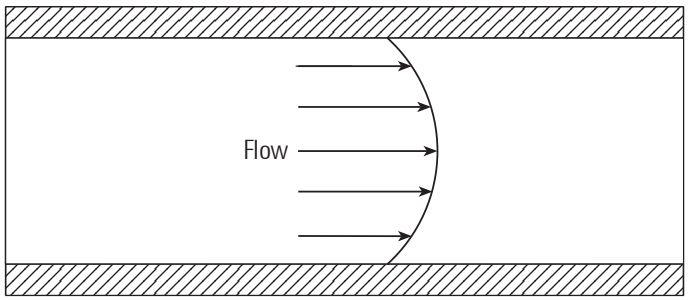


Fig. 4.1.4 Velocity profile with viscosity and friction

At low Reynolds numbers (2 300 and below) flow is termed 'laminar', that is, all motion occurs along the axis of the pipe. Under these conditions the friction of the fluid against the pipe wall means that the highest fluid velocity will occur at the centre of the pipe (see Figure 4.1.5).

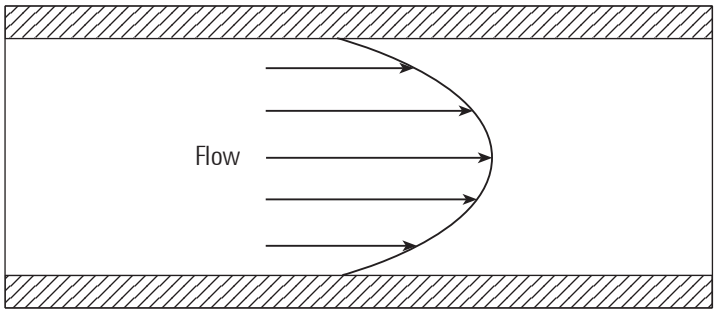


Fig. 4.1.5 Parabolic flow profile

As the velocity increases, and the Reynolds number exceeds 2 300, the flow becomes increasingly turbulent with more and more eddy currents, until at Reynolds number 10 000 the flow is completely turbulent (see Figure 4.1.6).

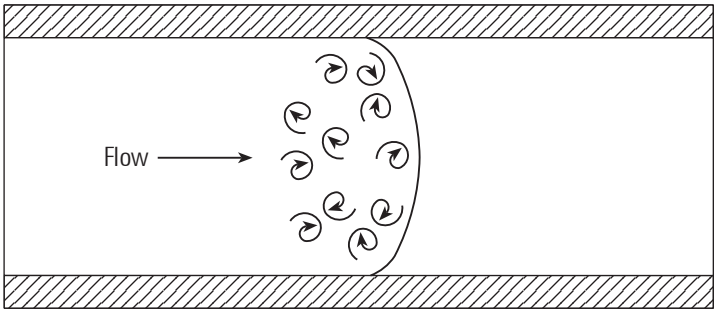


Fig. 4.1.6 Turbulent flow profile

Saturated steam, in common with most fluids, is transported through pipes in the 'turbulent flow' region.

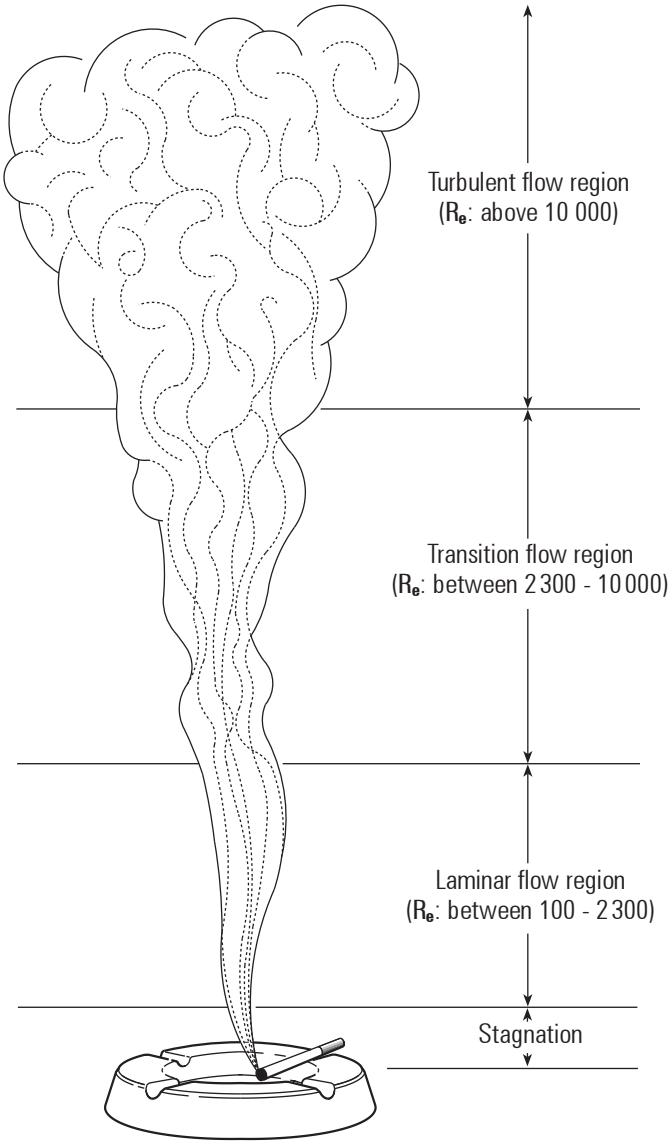


Fig. 4.1.7 Reynolds number

The examples shown in Figures 4.1.3 to 4.1.7 are useful in that they provide an understanding of fluid characteristics within pipes; however, the objective of the *Steam and Condensate Loop Book* is to provide specific information regarding saturated steam and water (or condensate).

Whilst these are two phases of the same fluid, their characteristics are entirely different. This has been demonstrated in the above Sections regarding Absolute Viscosity (μ) and Density (ρ). The following information, therefore, is specifically relevant to saturated steam systems.

Example 4.1.4
A 100 mm pipework system transports saturated steam at 10 bar g at an average velocity of 25 m/s.

Determine the Reynolds number.

The following data is available from comprehensive steam tables:

T_{sat} at 10 bar g = 184°C

Density (ρ) = 5.64 kg/m³

Dynamic viscosity of steam (μ) at 184°C = 15.2 x 10^{-6} Pa s

$$\text{Reynolds number } (R_e) = \frac{\rho \, u \, D}{\mu} \qquad \textbf{Equation 4.1.3}$$

Where:

ρ = Density = 5.64 kg/m³
u = Mean velocity in the pipe = 25 m/s
D = Internal pipe diameter = 100 mm = 0.1 m
μ = Dynamic viscosity = 15.2 x 10^{-6} Pa s

$$R_e = \frac{5.64 \times 25 \times 0.1}{15.2 \times 10^{-6}}$$

$$R_e = 927\,631 = 0.9 \times 10^6$$

- If the Reynolds number (R_e) in a saturated steam system is less than 10 000 (10^4) the flow may be laminar or transitional.

  Under laminar flow conditions, the pressure drop is directly proportional to flowrate.

- If the Reynolds number (R_e) is greater than 10 000 (10^4) the flow regime is turbulent.

  Under these conditions the pressure drop is proportional to the square root of the flow.

- For accurate steam flowmetering, consistent conditions are essential, and for saturated steam systems it is usual to specify the minimum Reynolds number (R_e) as 1 x 10^5 = 100 000.

- At the opposite end of the scale, when the Reynolds number (R_e) exceeds 1 x 10^6, the head losses due to friction within the pipework become significant, and this is specified as the maximum.

Example 4.1.5
Based on the information given above, determine the maximum and minimum flowrates for turbulent flow with saturated steam at 10 bar g in a 100 mm bore pipeline.

$$\text{Reynolds number } (R_e) = \frac{\rho \, u \, D}{\mu} \qquad \textbf{Equation 4.1.3}$$

Where:
ρ = Density = 5.64 kg/m³ $\left(v_g = \frac{1}{5.64} = 0.177 \text{ m}^3/\text{kg} \right)$
u = Mean velocity in the pipe (To be determined) m/s
D = Internal pipe diameter = 100 mm (0.1 m)
μ = Dynamic viscosity = 15.2 x 10^{-6} Pa s

For minimum turbulent flow, R_e of 1 x 10^5 should be considered:

$$\mathbf{R_e} = \frac{5.64 \text{ x } u \text{ x } 0.1}{15.2 \text{ x } 10^{-6}} = \mathbf{1 \text{ x } 10^5}$$

$$\mathbf{u} = \frac{1 \text{ x } 10^5 \text{ x } 15.2 \text{ x } 10^{-6}}{5.64 \text{ x } 0.1} = \mathbf{2.695 \text{ m/s}}$$

Volumetric flowrate may be determined using Equation 4.1.4:

$$q_v = A \, u \qquad \textbf{Equation 4.1.4}$$

Where:
q_v = Volume flow (m³/s)
A = Cross sectional area of the pipe (m²)
u = Velocity (m/s)

Mass flowrate may be determined using Equations 4.1.5 and 4.1.6:

$$q_m = \frac{q_v}{v_g} \qquad \textbf{Equation 4.1.5}$$

Where:
q_m = Mass flow (kg/s)
q_v = Volume flow (m³/s)
v_g = Specific volume (m³/kg)

Equation 4.1.6 is derived by combining Equations 4.1.4 and 4.1.5:

$$q_m = \frac{A \, u}{v_g} \qquad \textbf{Equation 4.1.6}$$

Where:
q_m = Mass flow (kg/s)
A = Cross sectional area of the pipe (m²)
u = Velocity (m/s)
v_g = Specific volume (m³/kg)

Returning to Example 4.1.5, and inserting values into Equation 4.1.6:

$$q_m = \frac{A\,u}{v_g} \quad \left(\text{where } A = \frac{\pi D^2}{4} \right)$$

$$q_m = \frac{\pi\, D^2\, u}{4\, v_g}$$

$$\mathbf{q_m} = \frac{\pi \times 0.1^2 \times 2.695}{4 \times 0.177} = \mathbf{430\ kg/h}\ (0.120\ kg/s)$$

Similarly, for maximum turbulent flow, $R_e = 1 \times 10^6$ shall be considered:

$$R_e = \frac{5.64 \times u \times 0.1}{15.2 \times 10^{-6}} = \mathbf{1 \times 10^6}$$

$$\mathbf{u} = \frac{1 \times 10^6 \times 15.2 \times 10^{-6}}{5.64 \times 0.1} = \mathbf{26.95\ m/s}$$

and:

$$q_m = \frac{A\,u}{v_g}$$

$$q_m = \frac{\pi\, D^2\, u}{4\, v_g}$$

$$\mathbf{q_m} = \frac{\pi \times 0.1^2 \times 2.695}{4 \times 0.177} = \mathbf{4\,300\ kg/h}\ (1.20\ kg/s)$$

Summary

- The mass flow of saturated steam through pipes is a function of density, viscosity and velocity.
- For accurate steam flowmetering, the pipe size selected should result in Reynolds numbers of between 1×10^5 and 1×10^6 at minimum and maximum conditions respectively.
- Since viscosity, etc., are fixed values for any one condition being considered, the correct Reynolds number is achieved by careful selection of the pipe size.
- If the Reynolds number increases by a factor of 10 (1×10^5 becomes 1×10^6), then so does the velocity (e.g. 2.695 m/s becomes 26.95 m/s respectively), providing pressure, density and viscosity remain constant.

Questions

1. 100 mm bore pipe carries 1000 kg/h of steam at 10 bar g. What is the Reynolds number at this flowrate?

a| 23.4×10^4 ☐

b| 49×10^5 ☐

c| 0.84×10^6 ☐

d| 16.8×10^4 ☐

2. If a flowrate has a Reynolds number of 32×10^4, what does it indicate?

a| Flow is turbulent and suitable for flowmetering ☐

b| Flow is laminar and any flowmeter reading would be inaccurate ☐

c| The pipe is oversized and a much smaller flowmeter would be necessary ☐

d| The steam must be superheated and unsuitable for flowmetering ☐

3. A 50 mm bore pipe carries 1100 kg/h of steam at 7 bar g. How would you describe the flow condition of the steam?

a| Laminar ☐

b| It has a dynamic viscosity of 130 Pa s ☐

c| Transitional ☐

d| Turbulent ☐

4. The dynamic viscosity of saturated steam:

a| Increases as pressure increases ☐

b| Remains constant at all temperatures ☐

c| Reduces as pressure increases ☐

d| Is directly proportional to velocity ☐

5. The Reynolds number (R_e) of steam:

a| Is directly proportional to the steam pressure and temperature ☐

b| Is directly proportional to the pipe diameter and velocity ☐

c| Is directly proportional to the pipe diameter and absolute viscosity, flowrate and density ☐

d| Is directly proportional to density, temperature and dynamic viscosity ☐

6. For accurate flowmetering of steam, flow should be:

a| Either turbulent or transitional ☐

b| Laminar ☐

c| Turbulent ☐

d| Either laminar or turbulent

Answers

1: a, 2: a, 3: d, 4: a, 5: c, 6: c

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Module 4.2

Principles of Flowmetering

Principles of Flowmetering

Terminology

When discussing flowmetering, a number of terms, which include **Repeatability**, **Uncertainty, Accuracy** and **Turndown**, are commonly used.

Repeatability

This describes the ability of a flowmeter to indicate the same value for an identical flowrate on more than one occasion. It should not be confused with accuracy i.e. its repeatability may be excellent in that it shows the same value for an identical flowrate on several occasions, but the reading might be consistently wrong (or inaccurate). Good repeatability is important, where steam flowmetering is required to monitor trends rather than accuracy. However, this does not dilute the importance of accuracy under any circumstances.

Uncertainty

The term 'uncertainty' is now becoming more commonly referred to than accuracy. This is because accuracy cannot be established, as the true value can never be exactly known. However 'uncertainty' can be estimated and an ISO standard exists offering guidance on this matter (EN ISO/IEC 17025). It is important to recognise that it is a statistical concept and not a guarantee. For example, it may be shown that with a large population of flowmeters, 95% would be at least as good as the uncertainty calculated. Most would be much better, but a few, 5% could be worse.

Accuracy

This is a measure of a flowmeter's performance when indicating a correct flowrate value against a 'true' value obtained by extensive calibration procedures. The subject of accuracy is dealt with in ISO 5725.

The following two methods used to express accuracy have very different meanings:

- **Percentage of measured value or actual reading**
  For example, a flowmeter's accuracy is given as ±3% of actual flow.

  At an indicated flowrate of 1 000 kg/h, the 'uncertainty' of actual flow is between:
  1 000 - 3% = 970 kg/h
  And
  1 000 + 3% = 1 030 kg/h

  Similarly, at an indicated flowrate of 500 kg/h, the error is still ±3%, and the 'uncertainty' is between:
  500 kg/h - 3% = 485 kg/h
  And
  500 kg/h + 3% = 515 kg/h

- **Percentage of full scale deflection (FSD)**
  A flowmeter's accuracy may also be given as ±3% of FSD. This means that the measurement error is expressed as a percentage of the maximum flow that the flowmeter can handle.

  As in the previous case, the maximum flow = 1 000 kg/h.

  At an indicated flowrate of 1 000 kg/h, the 'uncertainty' of actual flow is between:
  1 000 kg/h - 3% = 970 kg/h
  And
  1 000 kg/h + 3% = 1 030 kg/h

  At an indicated flowrate of 500 kg/h, the error is still ±30 kg/h, and the actual flow is between:
  500 kg/h - 30 kg/h = 470 kg/h an error of - 6%
  And
  500 kg/h + 30 kg/h = 530 kg/h an error of + 6%

  As the flowrate is reduced, the percentage error increases.

  A comparison of these measurement terms is shown graphically in Figure 4.2.1

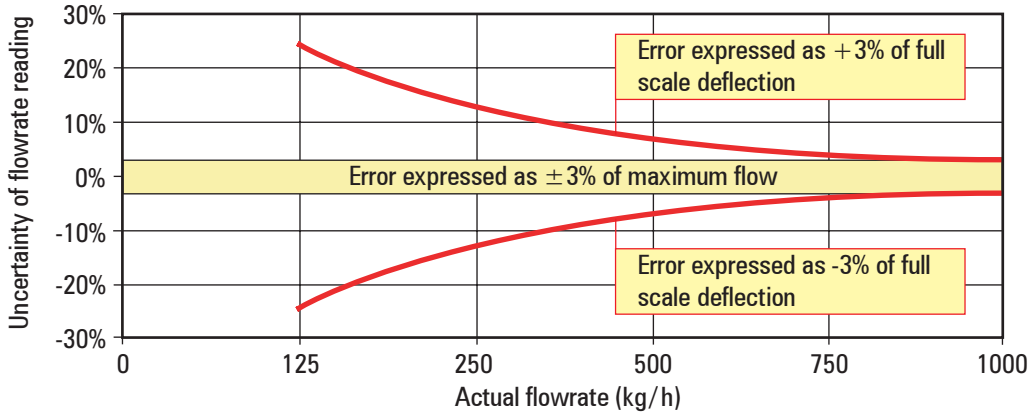


Fig. 4.2.1 Range of error

Turndown

When specifying a flowmeter, accuracy is a necessary requirement, but it is also essential to select a flowmeter with sufficient range for the application.

'Turndown' or 'turndown ratio', 'effective range' or 'rangeability' are all terms used to describe the range of flowrates over which the flowmeter will work within the accuracy and repeatability of the tolerances. Turndown is qualified in Equation 4.2.1.

$$\text{Turndown} = \frac{\text{Maximum flow}}{\text{Minimum flow}} \qquad \textbf{Equation 4.2.1}$$

Example 4.2.1

A particular steam system has a demand pattern as shown in Figure 4.2.2 The flowmeter has been sized to meet the maximum expected flowrate of 1 000 kg/h.

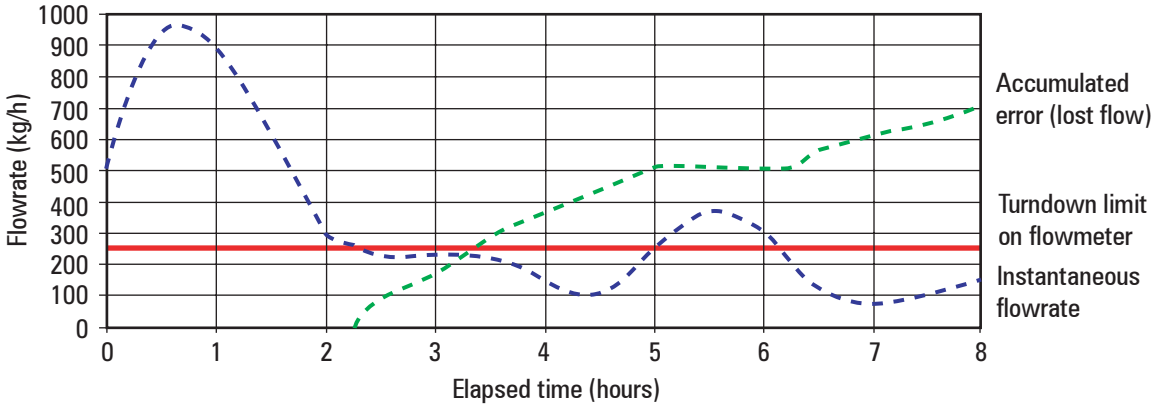


Fig. 4.2.2 Accumulated losses due to insufficient turndown

The turndown of the flowmeter selected is given as 4:1. i.e. The claimed accuracy of the flowmeter can be met at a minimum flowrate of 1 000 ÷ 4 = 250 kg/h.

When the steam flowrate is lower than this, the flowmeter cannot meet its specification, so large flow errors occur. At best, the recorded flows below 250 kg/h are inaccurate - at worst they are not recorded at all, and are 'lost'.

In the example shown in Figure 4.2.2, 'lost flow' is shown to amount to more than 700 kg of steam over an 8 hour period. The total amount of steam used during this time is approximately 2 700 kg, so the 'lost' amount represents an additional 30% of total steam use. Had the steam flowmeter been specified with an appropriate turndown capability, the steam flow to the process could have been more accurately measured and costed.

If steam flow is to be accurately metered, the user must make every effort to build up a true and complete assessment of demand, and then specify a flowmeter with:

- The capacity to meet maximum demand.
- A turndown sufficiently large to encompass all anticipated flow variations.

Flowmeter type	Turndown (operating) range
Orifice plate	**4:1** (Accurate measurement down to 25% of maximum flow)
Shunt flowmeter	**7:1** (Accurate measurement down to 14% of maximum flow)
Vortex flowmeters	**25:1** down to **4:1** (Accurate measurement from 25% to 4% of maximum flow depending on application)
Spring loaded variable area meter, position monitoring	Up to **50:1** (Accurate measurement down to 2% of maximum flow)
Spring loaded variable area meter, differential pressure monitoring	Up to **100:1** (Accurate measurement down to 1% of maximum flow)

Fig. 4.2.3 Table showing typical turndown ratios of commonly used flowmeters

Bernoulli's Theorem

Many flowmeters are based on the work of Daniel Bernoulli in the 1700s. Bernoulli's theorem relates to the Steady Flow Energy Equation (SFEE), and states that the sum of:

- Pressure energy,
- Kinetic energy and
- Potential energy

will be constant at any point within a piping system (ignoring the overall effects of friction). This is shown below, mathematically in Equation 4.2.2 for a unit mass flow:

$$\frac{P_1}{\rho g} + \frac{u_1^2}{2g} + h_1 = \frac{P_2}{\rho g} + \frac{u_2^2}{2g} + h_2 \qquad \textbf{Equation 4.2.2}$$

Where:

P_1 and P_2 = Pressure at points within a system (Pa)
u_1 and u_2 = Velocities at corresponding points within a system (m/s)
h_1 and h_2 = Relative vertical heights within a system (m)
ρ = Density (kg/m³)
g = Gravitational constant (9.81 m/s²)

Bernoulli's equation ignores the effects of friction and can be simplified as follows:

Pressure energy + Potential energy + Kinetic energy = Constant

Equation 4.2.3 can be developed from Equation 4.2.2 by multiplying throughout by 'ρg'.

$$P_1 + \rho g h_1 + \frac{1}{2}\rho u_1^2 = P_2 + \rho g h_2 + \frac{1}{2}\rho u_2^2 \qquad \textbf{Equation 4.2.3}$$

Friction is ignored in Equations 4.2.2 and 4.2.3, due to the fact that it can be considered negligible across the region concerned. Friction becomes more significant over longer pipe lengths. Equation 4.2.3 can be further developed by removing the 2nd term on either side when there is no change in reference height (h). This is shown in Equation 4.2.4:

$$P_1 + \frac{1}{2}\rho u_1^2 = P_2 + \frac{1}{2}\rho u_2^2 \qquad \textbf{Equation 4.2.4}$$

Example 4.2.2

Determine P_2 for the system shown in Figure 4.2.4, where water flows through a diverging section of pipe at a volumetric rate of 0.1 m³/s at 10°C.

The water has a density of 998.84 kg/m³ at 10°C and 2 bar g.

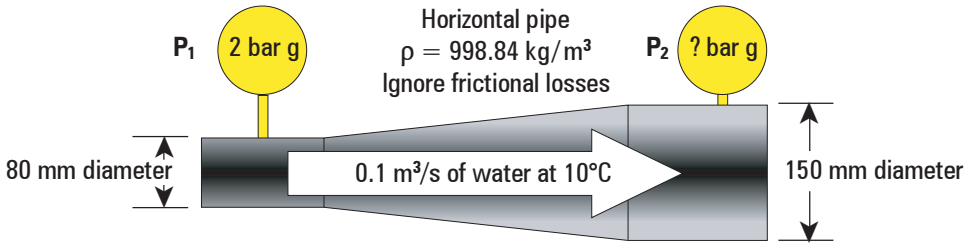


Fig. 4.2.4 System described in Example 4.2.2

From Equation 4.1.4:

$$q_v = A\ u \qquad \textbf{Equation 4.1.4}$$

Where:
q_v = Volumetric flowrate (m/s)
A = Cross-sectional area (m²)
u = Velocity (m/s)

By transposing the Equation 4.1.4, a figure for velocity can be calculated:

$$\text{Velocity } (u) = \frac{q_v}{A}$$

$$\text{Velocity in the 80 mm section of pipework } (u_1) = \frac{0.1 \times 4}{\pi \times 0.08^2} = 19.9 \text{ m/s}$$

$$\text{Velocity in the 150 mm section of pipework } (u_2) = \frac{0.1 \times 4}{\pi \times 0.15^2} = 5.66 \text{ m/s}$$

2 bar gauge pressure (P_1) = 3.013 25 bar absolute pressure (P_1)

3.013 25 bar a = 301.325 kPa = 301 325 Pa

Equation 4.2.4 is a development of Equation 4.2.3 as described previously, and can be used to predict the downstream pressure in this example.

$$P_1 + \frac{1}{2}\rho\ u_1^2 = P_2 + \frac{1}{2}\rho\ u_2^2 \qquad \textbf{Equation 4.2.4}$$

From Equation 4.2.4:

$$P_2 = P_1 + \rho \left(\frac{u_1^2 - u_2^2}{2} \right)$$

$$P_2 = 301\,325 + 998.84 \left(\frac{19.9^2 - 5.66^2}{2} \right)$$

$$P_2 = 483\,101 \text{ Pa}$$

$$P_2 = 4.83101 \text{ bar a}$$

$$P_2 = 3.818 \text{ bar g}$$

Example 4.2.2 highlights the implications of Bernoulli's theorem. It is shown that, in a diverging pipe, the downstream pressure will be higher than the upstream pressure. This may seem odd at first glance; it would normally be expected that the downstream pressure in a pipe is less than the upstream pressure for flow to occur in that direction. It is worth remembering that Bernoulli states, the sum of the energy at any point along a length of pipe is constant.

In Example 4.2.2, the increased pipe bore has caused the velocity to fall and hence the pressure to rise. In reality, friction cannot be ignored, as it is impossible for any fluid to flow along a pipe unless a pressure drop exists to overcome the friction created by the movement of the fluid itself. In longer pipes, the effect of friction is usually important, as it may be relatively large. A term, h_f, can be added to Equation 4.2.4 to account for the pressure drop due to friction, and is shown in Equation 4.2.5.

$$P_1 + \frac{1}{2}\rho u_1^2 = P_2 + \frac{1}{2}\rho u_2^2 + h_f \qquad \textbf{Equation 4.2.5}$$

With an incompressible fluid such as water flowing through the same size pipe, the density and velocity of the fluid can be regarded as constant and Equation 4.2.6 can be developed from Equation 4.2.5 ($P_1 = P_2 + h_f$).

$$P_1 - P_2 = h_f \qquad \textbf{Equation 4.2.6}$$

Equation 4.2.6 shows (for a constant fluid density) that the pressure drop along a length of the same size pipe is caused by the static head loss (h_f) due to friction from the relative movement between the fluid and the pipe. In a short length of pipe, or equally, a flowmetering device, the frictional forces are extremely small and in practice can be ignored. For compressible fluids like steam, the density will change along a relatively long piece of pipe. For a relatively short equivalent length of pipe (or a flowmeter using a relatively small pressure differential), changes in density and frictional forces will be negligible and can be ignored for practical purposes. This means that the pressure drop through a flowmeter can be attributed to the effects of the known resistance of the flowmeter rather than to friction.

Some flowmeters take advantage of the Bernoulli effect to be able to measure fluid flow, an example being the simple orifice plate flowmeter. Such flowmeters offer a resistance to the flowing fluid such that a pressure drop occurs over the flowmeter. If a relationship exists between the flow and this contrived pressure drop, and if the pressure drop can be measured, then it becomes possible to measure the flow.

Quantifying the relationship between flow and pressure drop

Consider the simple analogy of a tank filled to some level with water, and a hole at the side of the tank somewhere near the bottom which, initially, is plugged to stop the water from flowing out (see Figure 4.2.5). It is possible to consider a single molecule of water at the top of the tank (molecule 1) and a single molecule below at the same level as the hole (molecule 2).

With the hole plugged, the height of water (or head) above the hole creates a potential to force the molecules directly below molecule 1 through the hole. The potential energy of molecule 1 relative to molecule 2 would depend upon the height of molecule 1 above molecule 2, the mass of molecule 1, and the effect that gravitational force has on molecule 1's mass. The potential energy of all the water molecules directly between molecule 1 and molecule 2 is shown by Equation 4.2.7.

$$\text{Potential energy} = m\,g\,h \qquad \textbf{Equation 4.2.7}$$

Where:

m = Mass of all the molecules directly between and including molecule 1 and molecule 2.
g = Gravitational constant (9.81 m/s^2)
h = Cumulative height of molecules above the hole

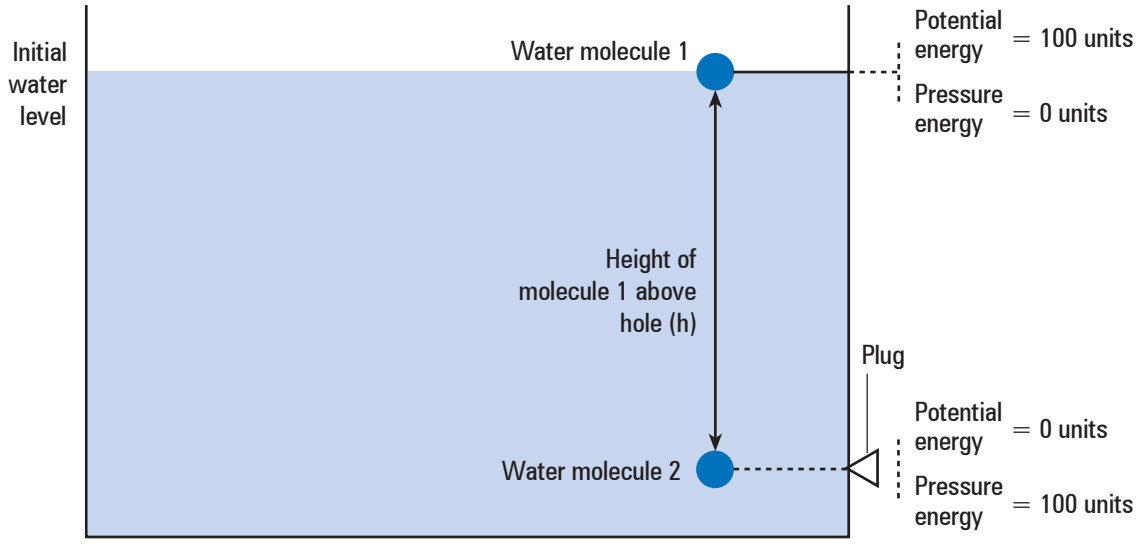


Fig. 4.2.5 A tank of water with a plugged hole near the bottom of the tank

Molecule 1 has no pressure energy (the nett effect of the air pressure is zero, because the plug at the bottom of the tank is also subjected to the same pressure), or kinetic energy (as the fluid in which it is placed is not moving). The only energy it possesses relative to the hole in the tank is potential energy.

Meanwhile, at the position opposite the hole, molecule 2 has a potential energy of zero as it has no height relative to the hole. However, the pressure at any point in a fluid must balance the weight of all the fluid above, plus any additional vertical force acting above the point of consideration. In this instance, the additional force is due to the atmospheric air pressure above the water surface, which can be thought of as zero gauge pressure. The pressure to which molecule 2 is subjected is therefore related purely to the weight of molecules above it.

Weight is actually a force applied to a mass due to the effect of gravity, and is defined as mass x acceleration. The weight being supported by molecule 2 is the mass of water (m) in a line of molecules directly above it multiplied by the constant of gravitational acceleration, (g). Therefore, molecule 2 is subjected to a pressure force m g.

But what is the **energy** contained in molecule 2? As discussed above, it has no potential energy; neither does it have kinetic energy, as, like molecule 1, it is not moving. It can only therefore possess pressure energy.

Mechanical energy is clearly defined as Force x Distance,
so the pressure energy held in molecule 2 = Force (m g) x Distance (h) = m g h, where:
m = Mass of all the molecules directly between and including molecule 1 and molecule 2
g = Gravitational acceleration 9.81 m/s^2
h = Cumulative height of molecules above the hole

It can therefore be seen that:

Potential energy in molecule 1 = m g h = Pressure energy in molecule 2.

This agrees with the principle of conservation of energy (which is related to the First Law of Thermodynamics) which states that energy cannot be created or destroyed, but it can change from one form to another. This essentially means that the loss in potential energy means an equal gain in pressure energy.

Consider now, that the plug is removed from the hole, as shown in Figure 4.2.6. It seems intuitive that water will pour out of the hole due to the head of water in the tank.

In fact, the rate at which water will flow through the hole is related to the difference in pressure energy between the molecules of water opposite the hole, inside and immediately outside the tank. As the pressure outside the tank is atmospheric, the pressure energy at any point outside the hole can be taken as zero (in the same way as the pressure applied to molecule 1 was zero). Therefore the difference in pressure energy across the hole can be taken as the pressure energy contained in molecule 2, and therefore, the rate at which water will flow through the hole is related to the pressure energy of molecule 2.

In Figure 4.2.6, consider molecule 2 with pressure energy of m g h, and consider molecule 3 having just passed through the hole in the tank, and contained in the issuing jet of water.

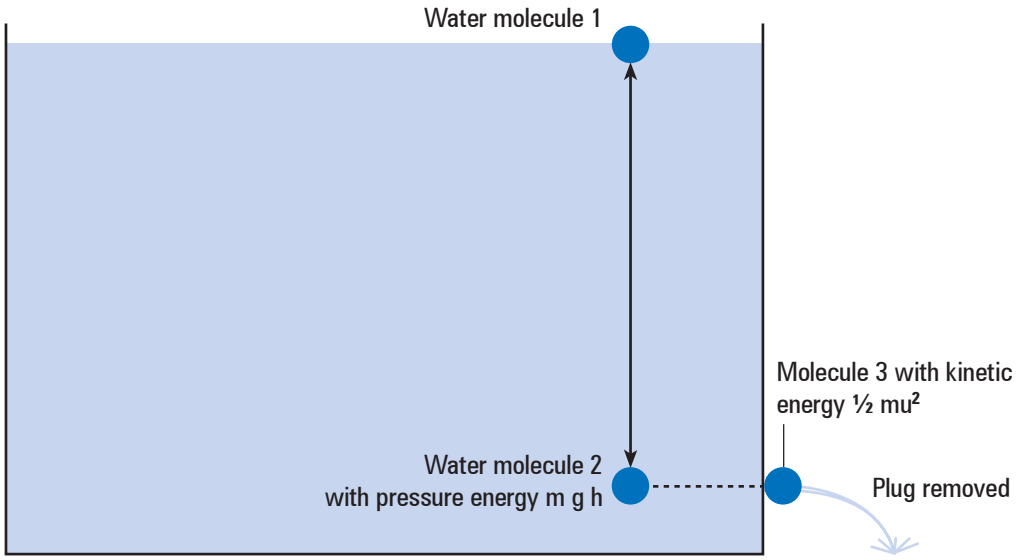


Fig. 4.2.6 The plug is removed from the tank

Molecule 3 has no pressure energy for the reasons described above, or potential energy (as the fluid in which it is placed is at the same height as the hole). The only energy it has can only be kinetic energy.

At some point in the water jet immediately after passing through the hole, molecule 3 is to be found in the jet and will have a certain velocity and therefore a certain kinetic energy. As energy cannot be created, it follows that the kinetic energy in molecule 3 is formed from that pressure energy held in molecule 2 immediately before the plug was removed from the hole.

It can therefore be concluded that the whole of the kinetic energy held in molecule 3 equals the pressure energy to which molecule 2 is subjected, which, in turn, equals the potential energy held in molecule 1.

The basic equation for kinetic energy is shown in Equation 4.2.8:

$$\text{Kinetic energy} = \frac{1}{2} m u^2 \qquad \textbf{Equation 4.2.8}$$

Where:
m = Mass of the object (kg)
u = Velocity of the object at any point (m/s)

If all the initial potential energy has changed into kinetic energy, it must be true that the potential energy at the start of the process equals the kinetic energy at the end of the process. To this end, it can be deduced that:

$$m\,g\,h = \frac{1}{2}\,m\,u^2 \qquad \textbf{Equation 4.2.9}$$

From Equation 4.2.9: $u^2 = \dfrac{2\,m\,g\,h}{m}$

Therefore: $u^2 = 2\,g\,h$

$$u = \sqrt{2\,g\,h} \qquad \textbf{Equation 4.2.10}$$

Equation 4.2.10 shows that the velocity of water passing through the hole is proportional to the square root of the height of water or pressure head (h) above the reference point, (the hole). The head 'h' can be thought of as a difference in pressure, also referred to as pressure drop or 'differential pressure'.

Equally, the same concept would apply to a fluid passing through an orifice that has been placed in a pipe. One simple method of metering fluid flow is by introducing an orifice plate flowmeter into a pipe, thereby creating a pressure drop relative to the flowing fluid. Measuring the differential pressure and applying the necessary square-root factor can determine the velocity of the fluid passing through the orifice.

The graph (Figure 4.2.7) shows how the flowrate changes relative to the pressure drop across an orifice plate flowmeter. It can be seen that, with a pressure drop of 25 kPa, the flowrate is the square root of 25, which is 5 units. Equally, the flowrate with a pressure drop of 16 kPa is 4 units, at 9 kPa is 3 units and so on.

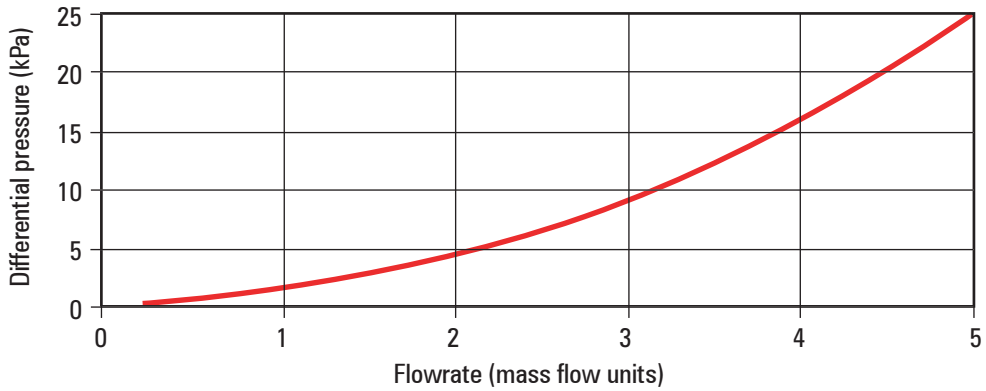


Fig. 4.2.7 The square-root relationship of an orifice plate flowmeter

Knowing the velocity through the orifice is of little use in itself. The prime objective of any flowmeter is to measure flowrate in terms of volume or mass. However, if the size of the hole is known, the volumetric flowrate can be determined by multiplying the velocity by the area of the hole. However, this is not as straightforward as it first seems.

It is a phenomenon of any orifice fitted in a pipe that the fluid, after passing through the orifice, will continue to constrict, due mainly to the momentum of the fluid itself. This effectively means that the fluid passes through a narrower aperture than the orifice. This aperture is called the 'vena contracta' and represents that part in the system of maximum constriction, minimum pressure, and maximum velocity for the fluid. The area of the vena contracta depends upon the physical shape of the hole, but can be predicted for standard sharp edged orifice plates used for such purposes. The ratio of the area of the vena contracta to the area of the orifice is usually in the region of 0.65 to 0.7; consequently if the orifice area is known, the area of the vena contracta can be established. The subject is discussed in further detail in the next Section.

The orifice plate flowmeter and Bernoulli's Theorem

When Bernoulli's theorem is applied to an orifice plate flowmeter, the difference in pressure across the orifice plate provides the kinetic energy of the fluid discharged through the orifice.

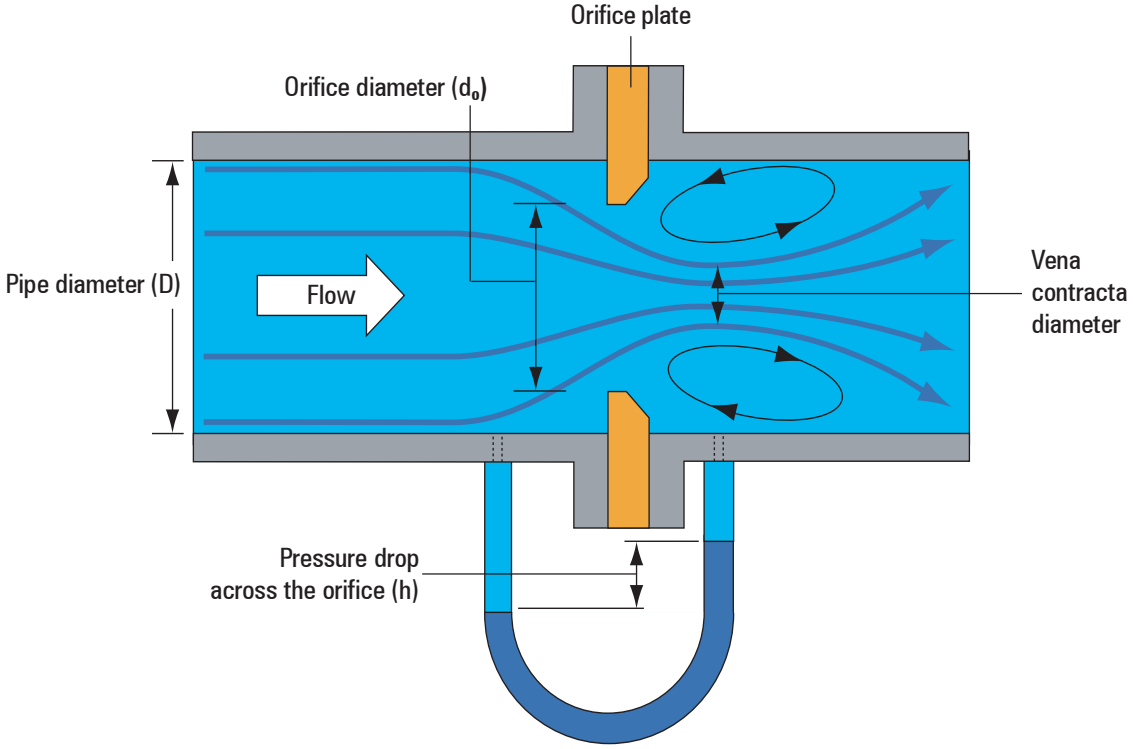


Fig. 4.2.8 An orifice plate with vena contracta

As seen previously, the velocity through the orifice can be calculated by use of Equation 4.2.10:

$$u = \sqrt{2\,g\,h} \qquad \textbf{Equation 4.2.10}$$

However, it has already been stated, volume flow is more useful than velocity (Equation 4.1.4):

$$q_v = A\,u \qquad \textbf{Equation 4.1.4}$$

Substituting for 'u' from Equation 4.2.10 into Equation 4.1.4:

$$q_v = A\sqrt{2\,g\,h}$$

In practice, the actual velocity through the orifice will be less than the theoretical value for velocity, due to friction losses. This difference between these theoretical and actual figures is referred to as the coefficient of velocity (C_v).

$$\text{Coefficient of velocity } (C_v) = \frac{\text{Actual velocity}}{\text{Theoretical velocity}}$$

Also, the flow area of the vena contracta will be less than the size of the orifice. The ratio of the area of the vena contracta to that of the orifice is called the coefficient of contraction.

$$\text{Coefficient of contraction } (C_c) = \frac{\text{Area of the vena contracta}}{\text{Area of the orifice}}$$

The coefficient of velocity and the coefficient of contraction may be combined to give a coefficient of discharge (C) for the installation. Volumetric flow will need to take the coefficient of discharge (C) into consideration as shown in Equation 4.2.11.

$$q_v = C A\sqrt{2 g h} \qquad \textbf{Equation 4.2.11}$$

Where:
q_v = Volumetric flowrate (m^3/s)
C = Coefficient of discharge (dimensionless)
A = Area of orifice (m^2)
g = Gravitational constant (9.8 m/s^2)
h = Differential pressure (m)

This may be further simplified by removing the constants as shown in Equation 4.2.12.

$$q_v \propto \sqrt{\Delta p} \qquad \textbf{Equation 4.2.12}$$

Equation 4.2.12 clearly shows that volume flowrate is proportional to the square root of the pressure drop.

Note:
The definition of C can be found in ISO 5167-2003, 'Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full'.

ISO 5167 offers the following information:

The equations for the numerical values of C given in ISO 5167 (all parts) are based on data determined experimentally.

The uncertainty in the value of C can be reduced by flow calibration in a suitable laboratory.

The Pitot tube and Bernoulli's Theorem

The Pitot tube is named after its French inventor Henri Pitot (1695 – 1771). The device measures a fluid velocity by converting the kinetic energy of the flowing fluid into potential energy at what is described as a 'stagnation point'. The stagnation point is located at the opening of the tube as in Figure 4.2.9. The fluid is stationary as it hits the end of the tube, and its velocity at this point is zero. The potential energy created is transmitted though the tube to a measuring device.

The tube entrance and the inside of the pipe in which the tube is situated are subject to the same dynamic pressure; hence the static pressure measured by the Pitot tube is in addition to the dynamic pressure in the pipe. The difference between these two pressures is proportional to the fluid velocity, and can be measured simply by a differential manometer.

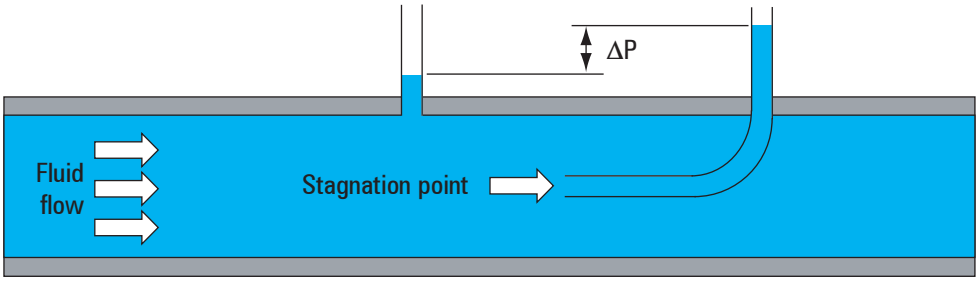


Fig. 4.2.9 The simple Pitot tube principle

Bernoulli's equation can be applied to the Pitot tube in order to determine the fluid velocity from the observed differential pressure (ΔP) and the known density of the fluid. The Pitot tube can be used to measure incompressible and compressible fluids, but to convert the differential pressure into velocity, different equations apply to liquids and gases. The details of these are outside the scope of this module, but the concept of the conservation of energy and Bernoulli's theorem applies to all; and for the sake of example, the following text refers to the relationship between pressure and velocity for an incompressible fluid flowing at less than sonic velocity. (Generally, a flow can be considered incompressible when its flow is less than 0.3 Mach or 30% of its sonic velocity).

From Equation 4.2.4, an equation can be developed to calculate velocity (Equation 4.2.13):

$$P_1 + \frac{1}{2}\rho u_1^2 = P_2 + \frac{1}{2}\rho u_2^2$$ **Equation 4.2.4**

Where:
P_1 = The dynamic pressure in the pipe
u_1 = The fluid velocity in the pipe
P_2 = The static pressure in the Pitot tube
u_2 = The stagnation velocity = zero
ρ = The fluid density

Because u_2 is zero, Equation 4.2.4 can be rewritten as Equation 4.2.13:

$$P_1 + \frac{1}{2}\rho u_1^2 = P_2$$

$$P_2 - P_1 = \frac{1}{2}\rho u_1^2$$

$$u_1^2 = \frac{2\,(\Delta P)}{\rho}$$

$$u_1 = \sqrt{\frac{2\,(\Delta P)}{\rho}}$$ **Equation 4.2.13**

The fluid volumetric flowrate can be calculated from the product of the pipe area and the velocity calculated from Equation 4.2.13.

The effect of the accuracy of the differential cell upon uncertainty

Example 4.2.3

In a particular orifice plate flowmetering system, the maximum flow of 1 000 kg/h equates to a differential pressure of 25 kPa, as shown in Figure 4.2.10.

The differential pressure cell has a guaranteed accuracy of ±0.1 kPa over the operating range of a particular installation.

Demonstrate the effect of the differential cell accuracy on the accuracy of the installation.

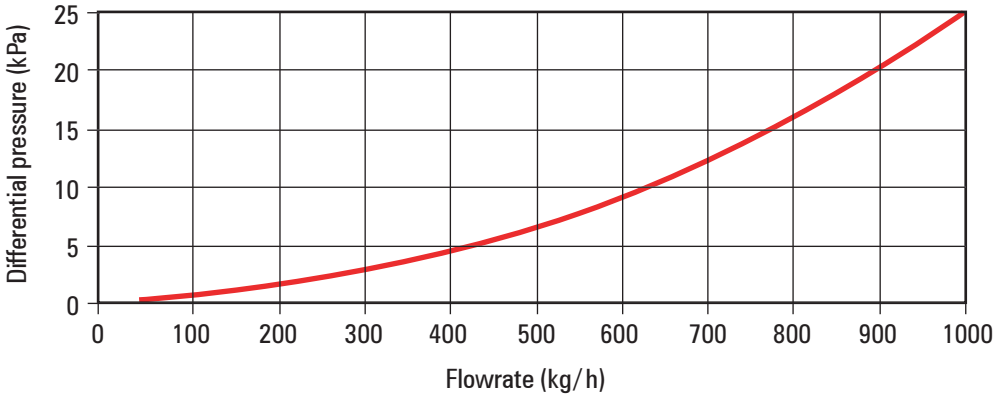


Fig. 4.2.10 Square root characteristic

Determine the flowmeter constant:

At maximum flow (1 000 kg/h), the differential pressure = 25 kPa

From Equation 4.2.12: $1\,000 \text{ kg/h} \propto \sqrt{25} \text{ kPa}$

or $1\,000 \text{ kg/h} = \text{Constant} \times \sqrt{25} \text{ kPa}$

$$\text{Constant} = \frac{1\,000 \text{ kg/h}}{\sqrt{25} \text{ kPa}} = 200$$

If the differential pressure cell is over-reading by 0.1 kPa, the actual flowrate (q_m):

$$q_m = \text{Constant} \times \sqrt{25 - 0.1} \text{ kPa}$$

$$q_m = 200 \times \sqrt{24.9} \text{ kPa} = 998 \text{ kg/h}$$

The percentage error at an actual flowrate of 1 000 kg/h:

$$\% \text{ error} = \frac{1000 - 998 \text{ kg/h}}{1000 \text{ kg/h}} = 0.2\%$$

Similarly, with an actual mass flowrate of 500 kg/h, the expected differential pressure:

$$500 \text{ kg/h} = 200 \times \sqrt{\Delta P} \text{ kPa}$$

$$\Delta P = 6.25 \text{ kPa}$$

If the differential pressure cell is over-reading by 0.1 kPa, the actual flowrate (q_m):

$$q_m = 200 \times \sqrt{6.25 - 0.1} \text{ kPa}$$

$$q_m = 496 \text{ kg/h}$$

The percentage error at an actual flowrate of 500 kg/h:

$$\% \text{ error} = \frac{500 - 496 \text{ kg/h}}{500 \text{ kg/h}} = 0.8\%$$

Figure 4.2.11 shows the effects over a range of flowrates:

Actual flowrate	kg/h	100	200	300	400	500	600	700	800	900	1000
Calculated flow using DP cell (Under-reading)	kg/h	77	190	293	395	496	597	697	797	898	998
Uncertainty (Negative)	%	22.5	5.13	2.25	1.26	0.80	0.56	0.41	0.31	0.25	0.20
Calculated flow using DP cell (Over-reading)	kg/h	118	210	307	405	504	603	703	302	902	1002
Uncertainty (Positive)	%	18.3	4.88	2.20	1.24	0.80	0.55	0.41	0.31	0.25	0.20

Fig. 4.2.11 Table showing percentage error in flow reading resulting from an accuracy limitation of 0.1 kPa on a differential pressure cell

Review of results:
At maximum flowrate, the 0.1 kPa uncertainty in the differential pressure cell reading represents only a small proportion of the total differential pressure, and the effect is minimal.

As the flowrate is reduced, the differential pressure is also reduced, and the 0.1 kPa uncertainty represents a progressively larger percentage of the differential pressure reading, resulting in the slope increasing slowly, as depicted in Figure 4.2.12.

At very low flowrates, the value of the uncertainty accelerates. At between 20 and 25% of maximum flow, the rate of change of the slope accelerates rapidly, and by 10% of maximum flow, the range of uncertainty is between +18.3% and -22.5%.

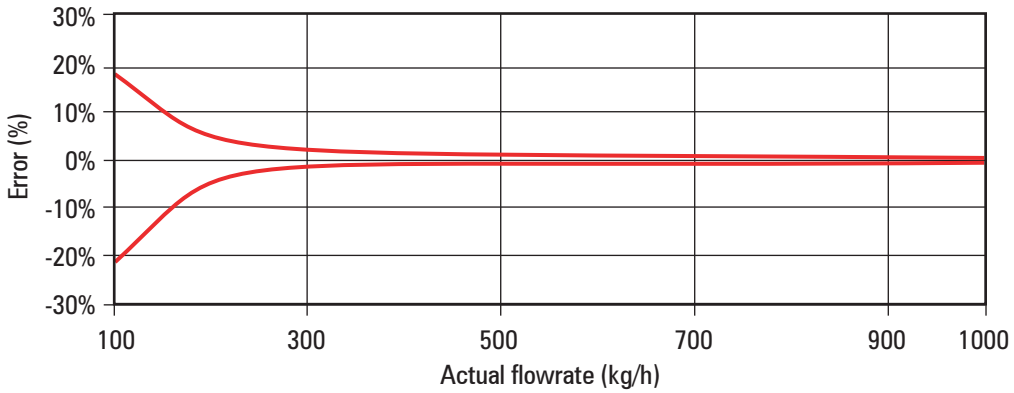


Fig. 4.2.12 Graph showing percentage uncertainty in flow reading resulting from an accuracy limitation of 0.1 kPa on a differential pressure cell

Conclusion

To have confidence in the readings of an orifice plate flowmeter system, the turndown ratio must not exceed 4 or 5:1.

Note:
- Example 4.2.3 examines only one element of a steam flowmetering installation.
- The overall confidence in the measured value given by a steam flowmetering system will include the installation, the accuracy of the orifice size, and the accuracy of the predicated coefficient of discharge (C) of the orifice.

Questions

1. An orifice plate flowmeter has been selected for a maximum flowrate of 2 500 kg/h. The flowmeter has a published accuracy of ±2% of actual flow. For a flow of 700 kg/h, over what range of flow will accuracy be maintained?

a| 650 - 750 kg/h ☐

b| 686 - 714 kg/h ☐

c| 675 - 725 kg/h ☐

d| 693 - 707 kg/h ☐

2. An orifice plate flowmeter has been selected for a maximum flowrate of 2 500 kg/h. The flowmeter has a published accuracy of ±2% of FSD. For a flow of 700 kg/h, over what range of flow will accuracy be maintained?

a| 675 - 725 kg/h ☐

b| 693 - 707 kg/h ☐

c| 650 - 750 kg/h ☐

d| 686 - 714 kg/h ☐

3. An orifice plate flowmeter is selected for a maximum flow of 3 000 kg/h. The minimum expected flow is 300 kg/h. The accuracy of the flowmeter is ±2% of actual flow. Over what range of flow at the minimum flow condition will accuracy be maintained?

a| Range unknown because the turndown is greater than 8:1 ☐

b| Range unknown because the turndown is greater than 4:1 ☐

c| 294 - 306 kg/h ☐

d| 240 - 360 kg/h ☐

4. Why is an orifice plate flowmeter limited to a turndown of 4:1?

a| At higher turndowns, the vena contracta has a choking effect on flow through an orifice ☐

b| At higher turndowns the differential pressure across an orifice is too small to be measured accurately ☐

c| At low flowrates, the accuracy of the differential pressure cell has a larger effect on the flowmeter accuracy ☐

d| The orifice is too large for flow at higher flowrates ☐

5. An orifice plate flowmeter is sized for a maximum flow of 2 000 kg/h. What is the effect on accuracy at a higher flow?

a| The accuracy is reduced because the turndown will be greater than 4:1 ☐

b| The flowmeter will be out of range so the indicated flow will be meaningless ☐

c| None ☐

d| The characteristics of an orifice plate flowmeter mean that the higher the flow, the greater the accuracy, consequently accuracy will be improved ☐

6. What would be the effect on accuracy of a DN100 orifice plate flowmeter if the downstream differential pressure tapping was 25 mm after the flowmeter, instead of the expected d/2 length.

a| Accuracy would be improved because the flow is now laminar ☐

b| Accuracy would be reduced due to a higher uncertainty effect caused by a lower differential pressure ☐

c| Accuracy would be much reduced because flow is now turbulent ☐

d| None ☐

Answers

1: b, 2: c, 3: b, 4: c, 5: b, 6: b

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Module 4.3

Types of Steam Flowmeter

Types of Steam Flowmeter

There are many types of flowmeter available, those suitable for steam applications include:

- Orifice plate flowmeters.
- Turbine flowmeters (including shunt or bypass types).
- Variable area flowmeters.
- Spring loaded variable area flowmeters.
- Direct in-line variable area (DIVA) flowmeter.
- Pitot tubes.
- Vortex shedding flowmeters.

Each of these flowmeter types has its own advantages and limitations. To ensure accurate and consistent performance from a steam flowmeter, it is essential to match the flowmeter to the application.

This Module will review the above flowmeter types, and discuss their characteristics, their advantages and disadvantages, typical applications and typical installations.

Orifice plate flowmeters

The orifice plate is one in a group known as head loss devices or differential pressure flowmeters. In simple terms the pipeline fluid is passed through a restriction, and the pressure differential is measured across that restriction. Based on the work of Daniel Bernoulli in 1738 (see Module 4.2), the relationship between the velocity of fluid passing through the orifice is proportional to the square root of the pressure loss across it. Other flowmeters in the differential pressure group include venturis and nozzles.

With an orifice plate flowmeter, the restriction is in the form of a plate which has a hole concentric with the pipeline. This is referred to as the primary element.

To measure the differential pressure when the fluid is flowing, connections are made from the upstream and downstream pressure tappings, to a secondary device known as a DP (Differential Pressure) cell.

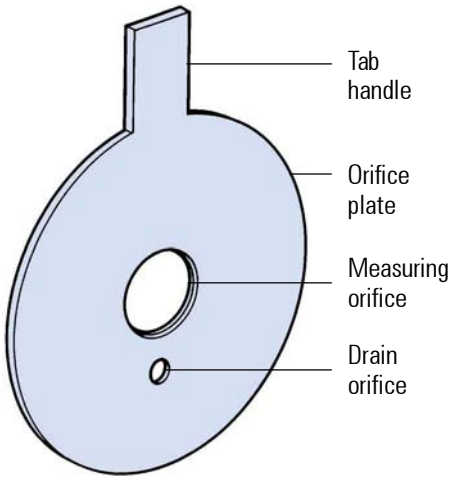


Fig. 4.3.1 Orifice plate

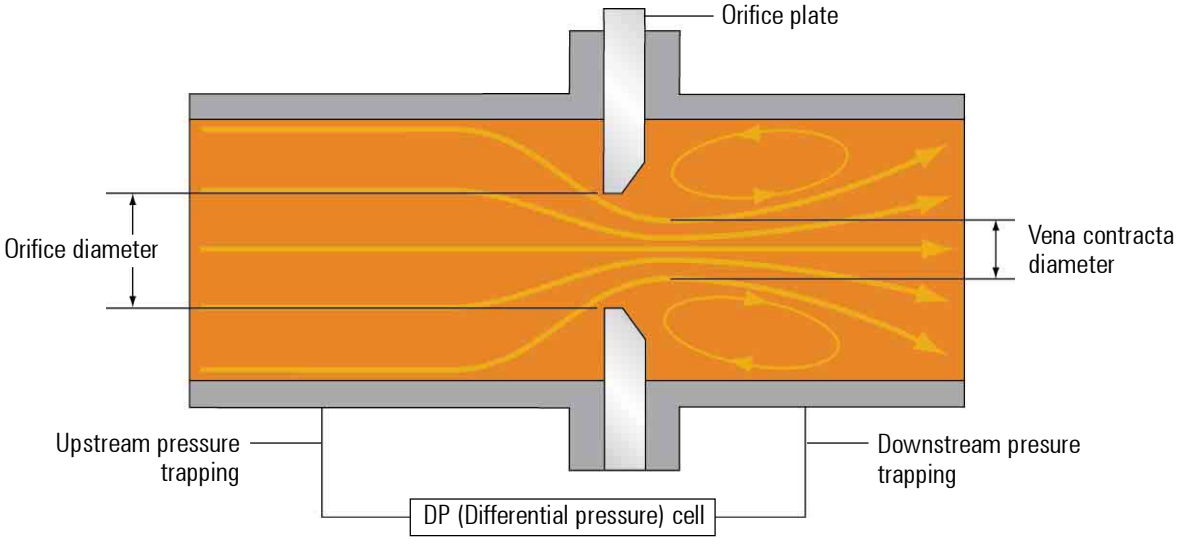


Fig. 4.3.2 Orifice plate flowmeter

From the DP cell, the information may be fed to a simple flow indicator, or to a flow computer along with temperature and/or pressure data, which enables the system to compensate for changes in fluid density.

In horizontal lines carrying vapours, water (or condensate) can build up against the upstream face of the orifice. To prevent this, a drain hole may be drilled in the plate at the bottom of the pipe. Clearly, the effect of this must be taken into account when the orifice plate dimensions are determined.

Correct sizing and installation of orifice plates is absolutely essential, and is well documented in the International Standard ISO 5167.

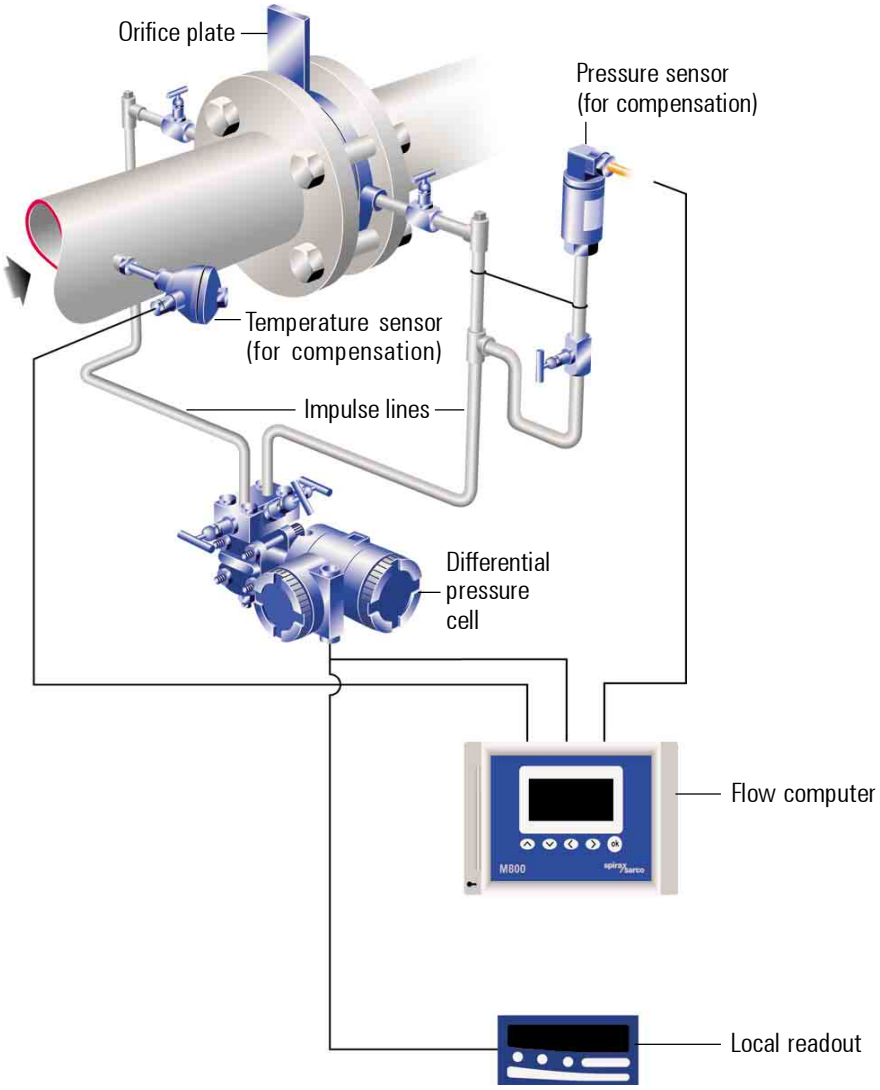


Fig. 4.3.3 Orifice plate flowmeter installation

Installation

A few of the most important points from ISO 5167 are discussed below:

Pressure tappings - Small bore pipes (referred to as impulse lines) connect the upstream and downstream pressure tappings of the orifice plate to a Differential Pressure or DP cell.

The positioning of the pressure tappings can be varied. The most common locations are:

- From the flanges (or carrier) containing the orifice plate as shown in Figure 4.3.3. This is convenient, but care needs to be taken with tappings at the bottom of the pipe,because they may become clogged.
- One pipe diameter on the upstream side and 0.5 x pipe diameter on the downstream side. This is less convenient, but potentially more accurate as the differential pressure measured is at its greatest at the vena contracta, which occurs at this position.

Corner tappings - These are generally used on smaller orifice plates where space restrictions mean flanged tappings are difficult to manufacture. Usually on pipe diameters including or below DN50.

From the DP cell, the information may be fed to a flow indicator, or to a flow computer along with temperature and/or pressure data, to provide density compensation.

Pipework - There is a requirement for a minimum of five straight pipe diameters downstream of the orifice plate, to reduce the effects of disturbance caused by the pipework.

The amount of straight pipework required upstream of the orifice plate is, however, affected by a number of factors including:

- The ß ratio; this is the relationship between the orifice diameter and the pipe diameter (see Equation 4.3.1), and would typically be a value of 0.7.

$$\beta = \frac{d\ (\text{orifice diameter})}{D\ (\text{pipe diameter})} \qquad \textbf{Equation 4.3.1}$$

- The nature and geometry of the preceding obstruction. A few obstruction examples are shown in Figure 4.3.4:

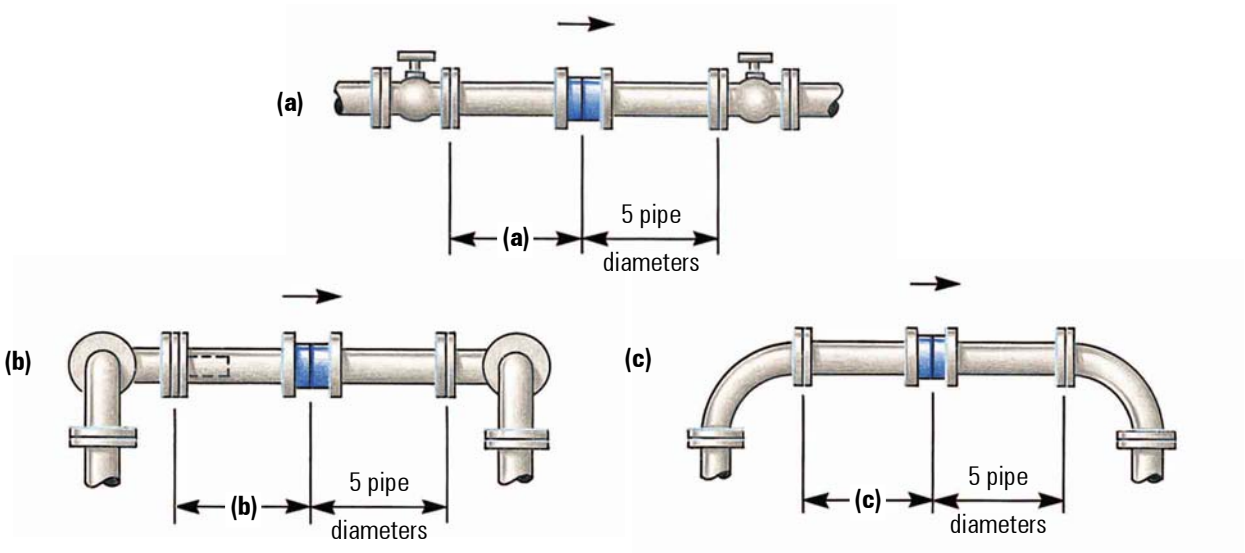


Fig. 4.3.4 Orifice plate installations

Table 4.3.1 brings the ß ratio and the pipework geometry together to recommend the number of straight diameters of pipework required for the configurations shown in Figure 4.3.4.

In particularly arduous situations, flow straighteners may be used. These are discussed in more detail in Module 4.5.

Table 4.3.1 Recommended straight pipe diameters upstream of an orifice plate for various ß ratios and preceding obstruction

See Figure 4.3.4	Recommended straight pipe diameters upstream of an orifice plate for various ß ratios and preceding obstruction						
	<0.32	0.45	0.55	0.63	0.70	0.77	0.84
a	18	20	23	27	32	40	49
b	15	18	22	28	36	46	57
c	10	13	16	22	29	44	56

Advantages of orifice plate steam flowmeters:

- Simple and rugged.
- Good accuracy.
- Low cost.
- No calibration or recalibration is required provided calculations, tolerances and installation comply with ISO 5167.

Disadvantages of orifice plate steam flowmeters:

- Turndown is limited to between 4:1 and 5:1 because of the square root relationship between flow and pressure drop.
- The orifice plate can buckle due to waterhammer and can block in a system that is poorly designed or installed.
- The square edge of the orifice can erode over time, particularly if the steam is wet or dirty. This will alter the characteristics of the orifice, and accuracy will be affected. Regular inspection and replacement is therefore necessary to ensure reliability and accuracy.
- The installed length of an orifice plate flowmetering system may be substantial; a minimum of 10 upstream and 5 downstream straight unobstructed pipe diameters may be needed for accuracy.

  This can be difficult to achieve in compact plants. Consider a system which uses 100 mm pipework, the ß ratio is 0.7, and the layout is similar to that shown in Figure 4.3.4(b):

  The upstream pipework length required would be = 36 x 0.1 m = 3.6 m

  The downstream pipework length required would be = 5 x 0.1 m = 0.5 m

  The total straight pipework required would be = 3.6 + 0.5 m = 4.1 m

Typical applications for orifice plate steam flowmeters:

- Anywhere the flowrate remains within the limited turndown ratio of between 4:1 and 5:1.

  This can include the boiler house and applications where steam is supplied to many plants, some on-line, some off-line, but the overall flowrate is within the range.

Turbine flowmeters

The primary element consists of a multi-bladed rotor which is mounted at right angles to the flow and suspended in the fluid stream on a free-running bearing. The diameter of the rotor is slightly less than the inside diameter of the flowmetering chamber, and its speed of rotation is proportional to the volumetric flowrate.

The speed of rotation of the turbine may be determined using an electronic proximity switch mounted on the outside of the pipework, which counts the pulses, as shown in Figure 4.3.5.

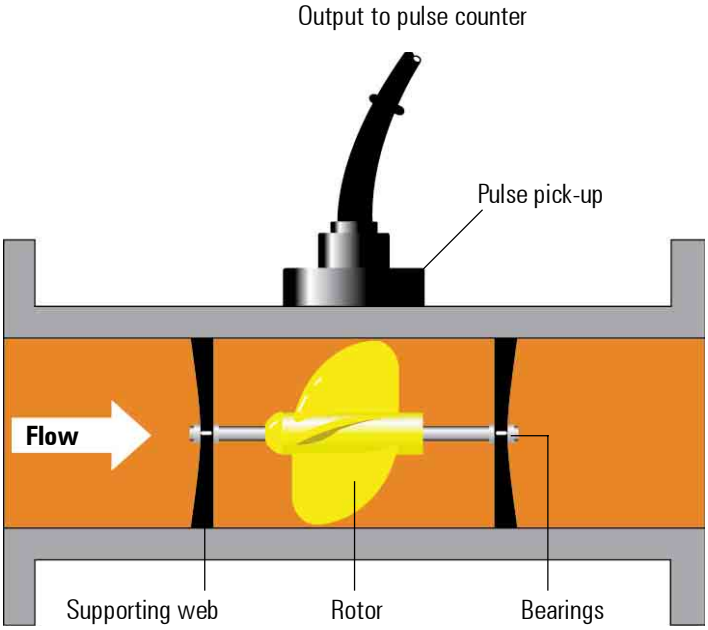


Fig. 4.3.5 Turbine flowmeter

Since a turbine flowmeter consists of a number of moving parts, there are several influencing factors that need to be considered:

- The temperature, pressure and viscosity of the fluid being measured.
- The lubricating qualities of the fluid.
- The bearing wear and friction.
- The conditional and dimensional changes of the blades.
- The inlet velocity profile and the effects of swirl.
- The pressure drop through the flowmeter.

Because of these factors, calibration of turbine flowmeters must be carried out under operational conditions.

In larger pipelines, to minimise cost, the turbine element can be installed in a pipework bypass, or even for the flowmeter body to incorporate a bypass or shunt, as shown in Figure 4.3.6.

Bypass flowmeters comprise an orifice plate, which is sized to provide sufficient restriction for a sample of the main flow to pass through a parallel circuit. Whilst the speed of rotation of the turbine may still be determined as explained previously, there are many older units still in existence which have a mechanical output as shown in Figure 4.3.6.

Clearly, friction between the turbine shaft and the gland sealing can be significant with this mechanical arrangement.

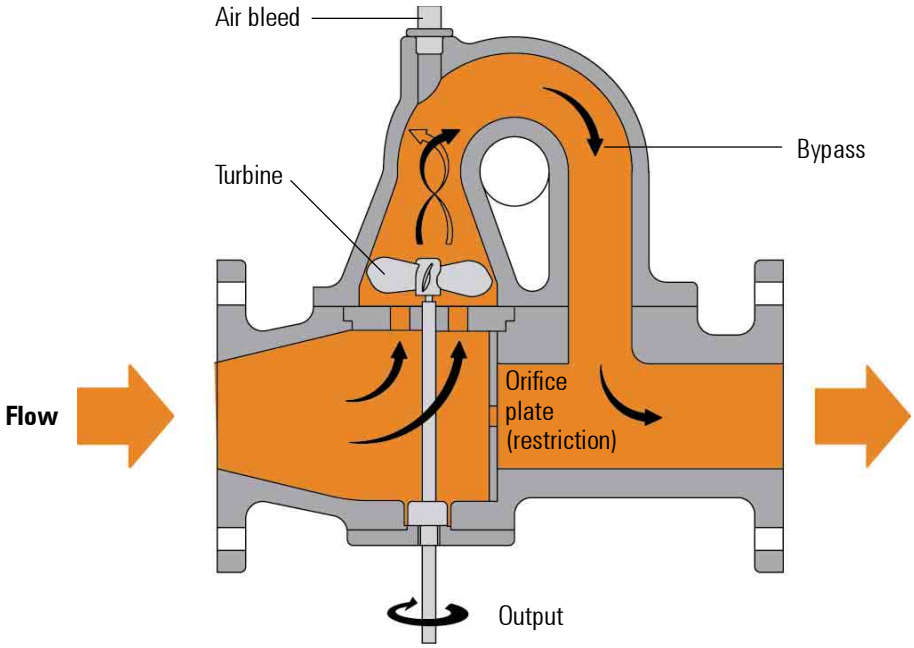


Fig. 4.3.6 Bypass or shunt turbine flowmeter

Advantages of turbine flowmeters:

- A turndown of 10:1 is achievable in a good installation with the turbine bearings in good condition.
- Accuracy is reasonable (± 0.5% of actual value).
- Bypass flowmeters are relatively low cost.

Disadvantages of turbine flowmeters:

- Generally calibrated for a specific line pressure. Any steam pressure variations will lead to inaccuracies in readout unless a density compensation package is included.
- Flow straighteners are essential (see Module 4.5).
- If the flow oscillates, the turbine will tend to over or under run, leading to inaccuracies due to lag time.
- Wet steam can damage the turbine wheel and affect accuracy.
- Low flowrates can be lost because there is insufficient energy to turn the turbine wheel.
- Viscosity sensitive: if the viscosity of the fluid increases, the response at low flowrates deteriorates giving a non-linear relationship between flow and rotational speed. Software may be available to reduce this effect.
- The fluid must be very clean (particle size not more than 100 µm) because:

  Clearances between the turbine wheel and the inside of the pipe are very small.

  Entrained debris can damage the turbine wheel and alter its performance.

  Entrained debris will accelerate bearing wear and affect accuracy, particularly at low flowrates.

Typical applications for turbine flowmeters:

- Superheated steam.
- Liquid flowmetering, particularly fluids with lubricating properties. As with all liquids, care must be taken to remove air and gases prior to them being metered.

Variable area flowmeters

The variable area flowmeter (Figure 4.3.7), often referred to as a rotameter, consists of a vertical, tapered bore tube with the small bore at the lower end, and a float that is allowed to freely move in the fluid. When fluid is passing through the tube, the float's position is in equilibrium with:

- The dynamic upward force of the fluid.
- The downward force resulting from the mass of the float.
- The position of the float, therefore, is an indication of the flowrate.

In practice, this type of flowmeter will be a mix of:

- **A float** selected to provide a certain weight, and chemical resistance to the fluid.

  The most common float material is grade 316 stainless steel, however, other materials such as Hastalloy C, aluminium or PVC are used for specific applications.

  On small flowmeters, the float is simply a ball, but on larger flowmeters special shaped floats are used to improve stability.

- **A tapered tube**, which will provide a measuring scale of typically between 40 mm and 250 mm over the design flow range.

  Usually the tube will be made from glass or plastic. However, if failure of the tube could present a hazard, then either a protective shroud may be fitted around the glass, or a metal tube may be used.

  With a transparent tube, flow readings are taken by observation of the float against a scale. For higher temperature applications where the tube material is opaque, a magnetic device is used to indicate the position of the float.

  Because the annular area around the float increases with flow, the differential pressure remains almost constant.

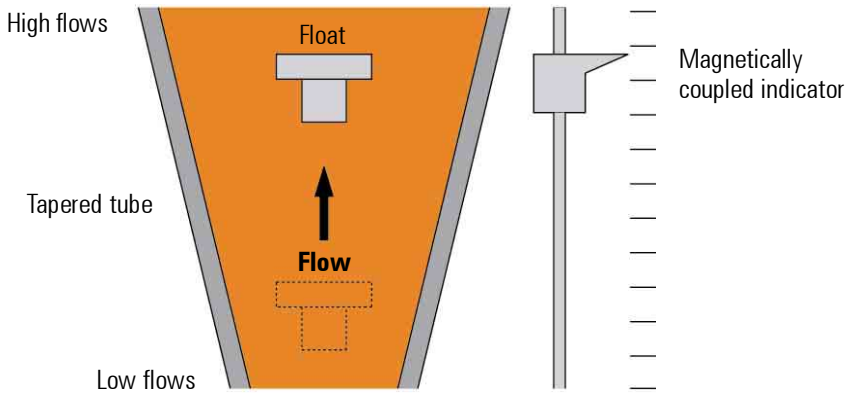


Fig. 4.3.7 Variable area flowmeter

Advantages of variable area flowmeters:

- Linear output.
- Turndown is approximately 10:1.
- Simple and robust.
- Pressure drop is minimal and fairly constant.

Disadvantages of variable area flowmeters:

- The tube must be mounted vertically (see Figure 4.3.8).
- Because readings are usually taken visually, and the float tends to move about, accuracy is only moderate. This is made worst by parallax error at higher flowrates, because the float is some distance away from the scale.
- Transparent taper tubes limit pressure and temperature.

Typical applications for variable area flowmeters:

- Metering of gases.
- Small bore airflow metering - In these applications, the tube is manufactured from glass, with calibrations marked on the outside. Readings are taken visually.
- Laboratory applications.
- Rotameters are sometimes used as a flow indicating device rather than a flow measuring device.

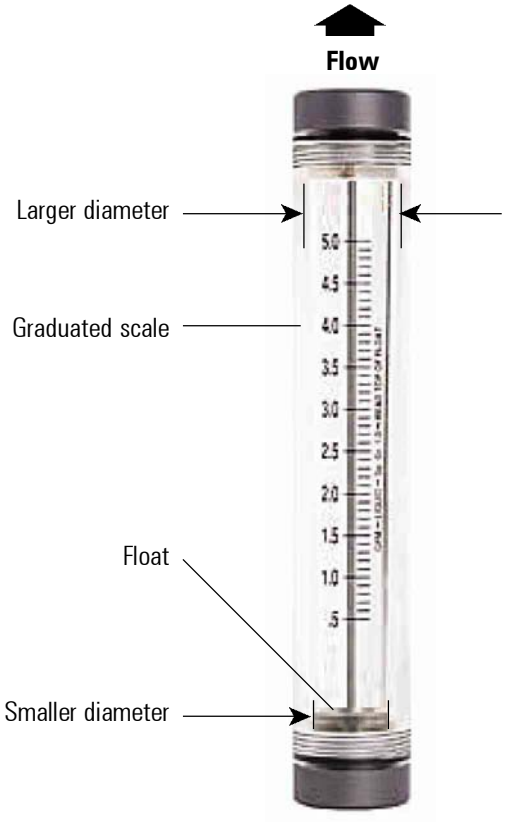


Fig. 4.3.8 Variable area flowmeter installed in a vertical plane

Spring loaded variable area flowmeters

The spring loaded variable area flowmeter (an extension of the variable area flowmeter) uses a spring as the balancing force. This makes the meter independent of gravity, allowing it to be used in any plane, even upside-down. However, in its fundamental configuration (as shown in Figure 4.3.9), there is also a limitation: the range of movement is constrained by the linear range of the spring, and the limits of the spring deformation.

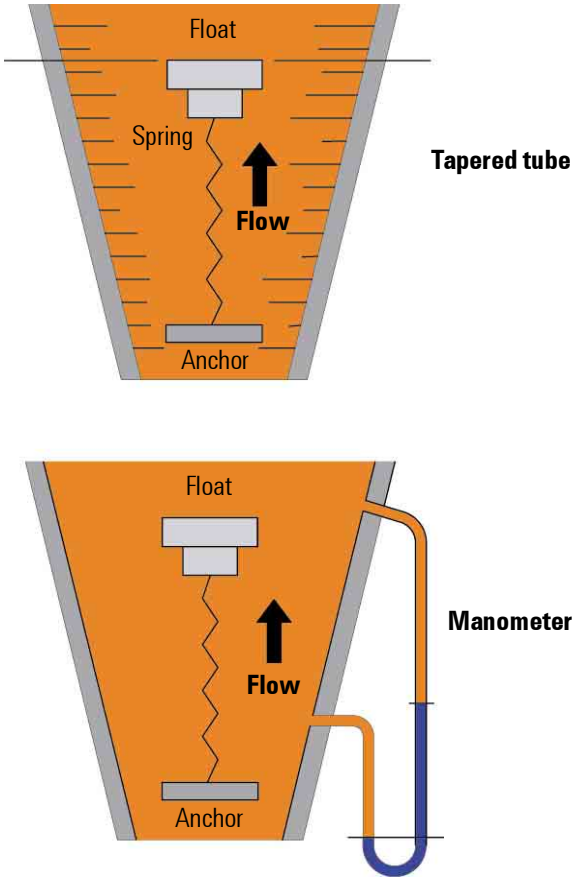


Fig. 4.3.9 Spring loaded variable area flowmeters

However, another important feature is also revealed: if the pass area (the area between the float and the tube) increases at an appropriate rate, then the differential pressure across the spring loaded variable area flowmeter can be directly proportional to flow.

To recap a few earlier statements

With orifice plates flowmeters:

- As the rate of flow increases, so does the differential pressure.
- By measuring this pressure difference it is possible to calculate the flowrate through the flowmeter.
- The pass area (for example, the size of the hole in the orifice plate) remains constant.

With any type of variable area flowmeter:

- The differential pressure remains almost constant as the flowrate varies.
- Flowrate is determine from the position of the float.
- The pass area (the area between the float and the tube) through which the flow passes increases with increasing flow.

Figure 4.3.10 compares these two principles.

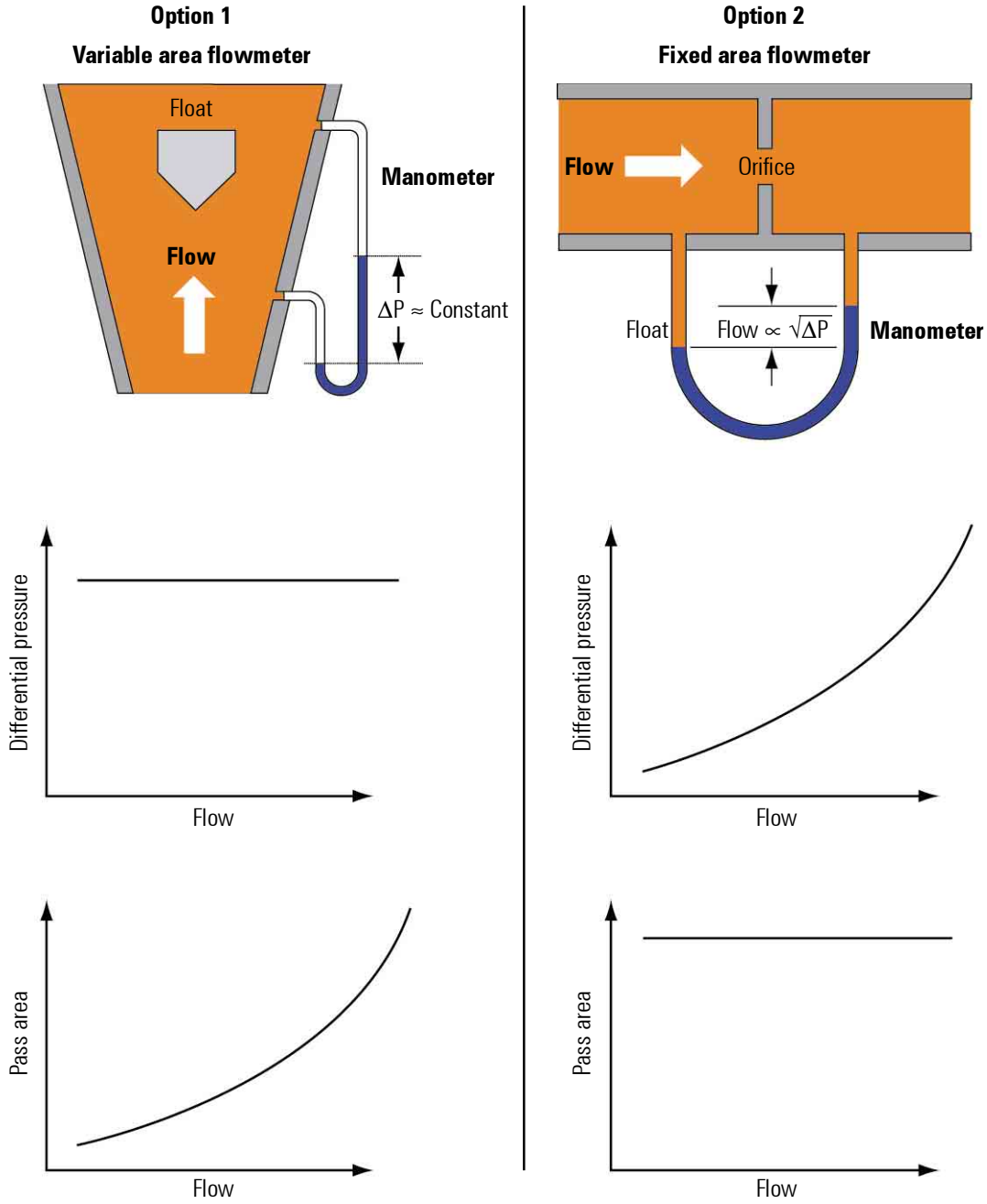


Fig. 4.3.10 Comparing the fixed area and variable area flowmeters

The spring loaded variable area principle is a hybrid between these two devices, and either:

- The displacement of the float - Option 1

or

- The differential pressure - Option 2

...may be used to determine the flowrate through the flowmeter.

In Option 1 (determining the displacement of the float or 'flap'). This can be developed for steam systems by:

- Using a torsion spring to give a better operating range.
- Using a system of coils to accurately determine the position of the float.

This will result in a very compact flowmeter. This may be further tailored for saturated steam applications by incorporating a temperature sensor and programming steam tables into the computer unit. See Figure 4.3.11 for an example of a flowmeter of this type.

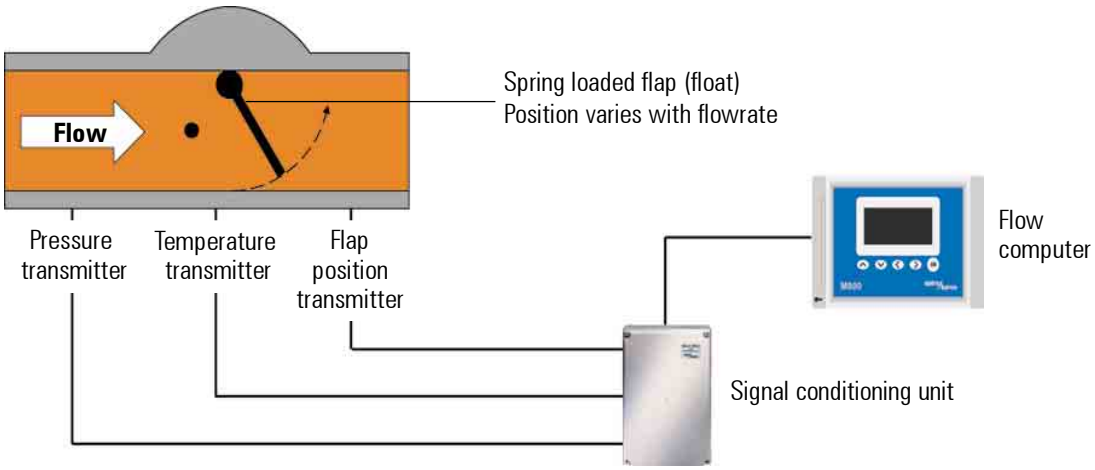


Fig. 4.3.11 Spring loaded variable area flowmeter monitoring the position of the float

Advantages of spring loaded variable area flowmeters:

- Robust.
- Turndowns of 25:1 are achievable with normal steam velocities (25 m/s), although high velocities can be tolerated on an intermittent basis, offering turndowns of up to 40:1.
- Accuracy is ±2% of actual value.
- Can be tailored for saturated steam systems with temperature and pressure sensors to provide pressure compensation.
- Relatively low cost.
- Short installation length.

Disadvantages of spring loaded variable area flowmeters:

- Size limited to DN100.
- Can be damaged over a long period by poor quality (wet and dirty) steam, at prolonged high velocity (>30 m/s).

Typical applications for spring loaded variable area flowmeters:

- Flowetering of steam to individual plants.
- Small boiler houses.

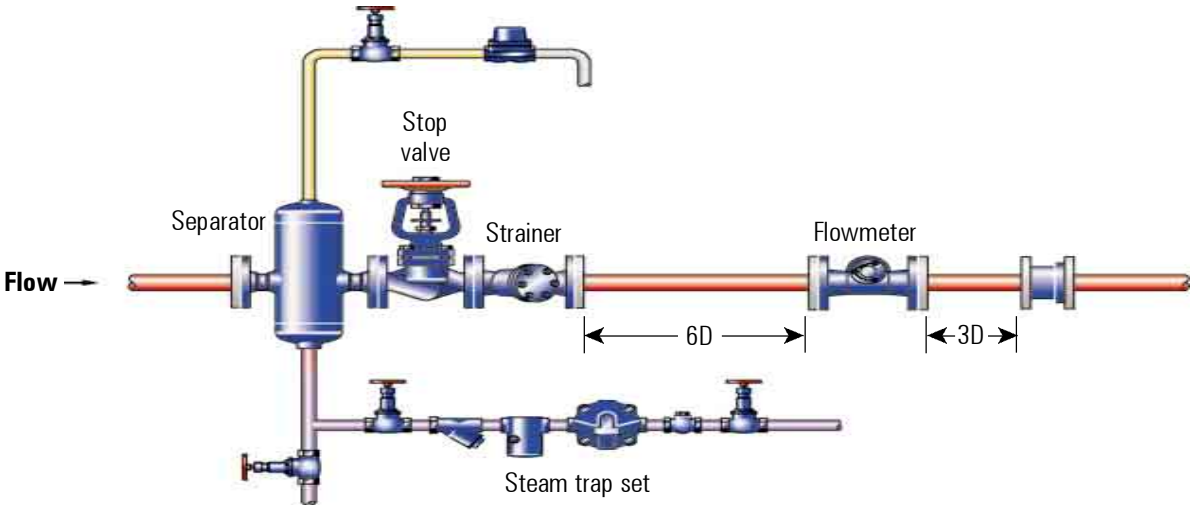


Fig. 4.3.12 Typical installation of a spring loaded variable area flowmeter measuring steam flow

In Option 2 (Figure 4.3.10), namely, determining the differential pressure, this concept can be developed further by shaping of the float to give a linear relationship between differential pressure and flowrate. See Figure 4.3.13 for an example of a spring loaded variable area flowmeter measuring differential pressure. The float is referred to as a cone due to its shape.

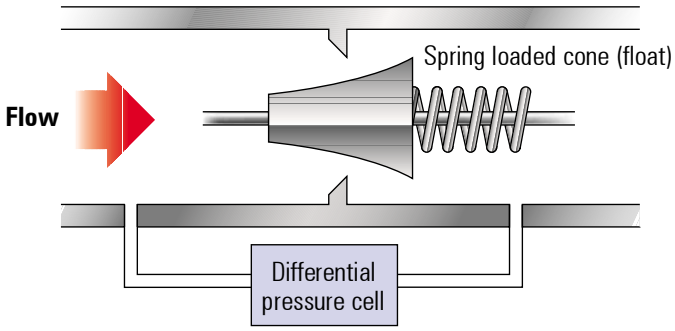


Fig. 4.3.13 Spring Loaded Variable Area flowmeter (SLVA) monitoring differential pressure

Advantages of a spring loaded variable area (SLVA) flowmeter:

- High turndown, up to 100:1.
- Good accuracy ±1% of reading for pipeline unit.
- Compact – a DN100 wafer unit requires only 60 mm between flanges.
- Suitable for many fluids.

Disadvantages of a variable area spring load flowmeter:

- Can be expensive due to the required accessories, such as the DP cell and flow computer.

Typical applications for a variable area spring load flowmeter:

- Boiler house flowmetering.
- Flowmetering of large plants.

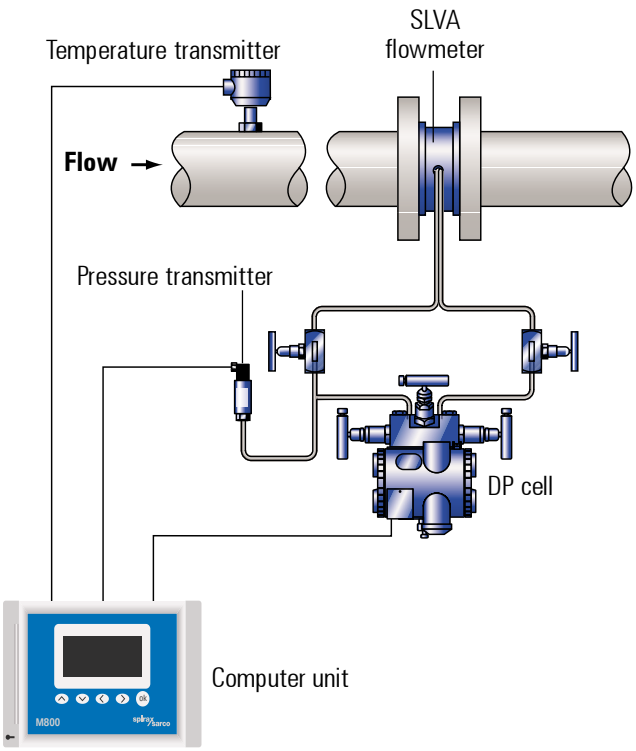


Fig. 4.3.14 Typical installation of a SVLA flowmeter monitoring differential pressure

Direct In-Line Variable Area (DIVA) flowmeter

The DIVA flowmeter operates on the well established spring loaded variable area (SLVA) principle, where the area of an annular orifice is continuously varied by a precision shaped moving cone. This cone is free to move axially against the resistance of a spring.

However, unlike other SLVA flowmeters, the DIVA does not rely on the measurement of differential pressure drop across the flowmeter to calculate flow, measuring instead the force caused by the deflection of the cone via a series of extremely high quality strain gauges. The higher the flow of steam the greater the force. This removes the need for expensive differential pressure transmitters, reducing installation costs and potential problems (Figure 4.3.15).

The DIVA has an internal temperature sensor, which provides full density compensation for saturated steam applications.

Flowmetering systems will:

- Check on the energy cost of any part of the plant.
- Cost energy as a raw material.
- Identify priority areas for energy savings.
- Enable efficiencies to be calculated for processes or power generation.

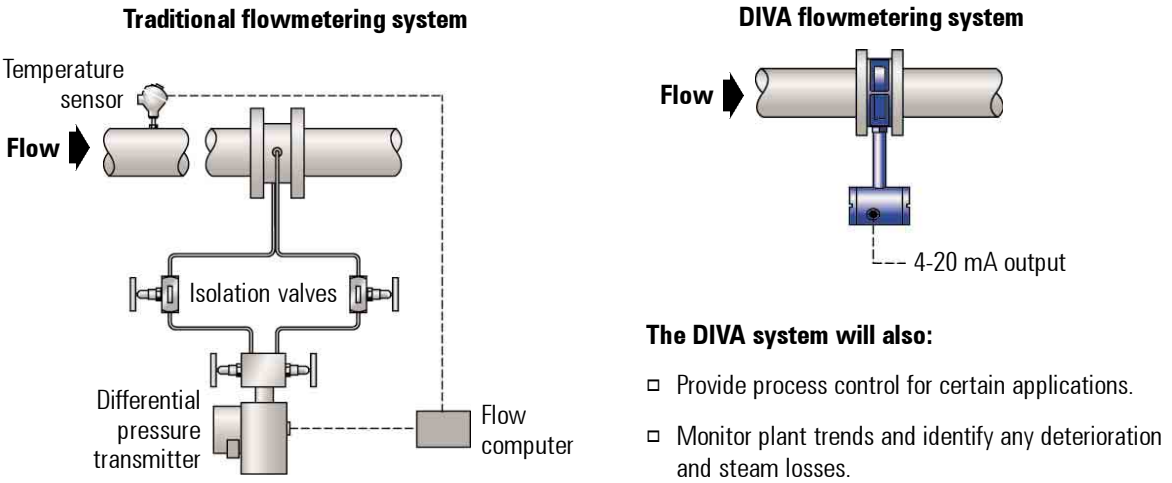


Fig. 4.3.15 Traditional flowmetering system versus a DIVA flowmetering system

The DIVA steam flowmeter (Figure 4.3.16) has a system uncertainty in accordance with EN ISO/IEC 17025, of:

- ± 2% of actual flow to a confidence of 95% (2 standard deviations) over a range of 10% to 100% of maximum rated flow.
- ± 0.2% FSD to a confidence of 95% (2 standard deviations) from 2% to 10% of the maximum rated flow.

As the DIVA is a self-contained unit the uncertainty quoted is for the complete system. Many flowmeters claim a pipeline unit uncertainty but, for the whole system, the individual uncertainty values of any associated equipment, such as DP cells, need to be taken into account.

The turndown of a flowmeter is the ratio of the maximum to minimum flowrate over which it will meet its specified performance, or its operational range. The DIVA flowmeter has a high turndown ratio of up to 50:1, giving an operational range of up to 98% of its maximum flow.

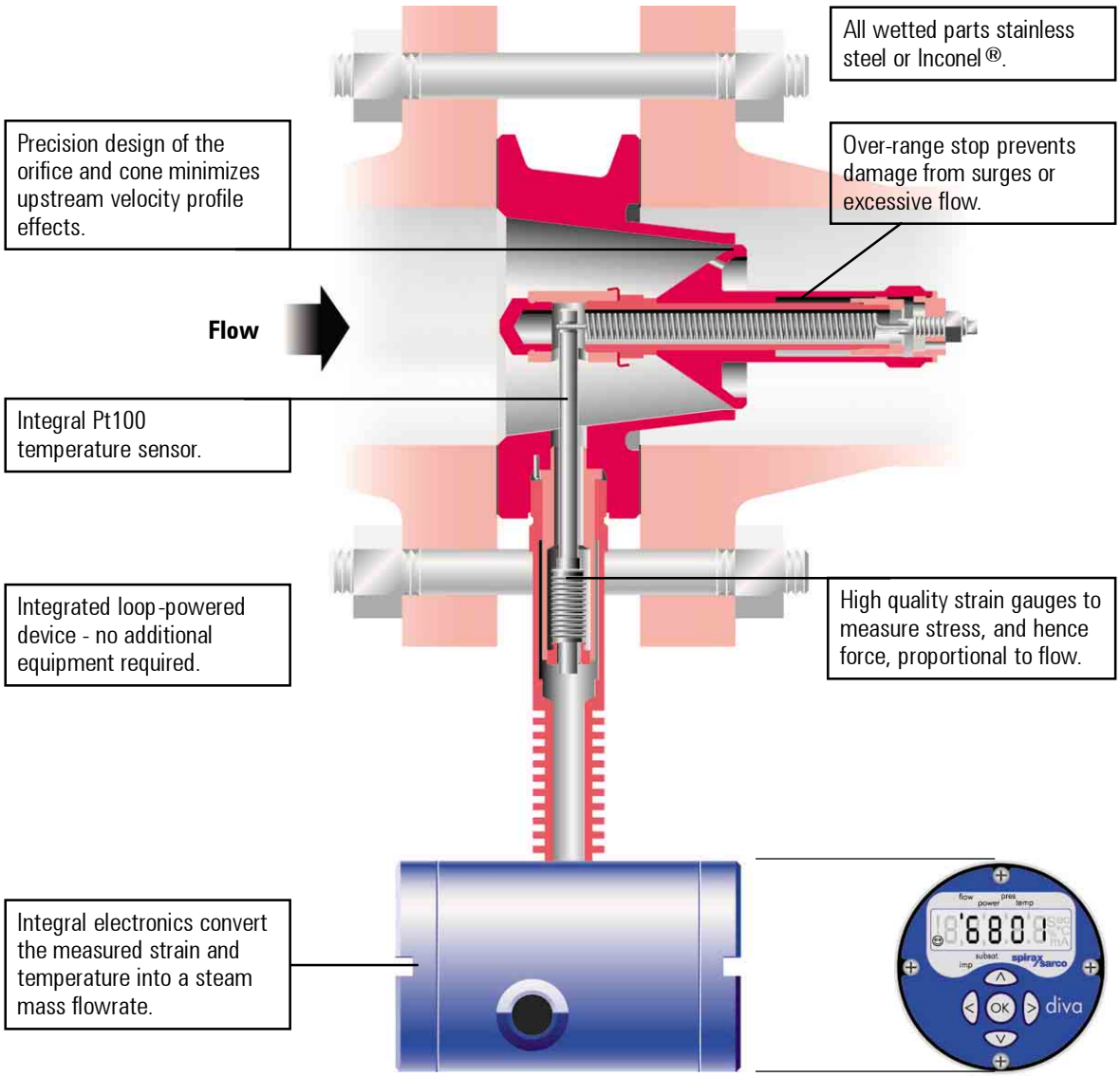
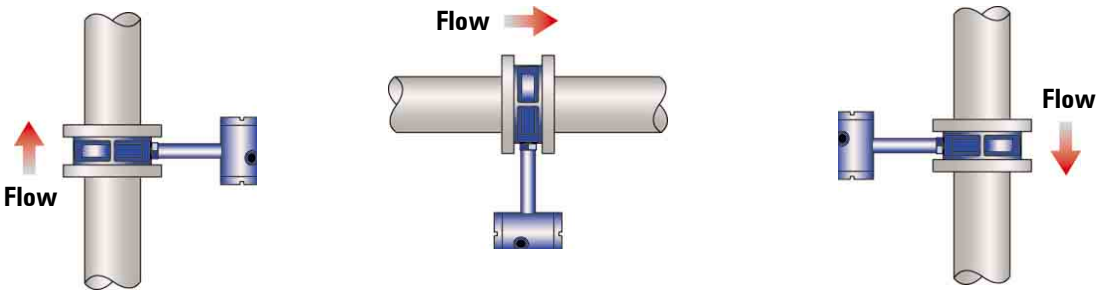


Fig. 4.3.16 The DIVA flowmeter

Flow orientations

The orientation of the DIVA flowmeter can have an effect on the operating performance. Installed in horizontal pipe, the DIVA has a steam pressure limit of 32 bar g, and a 50:1 turndown. As shown in Figure 4.3.17, if the DIVA is installed with a vertical flow direction then the pressure limit is reduced, and the turndown ratio will be affected if the flow is vertically upwards.



Flow orientation:
Vertically upwards
Turndown:
Up to 30:1
Pressure limitation:
11 bar g

Flow orientation:
Horizontal
Turndown:
Up to 50:1
Pressure limitation:
32 bar g

Flow orientation:
Vertically downwards
Turndown:
Up to 50:1
Pressure limitation:
11 bar g

Fig. 4.3.17 Flow orientation

Pitot tubes

In large steam mains, the cost of providing a full bore flowmeter can become extremely high both in terms of the cost of the flowmeter itself, and the installation work required.

A Piot tube flowmeter can be an inexpensive method of metering. The flowmeter itself is cheap, it is cheap to install, and one flowmeter may be used in several applications.

Pitot tubes, as introduced in Module 4.2, are a common type of insertion flowmeter. Figure 4.3.18 shows the basis for a Pitot tube, where a pressure is generated in a tube facing the flow, by the velocity of the fluid. This 'velocity' pressure is compared against the reference pressure (or static pressure) in the pipe, and the velocity can be determined by applying a simple equation.

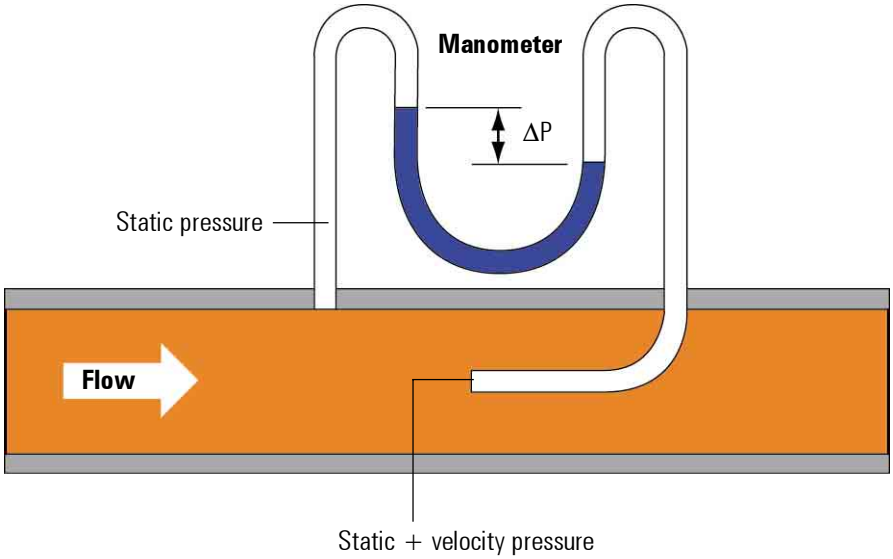


Fig. 4.3.18 A diagrammatic pitot tube

In practice, two tubes inserted into a pipe would be cumbersome, and a simple Pitot tube will consist of one unit as shown in Figure 4.3.19. Here, the hole measuring the velocity pressure and the holes measuring the reference or static pressure are incorporated in the same device.

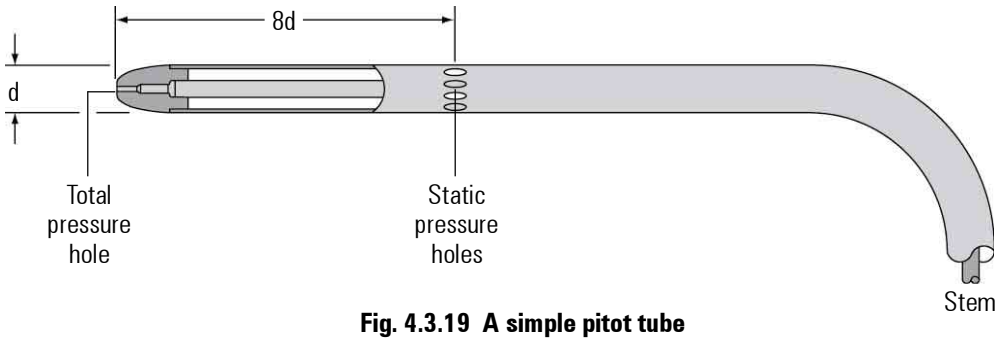


Fig. 4.3.19 A simple pitot tube

Because the simple Pitot tube (Figure 4.3.19) only samples a single point, and, because the flow profile of the fluid (and hence velocity profile) varies across the pipe, accurate placement of the nozzle is critical.

Note that a square root relationship exists between velocity and pressure drop (see Equation 4.2.13). This limits the accuracy to a small turndown range.

$$u_1 = \sqrt{\frac{2\ \Delta P}{\rho}}$$ **Equation 4.2.13**

Where:
u_1 = The fluid velocity in the pipe
Δp = Dynamic pressure - Static pressure
ρ = Density

The averaging Pitot tube

The averaging Pitot tube (Figure 4.3.20) was developed with a number of upstream sensing tubes to overcome the problems associated with correctly siting the simple type of Pitot tube. These sensing tubes sense various velocity pressures across the pipe, which are then averaged within the tube assembly to give a representative flowrate of the whole cross section.

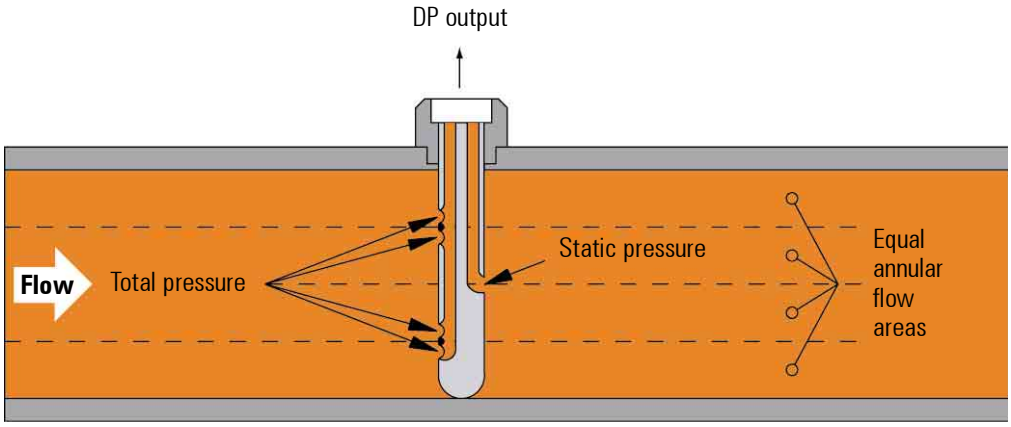


Fig. 4.3.20 The averaging pitot tube

Advantages of the Pitot tube:

- Presents little resistance to flow.
- Inexpensive to buy.
- Simple types can be used on different diameter pipes.

Disadvantages of the Pitot tube:

- Turndown is limited to approximately 4:1 by the square root relationship between pressure and velocity as discussed in Module 4.2.
- If steam is wet, the bottom holes can become effectively blocked. To counter this, some models can be installed horizontally.
- Sensitive to changes in turbulence and needs careful installation and maintenance.
- The low pressure drop measured by the unit, increases uncertainty, especially on steam.
- Placement inside the pipework is critical.

Typical applications for the Pitot tube:

- Occasional use to provide an indication of flowrate.
- Determining the range over which a more appropriate steam flowmeter may be used.

Vortex shedding flowmeters

These flowmeters utilise the fact that when a non-streamlined or 'bluff' body is placed in a fluid flow, regular vortices are shed from the rear of the body. These vortices can be detected, counted and displayed. Over a range of flows, the rate of vortex shedding is proportional to the flowrate, and this allows the velocity to be measured.

The bluff body causes a blockage around which the fluid has to flow. By forcing the fluid to flow around it, the body induces a change in the fluid direction and thus velocity. The fluid which is nearest to the body experiences friction from the body surface and slows down. Because of the area reduction between the bluff body and the pipe diameter, the fluid further away from the body is forced to accelerate to pass the necessary fluid through the reduced space. Once the fluid has passed the bluff body, it strives to fill the space produced behind it, which in turn causes a rotational motion in the fluid creating a spinning vortex.

The fluid velocity produced by the restriction is not constant on both sides of the bluff body. As the velocity increases on one side it decreases on the other. This also applies to the pressure. On the high velocity side the pressure is low, and on the low velocity side the pressure is high. As pressure attempts to redistribute itself, the high pressure region moving towards the low pressure region, the pressure regions change places and vortices of different strengths are produced on alternate sides of the body.

The shedding frequency and the fluid velocity have a near-linear relationship when the correct conditions are met.

The frequency of shedding is proportional to the Strouhal number (Sr), the flow velocity, and the inverse of the bluff body diameter. These factors are summarised in Equation 4.3.2.

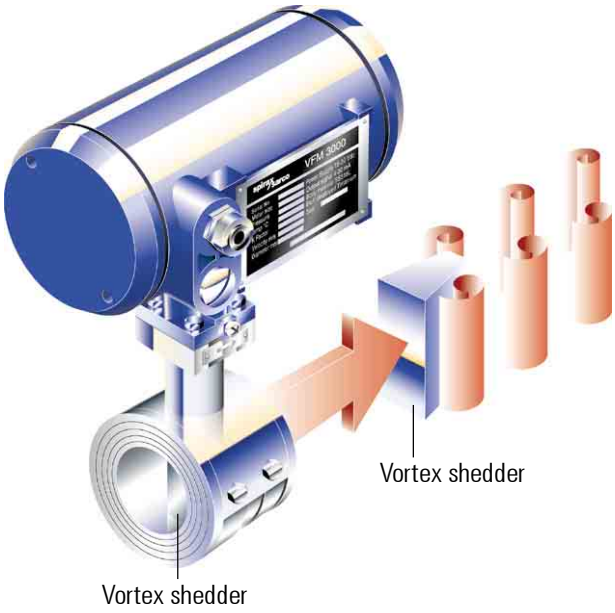


Fig. 4.3.21 Vortex shedding flowmeter

$$f \propto \frac{Sr\ u}{d}$$ **Equation 4.3.2**

Where:
f = Shedding frequency (Hz)
Sr = Strouhal number (dimensionless)
u = Mean pipe flow velocity (m/s)
d = Bluff body diameter (m)

The Strouhal number is determined experimentally and generally remains constant for a wide range of Reynolds numbers;which indicates that the shedding frequency will remain unaffected by a change in fluid density, and that it is directly proportional to the velocity for any given bluff body diameter. For example:

$f = k \times u$

Where:

k = A constant for all fluids on a given design of flowmeter.

Hence:

$$u = \frac{f}{k}$$

Then the volume flowrate q_v in a pipe can be calculated as shown in Equation 4.3.3:

$$q_v = A \frac{f}{k}$$ **Equation 4.3.3**

Where:
A = Area of the flowmeter bore (m²)

Advantages of vortex shedding flowmeters:

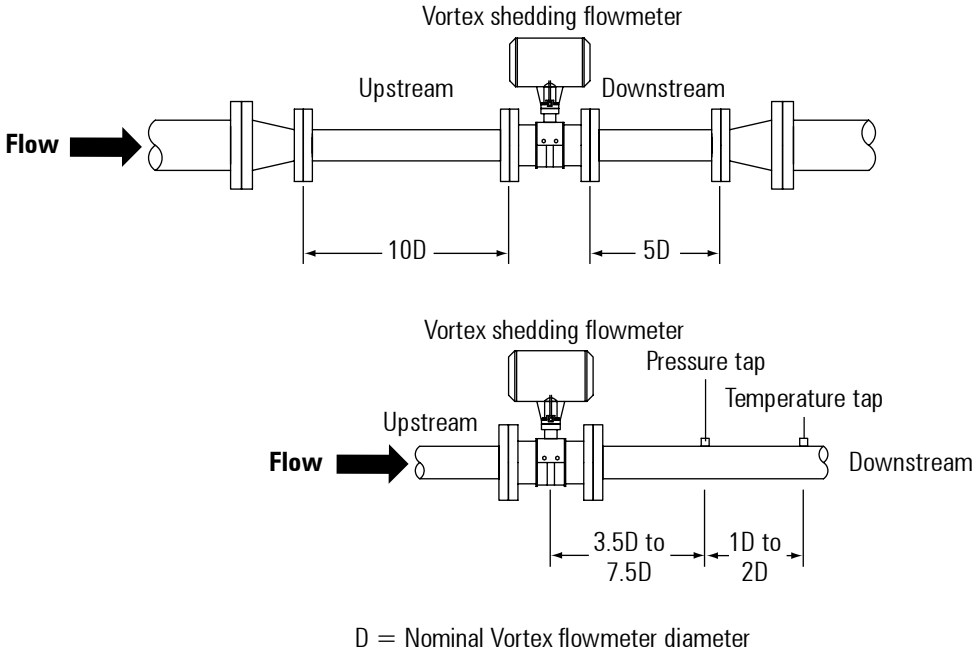
- Reasonable turndown (providing high velocities and high pressure drops are acceptable).
- No moving parts.
- Little resistance to flow.

Disadvantages of vortex shedding flowmeters:

- At low flows, pulses are not generated and the flowmeter can read low or even zero.
- Maximum flowrates are often quoted at velocities of 80 or 100 m/s, which would give severe problems in steam systems, especially if the steam is wet and/or dirty. Lower velocities found in steam pipes will reduce the capacity of vortex flowmeters.
- Vibration can cause errors in accuracy.
- Correct installation is critical as a protruding gasket or weld beads can cause vortices to form, leading to inaccuracy.
- Long, clear lengths of upstream pipework must be provided, as for orifice plate flowmeters.

Typical applications for vortex shedding flowmeters:

- Direct steam measurements at both boiler and point of use locations.
- Natural gas measurements for boiler fuel flow.



D = Nominal Vortex flowmeter diameter

Fig. 4.3.22 Vortex shedding flowmeter - typical installations

Questions

1. A 50 mm bore steam pipe lifts up and over a large industrial doorway. An orifice flowmeter is fitted in the horizontal pipe above the doorway, with a 1.6 m straight run before it. The β ratio is 0.7. What will be the effect of the straight run of pipe before the flowmeter?

a| No effect. 1.45 m is the recommended minimum length of upstream pipe ☐

b| The accuracy of the flowmeter will be reduced because the flow will be laminar, not turbulent ☐

c| The accuracy of the flowmeter will be reduced because of increased turbulence following the preceding pipe bend ☐

d| The accuracy will be reduced because of the swirling motion of the flow ☐

2. Why are turbine flowmeters frequently fitted in a bypass around an orifice plate flowmeter?

a| To minimise cost ☐

b| To improve accuracy ☐

c| To avoid the effects of suspended moisture particles in the steam ☐

d| Because in a bypass, turbine flowmeters will be less susceptible to inaccuracies due to low flowrates ☐

3. What is the likely effect of a spring loaded variable area flowmeter (installed as in Figure 4.3.14) on steam for long periods?

a| The cone (float) can be damaged by wet steam if no separator is fitted ☐

b| The turndown will be less than 25:1 ☐

c| No effect ☐

d| The differential pressure across the flowmeter will be higher, so accuracy will be reduced ☐

4. What feature makes the differential pressure type of spring loaded variable area flowmeter suitable for a turndown of 100:1?

a| The pass area, which remains constant under all flow conditions ☐

b| The pass area, which reduces with increasing flow ☐

c| The moving cone which gives a linear relationship between flow and pressure drop ☐

d| The moving cone which provides a decrease in flowrate as the differential pressure increases ☐

5. Which of the following is a feature of the Vortex shedding flowmeter against an orifice plate flowmeter?

a| It is suitable for steam with velocities up to 80 – 100 m/s ☐

b| It has a higher resistance to flow and therefore easier to measure differential pressure ☐

c| It has a higher turndown ☐

d| It has no moving parts ☐

6. Which of the following are an advantage of the spring loaded variable area flowmeter over the Vortex shedding flowmeter?

a| Shorter lengths of straight pipe before and after the flowmeter ☐

b| Higher turndown capability at practical working velocities ☐

c| Not susceptible to vibration or turbulence ☐

d| All of the above ☐

Answers

1: a, 2: d, 3: a, 4: c, 5: c, 6: d

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Module 4.4

Instrumentation

Instrumentation

A steam flowmeter comprises two parts:

1. The 'primary' device or pipeline unit, such as an orifice plate, located in the steam flow.
2. The 'secondary' device, such as a differential pressure cell, that translates any signals into a usable form.

In addition, some form of electronic processor will exist which can receive, process and display the information. This processor may also receive additional signals for pressure and/or temperature to enable density compensation calculations to be made.

Figure 4.4.1 shows a typical system.

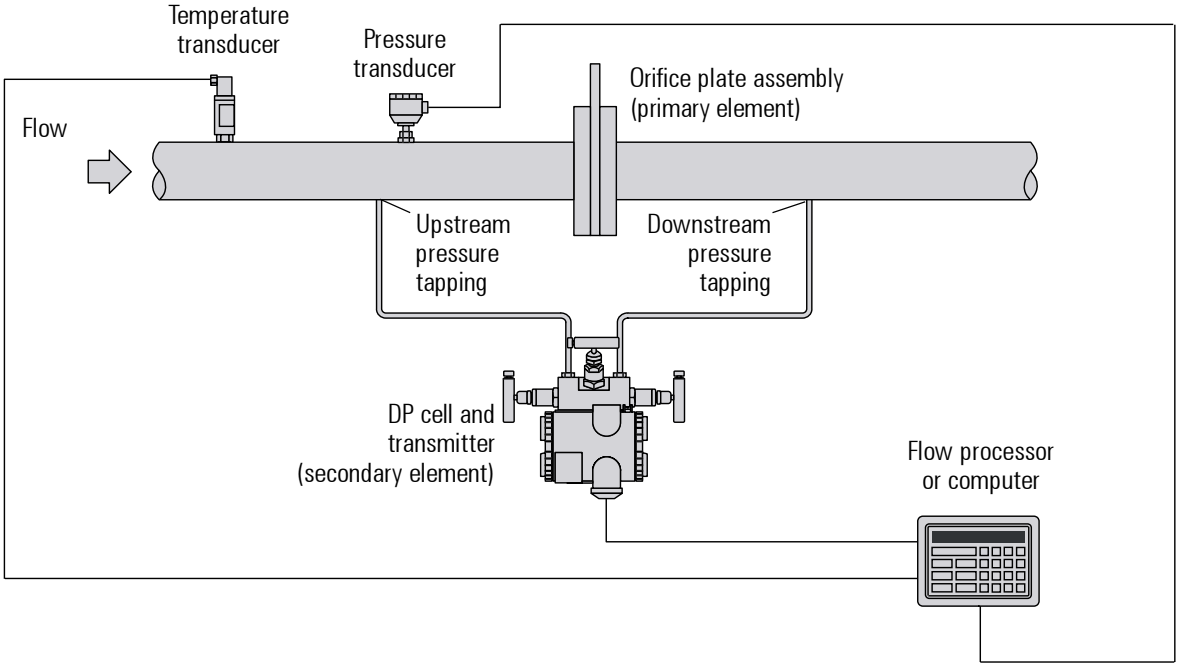
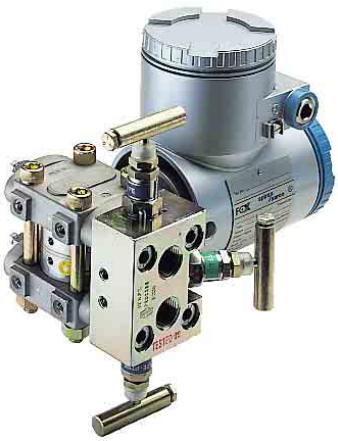


Fig. 4.4.1 A typical orifice plate steam flowmetering station

Differential pressure cells (DP cells)

If the pipeline unit is a differential pressure measuring device, for example an orifice plate flowmeter or Pitot tube, and an electronic signal is required, the secondary device will be a Differential Pressure (DP or ΔP) cell. This will change the pressure signal to an electrical signal. This signal can then be relayed on to an electronic processor capable of accepting, storing and processing these signals, as the user requires.



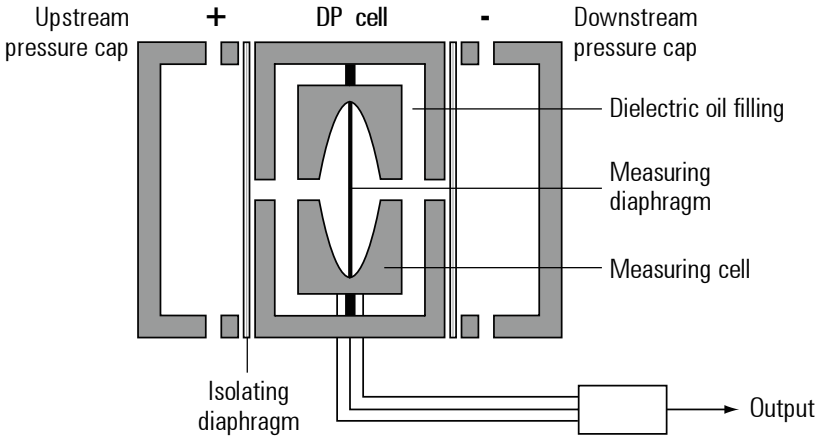


Fig. 4.4.2 Simple DP cell

A typical DP cell is an electrical capacitance device, which works by applying a differential pressure to either side of a metal diaphragm submerged in dielectric oil. The diaphragm forms one plate of a capacitor, and either side of the cell body form the stationary plates. The movement of the diaphragm produced by the differential pressure alters the separation between the plates, and alters the electrical capacitance of the cell, which in turn results in a change in the electrical output signal.

The degree of diaphragm movement is directly proportional to the pressure difference.

The output signal from the measuring cell is fed to an electronic circuit where it is amplified and rectified to a load-dependent 4-20 mA dc analogue signal. This signal can then be sent to a variety of devices to:

- Provide flowrate indication.
- Be used with other data to form part of a control signal.

The sophistication of this apparatus depends upon the type of data the user wishes to collect.

Advanced DP cells

The advancement of microelectronics, and the pursuit of increasingly sophisticated control systems has led to the development of more advanced differential pressure cells. In addition to the basic function of measuring differential pressure, cells can now be obtained which:

- Can indicate actual (as distinct from differential) pressure.
- Have communication capability, for example HART® or Fieldbus.
- Have self-monitoring or diagnostic facilities.
- Have 'on-board' intelligence allowing calculations to be carried out and displayed locally.
- Can accept additional inputs, such as temperature and pressure.

Data collection

Many different methods are available for gathering and processing of this data, these include:

- Dedicated computers.
- Stand alone PLCs (Programmable Logic Controller systems).
- Centralised DCSs (Distributed Control Systems).
- SCADAs (Supervisory Control And Data Acquisition systems).

One of the easier methods for data collection, storage, and display is a dedicated computer. With the advent of the microprocessor, extremely versatile flow monitoring computers are now available.

The display and monitoring facilities provided by these can include:

- Current flowrate.
- Total steam usage.
- Steam temperature/pressure.
- Steam usage over specified time periods.
- Abnormal flowrate, pressure or temperature, and trigger remote alarms.
- Compensate for density variations.
- Interface with chart recorders.
- Interface with energy management systems.

Some can more accurately be termed energy flowmeters since, in addition to the above variables, they can use time, steam tables, and other variables to compute and display both the power (kW or Btu/h) and heat energy usage (kJ or Btu).

In addition to the computer unit, it is sometimes beneficial to have a local readout of flowrate.

Data analysis

Data collection, whether it is manual, semi-automatic or fully automatic, will eventually be used as a management tool to monitor and control energy costs. Data may need to be gathered over a period of time to give an accurate picture of the process costs and trends. Some production processes will require data on a daily basis, although the period often preferred by industrial users is the production week.

Microcomputers with software capable of handling statistical calculations and graphics are commonly used to analyse data. Once the measuring system is in place, the first objective is to determine a relationship between the process (for example tonnes of product/hour) and energy consumption (for example kg of steam/hour). The usual means of achieving this is to plot consumption (or specific consumption) against production, and to establish a correlation. However, some caution is required in interpreting the precise nature of this relationship. There are two main reasons for this:

- Secondary factors may affect energy consumption levels.
- Control of primary energy use may be poor, obscuring any clear relationship.

Statistical techniques can be used to help identify the effect of multiple factors. It should be noted that care should be taken when using such methods, as it is quite easy to make a statistical relationship between two or more variables that are totally independent.

Once these factors have been identified and taken into account, the standard energy consumption can then be determined. This is the minimum energy consumption that is achievable for the current plant and operating practices.

The diagram in Figure 4.4.3 plots a typical relationship between production and consumption.

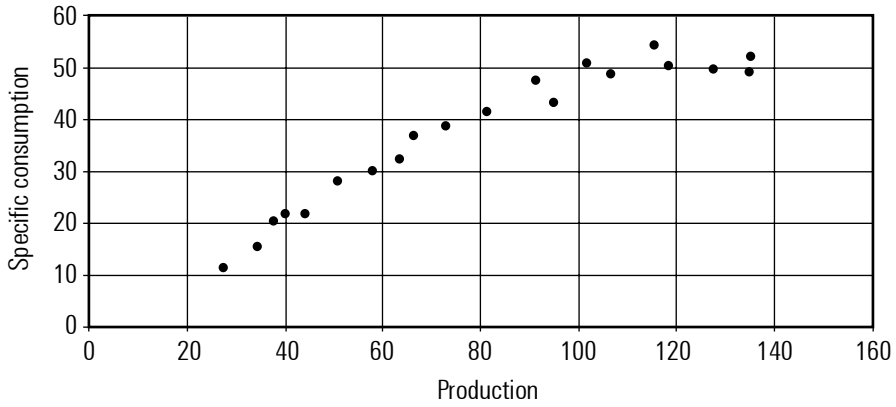


Fig. 4.4.3 Typical relationship between production and steam consumption

Once the relationship between steam consumption and factory production has been established, it becomes the basis/standard to which all future production can be measured.

Using the standard, the managers of individual sections can then receive regular reports of their energy consumption and how this compares to the standard. The individual manager can then analyse his/her plant performance by asking:

- How does consumption compare with the standard?
- Is the consumption above or below the standard, and by how much does it vary?
- Are there any trends in the consumption?

If there is a variation in consumption it may be for a number of reasons, including:

- Poor control of energy consumption.
- Defective equipment, or equipment requiring maintenance.
- Seasonal variations.

To isolate the cause, it is necessary to first check past records, to determine whether the change is a trend towards increased consumption or an isolated case. In the latter case, checks should then be carried out around the plant for leaks or faulty pieces of equipment. These can then be repaired as required.

Standard consumption has to be an achievable target for plant managers, and a common approach is to use the line of best fit based on the average rather than the best performance that can be achieved (see Figure 4.4.4).

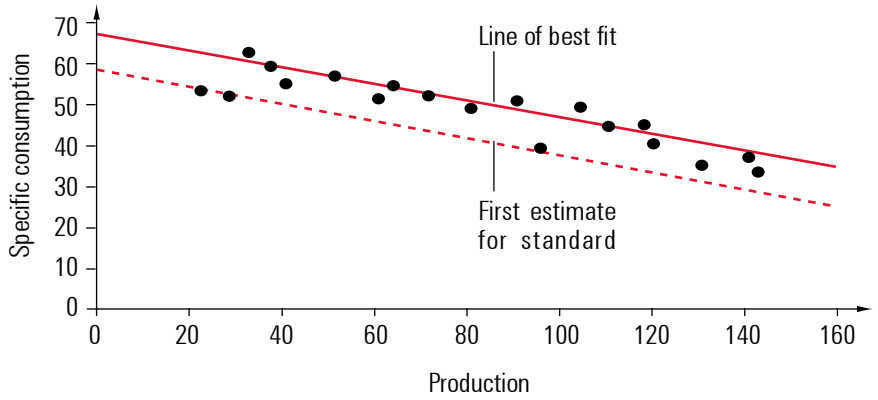


Fig. 4.4.4 Relationship between production and specific steam consumption

Once the standard has been determined, this will be the new energy consumption datum line.

This increase in energy consciousness will inevitably result in a decrease in energy costs and overall plant running costs, consequently, a more energy efficient system.

Special requirements for accurate steam flow measurement

As mentioned earlier in Block 4, flowmeters measure velocity; additional values for cross sectional area (A) and density (ρ) are required to enable the mass flowrate (q_m) to be calculated. For any installation, the cross sectional area will remain constant, the density (ρ) however will vary with pressure and dryness fraction.

The next two sections examine the effect of pressure and dryness fraction variation on the accuracy on steam flowmeter installations.

Pressure variation

In an ideal world, the pressure in process steam lines would remain absolutely constant. Unfortunately, this is very rarely the case with varying loads, boiler pressure control dead-bands, frictional pressure losses, and process parameters all contributing to pressure variations in the steam main.

Figure 4.4.5 shows the duty cycle for a saturated steam application. Following start-up, the system pressure gradually rises to the nominal 5 bar g but due to process load demands the pressure varies throughout the day. With a non-pressure compensated flowmeter, the cumulative error can be significant.

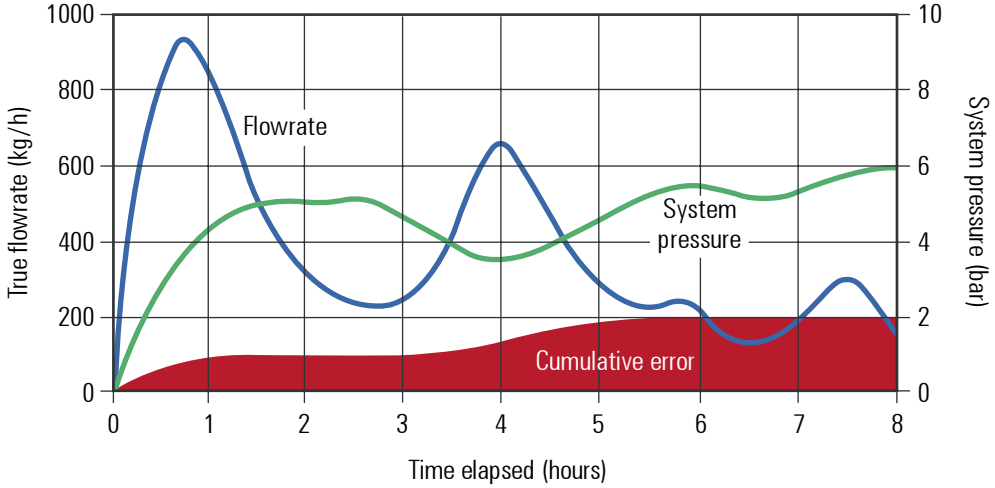


Fig. 4.4.5 Steam usage with flowrate and pressure

Some steam flowmetering systems do not have inbuilt density compensation, and are specified to operate at a single, fixed line pressure. If the line pressure is actually constant, then this is acceptable. However, even relatively small pressure variations can affect flowmeter accuracy. It may be worth noting at this point that different types of flowmeter may be affected in different ways.

Velocity flowmeters

The output signal from a vortex shedding flowmeter is a function of the velocity of flow only. It is independent of the density, pressure and temperature of the fluid that it is monitoring. Given the same flow velocity, the uncompensated output from a vortex shedding flowmeter is the same whether it is measuring 3 bar g steam, 17 bar g steam, or water.

Flow errors, therefore are a function of the error in density and may be expressed as shown in Equation 4.4.1.

$$\varepsilon = \left[\left(\frac{\text{Specified } \rho}{\text{Actual } \rho}\right) - 1\right] \times 100\% \qquad \textbf{Equation 4.4.1}$$

Where:

ε = Flow error expressed as a percentage of the actual flow

Specified ρ = Density of steam at the specified steam line pressure

Actual ρ = Density of steam at the actual line pressure

Example 4.4.1

As a basis for the following examples, determine the density (ρ) of dry saturated steam at 4.2 bar g and 5.0 bar g.

Pressure bar g	Specific volume (from steam tables) m^3/kg	Density (ρ) kg/m^3
4.2	0.360 4	$= \frac{1}{0.360\,4} = 2.774\,8\ kg/m^3$
5.0	0.315	$= \frac{1}{0.315} = 3.174\,9\ kg/m^3$

Example 4.4.2

A vortex shedding steam flowmeter specified to be used at 5 bar g is used at 4.2 bar g.

Use Equation 4.4.1 and the data from Example 4.4.1 to determine the resulting error (ε).

Where:

Actual ρ = 2.774 8 kg/m³

Specified ρ = 3.174 9 kg/m³

$$\varepsilon = \left[\left(\frac{3.174\,9}{2.774\,8}\right) - 1\right] \times 100\% = \mathbf{+14.42\%}$$

Therefore, the uncompensated vortex flowmeter will over read by 14.42%

As one of the characteristics of saturated steam (particularly at low pressures up to about 6 bar g) is that the density varies greatly for a small change in pressure, density compensation is essential to ensure accurate readings.

Equation 4.4.1 may be used to generate a chart showing the expected error in flow for an error in pressure, as shown in Figure 4.4.6.

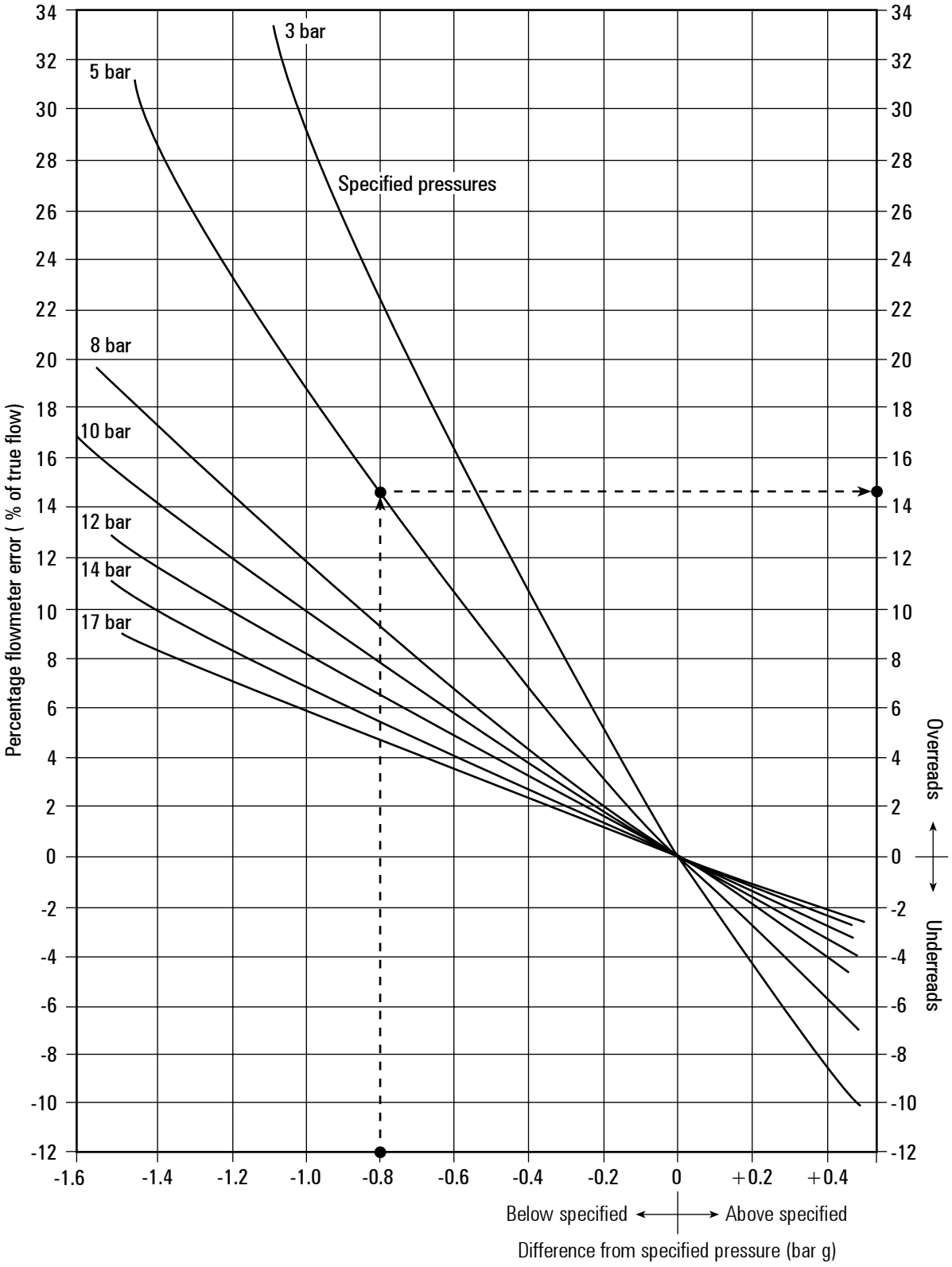


Fig. 4.4.6 Vortex shedding flowmeter - % errors due to lack of density compensation

Differential pressure flowmeters

The output signal from an orifice plate and cell takes the form of a differential pressure signal. The measured mass flowrate is a function of the shape and size of the hole, the square root of the differential pressure and the square root of the density of the fluid. Given the same observed differential pressure across an orifice plate, the derived mass flowrate will vary with the square root of the density.

As for vortex flowmeters, running an orifice plate flowmeter at a pressure other than the specified pressure will give rise to errors.

The percentage error may be calculated using Equation 4.4.2.

$$\% \text{ error } (\varepsilon) = \left(\frac{\sqrt{\text{Specified } \rho}}{\sqrt{\text{Actual } \rho}} - 1 \right) \times 100 \qquad \textbf{Equation 4.4.2}$$

Example 4.4.3.
An orifice plate steam flowmeter specified to be used at 5 bar g is used at 4.2 bar g.

Use Equation 4.4.2 to determine the resulting percentage error (ε).

Actual ρ = 2.774 8 kg/m³

Specified ρ = 3.174 9 kg/m³

$$\varepsilon = \left[\left(\frac{\sqrt{3.1749}}{\sqrt{2.7748}} \right) - 1 \right] \times 100\%$$

$$\varepsilon = \left[\left(\frac{1.781\,9}{1.665\,8} \right) - 1 \right] \times 100\% = +6.96\%$$

The positive error means the flowmeter is overreading, in this instance, for every 100 kg of steam passing through, the flowmeter registers 106.96 kg.

Equation 4.4.2 may be used to generate a chart showing the expected error in flow for an error in pressure, as shown in Figure 4.4.7.

When comparing Figure 4.4.6 with Figure 4.4.7, it can be seen that the % error due to lack of density compensation for the vortex flowmeter is approximately double the % error for the orifice plate flowmeter. Therefore, density compensation is essential if steam flow is to be measured accurately. If the steam flowmeter does not include an inbuilt density compensation feature then extra pressure and/or temperature sensors must be provided, linked back to the instrumentation system.

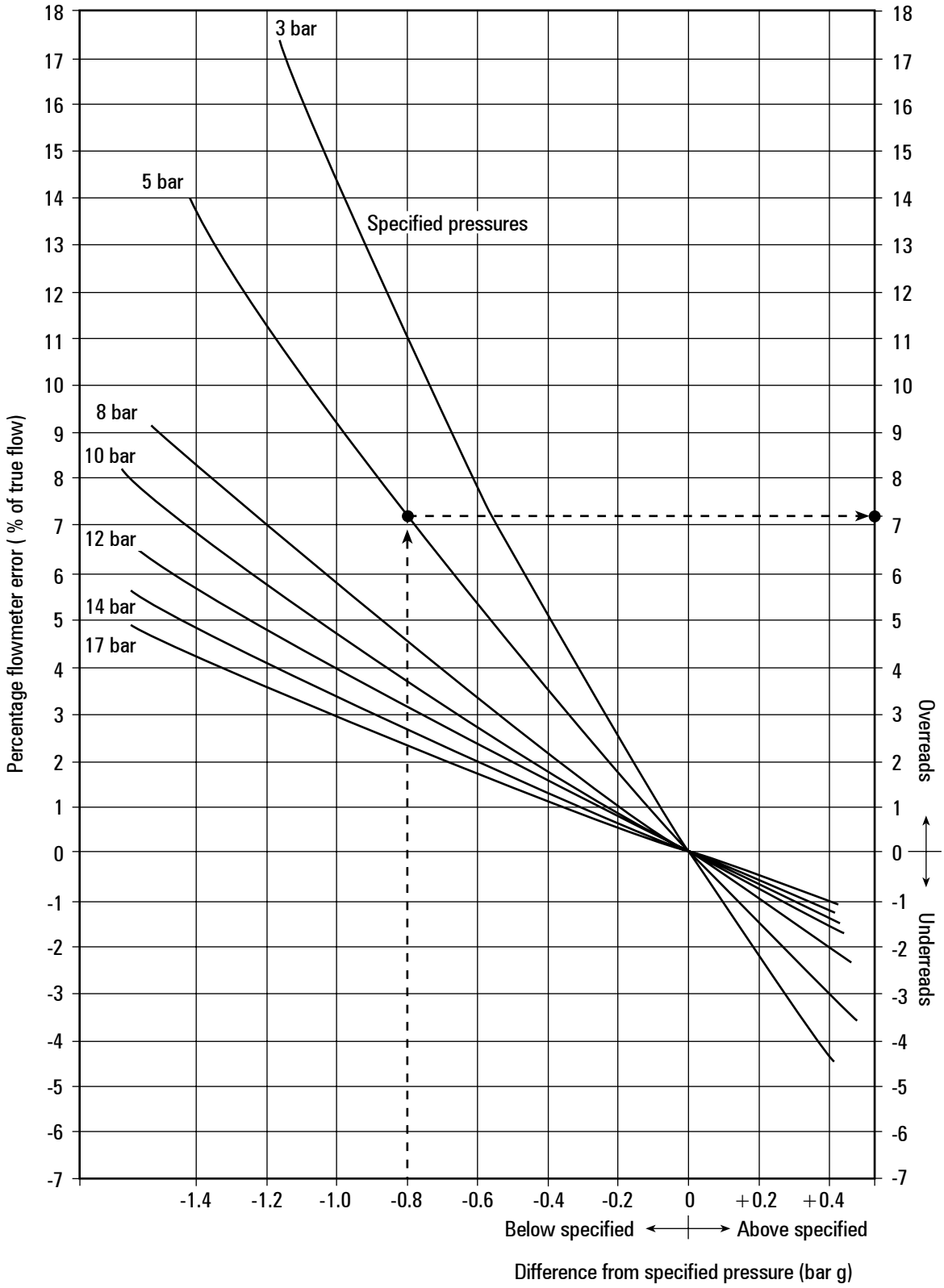


Fig. 4.4.7 Orifice plate flowmeter - % errors due to lack of density compensation

Dryness fraction variation

The density of a cubic metre of wet steam is higher than that of a cubic metre of dry steam. If the quality of steam is not taken into account as the steam passes through the flowmeter, then the indicated flowrate will be lower than the actual value.

Dryness fraction (χ) has already been discussed in Module 2.2, but to reiterate; dryness fraction is an expression of the proportions of saturated steam and saturated water. For example, a kilogram of steam with a dryness fraction of 0.95, contains 0.95 kilogram of steam and 0.05 kilogram of water.

Example 4.4.4
As a basis for the following examples, determine the density (ρ) of dry saturated steam at 10 bar g with dryness fractions of 1.0 and 0.95.

Dryness fraction (χ) = 1.0

Specific volume of dry steam (v_g) at 10 bar g (from steam tables) = 0.1773 m³/kg

$$\text{Density } (\rho) = \frac{1}{0.1773 \text{ m}^3/\text{kg}}$$

With χ having a dryness fraction of 1.0, density (ρ) = 5.6414 kg/m³

Dryness fraction (χ) = 0.95

Specific volume of dry steam (v_g) at 10 bar g (from steam tables) = 0.1773 m³/kg

Specific volume of water (v_f) at 10 bar g (from steam tables) = 0.0011

Volume occupied by steam @ χ = 0.95 = 0.95 x 0.1773 = 0.1684 m³

Volume occupied by water @ χ = 0.95 = 0.05 x 0.0011 = 0.000055 m³

Total volume occupied by steam and water = 0.1684 + 0.000055 m³ = 0.168 455 m³

$$\textbf{Density } (\rho) \textbf{ of mixture} = \frac{1}{0.168\,455 \text{ m}^3/\text{kg}} = \textbf{5.936 3 kg/m}^3$$

Difference in density = 5.936 3 kg/m³ - 5.641 4 kg/m³ **= 0.2949 kg/m³**

Therefore, a reduction in volume is calculated to be 4.97%.

Important note:
The proportion of the volume occupied by the water is approximately 0.03% of that occupied by the steam. For most practical purposes the volume occupied by the water can be ignored and the density (ρ) of wet steam can be defined as shown in Equation 4.4.3.

$$\text{Density of steam} = \frac{1}{v_g \chi} \qquad \textbf{Equation 4.4.3}$$

Where:
v_g = Specific volume of dry steam
χ = Dryness fraction

Using Equation 4.4.3, find the density of wet steam at 10 bar g with a dryness fraction (χ) of 0.95.

The specific volume of dry steam at 10 bar g (v_g) = 0.177 3 m³/kg

$$\text{Density} = \frac{1}{v_g \times \chi} = \frac{1}{0.1773 \times 0.95} = 5.937 \text{ kg/m}^3$$

This compares to 5.936 3 kg/m³ when calculated as a mixture.

The effect of dryness fraction on flowmeters that measure differential pressure

To reiterate earlier comments regarding differential pressure flowmeter errors, mass flowrate (q_m) will be proportional to the square root of the density (ρ), and density is related to the dryness fraction. Changes in dryness fraction will have an effect on the flow indicated by the flowmeter.

Equation 4.4.4 can be used to determine the relationship between actual flow and indicated flow:

$$\frac{\text{Indicated mass flowrate}}{\text{Actual flowrate}} = \frac{\sqrt{\text{density at calibrated dryness fraction}}}{\sqrt{\text{density at actual dryness fraction}}} \qquad \textbf{Equation 4.4.4}$$

All steam flowmeters will be calibrated to read at a pre-determined dryness fraction (χ), the typically value is 1. Some steam flowmeters can be recalibrated to suit actual conditions.

Example 4.4.5

Using the data from Example 4.4.4, determine the percentage error if the actual dryness fraction is 0.95 rather than the calibrated value of 1.0, and the steam flowmeter was indicating a flowrate of 1 kg/s.

$$\frac{\text{Indicated flowrate}}{\text{Actual flowrate}} = \frac{\sqrt{\text{density at } \chi = 1.00}}{\sqrt{\text{density at } \chi = 0.95}}$$

$$\frac{1 \text{ kg/s}}{\text{Actual flowrate}} = \frac{\sqrt{5.6414}}{\sqrt{5.9294}} = \frac{2.3752}{2.4350}$$

$$\text{Actual flowrate} = 1.0252 \text{ kg/s}$$

$$\text{Percentage error} = \frac{\text{Indicated flow - Actual flow}}{\text{Actual flow}} \times 100\%$$

$$\textbf{Percentage error} = \frac{1 - 1.0252}{1.0252} \times 100\% = \textbf{-2.46\%}$$

Therefore, the negative sign indicates that the flowmeter under-reads by 2.46%.

Equation 4.4.4 is used to compile the graph shown in Figure 4.4.8.

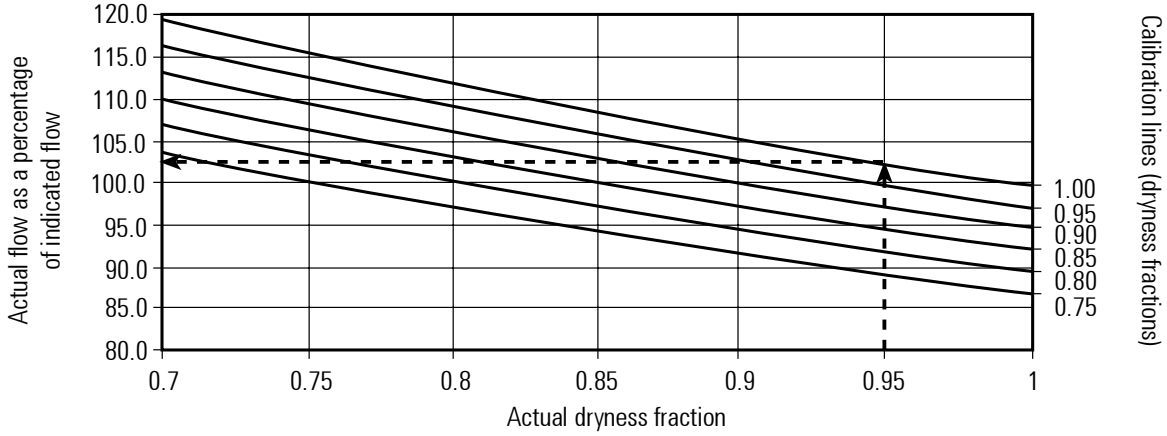


Fig. 4.4.8 Effect of dryness fraction on differential pressure flowmeters

The effect of dryness fraction on vortex flowmeters

It can be argued that dryness fraction, within sensible limitations, is of no importance because:

- Vortex flowmeters measure velocity.
- The volume of water in steam with a dryness fraction of, for example, 0.95, in proportion to the steam is very small.
- It is the condensation of dry steam that needs to be measured.

However, independent research has shown that the water droplets impacting the bluff body will cause errors and as vortex flowmeters tend to be used at higher velocities, erosion by the water droplets is also to be expected. Unfortunately, it is not possible to quantify these errors.

Conclusion

Accurate steam flowmetering depends on:

- **Taking pressure variations into account -** Pressure will vary in any steam system, and it is clearly futile to specify a flowmeter with an accuracy of ±2% if pressure variations alone can give errors of ±10%. The steam flowmetering package must include density compensation.
- **Predictable dryness fraction -** Measurement of dryness fraction is very complex; a much easier and better option is to install a steam separator prior to any steam flowmeter. This will ensure that the dryness fraction is always close to 1.0, irrespective of the condition of the steam supplied.

Superheated steam

With saturated steam there is a fixed relationship between steam pressure and steam temperature. Steam tables provide detailed information on this relationship. To apply density compensation on saturated steam, it is only necessary to sense either steam temperature or steam pressure to determine the density (ρ). This signal can then be fed, along with the flow signal, to the flow computer, where, assuming the computer contains a steam table algorithm, it will then do the calculations of mass flowrate.

However, superheated steam is close to being a gas and no obvious relationship exists between temperature and pressure. When measuring superheated steam flowrates, both steam pressure and steam temperature must be sensed and signalled simultaneously. The flowmeter instrumentation must also include the necessary steam table software to enable it to compute superheated steam conditions and to indicate correct values.

If a differential pressure type steam flowmeter is installed which does not have this instrumentation, a flow measurement error will always be displayed if superheat is present. Figure 4.4.9 shows the percentage errors for various degrees of superheat for flowmeters not fitted with temperature compensation.

Pressure	Amount of superheat			
bar g	1°C	5°C	10°C	50°C
1	1.5	8.3	17.0	105.0
2	1.4	7.6	16.1	95.9
3	1.4	7.5	15.0	90.5
4	1.3	7.0	14.5	86.6
5	1.3	6.8	14.1	83.5
6	1.3	6.8	13.8	81.4
7	1.3	6.5	13.7	79.0
8	1.3	6.5	13.3	77.8
9	1.3	6.4	12.9	76.5
10	1.3	6.3	12.8	75.0
11	1.2	6.3	12.7	73.9
12	1.2	6.1	12.3	72.9
13	1.2	6.0	12.3	71.0
14	1.2	6.0	12.2	71.4
15	1.2	6.0	12.1	70.7
16	1.2	5.9	12.1	70.0
17	1.1	5.9	12.1	69.5

Fig. 4.4.9 Percentage errors for over-reading various degrees of superheat for flowmeters not fitted with temperature compensation

Example 4.4.6
Consider a steam flowmeter fitted with pressure reading equipment, but not temperature reading equipment. The flowmeter thinks it is reading saturated steam at its corresponding temperature. With superheated steam at 4 bar g and 10°C superheat passing through the flowmeter, determine the actual flowrate if the flowmeter displays a flowrate of 250 kg/h.

Equation 4.4.5 can be used to calculate the actual value from the displayed value.

$$\text{Actual value} = \frac{\text{Displayed value}}{\left[1+\left(\frac{\%\ \text{error}}{100}\right)\right]}$$

Equation 4.4.5

With steam at a line pressure of 4 bar g and 10°C superheat, the displayed value of mass flow will be 14.5% higher than the actual value.

For example, if the display shows 250 kg/h under the above conditions, then the actual flowrate is given by:

$$\text{Actual value} = \frac{250}{[1 + 0.145]} = 218\ \text{kg/h}$$

Questions

1. A flowmeter used on superheated steam at 10 bar g and 234°C displays a flow of 1 000 kg/h. If the flowmeter does not incorporate temperature and pressure compensation what is the actual flowrate?

a| 1 000 kg/h ☐
b| 571 kg/h ☐
c| 1 339 kg/h ☐
d| 822 kg/h ☐

2. A flowmeter measuring differential pressure calibrated for saturated steam at 7 bar g displays a flowrate of 800 kg/h. What will be the effect of the steam being 3% wet?

a| The actual flow will remain the same as that indicated ☐
b| The actual flow will be 406 kg/h ☐
c| The actual flow will be 788 kg/h ☐
d| The actual flow will be 812 kg/h ☐

3. A typical DP cell used with a measuring differential pressure flowmeter……

a| Senses the pressure either side of the flowmetering device and relays a corresponding electrical signal to a display processor ☐
b| Compares the pressure downstream of the flowmetering device with a fixed upstream pressure and volume, and relays the difference by means of a corresponding electrical signal to a display processor ☐
c| Senses differential pressure across the flowmetering device, and density of the steam at the designed upstream pressure and passes this information to a display processor ☐
d| Senses changes in pressure upstream of the flowmetering device and relays a corresponding electrical signal to a display processor ☐

4. An orifice plate flowmeter is designed for use on saturated steam at 5 bar g but for much of its life it operates on steam at 4 bar g and displays a flowrate of 1 200 kg/h. Will the display at 4 bar g be accurate if the flowmeter is not fitted with density compensation?

a| No, the actual flowrate will be 1 316 kg/h ☐
b| No, the actual flowrate will be 1 100 kg/h ☐
c| Yes ☐
d| No, the flowmeter will be outside its turndown ratio ☐

5. The steam in question 4 is thought to be very wet. What effect will this have?

a| The orifice will erode resulting in the actual flow being less than that indicated ☐
b| The effect will be insignificant ☐
c| The actual flowrate will be higher than the indicated flowrate ☐
d| The actual flowrate will be less than the indicated flowrate ☐

6. A flowmeter measuring differential pressure is installed on a system where the pressure can vary between 20 bar g and 1 bar g. Which of the following could cause inaccuracy of the flowmeter?

a| The steam becoming superheated because of the pressure drop ☐
b| Density compensation not being incorporated ☐
c| The high pressure turndown ☐
d| All of the above ☐

Answers

1: b, 2: d, 3: a, 4: b, 5: c, 6: b

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Module 4.5

Installation

Installation

The manufacturer should always supply installation data with the product as this will lay down specific requirements such as the minimum lengths of unobstructed pipe to be provided upstream and downstream of the flowmeter. It is usual for the flowmeter supplier to be able to offer advice and relay recommendations regarding the installation requirements of his particular flowmeter.

Statistics show that over a third of flowmeter problems are due to poor installation. No steam flowmeter, however good its design and thorough its manufacture, can cope if little attention is paid to its installation and the layout of the steam system.

Steam quality

Dry steam

Steam should always be provided in as dry a condition as possible at the point of metering. Module 4.4 has already demonstrated that wet steam will cause inaccuracies and can physically damage some types of flowmeter.

A simple but effective method of drying wet steam is to install a separator upstream of the flowmeter. Entrained moisture impinges on the baffle plates and the heavy droplets fall to the bottom and are drained away via a properly sized and selected steam trap set. Independent tests show that it is possible to achieve a 99% dryness fraction over a wide range of flows by use of a high efficiency separator as shown in Figure 4.5.1.

The separator has one other important benefit: Slugs of water impacting on any steam flowmeter (i.e. waterhammer) can cause severe mechanical damage. Fitting a separator before a steam flowmeter will reduce the resulting impact pressure from water slugs by up to 90%, affording considerable protection to any expensive flowmetering device.

The separator with its drain trap ensures efficient condensate removal ahead of the flowmeter. But any low points where the steam main rises to a higher level should also have drain trap points that are adequately sized and correctly selected. It is also worthwhile ensuring that air and other entrained gases are removed by fitting an air vent in the steam line.

The separator shown in Figure 4.5.1 has a top connection suitable for an automatic air vent that will help to remove incondensable gases prior to the flowmetering station. Figure 4.5.2 illustrates a combined drain trap point and venting station at the end of a steam main.

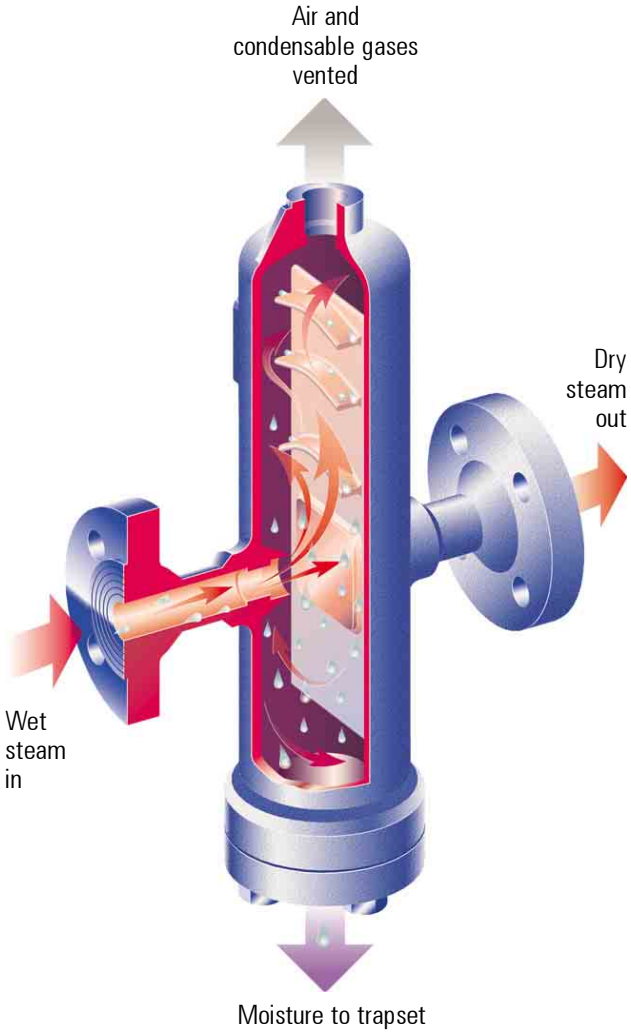


Fig. 4.5.1 Typical separator

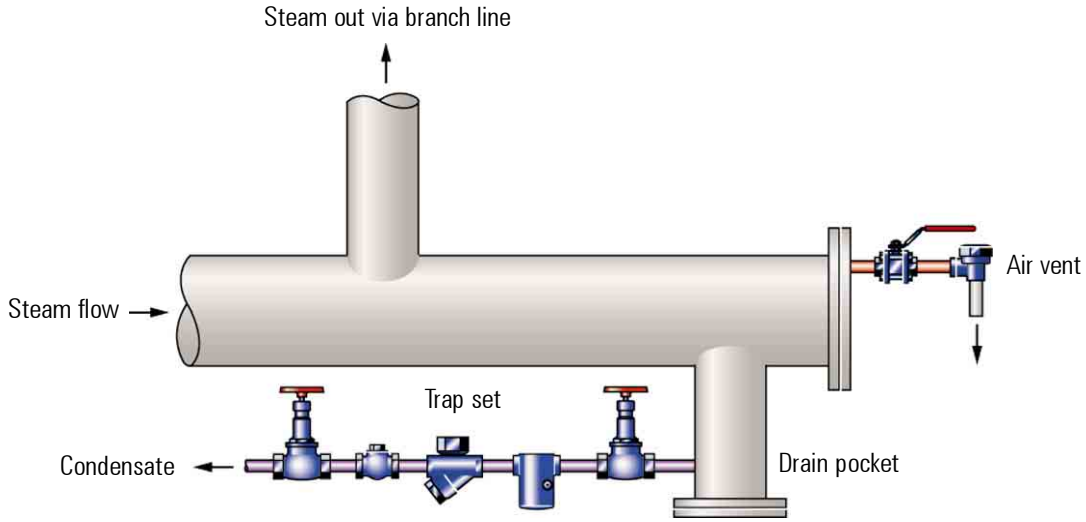


Fig. 4.5.2 Condensate and air removal at the end of a steam main

Clean steam

A pipeline strainer (Figure 4.5.3) should be fitted ahead of the flowmeter. This will remove any larger pieces of scale, swarf or other pipeline debris, which would otherwise damage the primary device. The internal strainer device should be cleaned periodically, particularly during the initial start-up of a new installation.

As with any steam pipeline strainer, the strainer should be installed with the body horizontal to avoid creating an accumulation of condensate and hence a reduction in the screening area (Figure 4.5.4).

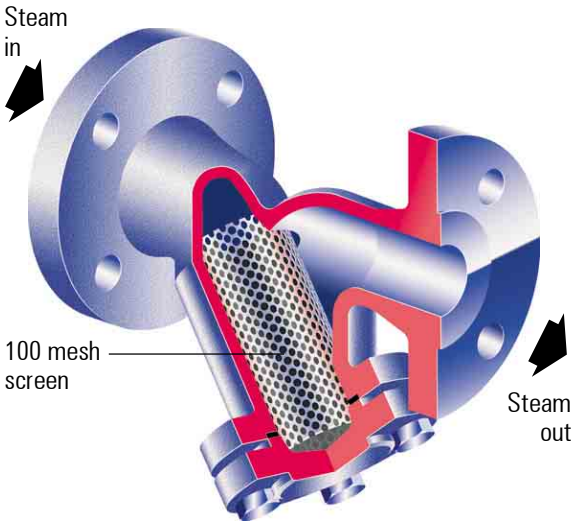


Fig. 4.5.3 Cut section of a typical pipeline strainer

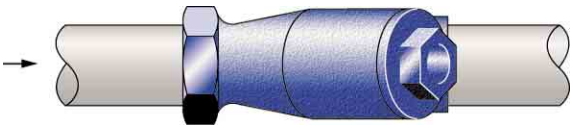


Fig. 4.5.4 Correct strainer orientation for steam or gas applications

Maintenance

The provision of valves either side of the flowmeter should be considered for isolation purposes, since inspection, maintenance and perhaps even 'removal for calibration' will sometimes be necessary. Such valves should be of the fully open or fully closed type, which present the least resistance to flow, such as full bore ball valves. In addition, a valved bypass, or a make-up piece to act as a temporary replacement if the flowmeter is removed from the pipeline, will solve the problem of interrupting the steam supply during maintenance procedures. Both pipework and flowmeter must be adequately supported and properly aligned with a slight fall to the last drain point ahead of the flowmeter. Pipework should also be properly and effectively insulated to minimise radiation losses and further condensation.

Installation recommendations

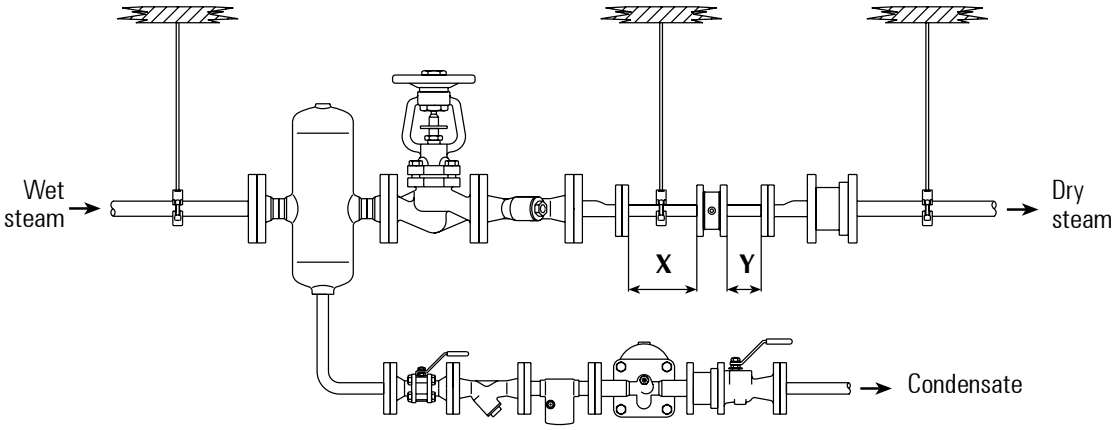


Fig. 4.5.5 Clear, unobstructed pipeline lengths

1. Ensure all pipework is adequately supported and properly aligned.

   This will prevent waterlogging during shutdown periods and possible problems on 'start-up'.
2. Size the flowmeter on capacity rather than line size.

   Where a pipe size reduction is necessary, use eccentric reducing sockets.
3. Take care to observe the correct direction of flow.

   An arrow on the flowmeter body should show this.
4. It is advisable to fit a check valve downstream of the transducer

   This will avoid possible damage by reverse flow.
5. Do not close-couple the flowmeter **immediately** downstream to a pressure reducing valve.

   This comment is particularly relevant to pilot operated self-acting pressure controllers with a narrow proportional band; these may cause pressure oscillations leading to inaccuracies and/or possible damage of the primary unit.

   As a general rule, a self-acting pressure control should be at least 10, and preferably 25 pipe diameters upstream of the flowmeter.
6. Do not install the flowmeter downstream of a partially open stop valve.

   This can lead to swirl, which may lead to inaccuracies.
7. A separator should always be fitted upstream of the flowmeter.

   This will remove entrained moisture from the steam. Dry steam is required for accurate steam flowmetering. It will also provide some degree of protection against waterhammer impact damage.

   The separator should be drained using a float thermostatic steam trap.
8. A full line size strainer with 100 mesh stainless steel screen must be fitted.

   This will prevent dirt and scale reaching the transducer. This is especially advisable on old or dirty systems where dirt or corrosion is present.
9. Ensure gasket faces do not protrude into the pipeline.
10. A bellows sealed stop valve may be fitted upstream of the flowmeter.
11. Recommended lengths of clear, unobstructed pipe must be provided upstream and downstream of the flowmeter.

    X + **Y** is known as the 'Flowmeter run' (Figure 4.5.5).

    The question of leaving sufficient length of clear, unobstructed pipework upstream and downstream of the flowmeter is most important. This is to prevent the risk of swirl, which can be produced by bends and partially open valves.

Some types of flowmeter are more susceptible to swirl than others. Some manufacturers recommend the use of flow straighteners to remove swirl (Figure 4.5.6). However, it is preferable to do all that is possible to prevent the risk of swirl by providing an adequate flowmeter run since flow straighteners in steam systems can entrain surface water. It may even be preferable to select a steam flowmeter that is less susceptible to the effects of swirl.

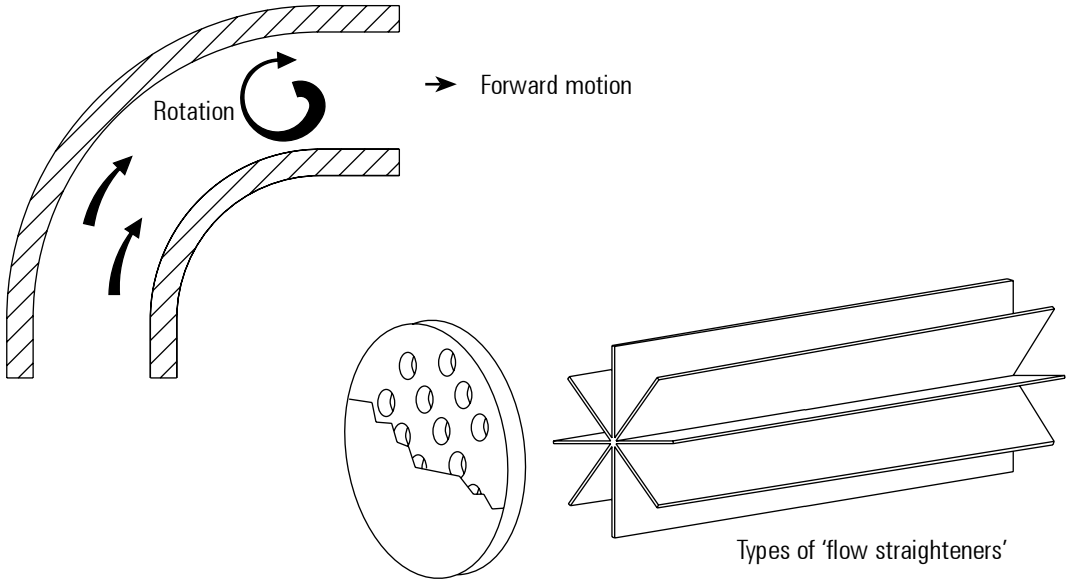


Fig. 4.5.6 Flow straighteners

Correct sizing of the flowmeter is also essential and most manufacturers will recommend maximum and minimum flowrates for each size of flowmeter.

If the flowmeter to be used is smaller than the pipeline into which it is to be fitted, reductions in pipe size should be achieved by using eccentric reducers (Figure 4.5.7). This will prevent the collection of condensate at a lowpoint - as would be the result if concentric reducers were used. The reduction in pipe size should be achieved at the nearest point to the flowmeter consistent with maintaining the required flowmeter run.

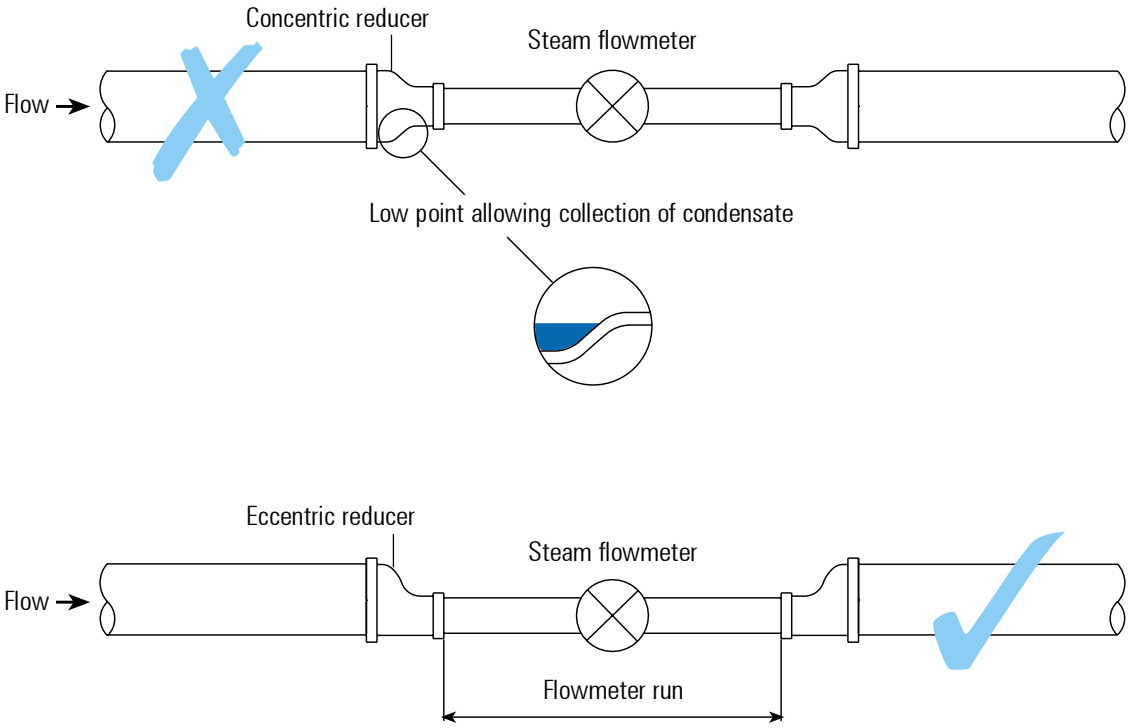


Fig. 4.5.7 Pipe size reduction

System design considerations

Adopting a structured approach to steam flowmetering will help to ensure that:

- The design objectives are achieved.
- No elements of the design are omitted.
- The benefits are maximised.
- The financial outlay is minimised.

There are two main elements to such an approach:

1. Consideration of the existing steam supply system
The planner should identify any future changes to the plant or process that may affect the installation of steam flowmeters, and should consider whether the installation of flowmeters is likely to act as a catalyst for such changes. Alterations to the system, for example, may involve blanking off redundant sections of steam mains, rerouting pipework, or generally improving the condition of pipe layout and/or insulation.

2. Identifying the aim of installing steam flowmetering
Typically, one or more of the following design criteria will be clearly defined:

- To provide information for accounting purposes, such as departmental allocation of costs.
- To facilitate custody transfer, for example where a central station sells steam to a range of clients.
- To facilitate Monitoring and Targeting (M and T) policies and observe trends.
- To determine and monitor energy utilisation and efficiency.

Each of the above criteria imposes different limitations on the design of the steam flowmetering system.

If flowmetering is to be used for accounting purposes or for custody transfer, it will be necessary to install a sufficient number of flowmeters for consumption to be assigned to each of the cost centres. Also, if the product being sold is **energy** not steam, flowmeters will also have to be installed on the condensate return lines, as this hot water will have a heat value. For both applications, the highest possible standard of flowmetering will be required, particularly with respect to accuracy, turndown ratio, and repeatability.

The system may also require check flowmetering so that consumption can be proven correct. It should be noted that confidence in any monitoring system, once lost, is very difficult to restore. A system should also include measurement of the system losses incurred as a result of supplying steam to a particular location. This implies that flowmeter positions should be located as near to the boiler house as possible.

In M and T applications and in the determining of energy efficiency, the important flowmetering criterion is repeatability. The user will be more interested in trends in consumption rather than absolute values.

Determining flowmeter arrangements

Once the system layout has been determined, and the data required to accurately measure the energy consumption of the system / plant has been decided, the number and location of required flowmeters can be contemplated. This requires consideration of the site as a whole including the steam main from the boiler house.

Figure 4.5.8 shows four possible layouts for the same system.

The four diagrams shown in Figure 4.5.8 illustrate how the connection of multiple steam flowmeters can affect the results obtained and ultimately influence the data analysis.

Diagram 1

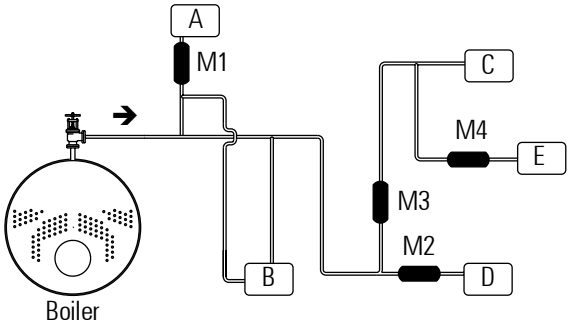


Diagram 1 shows that the individual usage by each section can be measured directly, except that of area B, which is obtained by difference. This means that the majority of the system losses will be included in B's figures whilst not giving a representative illustration of where the system losses are occurring.

Diagram 2

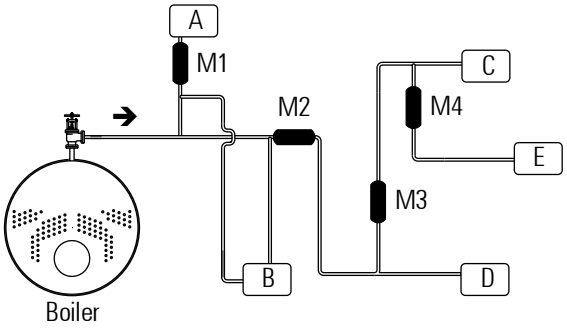


Diagram 2 shows a layout that allows the system losses to be more fairly distributed across the areas. Although the same number of flowmeters are being used as in the first option, the flowmeter losses are those inherent to each supply.

Steam flowmeters

Diagram 3

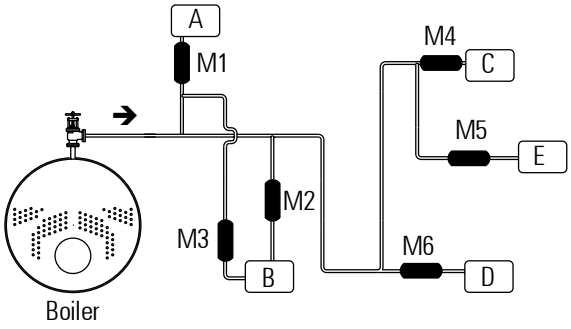


Diagram 3 shows the simplest way to measure the steam consumption with each individual steam supply being metered and the losses being calculated through difference. It does, however, use two flowmeters more than the previous two options and will therefore be more expensive.

Diagram 4

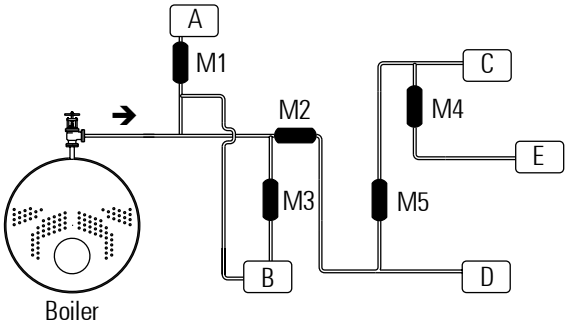


Diagram 4 shows the benefits from Diagrams 1 and 2 in that it uses five flowmeters yet allows flowrate in the individual steam mains to be determined and allocates the distribution losses fairly.

Fig. 4.5.8 Four possible layouts for the same system

Specifying a steam flowmeter

Some of the factors which need to be taken into account when selecting a steam flowmeter include:

Performance

- Accuracy.
- Repeatability.
- Turndown.
- Pressure drop.
- Display unit facilities.

Maintenance

- Reliability.
- Calibration needs.
- Spare parts requirement or service exchange scheme.
- Ease of maintenance.

Cost

- Cost of flowmeter.
- Cost of associated instruments.
- Cost of installation.
- Overall lifetime costs.

Other factors

- Reputation of manufacturer.
- Back-up provided by the manufacturer.
- Initial calibration requirements.
- Density compensation.
- Ability to interface.
- Availability of associated equipment.
- Quality of literature and information provided.

The above points should be considered collectively. For example, it can be a mistake to simply select a flowmeter on accuracy when, often, there is a balance between accuracy and reliability.

The most accurate flowmeters are often the most delicate and can suffer badly when used with steam. A more sensible approach will be to look for reasonable accuracy with good repeatability and proven reliability with steam.

Useful checklist to help in the selection of a steam flowmeter

The following is offered to help in the selection of a steam flowmeter and gives a useful check list and prompt for the questions that need to be raised:

- What is the application? (Boiler house flowmeter, departmental flowmeter, or plant flowmeter.)
- What is the pipeline size and configuration?
- What is the steam pressure and temperature?
- What is the object of flowmetering? (Cost allocation, plant efficiency check, energy saving scheme monitor.)
- What is the flowmeter required to indicate? (Flowrate, quantity, mass or volume.)
- Is there a need to measure maximum, minimum, and/or average flowrates?
- What accuracy, repeatability and turndown is needed?
- What is the purchase budget allowed?
- How much of this is allocated to installation costs and ancillary equipment costs?
- Who will install the flowmeter?
- Who will commission the flowmeter?
- Who will maintain the flowmeter?
- Is there a need to interface the flowmeter with any local chart recorders or central energy management systems?
- Is physical size a constraint?
- Is the flowmeter designed for operation with steam?
- Are any other features required, such as remote alarms on timers?

Once this evaluation has been completed, the Steps in Figure 4.5.9 need to be followed before making a final selection.

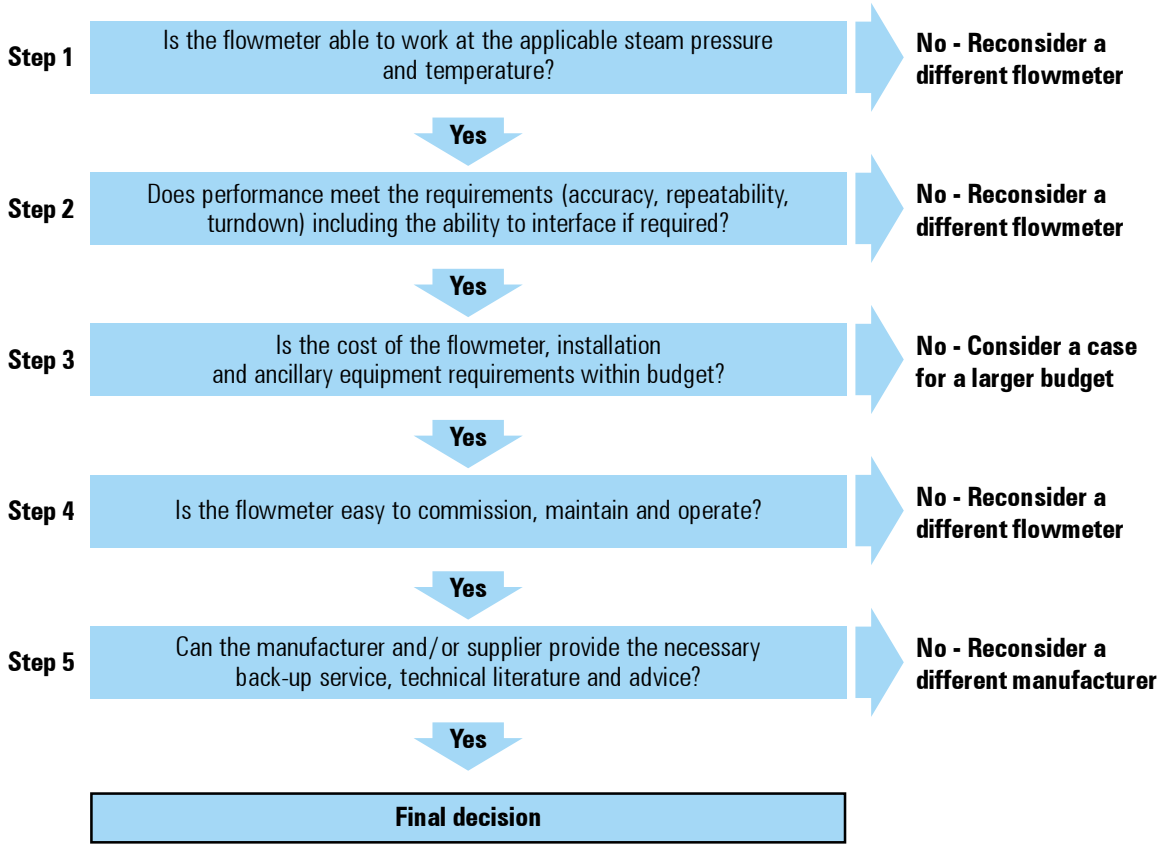


Fig. 4.5.9 Typical decision table for a steam flowmeter

Conclusion

Difficulties in the energy management of steam arise from the fact that it is often perceived as a 'free' (unmetered) service.

Measurement is essential if savings are to be made

Most plants have figures on the annual cost of fuel. However, even these figures can become doubtful when a supply provides fuel to multi-users. Again, measuring the total fuel consumption of two or more perhaps dissimilar boilers can hide useful information.

Gas or oil can be measured quite easily. Measurement of steam is more difficult - which explains why steam is often perceived as being free. If steam is metered, then is the measurement accurate? Most flowmeters depend on a measurement of volume, whilst steam is traditionally costed on a mass basis. To ensure the correct volumetric flowrate is measured for conversion to mass flow, density compensation is essential.

It is easy to accept the instrument reading as shown by the integrator or chart. Most flowmeters, however, are calibrated on media other than steam, with a correction factor to convert the scale reading to an actual amount. It is important the manufacturer can provide test details if required.

Flowmeters should be checked from time to time to make sure that there is no erosion to any measuring orifice or any similar change to an alternative type of primary device.

Although steam flowmetering is often confined to the boiler house, it can be extremely useful in other parts of the system. It is essential where steam has to be costed. It is essential information for the plant manager charged with conserving energy or improving production efficiency or quality.

Steam flowmeters will provide useful information on plant performance, fouling of heat transfer surfaces or the malfunction of steam traps.

Flowmeter readings provide the only positive approach when schemes or improvements are introduced to save steam.

Questions

1. Where should the separator be fitted in relation to any steam flowmeter?

a| As near as possible to the flowmeter ☐

b| Ten pipe diameters before the flowmeter ☐

c| Beyond five pipe diameters after the flowmeter ☐

d| Immediately before the upstream isolation valve and strainer ☐

2. What size of separator should be fitted as part of a DN100 orifice plate flowmeter system? The straight run of pipe each side of the flowmeter is 100 mm diameter. The pipe either side of that has a diameter of 125 mm.

a| DN125 ☐

b| DN80 ☐

c| DN100 ☐

d| DN150 ☐

3. Which of the following is true of a strainer protecting a steam flowmeter?

a| It should be fitted immediately before the upstream isolating valve so that the valve is protected ☐

b| It should be fitted with a 1.6 mm mesh screen to minimise the pressure drop across it ☐

c| It should be fitted with a 100 mesh screen and with the basket pointed down to collect debris ☐

d| It should be fitted with a 100 mesh screen and with the basket on its side ☐

4. A factory buys its steam from a power station and is charged for it on the basis of energy used. Credit is given for condensate returned to the power station. The factory wants to be able to check its invoices. How could this be done?

a| By metering the energy in the steam supply, in the condensate returned and in the flash steam vented from the pump receivers ☐

b| By metering the energy in the steam supply and deducting this from the calculated heat content of the condensate entering each steam trap ☐

c| By metering the flowrate in the steam supply and condensate return and converting these figures to energy flow ☐

d| By metering the energy in the steam supply ☐

5. Which of the following contributes most to the high standard of flowmetering?

a| Accuracy, pressure, turndown ratio and installation ☐

b| Accuracy, repeatability, turndown ratio and installation ☐

c| Density compensation, when metering water ☐

d| Turndown ratio, rangeability and constant pressure ☐

6. What personnel are likely to benefit from steam flowmetering

a| The Managing Director ☐

b| The Engineering Director ☐

c| The Finance Director ☐

d| All of them

Answers

1: d, 2: a, 3: d, 4: c, 5: b, 6: d

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Module 5.1

An Introduction to Controls

An Introduction to Controls

The subject of automatic controls is enormous, covering the control of variables such as temperature, pressure, flow, level, and speed.

The objective of this Block is to provide an introduction to automatic controls. This too can be divided into two parts:

- The control of Heating, Ventilating and Air Conditioning systems (commonly known as HVAC); and
- Process control.

Both are immense subjects, the latter ranging from the control of a simple domestic cooker to a complete production system or process, as may be found in a large petrochemical complex.

The Controls Engineer needs to have various skills at his command - knowledge of mechanical engineering, electrical engineering, electronics and pneumatic systems, a working understanding of HVAC design and process applications and, increasingly today, an understanding of computers and digital communications.

The intention of this Block is to provide a basic insight into the practical and theoretical facets of automatic control, to which other skills can be added in the future, **not** to transform an individual into a Controls Engineer

This Block is confined to the control of processes that utilise the following fluids: steam, water, compressed air and hot oils.

Control is generally achieved by varying fluid flow using actuated valves. For the fluids mentioned above, the usual requirement is to measure and respond to changes in temperature, pressure, level, humidity and flowrate. Almost always, the response to changes in these physical properties must be within a given time. The combined manipulation of the valve and its actuator with time, and the close control of the measured variable, will be explained later in this Block.

The control of fluids is not confined to valves. Some process streams are manipulated by the action of variable speed pumps or fans.

The need for automatic controls

There are three major reasons why process plant or buildings require automatic controls:

- **Safety -** The plant or process must be safe to operate.
  The more complex or dangerous the plant or process, the greater is the need for automatic controls and safeguard protocol.
- **Stability -** The plant or processes should work steadily, predictably and repeatably, without fluctuations or unplanned shutdowns.
- **Accuracy -** This is a primary requirement in factories and buildings to prevent spoilage, increase quality and production rates, and maintain comfort. These are the fundamentals of economic efficiency.

Other desirable benefits such as economy, speed, and reliability are also important, but it is against the three major parameters of safety, stability and accuracy that each control application will be measured.

Automatic control terminology

Specific terms are used within the controls industry, primarily to avoid confusion. The same words and phrases come together in all aspects of controls, and when used correctly, their meaning is universal.

The simple manual system described in Example 5.1.1 and illustrated in Figure 5.1.1 is used to introduce some standard terms used in control engineering.

Example 5.1.1 A simple analogy of a control system

In the process example shown (Figure5.1.1), the operator manually varies the flow of water by opening or closing an inlet valve to ensure that:

- The water level is not too high; or it will run to waste via the overflow.
- The water level is not too low; or it will not cover the bottom of the tank.

The outcome of this is that the water runs out of the tank at a rate within a required range. If the water runs out at too high or too low a rate, the process it is feeding cannot operate properly.

At an initial stage, the outlet valve in the discharge pipe is fixed at a certain position.

The operator has marked three lines on the side of the tank to enable him to manipulate the water supply via the inlet valve. The 3 levels represent:

1. The lowest allowable water level to ensure the bottom of the tank is covered.
2. The highest allowable water level to ensure there is no discharge through the overflow.
3. The ideal level between 1 and 2.

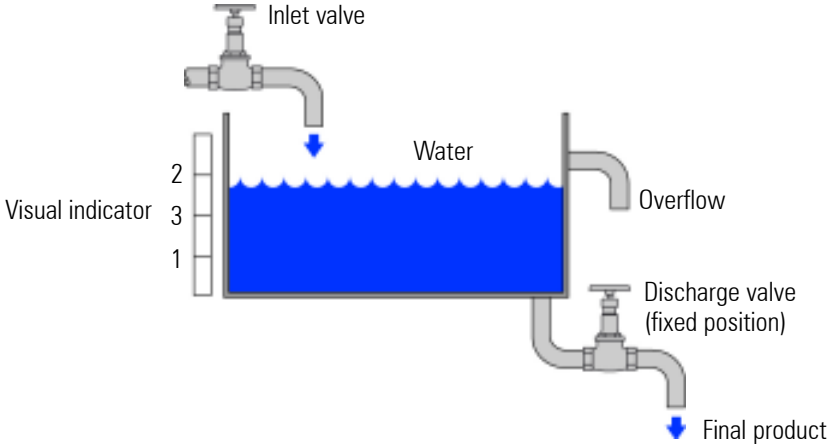


Fig. 5.1.1 Manual control of a simple process

The Example (Figure 5.1.1) demonstrates that:

1. The operator is aiming to maintain the water in the vessel between levels 1 and 2. The water level is called the **Controlled condition**.

2. The controlled condition is achieved by controlling the flow of water through the valve in the inlet pipe. The flow is known as the **Manipulated Variable**, and the valve is referred to as the **Controlled Device**.

3. The water itself is known as the **Control Agent**.

4. By controlling the flow of water into the tank, the level of water in the tank is altered. The change in water level is known as the **Controlled Variable**.

5. Once the water is in the tank it is known as the **Controlled Medium**.

6. The level of water trying to be maintained on the visual indicator is known as the **Set Value** (also known as the **Set Point**).

7. The water level can be maintained at any point between 1 and 2 on the visual indicator and still meet the control parameters such that the bottom of the tank is covered and there is no overflow. Any value within this range is known as the **Desired Value**.

8. Assume the level is strictly maintained at any point between 1 and 2. This is the water level at steady state conditions, referred to as the **Control Value** or **Actual Value**.

Note: With reference to (**7**) and (**8**) above, the ideal level of water to be maintained was at point 3. But if the actual level is at any point between 1 and 2, then that is still satisfactory. The difference between the **Set Point** and the **Actual Value** is known as **Deviation**.

9. If the inlet valve is closed to a new position, the water level will drop and the deviation will change. A sustained deviation is known as **Offset**.

Elements of automatic control

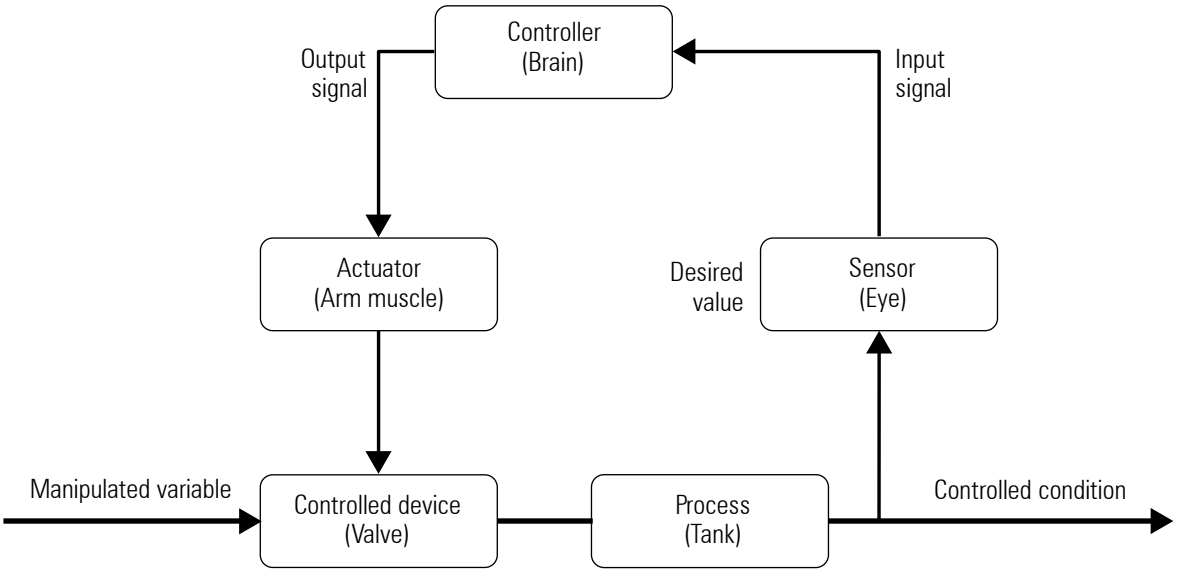


Fig. 5.1.2 Elements of automatic control

Example 5.1.2 Elements of automatic control

- The operator's eye detects movement of the water level against the marked scale indicator. His eye could be thought of as a **Sensor**.
- The eye (sensor) signals this information back to the brain, which notices a deviation. The brain could be thought of as a **Controller**.
- The brain (controller) acts to send a signal to the arm muscle and hand, which could be thought of as an **Actuator**.
- The arm muscle and hand (actuator) turn the valve, which could be thought of as a **Controlled Device**.

It is worth repeating these points in a slightly different way to reinforce Example 5.1.2:

In simple terms the operator's aim in Example 5.1.1 is to hold the water within the tank at a pre-defined level. Level 3 can be considered to be his target or **Set Point**.

The operator physically manipulates the level by adjusting the inlet valve (the control device). Within this operation it is necessary to take the operator's competence and concentration into account. Because of this, it is unlikely that the water level will be exactly at Level 3 at all times. Generally, it will be at a point above or below Level 3. The position or level at any particular moment is termed the **Control Value** or **Actual Value**.

The amount of error or difference between the **Set Point** and the **Actual Value** is termed deviation. When a deviation is constant, or steady state, it is termed **Sustained Deviation** or **Offset**.

Although the operator is manipulating the water level, the final aim is to generate a proper outcome, in this case, a required flow of water from the tank.

Assessing safety, stability and accuracy

It can be assumed that a process typical of that in Example 5.1.1 contains neither valuable nor harmful ingredients. Therefore, overflow or water starvation will be safe, but not economic or productive.

In terms of stability, the operator would be able to handle this process providing he pays full and constant attention.

Accuracy is not a feature of this process because the operator can only respond to a visible and recognisable error.

Summary of terminology

Set point	The value set on the scale of the control system in order to obtain the required condition. If the controller was set at 60°C for a particular application: 60°C would be termed as the 'set point'.
Desired value	The required value that should be sustained under ideal conditions.
Control value	The value of the control condition actually maintained under steady state conditions.
Deviation	The difference between the set point and the control value.
Offset	Sustained deviation.
Sensor	The element that responds directly to the magnitude of the controlled condition.
Controlled medium	The medium being controlled by the system. The controlled medium in Figure 5.1.1 is the water in the tank.
Controlled condition	The physical condition of the controlled medium. The controlled condition in Figure 5.1.1 is the water level.
Controller	A device which accepts the signal from the sensor and sends a corrective (or controlling) signal to the actuator.
Actuator	The element that adjusts the controlled device in response to a signal from the controller.
Controlled device	The final controlling element in a control system, such as a control valve or a variable speed pump.

There are many other terms used in Automatic Controls; these will be explained later in this Block.

Elements of a temperature control system

Example 5.1.1 depicted a simple manual level control system. This can be compared with a simple temperature control example as shown in Example 5.1.3 (manually controlled) and Figure 5.1.3. All the previous factors and definitions apply.

Example 5.1.3 Depicting a simple manual temperature control system

The task is to admit sufficient steam (the heating medium) to heat the incoming water from a temperature of T_1; ensuring that hot water leaves the tank at a required temperature of T_2.

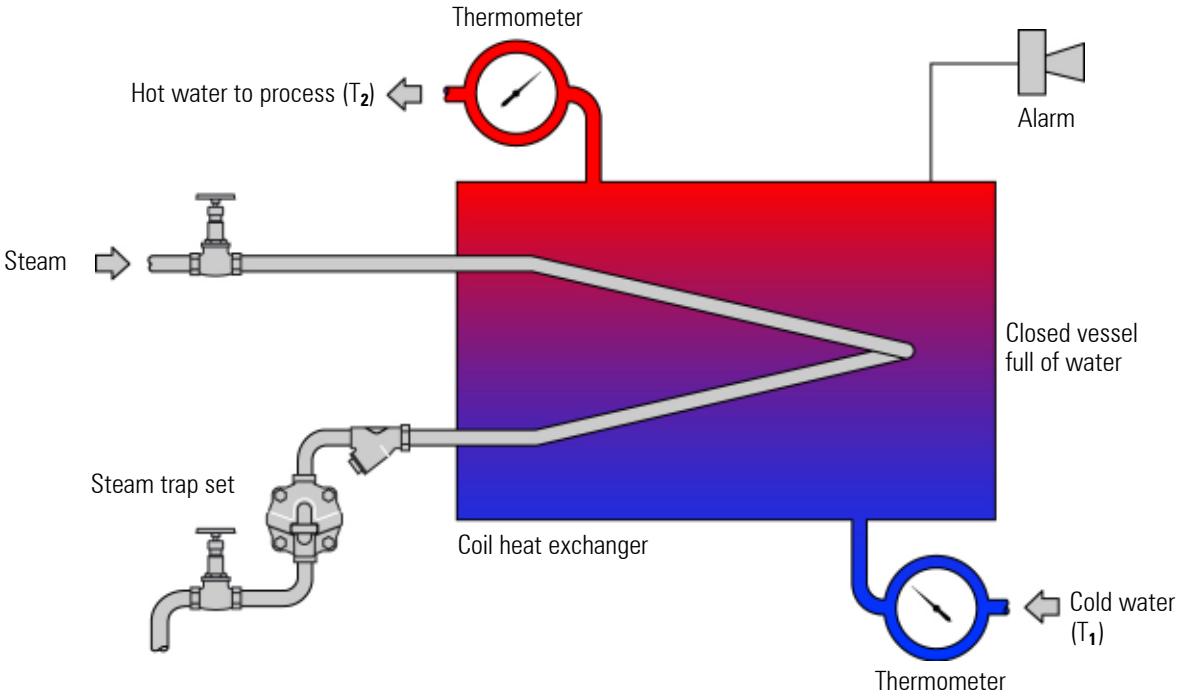


Fig. 5.1.3 Simple manual temperature control

Assessing safety, stability and accuracy

Whilst manual operation could probably control the water level in Example 5.1.1, the manual control of temperature is inherently more difficult in Example 5.1.3 for various reasons.

If the flow of water varies, conditions will tend to change rapidly due to the large amount of heat held in the steam. The operator's response in changing the position of the steam valve may simply not be quick enough. Even after the valve is closed, the coil will still contain a quantity of residual steam, which will continue to give up its heat by condensing.

Anticipating change

Experience will help but in general the operator will not be able to anticipate change. He must observe change before making a decision and performing an action.

This and other factors, such as the inconvenience and cost of a human operator permanently on duty, potential operator error, variations in process needs, accuracy, rapid changes in conditions and the involvement of several processes, all lead to the need for automatic controls.

With regards to safety, an audible alarm has been introduced in Example 5.1.3 to warn of overtemperature - another reason for automatic controls.

Automatic control

A controlled condition might be temperature, pressure, humidity, level, or flow. This means that the measuring element could be a temperature sensor, a pressure transducer or transmitter, a level detector, a humidity sensor or a flow sensor.

The manipulated variable could be steam, water, air, electricity, oil or gas, whilst the controlled device could be a valve, damper, pump or fan.

For the purposes of demonstrating the basic principles, this Module will concentrate on valves as the controlled device and temperature as the controlled condition, with temperature sensors as the measuring element.

Components of an automatic control

Figure 5.1.4 illustrates the component parts of a basic control system. The sensor signals to the controller. The controller, which may take signals from more than one sensor, determines whether a change is required in the manipulated variable, based on these signal(s). It then commands the actuator to move the valve to a different position; more open or more closed depending on the requirement.

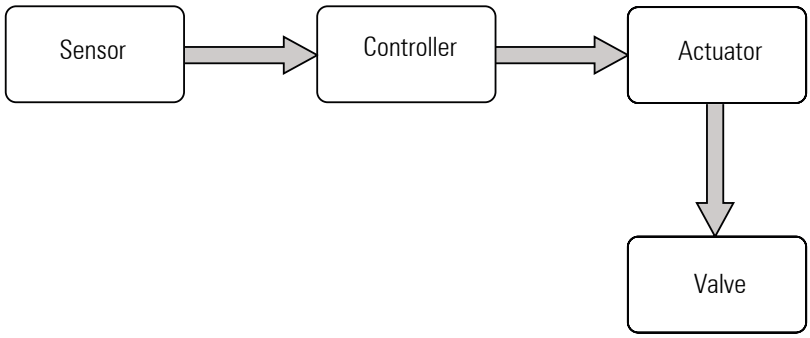


Fig. 5.1.4 Components of an automatic control

Controllers are generally classified by the sources of energy that power them, electrical, pneumatic, hydraulic or mechanical.

An actuator can be thought of as a motor. Actuators are also classified by the sources of energy that power them, in the same way as controllers.

Valves are classified by the action they use to effect an opening or closing of the flow orifice, and by their body configurations, for example whether they consist of a sliding spindle or have a rotary movement.

If the system elements are combined with the system parts (or devices) the relationship between 'What needs to be done?' with 'How does it do it?', can be seen.

Some of the terms used may not yet be familiar. However, in the following parts of Block 5, all the individual components and items shown on the previous drawing will be addressed.

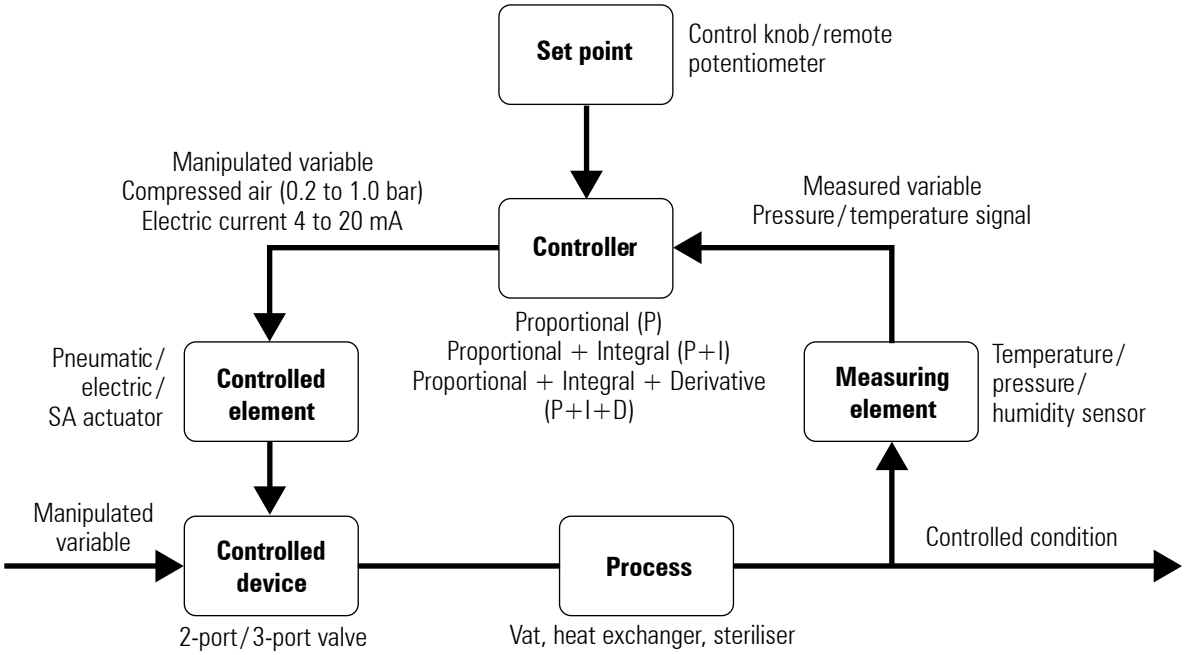


Fig. 5.1.5 Typical mix of process control devices with system elements

Questions

1. Air temperature in a room is controlled at 25°C. If the actual temperature varies from this, what term is used to define the difference?

a| Offset ☐
b| Deviation ☐
c| Sustained deviation ☐
d| Desired value ☐

2. A pneumatic temperature control is used on the steam supply to a non-storage heat exchanger that heats water serving an office heating system. What is referred to as the 'manipulated variable'?

a| The water being heated ☐
b| The steam supply ☐
c| The air signal from the controller to the valve actuator ☐
d| The temperature of the air being heated ☐

3. If an automatic control is to be selected and sized, what is the most important aspect to consider?

a| Safety in the event of a power failure ☐
b| Accuracy of control ☐
c| Stability of control ☐
d| All of them ☐

4. Define 'control value'?

a| The value set on the scale of the control system in order to obtain the required condition ☐
b| The quantity or condition of the controlled medium ☐
c| The flow or pressure of the steam (or fluid) being manipulated ☐
d| The value of the controlled condition actually maintained under steady state conditions ☐

5. An electronic controller sends a signal to an electric actuator fitted to a valve on the steam supply to a coil in a tank of water. In control terms, how is the water described?

a| Control agent ☐
b| Manipulated variable ☐
c| Controlled medium ☐
d| Controlled variable ☐

6. With reference to Question 5, the controller is set to maintain the water temperature at 80°C, but at a particular time it is 70°C. In control terms how is the temperature of 80°C described?

a| Controlled condition ☐
b| Control value ☐
c| Set value ☐
d| Control point ☐

Answers

1: b 2: b, 3: d, 4: d, 5: a, 6: c

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Module 5.2

Basic Control Theory

Basic Control Theory

Modes of control

An automatic temperature control might consist of a valve, actuator, controller and sensor detecting the space temperature in a room. The control system is said to be 'in balance' when the space temperature sensor does not register more or less temperature than that required by the control system. What happens to the control valve when the space sensor registers a change in temperature (a temperature deviation) depends on the type of control system used. The relationship between the movement of the valve and the change of temperature in the controlled medium is known as the mode of control or control action.

There are two basic modes of control:

- **On/Off -** The valve is either fully open or fully closed, with no intermediate state.
- **Continuous -** The valve can move between fully open or fully closed, or be held at any intermediate position.

Variations of both these modes exist, which will now be examined in greater detail.

On/off control

Occasionally known as two-step or two-position control, this is the most basic control mode. Considering the tank of water shown in Figure 5.2.1, the objective is to heat the water in the tank using the energy given off a simple steam coil. In the flow pipe to the coil, a two port valve and actuator is fitted, complete with a thermostat, placed in the water in the tank.

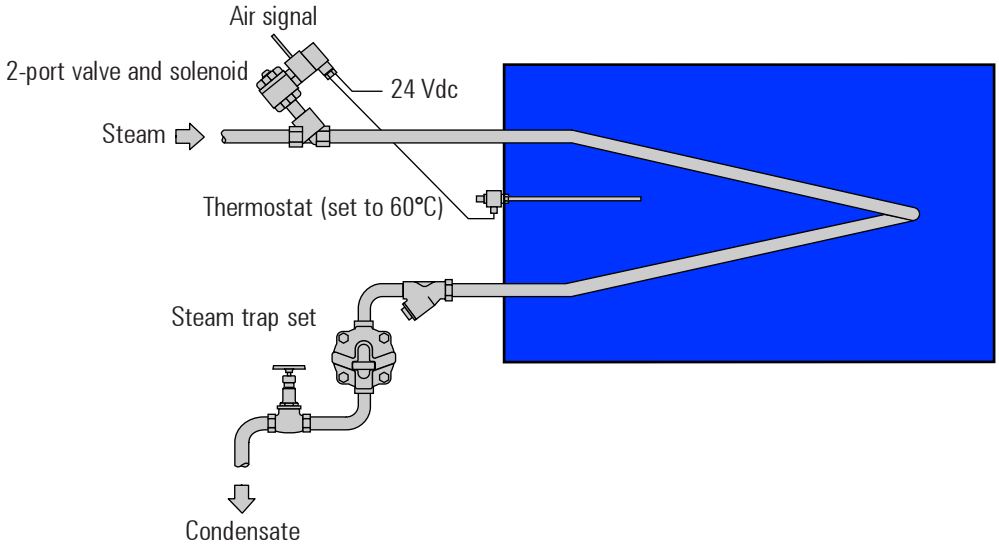


Fig. 5.2.1 On/off temperature control of water in a tank

The thermostat is set to 60°C, which is the required temperature of the water in the tank. Logic dictates that if the switching point were actually at 60°C the system would never operate properly, because the valve would not know whether to be open or closed at 60°C. From then on it could open and shut rapidly, causing wear.

For this reason, the thermostat would have an upper and lower switching point. This is essential to prevent over-rapid cycling. In this case the upper switching point might be 61°C (the point at which the thermostat tells the valve to shut) and the lower switching point might be 59°C (the point when the valve is told to open). Thus there is an in-built switching difference in the thermostat of ±1°C about the 60°C set point.

This 2°C (±1°C) is known as the switching differential. (This will vary between thermostats). A diagram of the switching action of the thermostat would look like the graph shown in Figure 5.2.2. The temperature of the tank contents will fall to 59°C before the valve is asked to open and will rise to 61°C before the valve is instructed to close.

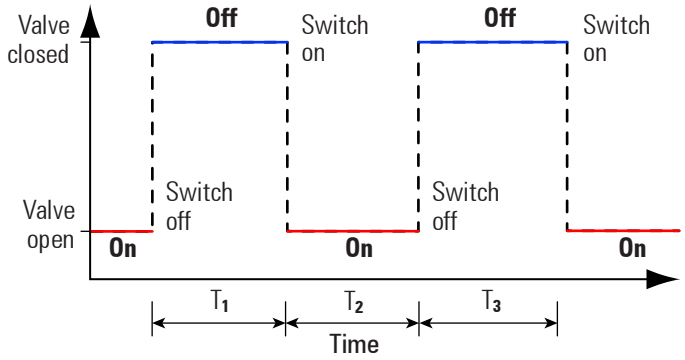


Fig. 5.2.2 On/off switching action of the thermostat

Figure 5.2.2 shows straight switching lines but the effect on heat transfer from coil to water will not be immediate. It will take time for the steam in the coil to affect the temperature of the water in the tank. Not only that, but the water in the tank will rise above the 61°C upper limit and fall below the 59°C lower limit. This can be explained by cross referencing Figures 5.2.2 and 5.2.3. First however it is necessary to describe what is happening.

At point **A** (59°C, Figure 5.2.3) the thermostat switches on, directing the valve wide open. It takes time for the transfer of heat from the coil to affect the water temperature, as shown by the graph of the water temperature in Figure 5.2.3. At point **B** (61°C) the thermostat switches off and allows the valve to shut. However the coil is still full of steam, which continues to condense and give up its heat. Hence the water temperature continues to rise above the upper switching temperature, and 'overshoots' at **C**, before eventually falling.

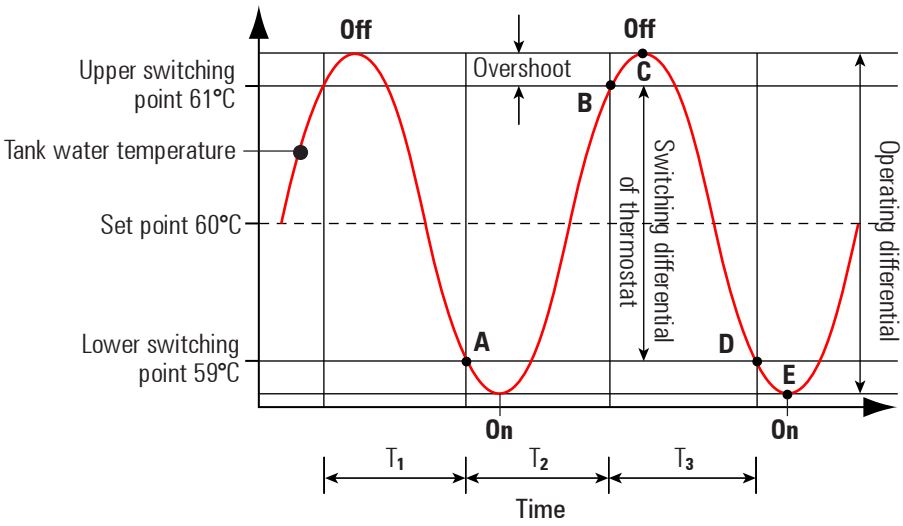


Fig. 5.2.3 Tank temperature versus time

From this point onwards, the water temperature in the tank continues to fall until, at point **D** (59°C), the thermostat tells the valve to open. Steam is admitted through the coil but again, it takes time to have an effect and the water temperature continues to fall for a while, reaching its trough of undershoot at point **E**.

The difference between the peak and the trough is known as the operating differential. The switching differential of the thermostat depends on the type of thermostat used. The operating differential depends on the characteristics of the application such as the tank, its contents, the heat transfer characteristics of the coil, the rate at which heat is transferred to the thermostat, and so on.

Essentially, with on/off control, there are upper and lower switching limits, and the valve is either fully open or fully closed - there is no intermediate state.

However, controllers are available that provide a proportioning time control, in which it is possible to alter the ratio of the 'on' time to the 'off' time to control the controlled condition. This proportioning action occurs within a selected bandwidth around the set point; the set point being the bandwidth mid point.

If the controlled condition is outside the bandwidth, the output signal from the controller is either fully on or fully off, acting as an on/off device. If the controlled condition is within the bandwidth, the controller output is turned on and off relative to the deviation between the value of the controlled condition and the set point.

With the controlled condition being at set point, the ratio of 'on' time to 'off' time is 1:1, that is, the 'on' time equals the 'off' time. If the controlled condition is below the set point, the 'on' time will be longer than the 'off' time, whilst if above the set point, the 'off' time will be longer, relative to the deviation within the bandwidth.

The main advantages of on/off control are that it is simple and very low cost. This is why it is frequently found on domestic type applications such as central heating boilers and heater fans.

Its major disadvantage is that the operating differential might fall outside the control tolerance required by the process. For example, on a food production line, where the taste and repeatability of taste is determined by precise temperature control, on/off control could well be unsuitable.

By contrast, in the case of space heating there are often large storage capacities (a large area to heat or cool that will respond to temperature change slowly) and slight variation in the desired value is acceptable. In many cases on/off control is quite appropriate for this type of application.

If on/off control is unsuitable because more accurate temperature control is required, the next option is continuous control.

Continuous control

Continuous control is often called modulating control. It means that the valve is capable of moving continually to change the degree of valve opening or closing. It does not just move to either fully open or fully closed, as with on-off control.

There are three basic control actions that are often applied to continuous control:

- Proportional (P)
- Integral (I)
- Derivative (D)

It is also necessary to consider these in combination such as P + I, P + D, P + I + D. Although it is possible to combine the different actions, and all help to produce the required response, it is important to remember that both the integral and derivative actions are usually corrective functions of a basic proportional control action.

The three control actions are considered below.

Proportional control

This is the most basic of the continuous control modes and is usually referred to by use of the letter 'P'. The principle aim of proportional control is to control the process as the conditions change.

This section shows that:

- The larger the proportional band, the more stable the control, but the greater the offset.
- The narrower the proportional band, the less stable the process, but the smaller the offset.

The aim, therefore, should be to introduce the smallest acceptable proportional band that will always keep the process stable with the minimum offset.

In explaining proportional control, several new terms must be introduced.

To define these, a simple analogy can be considered - a cold water tank is supplied with water via a float operated control valve and with a globe valve on the outlet pipe valve 'V', as shown in Figure 5.2.4. Both valves are the same size and have the same flow capacity and flow characteristic. The desired water level in the tank is at point **B** (equivalent to the set point of a level controller).

It can be assumed that, with valve 'V' half open, (50% load) there is just the right flowrate of water entering via the float operated valve to provide the desired flow out through the discharge pipe, and to maintain the water level in the tank at point at **B**.

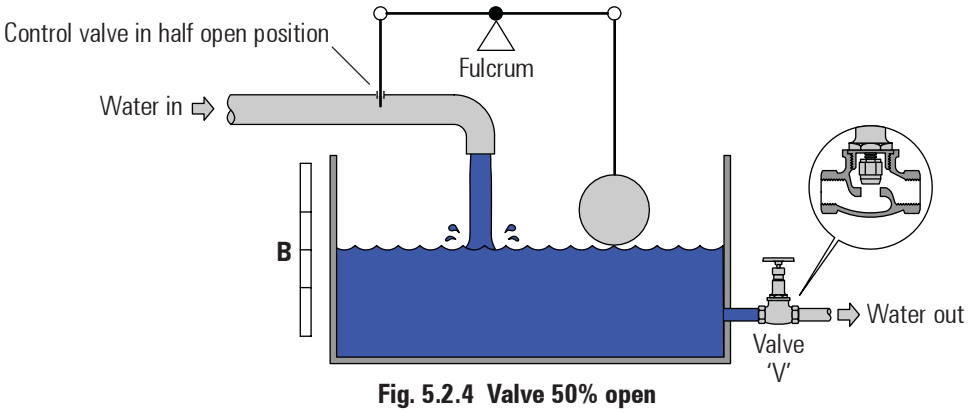


Fig. 5.2.4 Valve 50% open

The system can be said to be in balance (the flowrate of water entering and leaving the tank is the same); under control, in a stable condition (the level is not varying) and at precisely the desired water level (**B**); giving the required outflow.

With the valve 'V' closed, the level of water in the tank rises to point **A** and the float operated valve cuts off the water supply (see Figure 5.2.5 below).

The system is still under control and stable but control is above level **B**. The difference between level **B** and the actual controlled level, **A**, is related to the proportional band of the control system.

Once again, if valve 'V' is half opened to give 50% load, the water level in the tank will return to the desired level, point **B**.

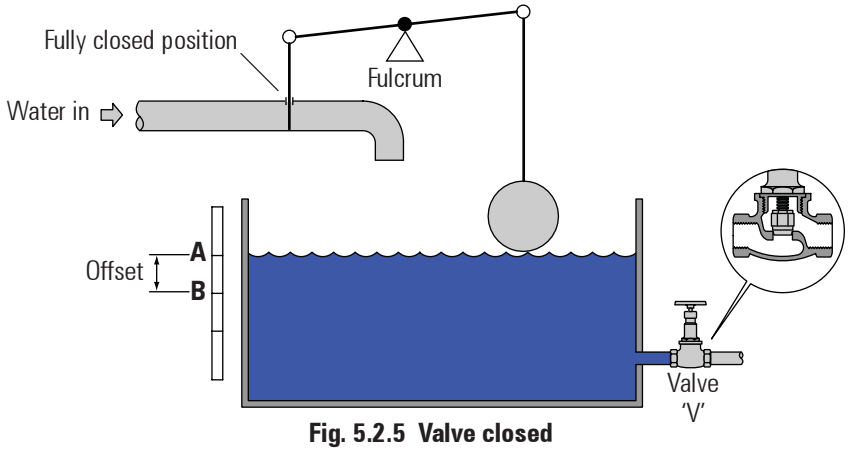


Fig. 5.2.5 Valve closed

In Figure 5.2.6 below, the valve 'V' is fully opened (100% load). The float operated valve will need to drop to open the inlet valve wide and admit a higher flowrate of water to meet the increased demand from the discharge pipe. When it reaches level **C**, enough water will be entering to meet the discharge needs and the water level will be maintained at point **C**.

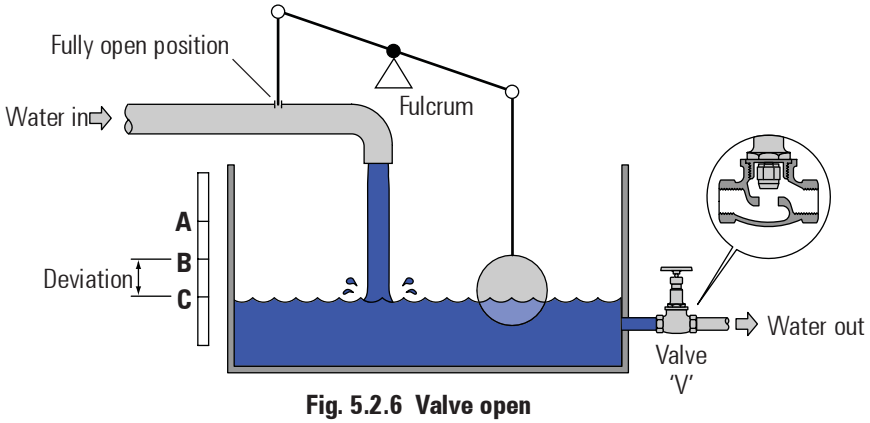


Fig. 5.2.6 Valve open

The system is under control and stable, but there is an offset; the deviation in level between points **B** and **C**. Figure 5.2.7 combines the three conditions used in this example.

The difference in levels between points **A** and **C** is known as the Proportional Band or P-band, since this is the change in level (or temperature in the case of a temperature control) for the control valve to move from fully open to fully closed.

One recognised symbol for Proportional Band is X_P.

The analogy illustrates several basic and important points relating to proportional control:

- The control valve is moved in proportion to the error in the water level (or the temperature deviation, in the case of a temperature control) from the set point.
- The set point can only be maintained for one specific load condition.
- Whilst stable control will be achieved between points **A** and **C**, any load causing a difference in level to that of **B** will always provide an offset.

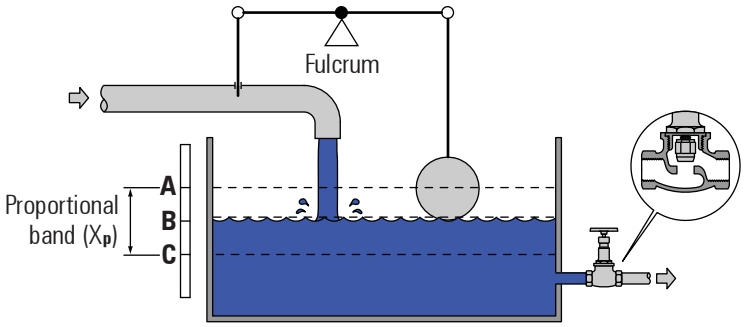


Fig. 5.2.7 Proportional band

Note: By altering the fulcrum position, the system Proportional Band changes. Nearer the float gives a narrower P-band, whilst nearer the valve gives a wider P-band. Figure 5.2.8 illustrates why this is so. Different fulcrum positions require different changes in water level to move the valve from fully open to fully closed. In both cases, It can be seen that level **B** represents the 50% load level, **A** represents the 0% load level, and **C** represents the 100% load level. It can also be seen how the offset is greater at any same load with the wider proportional band.

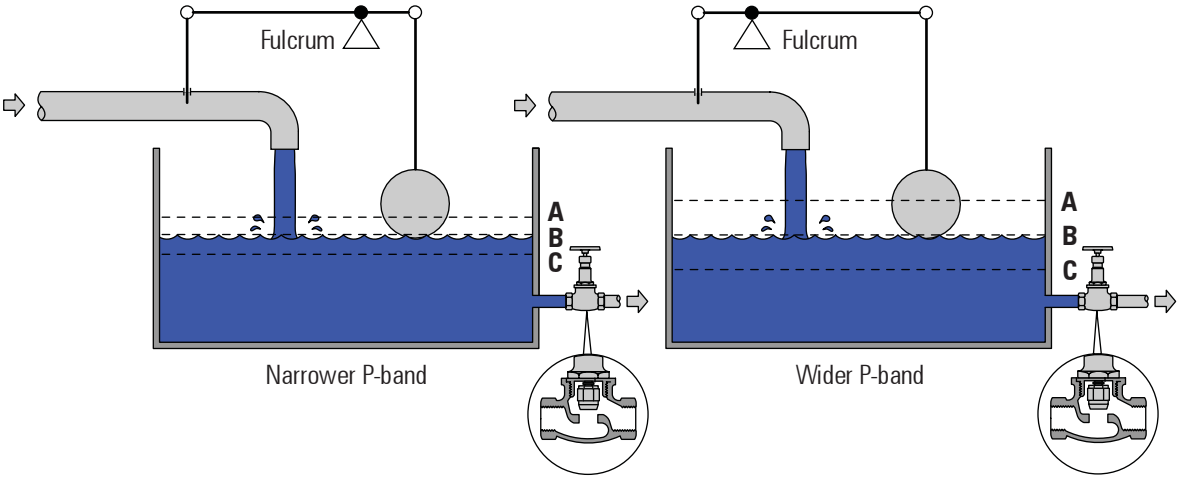


Fig. 5.2.8 Demonstrating the relationship between P-band and offset

The examples depicted in Figures 5.2.4 through to 5.2.8 describe proportional band as the level (or perhaps temperature or pressure etc.) change required to move the valve from fully open to fully closed. This is convenient for mechanical systems, but a more general (and more correct) definition of proportional band is the percentage change in measured value required to give a 100% change in output. It is therefore usually expressed in percentage terms rather than in engineering units such as degrees centigrade.

For electrical and pneumatic controllers, the set value is at the middle of the proportional band. The effect of changing the P-band for an electrical or pneumatic system can be described with a slightly different example, by using a temperature control.

The space temperature of a building is controlled by a water (radiator type) heating system using a proportional action control by a valve driven with an electrical actuator, and an electronic controller and room temperature sensor. The control selected has a proportional band (P-band or X_p) of 6% of the controller input span of 0° - 100°C, and the desired internal space temperature is 18°C. Under certain load conditions, the valve is 50% open and the required internal temperature is correct at 18°C.

A fall in outside temperature occurs, resulting in an increase in the rate of heat loss from the building. Consequently, the internal temperature will decrease. This will be detected by the room temperature sensor, which will signal the valve to move to a more open position allowing hotter water to pass through the room radiators.

The valve is instructed to open by an amount proportional to the drop in room temperature. In simplistic terms, if the room temperature falls by 1°C, the valve may open by 10%; if the room temperature falls by 2°C, the valve will open by 20%.

In due course, the outside temperature stabilises and the inside temperature stops falling. In order to provide the additional heat required for the lower outside temperature, the valve will stabilise in a more open position; but the actual inside temperature will be slightly lower than 18°C.

Example 5.2.1 and Figure 5.2.9 explain this further, using a P-band of 6°C.

Example 5.2.1 Consider a space heating application with the following characteristics:

1. The required temperature in the building is 18°C.

2. The room temperature is currently 18°C, and the valve is 50% open.

3. The proportional band is set at 6% of 100°C = 6°C, which gives 3°C either side of the 18°C set point.

Figure 5.2.9 shows the room temperature and valve relationship:

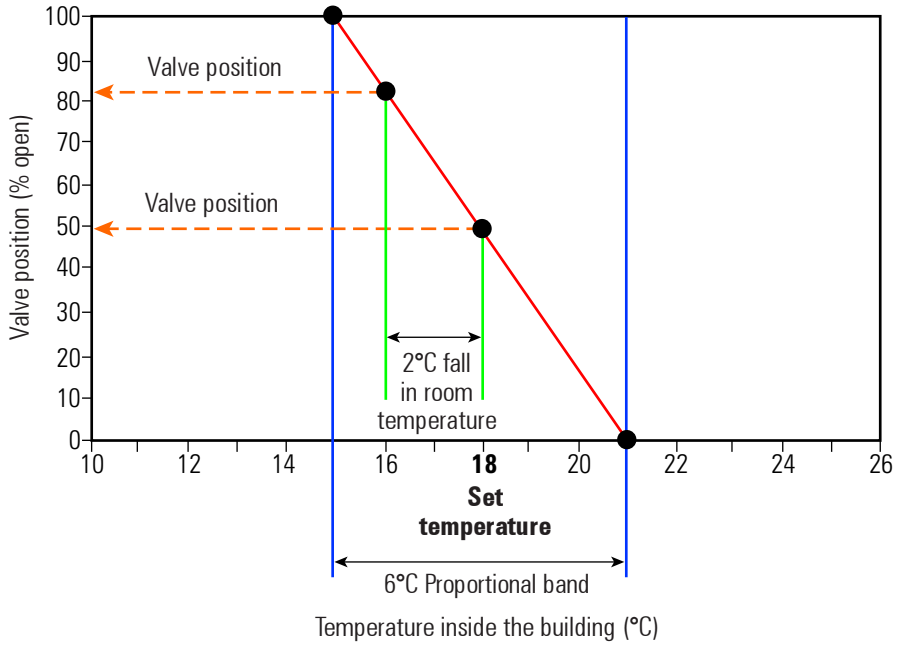


Fig. 5.2.9 Room temperature and valve relationship - 6°C proportional band

As an example, consider the room temperature falling to 16°C. From the chart it can be seen that the new valve opening will be approximately 83%.

With proportional control, if the load changes, so too will the offset:

- A load of less than 50% will cause the room temperature to be above the set value.
- A load of more than 50% will cause the room temperature to be below the set value.

The deviation between the set temperature on the controller (the set point) and the actual room temperature is called the 'proportional offset'.

In Example 5.2.1, as long as the load conditions remain the same, the control will remain steady at a valve opening of 83.3%; this is called 'sustained offset'.

The effect of adjusting the P-band

In electronic and pneumatic controllers, the P-band is adjustable. This enables the user to find a setting suitable for the individual application.

Increasing the P-band - For example, if the previous application had been programmed with a 12% proportional band equivalent to 12°C, the results can be seen in Figure 5.2.10. Note that the wider P-band results in a less steep 'gain' line. For the same change in room temperature the valve movement will be smaller. The term 'gain' is discussed in a following section.

In this instance, the 2°C fall in room temperature would give a valve opening of about 68% from the chart in Figure 5.2.10.

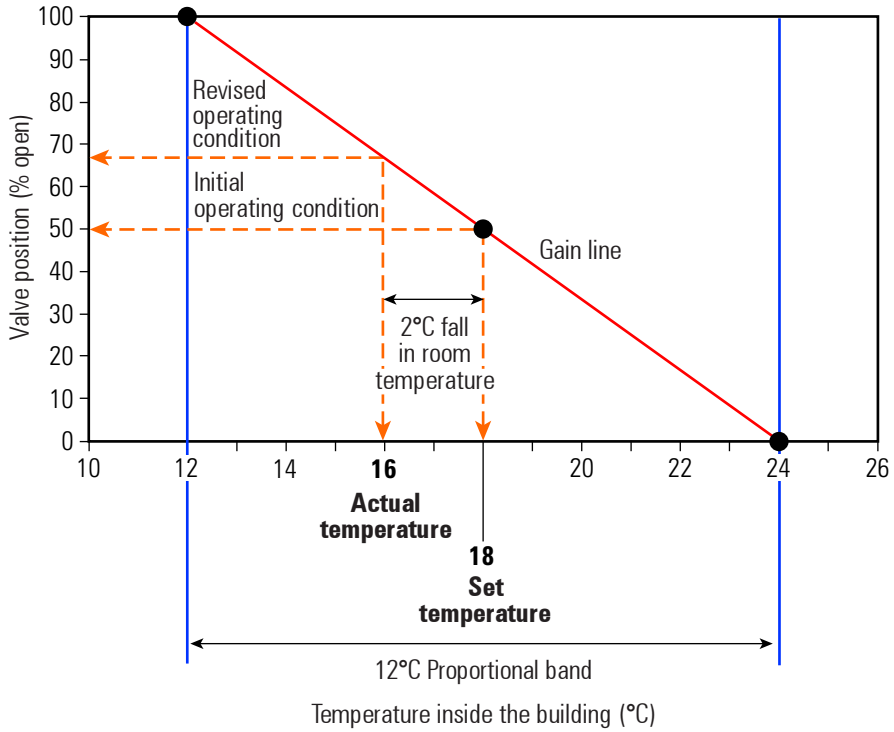


Fig. 5.2.10 Room temperature and valve relationship - 12°C Proportional band

Reducing the P-band - Conversely, if the P-band is reduced, the valve movement per temperature increment is increased. However, reducing the P-band to zero gives an on/off control. The ideal P-band is as narrow as possible without producing a noticeable oscillation in the actual room temperature.

Gain

The term 'gain' is often used with controllers and is simply the reciprocal of proportional band.

The larger the controller gain, the more the controller output will change for a given error. For instance for a gain of 1, an error of 10% of scale will change the controller output by 10% of scale, for a gain of 5, an error of 10% will change the controller output by 50% of scale, whilst for a gain of 10, an error of 10% will change the output by 100% of scale.

The proportional band in 'degree terms' will depend on the controller input scale. For instance, for a controller with a 200°C input scale: An X_p of 20% = 20% of 200°C = 40°C
An X_p of 10% = 10% of 200°C = 20°C

Example 5.2.2

Let the input span of a controller be 100°C.

If the controller is set so that full change in output occurs over a proportional band of 20% the controller gain is:

$$\frac{100\%}{20\%} = 5$$

Equally it could be said that the proportional band is 20% of 100°C = 20°C and the gain is:

$$\frac{100°C}{20°C} = 5$$

The controller in Example 5.2.1 had a gain of: $\frac{100°C}{10°C} = 10$

Therefore the relationship between P-band and Gain is:

$$\text{Gain} = \frac{100\%}{\%\ \text{P-band}} = \text{a number}$$

$$\text{or Gain} = \frac{\text{Input span °C}}{\text{P-band °C}} = \text{a number}$$

As a reminder:

- A wide proportional band (small gain) will provide a less sensitive response, but a greater stability.
- A narrow proportional band (large gain) will provide a more sensitive response, but there is a practical limit to how narrow the X_p can be set.
- Too narrow a proportional band (too much gain) will result in oscillation and unstable control.

For any controller for various P-bands, gain lines can be determined as shown in Figure 5.2.11, where the controller input span is 100°C.

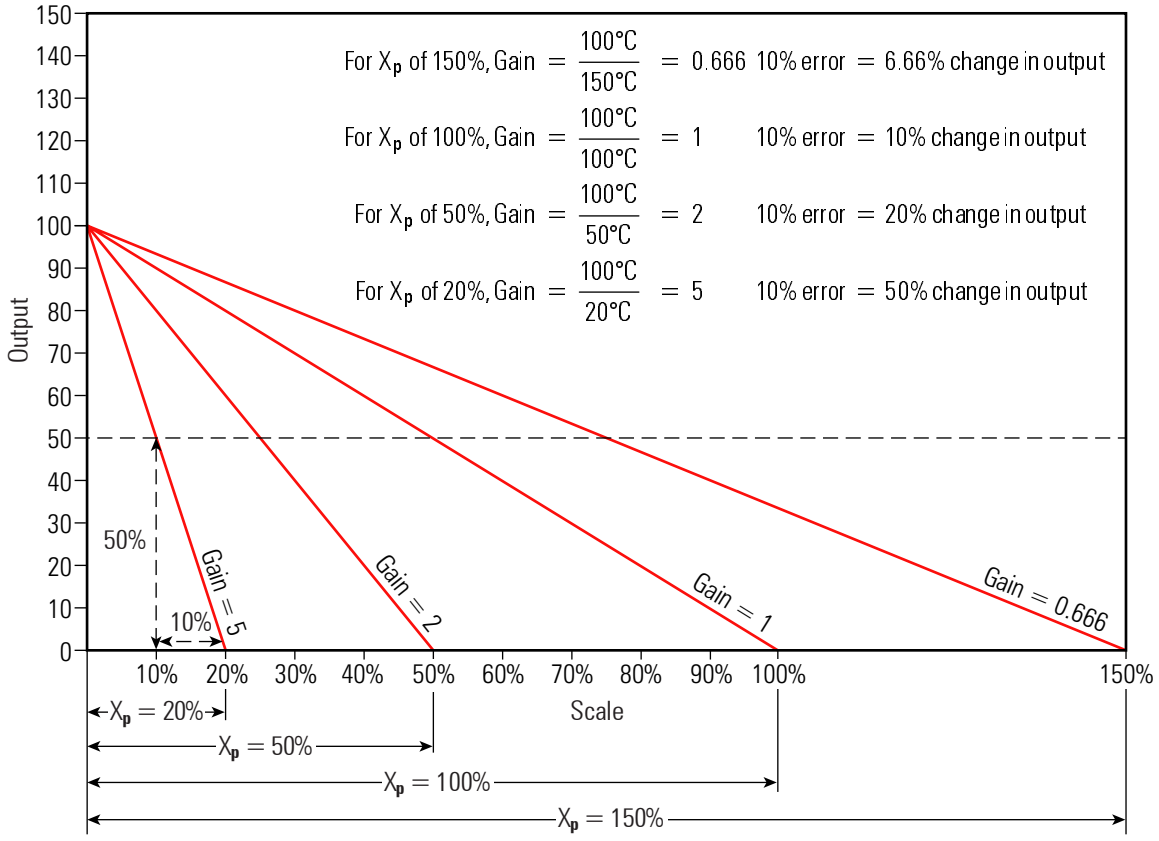


Fig. 5.2.11 Proportional band and gain

Reverse or direct acting control signal

A closer look at the figures used so far to describe the effect of proportional control shows that the output is assumed to be reverse acting. In other words, a rise in process temperature causes the control signal to fall and the valve to close. This is usually the situation on heating controls. This configuration would not work on a cooling control; here the valve must open with a rise in temperature. This is termed a direct acting control signal. Figures 5.2.12 and 5.2.13 depict the difference between reverse and direct acting control signals for the same valve action.

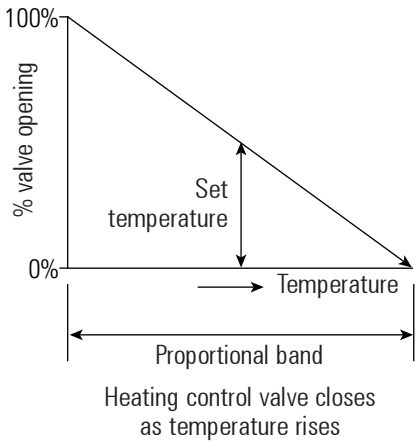


Fig. 5.2.12 Reverse acting signal

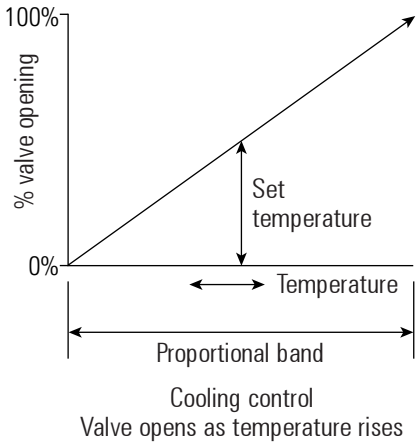


Fig. 5.2.13 Direct acting signal

On mechanical controllers (such as a pneumatic controller) it is usual to be able to invert the output signal of the controller by rotating the proportional control dial. Thus, the magnitude of the proportional band and the direction of the control action can be determined from the same dial.

On electronic controllers, reverse acting (RA) or direct acting (DA) is selected through the keypad.

Gain line offset or proportional effect

From the explanation of proportional control, it should be clear that there is a control offset or a deviation of the actual value from the set value whenever the load varies from 50%.

To further illustrate this, consider Example 5.2.1 with a 12°C P-band, where an offset of 2°C was expected. If the offset cannot be tolerated by the application, then it must be eliminated.

This could be achieved by relocating (or resetting) the set point to a higher value. This provides the same valve opening after manual reset but at a room temperature of 18°C not 16°C.

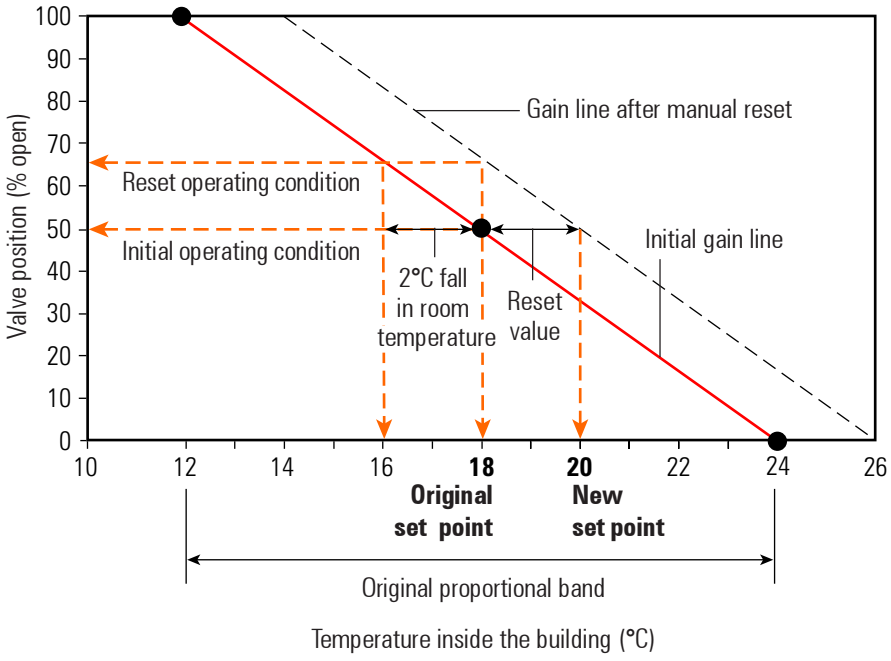


Fig. 5.2.14 Gain line offset

Manual reset

The offset can be removed either manually or automatically. The effect of manual reset can be seen in Figure 5.2.14, and the value is adjusted manually by applying an offset to the set point of 2°C.

It should be clear from Figure 5.2.14 and the above text that the effect is the same as increasing the set value by 2°C. The same valve opening of 66.7% now coincides with the room temperature at 18°C.

The effects of manual reset are demonstrated in Figure 5.2.15

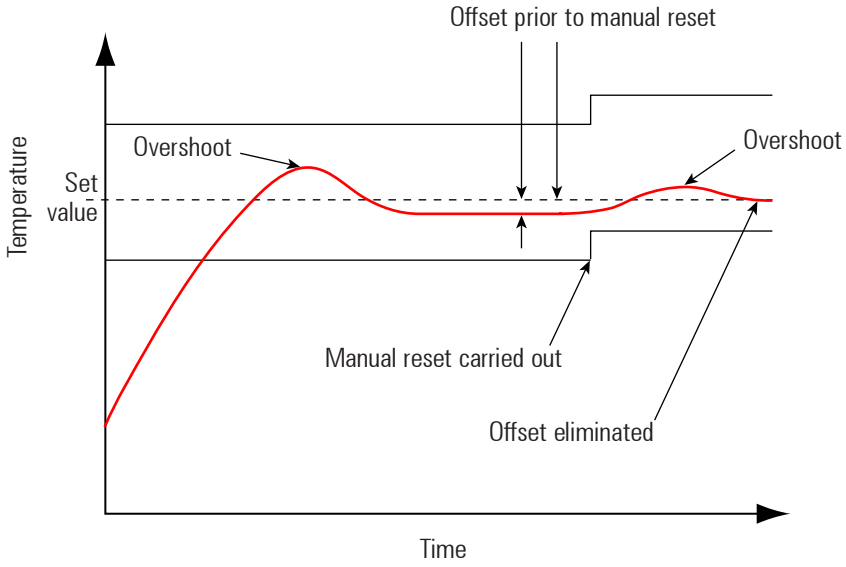


Fig. 5.2.15 Effect of manual reset

Integral control - automatic reset action

'Manual reset' is usually unsatisfactory in process plant where each load change will require a reset action. It is also quite common for an operator to be confused by the differences between:

- Set value - What is on the dial.
- Actual value - What the process value is.
- Required value - The perfect process condition.

Such problems are overcome by the reset action being contained within the mechanism of an automatic controller.

Such a controller is primarily a proportional controller. It then has a reset function added, which is called 'integral action'. Automatic reset uses an electronic or pneumatic integration routine to perform the reset function. The most commonly used term for automatic reset is integral action, which is given the letter I.

The function of integral action is to eliminate offset by continuously and automatically modifying the controller output in accordance with the control deviation integrated over time. The Integral Action Time (**IAT**) is defined as the time taken for the controller output to change due to the integral action to equal the output change due to the proportional action. Integral action gives a steadily increasing corrective action as long as an error continues to exist. Such corrective action will increase with time and must therefore, at some time, be sufficient to eliminate the steady state error altogether, providing sufficient time elapses before another change occurs. The controller allows the integral time to be adjusted to suit the plant dynamic behaviour.

Proportional plus integral (**P + I**) becomes the terminology for a controller incorporating these features.

The integral action on a controller is often restricted to within the proportional band. A typical P + I response is shown in Figure 5.2.16, for a step change in load.

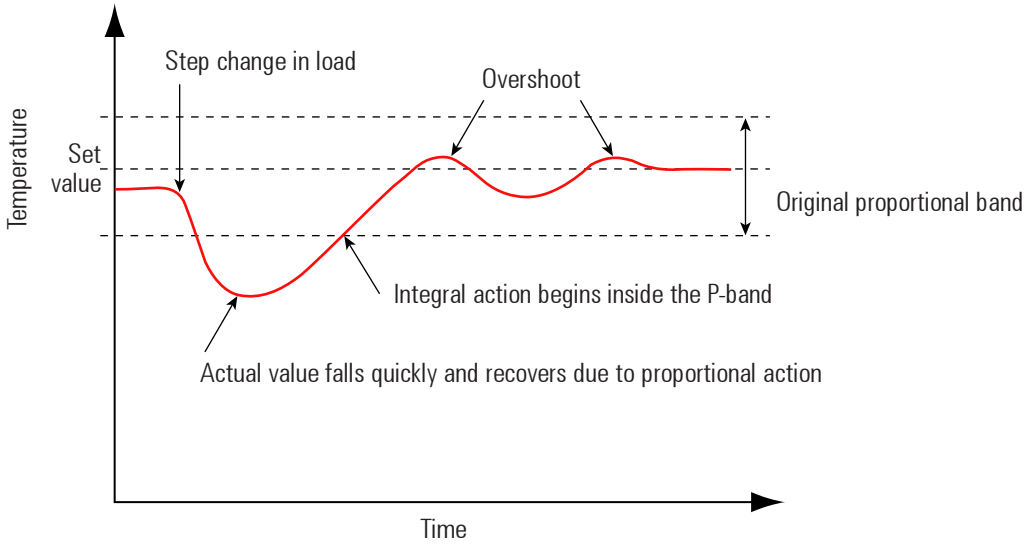


Fig. 5.2.16 P+I Function after a step change in load

The IAT is adjustable within the controller:

- If it is too short, over-reaction and instability will result.
- If it is too long, reset action will be very slow to take effect.

IAT is represented in time units. On some controllers the adjustable parameter for the integral action is termed 'repeats per minute', which is the number of times per minute that the integral action output changes by the proportional output change.

- Repeats per minute $= {}^{1}/_{\text{(IAT in minutes)}}$
- IAT = Infinity – Means no integral action
- IAT = 0 – Means infinite integral action

It is important to check the controller manual to see how integral action is designated.

Overshoot and 'wind up'

With P+ I controllers (and with P controllers), overshoot is likely to occur when there are time lags on the system.

A typical example of this is after a sudden change in load. Consider a process application where a process heat exchanger is designed to maintain water at a fixed temperature.

The set point is 80°C, the P-band is set at 5°C (±2.5°C), and the load suddenly changes such that the returning water temperature falls almost instantaneously to 60°C.

Figure 5.2.16 shows the effect of this sudden (step change) in load on the actual water temperature. The measured value changes almost instantaneously from a steady 80°C to a value of 60°C.

By the nature of the integration process, the generation of integral control action must lag behind the proportional control action, introducing a delay and more dead time to the response. This could have serious consequences in practice, because it means that the initial control response, which in a proportional system would be instantaneous and fast acting, is now subjected to a delay and responds slowly. This may cause the actual value to run out of control and the system to oscillate. These oscillations may increase or decrease depending on the relative values of the controller gain and the integral action. If applying integral action it is important to make sure, that it is necessary and if so, that the correct amount of integral action is applied.

Integral control can also aggravate other situations. If the error is large for a long period, for example after a large step change or the system being shut down, the value of the integral can become excessively large and cause overshoot or undershoot that takes a long time to recover. To avoid this problem, which is often called 'integral wind-up', sophisticated controllers will inhibit integral action until the system gets fairly close to equilibrium.

To remedy these situations it is useful to measure the rate at which the actual temperature is changing; in other words, to measure the rate of change of the signal. Another type of control mode is used to measure how fast the measured value changes, and this is termed Rate Action or Derivative Action.

Derivative control - rate action

A Derivative action (referred to by the letter D) measures and responds to the rate of change of process signal, and adjusts the output of the controller to minimise overshoot.

If applied properly on systems with time lags, derivative action will minimise the deviation from the set point when there is a change in the process condition. It is interesting to note that derivative action will only apply itself when there is a change in process signal. If the value is steady, whatever the offset, then derivative action does not occur.

One useful function of the derivative function is that overshoot can be minimised especially on fast changes in load. However, derivative action is not easy to apply properly; if not enough is used, little benefit is achieved, and applying too much can cause more problems than it solves.

D action is again adjustable within the controller, and referred to as T_D in time units:

T_D = 0 – Means no D action.

T_D = Infinity – Means infinite D action.

P + D controllers can be obtained, but proportional offset will probably be experienced. It is worth remembering that the main disadvantage with a P control is the presence of offset. To overcome and remove offset, 'I' action is introduced. The frequent existence of time lags in the control loop explains the need for the third action D. The result is a P + I + D controller which, if properly tuned, can in most processes give a rapid and stable response, with no offset and without overshoot.

PID controllers

P and I and D are referred to as 'terms' and thus a P + I + D controller is often referred to as a three term controller.

Summary of modes of control

A three-term controller contains three modes of control:

- Proportional (P) action with adjustable gain to obtain stability.
- Reset (Integral) (I) action to compensate for offset due to load changes.
- Rate (Derivative) (D) action to speed up valve movement when rapid load changes take place.

The various characteristics can be summarised, as shown in Figure 5.2.17.

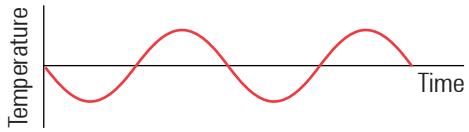
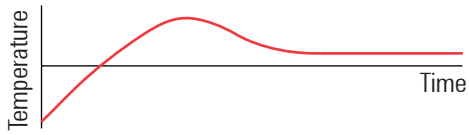
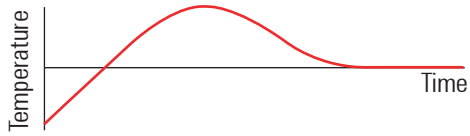
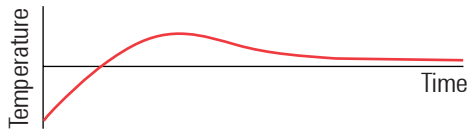
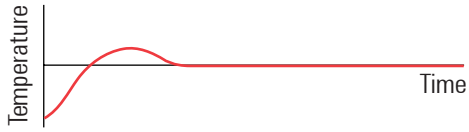
Control mode	Typical system responses	Advantages/disadvantages
On / off		▪ Inexpensive ▪ Simple ▪ Operating differential can be outside of process requirements
Proportional P		▪ Simple and stable ▪ Fairly high initial deviation (unless a large P-band is chosen), then sustained offset ▪ Easy to set up ▪ Offset occurs
Proportional plus Integral P + I		▪ No sustained offset ▪ Increase in proportional band usually required to overcome instability ▪ Possible increased overshoot on start-up
Proportional plus Derivative P+D		▪ Stable ▪ Some offset ▪ Rapid response to changes
Proportional plus Integral plus Derivative P+I+D		▪ Will give best control, no offset and minimal overshoot ▪ More complex to set up manually but most electronic controllers have an 'autotune' facility. ▪ More expensive where pneumatic controllers are concerned

Fig. 5.2.17 Summary of control modes and responses

Finally, the controls engineer must try to avoid the danger of using unnecessarily complicated controls for a specific application. The least complicated control action, which will provide the degree of control required, should always be selected.

Further terminology

Time constant

This is defined as: 'The time taken for a controller output to change by 63.2% of its total due to a step (or sudden) change in process load'.

In reality, the explanation is more involved because the time constant is really the time taken for a signal or output to achieve its final value from its initial value, had the original rate of increase been maintained. This concept is depicted in Figure 5.12.18.

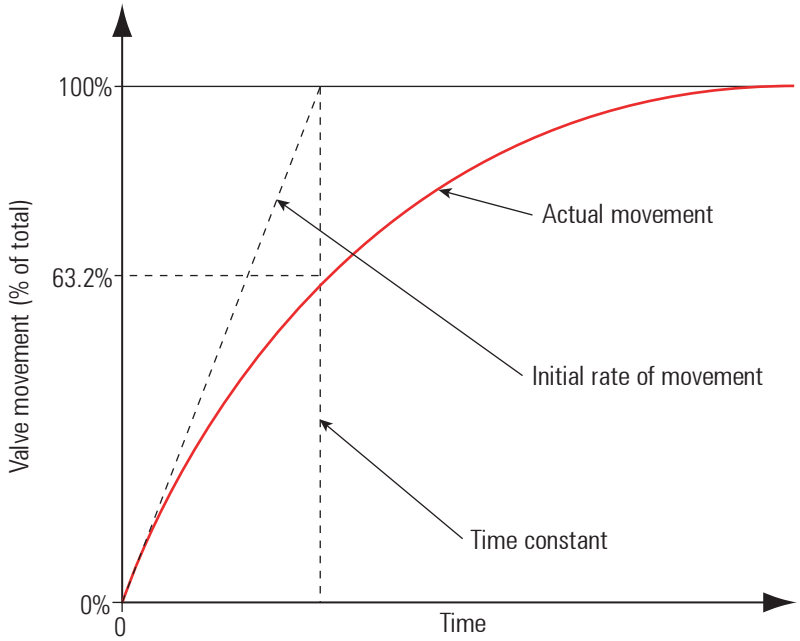


Fig. 5.2.18 Time constant

Example 5.2.2 A practical appreciation of the time constant

Consider two tanks of water, tank A at a temperature of 25°C, and tank B at 75°C. A sensor is placed in tank A and allowed to reach equilibrium temperature. It is then quickly transferred to tank B. The temperature difference between the two tanks is 50°C, and 63.2% of this temperature span can be calculated as shown below:

$$63.2\% \text{ of } 50°C = 31.6°C$$

The initial datum temperature was 25°C, consequently the time constant for this simple example is the time required for the sensor to reach 56.6°C, as shown below:

$$25°C + 31.6°C = 56.6°C$$

Hunting

Often referred to as instability, cycling or oscillation. Hunting produces a continuously changing deviation from the normal operating point. This can be caused by:

- The proportional band being too narrow.
- The integral time being too short.
- The derivative time being too long.
- A combination of these.
- Long time constants or dead times in the control system or the process itself.

In Figure 5.2.19 the heat exchanger is oversized for the application. Accurate temperature control will be difficult to achieve and may result in a large proportional band in an attempt to achieve stability.

If the system load suddenly increases, the two port valve will open wider, filling the heat exchanger with high temperature steam. The heat transfer rate increases extremely quickly causing the water system temperature to overshoot. The rapid increase in water temperature is picked up by the sensor and directs the two port valve to close quickly. This causes the water temperature to fall, and the two port valve to open again. This cycle is repeated, the cycling only ceasing when the PID terms are adjusted. The following example (Example 5.2.3) gives an idea of the effects of a hunting steam system.

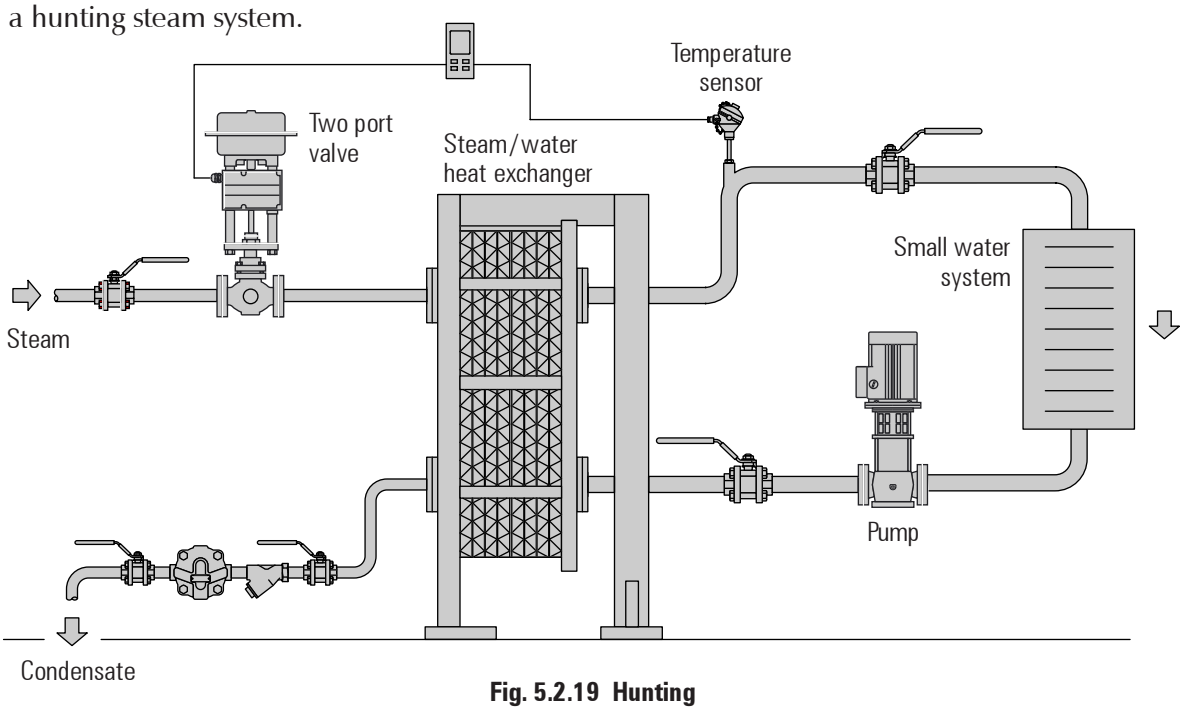


Fig. 5.2.19 Hunting

Example 5.2.3 The effect of hunting on the system in Figure 5.2.19

Consider the steam to water heat exchanger system in Figure 5.2.19. Under minimum load conditions, the size of the heat exchanger is such that it heats the constant flowrate secondary water from 60°C to 65°C with a steam temperature of 70°C. The controller has a set point of 65°C and a P-band of 10°C.

Consider a sudden increase in the secondary load, such that the returning water temperature almost immediately drops by 40°C. The temperature of the water flowing out of the heat exchanger will also drop by 40°C to 25°C. The sensor detects this and, as this temperature is below the P-band, it directs the pneumatically actuated steam valve to open fully.

The steam temperature is observed to increase from 70°C to 140°C almost instantaneously. What is the effect on the secondary water temperature and the stability of the control system?

As demonstrated in Module 13.2 (The heat load, heat exchanger and steam load relationship), the heat exchanger temperature design constant, TDC, can be calculated from the observed operating conditions and Equation 13.2.2:

$$TDC = \frac{T_s - T_1}{T_s - T_2} \qquad \textbf{Equation 13.2.2}$$

Where:

TDC = Temperature Design Constant
T_s = Steam temperature
T_1 = Secondary fluid inlet temperature
T_2 = Secondary fluid outlet temperature

In this example, the observed conditions (at minimum load) are as follows:

The inlet water temperature (T1) = 60°C

The outlet water temperature (T2) = 65°C

Steam temperature (Ts) = 70°C

$$TDC = \frac{T_s - T_1}{T_s - T_2}$$

$$TDC = \frac{70 - 60}{70 - 65}$$

$$TDC = \frac{10}{5}$$

TDC = 2

When the steam temperature rises to 140°C, it is possible to predict the outlet temperature from Equation 13.2.5:

$$T_2 = T_s - \left[\frac{T_s - T_1}{TDC}\right]$$ **Equation 13.2.5**

Where:

T_s = 140°C
T_1 = 60°C - 40°C = 20°C
TDC = 2

$$T_2 = 140 - \frac{(140 - 20)}{2}$$

$$T_2 = 140 - 60$$

T_2 = 80°C

The heat exchanger outlet temperature is 80°C, which is now above the P-band, and the sensor now signals the controller to shut down the steam valve.

The steam temperature falls rapidly, causing the outlet water temperature to fall; and the steam valve opens yet again. The system cycles around these temperatures until the control parameters are changed. These symptoms are referred to as 'hunting'. The control valve and its controller are hunting to find a stable condition. In practice, other factors will add to the uncertainty of the situation, such as the system size and reaction to temperature change and the position of the sensor.

Hunting of this type can cause premature wear of system components, in particular valves and actuators, and gives poor control.

Example 5.2.3 is not typical of a practical application. In reality, correct design and sizing of the control system and steam heated heat exchanger would not be a problem.

Lag

Lag is a delay in response and will exist in both the control system and in the process or system under control.

Consider a small room warmed by a heater, which is controlled by a room space thermostat. A large window is opened admitting large amounts of cold air. The room temperature will fall but there will be a delay while the mass of the sensor cools down to the new temperature - this is known as control lag. The delay time is also referred to as dead time.

Having then asked for more heat from the room heater, it will be some time before this takes effect and warms up the room to the point where the thermostat is satisfied. This is known as system lag or thermal lag.

Rangeability

This relates to the control valve and is the ratio between the maximum controllable flow and the minimum controllable flow, between which the characteristics of the valve (linear, equal percentage, quick opening) will be maintained. With most control valves, at some point before the fully closed position is reached, there is no longer a defined control over flow in accordance with the valve characteristics. Reputable manufacturers will provide rangeability figures for their valves.

Turndown ratio

Turndown ratio is the ratio between the maximum flow and the minimum controllable flow. It will be substantially less than the valve's rangeability if the valve is oversized.

Although the definition relates only to the valve, it is a function of the complete control system.

Questions

1. In an on/off control the upper limit is 80°C and the lower limit 76°C. What term is used for the 4°C difference?

a| Offset ☐

b| Deviation ☐

c| Switching differential ☐

d| Proportional band ☐

2. In an on/off application the upper switching point is 50°C and the lower switching point is 48°C. The process temperature actually overshoots to 52°C and undershoots to 46°C. What term is used to describe the 46 - 52°C range?

a| Operating differential ☐

b| Switching differential ☐

c| Controlled condition ☐

d| Sustained deviation ☐

3. A controller is adjusted to give a larger proportional band. What is the likely effect?

a| Stable process conditions with a larger offset ☐

b| Unstable process conditions with a smaller or offset ☐

c| Unstable process conditions with a larger offset ☐

d| Stable process conditions with a smaller offset ☐

4. A pneumatic pressure controller on a pressure reducing application has proportional action only. It has a set point of 4 bar g and a proportional band of 0.4 bar. What position will the valve be in at 4 bar g, and at what sensed pressure will the valve be wide open?

a| Closed and 3.6 bar ☐

b| 50% open and 3.6 bar ☐

c| 100% open and 4 bar ☐

d| 50% open and 3.8 bar ☐

5. Which of the following is true of a proportional control?

a| The valve is moved in proportion to the time the error occurs ☐

b| The set point can be maintained for all load conditions ☐

c| Proportional control will tend to give an offset ☐

d| Proportional control will never result in an offset ☐

6. A proportional temperature controller provides a direct acting signal to an actuator. What is the effect on the controller output of a rise in process temperature?

a| The signal will fall ☐

b| The gain line will be relocated ☐

c| The proportional band will be reduced ☐

d| The signal will increase ☐

Answers

1: c, 2: a, 3: a, 4: d, 5: c, 6: d

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Module 5.3

Control Loops and Dynamics

Control Loops and Dynamics

This Module introduces discussion on complete control systems, made up of the valve, actuator, sensor, controller and the dynamics of the process itself.

Control loops

An open loop control system

Open loop control simply means there is no direct feedback from the controlled condition; in other words, no information is sent back from the process or system under control to advise the controller that corrective action is required. The heating system shown in Figure 5.3.1 demonstrates this by using a sensor outside of the room being heated. The system shown in Figure 5.3.1 is not an example of a practical heating control system; it is simply being used to depict the principle of open loop control.

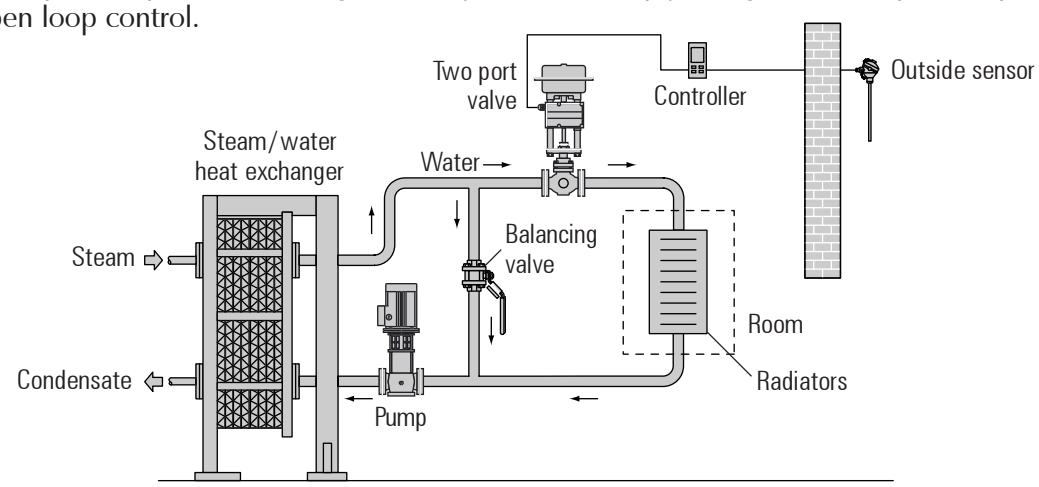


Fig. 5.3.1 Open loop control

The system consists of a proportional controller with an outside sensor sensing ambient air temperature. The controller might be set with a fairly large proportional band, such that at an ambient temperature of -1°C the valve is full open, and at an ambient of 19°C the valve is fully closed. As the ambient temperature will have an effect on the heat loss from the building, it is hoped that the room temperature will be controlled.

However, there is no feedback regarding the room temperature and heating due to other factors. In mild weather, although the flow of water is being controlled, other factors, such as high solar gain, might cause the room to overheat. In other words, open control tends only to provide a coarse control of the application.

Figure 5.3.2 depicts a slightly more sophisticated control system with two sensors.

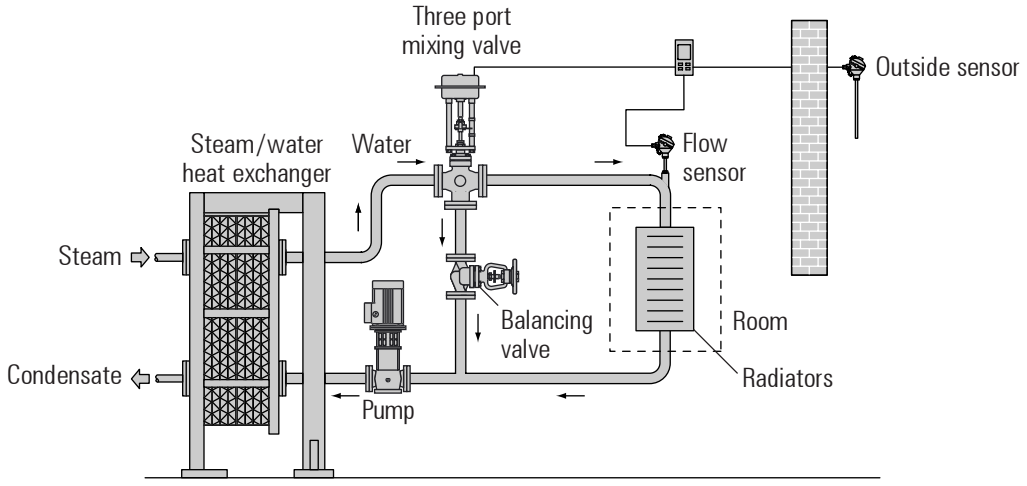


Fig. 5.3.2 Open loop control system with outside temperature sensor and water temperature sensor

The system uses a three port mixing valve with an actuator, controller and outside air sensor, plus a temperature sensor in the water line.

The outside temperature sensor provides a remote set point input to the controller, which is used to offset the water temperature set point. In this way, closed loop control applies to the water temperature flowing through the radiators.

When it is cold outside, water flows through the radiator at its maximum temperature. As the outside temperature rises, the controller automatically reduces the temperature of the water flowing through the radiators.

However, this is still open loop control as far as the room temperature is concerned, as there is no feedback from the building or space being heated. If radiators are oversized or design errors have occurred, overheating will still occur.

Closed loop control

Quite simply, a closed loop control requires feedback; information sent back direct from the process or system. Using the simple heating system shown in Figure 5.3.3, the addition of an internal space temperature sensor will detect the room temperature and provide closed loop control with respect to the room.

In Figure 5.3.3, the valve and actuator are controlled via a space temperature sensor in the room, providing feedback from the actual room temperature.

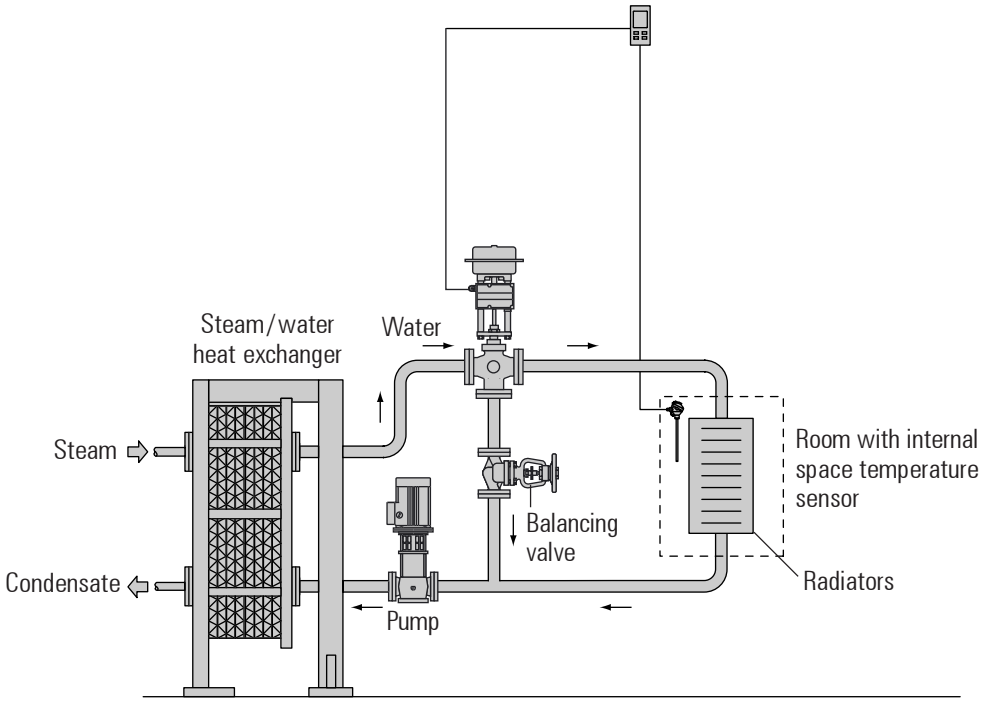


Fig. 5.3.3 Closed loop control system with sensor for internal space temperature

Disturbances

Disturbances are factors, which enter the process or system to upset the value of the controlled medium. These disturbances can be caused by changes in load or by outside influences.

For example; if in a simple heating system, a room was suddenly filled with people, this would constitute a disturbance, since it would affect the temperature of the room and the amount of heat required to maintain the desired space temperature.

Feedback control

This is another type of closed loop control. Feedback control takes account of disturbances and feeds this information back to the controller, to allow corrective action to be taken. For example, if a large number of people enter a room, the space temperature will increase, which will then cause the control system to reduce the heat input to the room.

Feed-forward control

With feed-forward control, the effects of any disturbances are anticipated and allowed for before the event actually takes place.

An example of this is bringing the boiler up to high fire before bringing a large steam-using process plant on line. The sequence of events might be that the process plant is switched on. This action, rather than opening the steam valve to the process, instructs the boiler burner to high fire. Only when the high fire position is reached is the process steam valve allowed to open, and then in a slow, controlled way.

Single loop control

This is the simplest control loop involving just one controlled variable, for instance, temperature. To explain this, a steam-to-water heat exchanger is considered as shown in Figure 5.3.4.

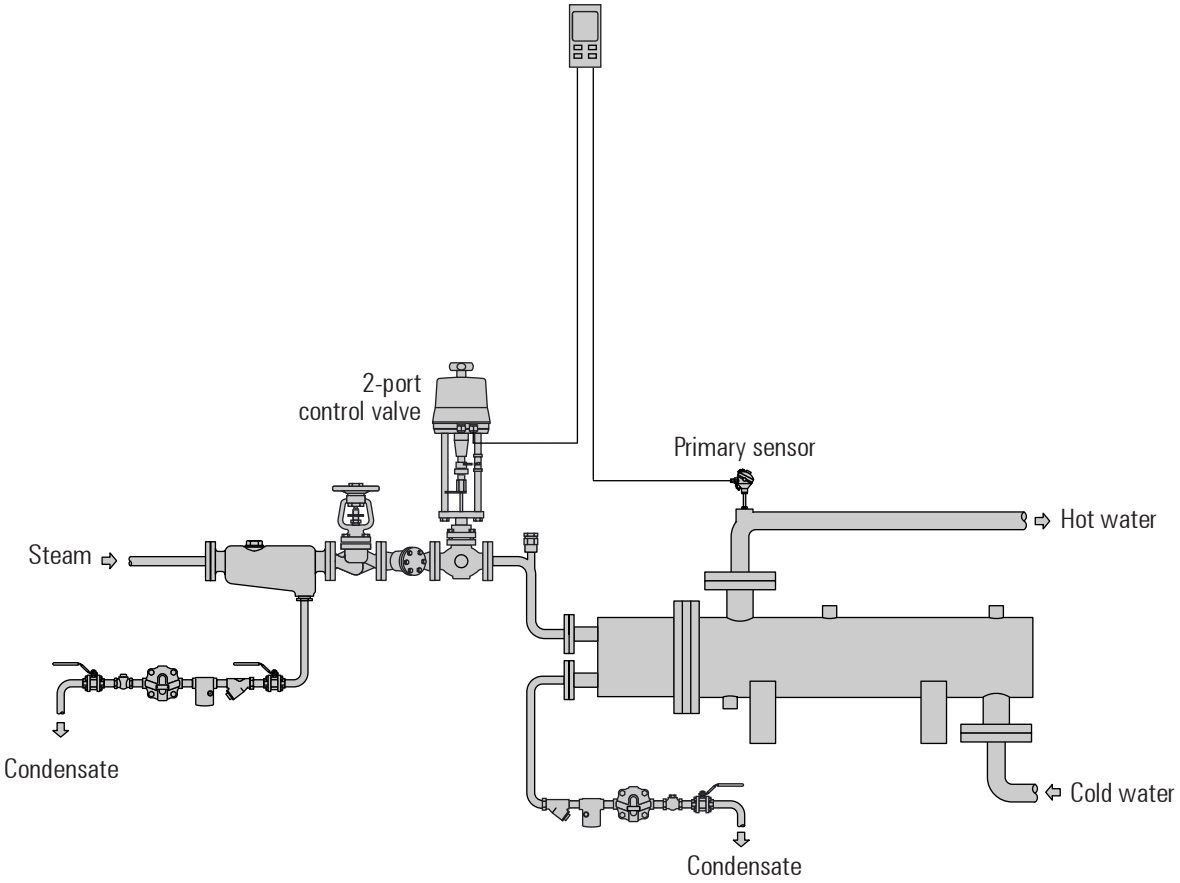


Fig. 5.3.4 Single loop control on a heating calorifier

The only one variable controlled in Figure 5.3.4 is the temperature of the water leaving the heat exchanger. This is achieved by controlling the 2-port steam valve supplying steam to the heat exchanger. The primary sensor may be a thermocouple or PT100 platinum resistance thermometer sensing the water temperature.

The controller compares the signal from the sensor to the set point on the controller. If there is a difference, the controller sends a signal to the actuator of the valve, which in turn moves the valve to a new position. The controller may also include an output indicator, which shows the percentage of valve opening.

Single control loops provide the vast majority of control for heating systems and industrial processes.

Other terms used for single control loops include:

- Set value control.
- Single closed loop control.
- Feedback control.

Multi-loop control

The following example considers an application for a slow moving timber-based product, which must be controlled to a specific humidity level (see Figures 5.3.5 and 5.3.6).

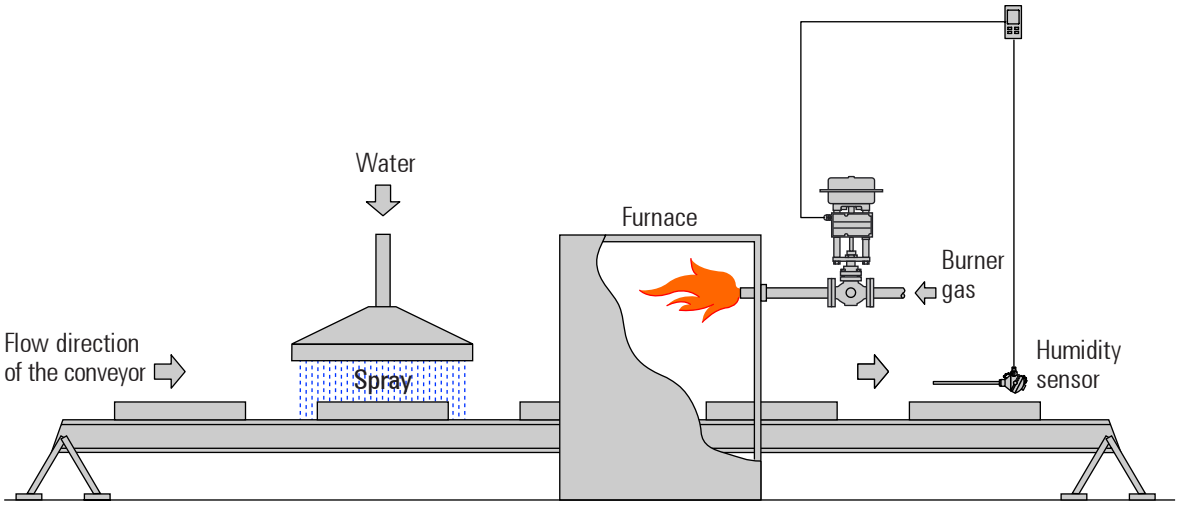


Fig. 5.3.5 Single humidity sensor

In Figure 5.3.5, the single humidity sensor at the end of the conveyor controls the amount of heat added by the furnace. But if the water spray rate changes due, for instance, to fluctuations in the water supply pressure, it may take perhaps 10 minutes before the product reaches the far end of the conveyor and the humidity sensor reacts. This will cause variations in product quality.

To improve the control, a second humidity sensor on another control loop can be installed immediately after the water spray, as shown in Figure 5.3.6. This humidity sensor provides a remote set point input to the controller which is used to offset the local set point. The local set point is set at the required humidity after the furnace. This, in a simple form, illustrates multi-loop control.

This humidity control system consists of two control loops:

- Loop 1 controls the addition of water.
- Loop 2 controls the removal of water.

Within this process, factors will influence both loops. Some factors such as water pressure will affect both loops. Loop 1 will try to correct for this, but any resulting error will have an impact on Loop 2.

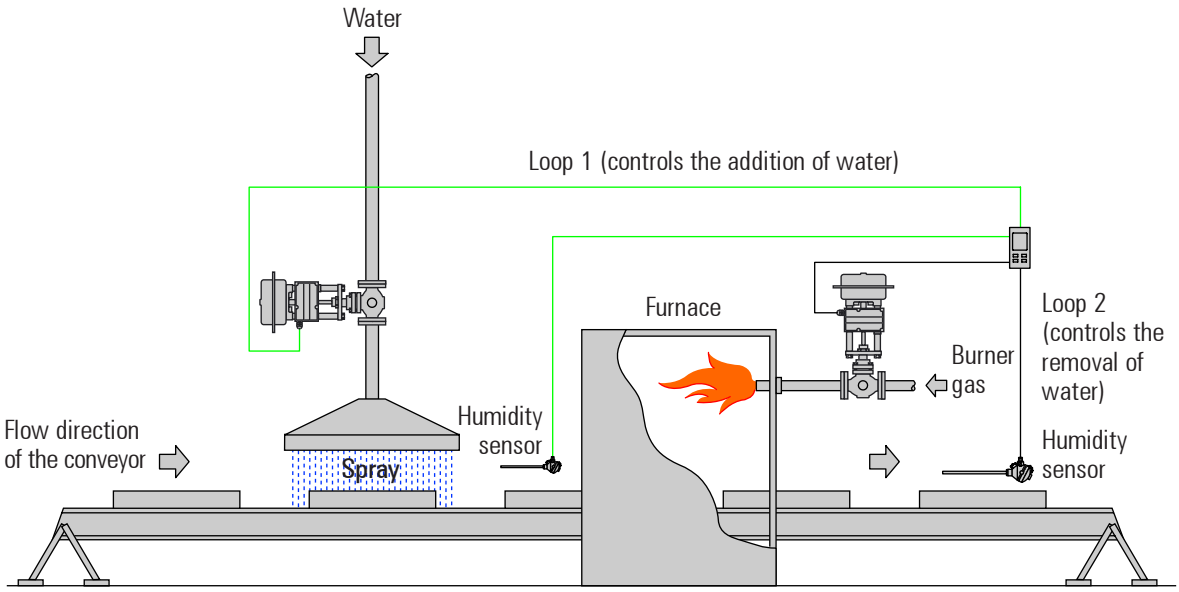


Fig. 5.3.6 Dual humidity sensors

Cascade control

Where two independent variables need to be controlled with one valve, a cascade control system may be used.

Figure 5.3.7 shows a steam jacketed vessel full of liquid product. The essential aspects of the process are quite rigorous:

- The product in the vessel must be heated to a certain temperature.
- The steam must not exceed a certain temperature or the product may be spoiled.
- The product temperature must not increase faster than a certain rate or the product may be spoiled.

If a normal, single loop control was used with the sensor in the liquid, at the start of the process the sensor would detect a low temperature, and the controller would signal the valve to move to the fully open position. This would result in a problem caused by an excessive steam temperature in the jacket.

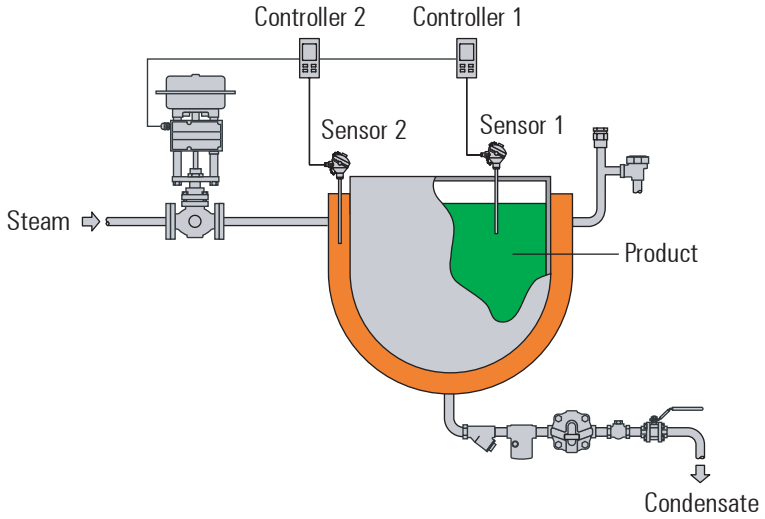


Fig. 5.3.7 Jacketed vessel

The solution is to use a cascade control using two controllers and two sensors:

- A slave controller (Controller 2) and sensor monitoring the steam temperature in the jacket, and outputting a signal to the control valve.
- A master controller (Controller 1) and sensor monitoring the product temperature with the controller output directed to the slave controller.
- The output signal from the master controller is used to vary the set point in the slave controller, ensuring that the steam temperature is not exceeded.

Example 5.3.1 An example of cascade control applied to a process vessel

The liquid temperature is to be heated from 15°C to 80°C and maintained at 80°C for two hours.

The steam temperature cannot exceed 120°C under any circumstances.

The product temperature must not increase faster than 1°C /minute.

The master controller can be ramped so that the rate of increase in water temperature is not higher than that specified.

The master controller is set in reverse acting mode, so that its output signal to the slave controller is 20 mA at low temperature and 4 mA at high temperature.

The remote set point on the slave controller is set so that its output signal to the valve is 4 mA when the steam temperature is 80°C, and 20 mA when the steam temperature is 120°C. In this way, the temperature of the steam cannot be higher than that tolerated by the system, and the steam pressure in the jacket cannot be higher than the, 1 bar g, saturation pressure at 120°C.

Dynamics of the process

This is a very complex subject but this part of the text will cover the most basic considerations.

The term 'time constant', which deals with the definition of the time taken for actuator movement, has already been outlined in Module 5.1; but to reiterate, it is the time taken for a control system to reach approximately two-thirds of its total movement as a result of a given step change in temperature, or other variable.

Other parts of the control system will have similar time based responses - the controller and its components and the sensor itself. All instruments have a time lag between the input to the instrument and its subsequent output. Even the transmission system will have a time lag - not a problem with electric/electronic systems but a factor that may need to be taken into account with pneumatic transmission systems.

Figures 5.3.8 and 5.3.9 show some typical response lags for a thermocouple that has been installed into a pocket for sensing water temperature.

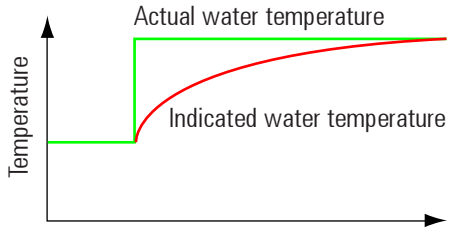


Fig. 5.3.8 Step change 5°C

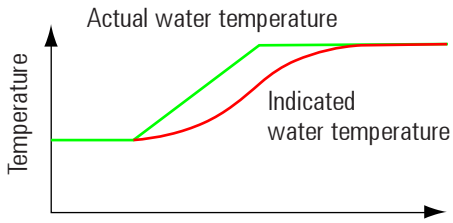


Fig. 5.3.9 Ramp change 5°C

Apart from the delays in sensor response, other parts of the control system also affect the response time. With pneumatic and self-acting systems, the valve/actuator movement tends to be smooth and, in a proportional controller, directly proportional to the temperature deviation at the sensor.

With an electric actuator there is a delay due to the time it takes for the motor to move the control linkage. Because the control signal is a series of pulses, the motor provides bursts of movement followed by periods where the actuator is stationary. The response diagram (Figure 5.3.10) depicts this. However, because of delays in the process response, the final controlled temperature can still be smooth.

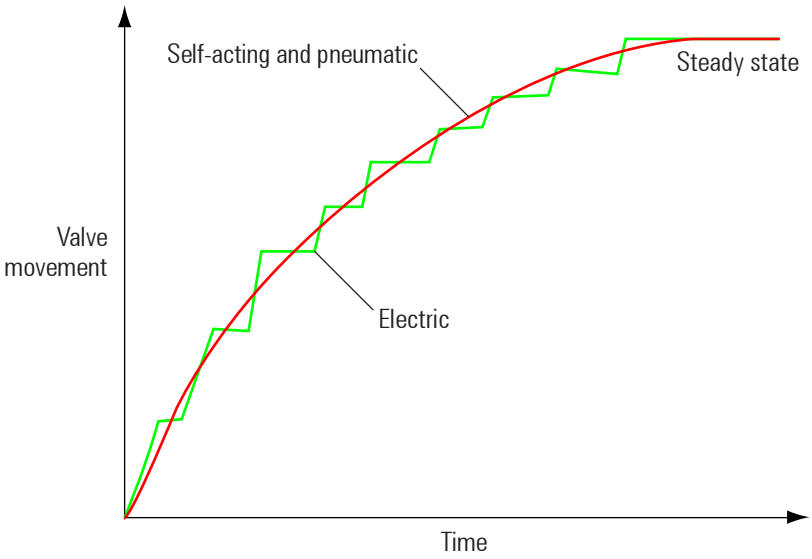


Fig. 5.3.10 Comparison of response by different actuators

The control systems covered in this Module have only considered steady state conditions. However the process or plant under control may be subject to variations following a certain behaviour pattern. The control system is required to make the process behave in a predictable manner. If the process is one which changes rapidly, then the control system must be able to react quickly. If the process undergoes slow change, the demands on the operating speed of the control system are not so stringent.

Much is documented about the static and dynamic behaviour of controllers and control systems - sensitivity, response time and so on. Possibly the most important factor of consideration is the time lag of the complete control loop.

The dynamics of the process need consideration to select the right type of controller, sensor and actuator.

Process reactions

These dynamic characteristics are defined by the reaction of the process to a sudden change in the control settings, known as a step input. This might include an immediate change in set temperature, as shown in Figure 5.3.11.

The response of the system is depicted in Figure 5.3.12, which shows a certain amount of dead time before the process temperature starts to increase. This dead time is due to the control lag caused by such things as an electrical actuator moving to its new position. The time constant will differ according to the dynamic response of the system, affected by such things as whether or not the sensor is housed in a pocket.

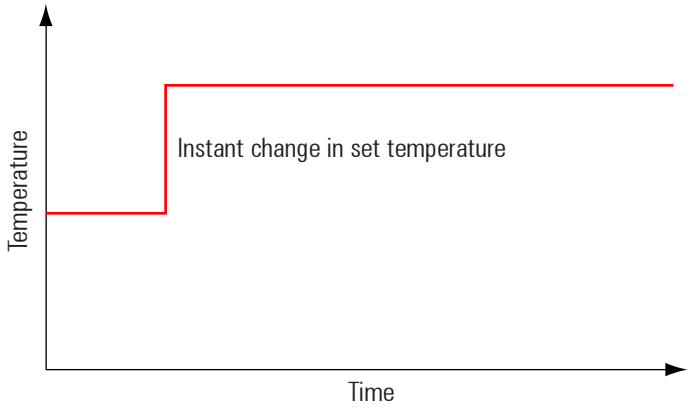


Fig. 5.3.11 Step input

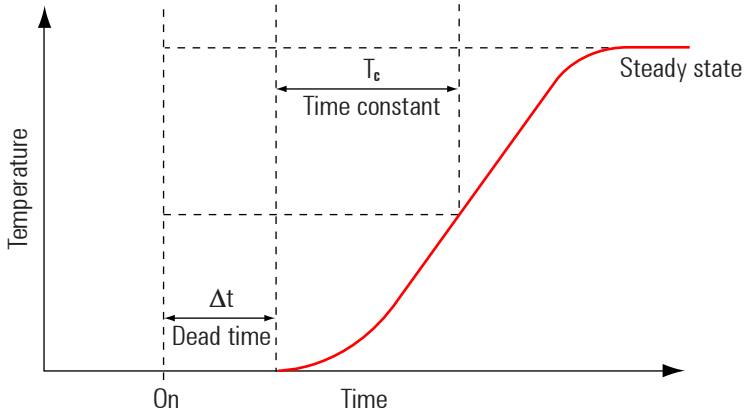


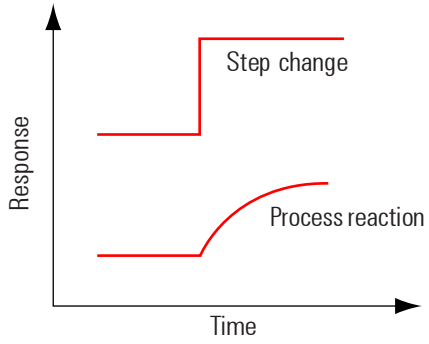
Fig. 5.3.12 Components of process response to step changes

The response of any two processes can have different characteristics because of the system. The effects of dead time and the time constant on the system response to a sudden input change are shown graphically in Figure 5.3.12.

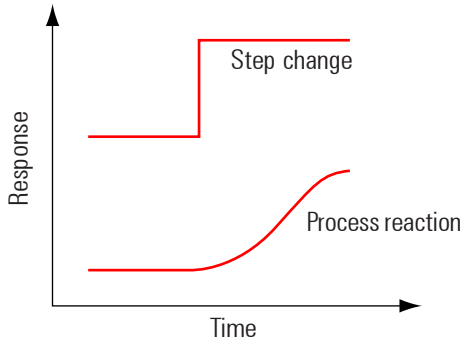
Systems that have a quick initial rate of response to input changes are generally referred to as possessing a first order response.

Systems that have a slow initial rate of response to input changes are generally referred to as possessing a second order response.

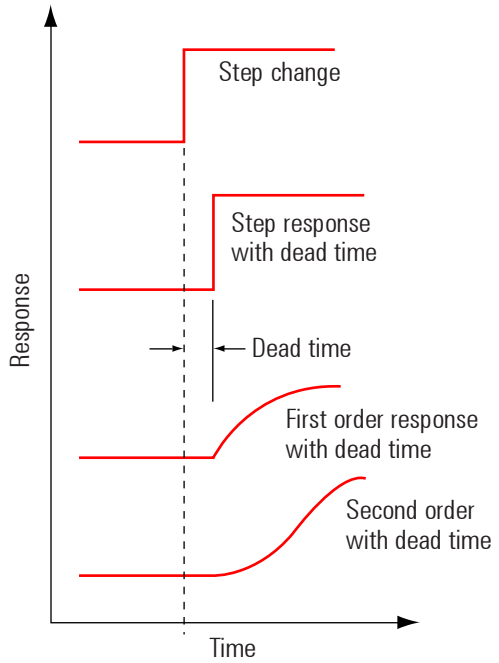
An overview of the basic types of process response (effects of dead time, first order response, and second order response) is shown in Figure 5.3.13.



First order response with no dead time
In basic terms, the rate of response is at a maximum at the start and gradually decreases from that point onwards.



Second order response with no dead time
In basic terms, the maximum rate of response does not occur at the very beginning (when the step change happened) but some time later.



Dead time
The process response may be such that, with any of the types so far discussed, there is no immediate dynamic response at first.

In other words, there is a period of dead time.

In basic terms, if the time constant is greater than the dead time, control should not be difficult. If, however, the dead time is greater than the time constant, satisfactory control may be difficult to achieve.

Fig. 5.3.13 Response curves

Questions

1. What factors affect the response of a process to any input change?

a| P + I + D ☐

b| Time constant and actuator voltage ☐

c| Size of valve and actuator ☐

d| Time constant and dead time ☐

2. What is meant by the term 'time constant'?

a| It is the time for the valve to move from its fully open to fully closed position ☐

b| It is the time for the valve to move 63.2% of its full movement due to a sudden change in process load ☐

c| It is the time taken for a controller output to change by 63.2% of its total due to a sudden change in process load ☐

d| It is the time taken for a controller output to achieve 63.2% of the time required to reach set point ☐

3. What is meant by cascade control?

a| The control of water flowing over a weir ☐

b| Two valves are used to control two independent variables ☐

c| Two independent variables are controlled by one valve ☐

d| Two controllers are used to average the output from one sensor ☐

4. What is meant by feedback control on a steam jacketed vessel?

a| When the controller of the vessel contents feeds back a signal to a controller of the steam temperature in the jacket ☐

b| It is a control in which a sensor in the steam jacket only indirectly controls the temperature of the vessel contents ☐

c| It is another name for a multi-loop control in which one controller loop will maintain the temperature of the vessel contents and another will maintain the steam jacket pressure/temperature ☐

d| It is a closed loop control system in which the condition of the vessel contents is fed back to a controller operating on a valve in the steam supply to the jacket ☐

5. What is the disadvantage of an open loop control system?

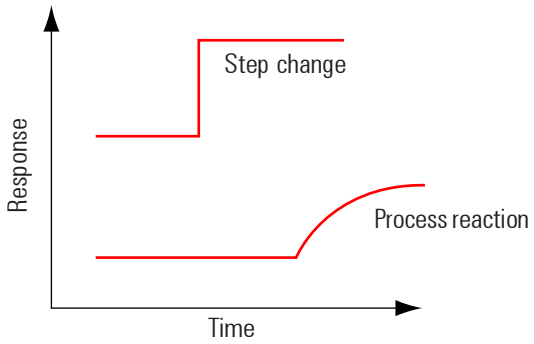
a| Only one variable can be controlled ☐

b| It tends to provide a coarse control as there is no feedback from the plant being heated ☐

c| It is proportional control only ☐

d| It can only be used with a thermostat ☐

6. What can be derived from the process response shown below, in response to a step change signal change?



a| It is a second order response, the maximum response not occurring at the time of the step change but sometime later ☐

b| It indicates the use of an open loop control system ☐

c| There is a significant delay in the whole system responding to a step change and a quick opening valve is being used with a P + D controller ☐

d| It is a first order response following a dead time and the rate of response starts at the maximum and then gradually decreases ☐

Answers

1: d, 2: c, 3: c, 4: d, 5: b, 6: d

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Module 5.4

Choice and Selection of Controls

Choice and Selection of Controls

This Module will concentrate on available automatic control choices and the decisions which must be made before selection. Guidance is offered here rather than a set of rules, because actual decisions will depend upon varying factors; some of which, such as cost, personal preferences and current fashions, cannot be included here.

Application

It is important to reflect on the three basic parameters discussed at the beginning of Module 5.1: Safety, Stability and Accuracy.

In order to select the correct control valve, details of the application and the process itself are required. For example:

- Are any safety features involved? For instance, should the valve fail-open or fail-closed in the event of power failure? Is separate control required for high and low limit?
- What property is to be controlled? For instance, temperature, pressure, level, flow?
- What is the medium and its physical properties. What is the flowrate?
- What is the differential pressure across a control valve across the load range?
- What are the valve materials and end connections?
- What type of process is being controlled? For instance, a heat exchanger used for heating or process purposes?
- For temperature control, is the set point temperature fixed or variable?
- Is the load steady or variable and, if it is variable, what is the time scale for change, fast or slow?
- How critical is the temperature to be maintained?
- Is a single loop or multi-loop control required?
- What other functions (if any) are to be carried out by the control? For instance, normal temperature control of a heating system, but with added frost protection during 'off' periods?
- Is the plant or process in a hazardous area?
- Is the atmosphere or environment corrosive by nature or is the valve to be fitted externally or in a 'dirty' area?
- What motive power is available, such as electricity or compressed air, and at what voltage and pressure?

Motive power

This is the power source to operate the control and drive the valve or other controlled device. This will usually be electricity, or compressed air for a pneumatic system, or a mixture of both for an electropneumatic system. Self-acting control systems require no external form of power to operate; they generate their own power from an enclosed hydraulic or vapour pressure system.

To some extent, the details of the application itself may determine the choice of control power. For example, if the control is in a hazardous area, pneumatic or self-acting controls may be preferable to expensive intrinsically safe or explosion-proof electric/electronic controls.

The following features are listed as a general comment on the various power source options:

Self-acting controls

Advantages:

- Robust, simple, tolerant of 'unfriendly' environments.
- Easy to install and commission.
- Provide proportional control with very high rangeability.
- Controls can be obtained which fail-open or fail-closed in the event of an unacceptable overrun in temperature.
- They are safe in hazardous areas.
- Relatively maintenance free.

Disadvantages:

- Self-acting temperature controls can be relatively slow to react, and Integral and Derivative control functions cannot be provided.
- Data cannot be re-transmitted.

Pneumatic controls

Advantages:

- Robust.
- They operate very quickly, making them suitable for processes where the process variables change rapidly.
- The actuators can provide a high closing or opening force to operate valves against high differential pressures.
- The use of valve positioners will ensure accurate, repeatable control.
- Pure pneumatic controls are inherently safe and actuators provide smooth operation.
- Can be arranged to provide fail-open or fail-closed operation without additional cost or difficulty.

Disadvantages:

- The necessary compressed air system can be expensive to install, if no supply already exists.
- Regular maintenance of the compressed air system may be required.
- Basic control mode is on/off or proportional although combinations of P+I and P+ I +D are available, but usually at greater cost than an equivalent electronic control system.
- Installation and commissioning is straightforward and of a mechanical nature.

Electric controls

Advantages:

- Highly accurate positioning.
- Controllers are available to provide high versatility with on-off or P+I+D combinations of control mode, and multi-function outputs.

Disadvantages:

- Electric valves operate relatively slowly, meaning they are not always suitable for rapidly changing process parameters such as pressure control on loads that change quickly.
- Installation and commissioning involves both electrical and mechanical trades and the cost of wiring and installation of a separate power supply must be taken into account.
- Electric actuators tend to be less smooth than their pneumatic counterparts. Spring return actuators are required for fail open or fail closed functions: This can substantially reduce the closing force available and they usually cost more.
- Intrinsically safe or explosion-proof electric controls are needed for use in hazardous areas; they are an expensive proposition and, as such, a pneumatic or electropneumatic solution may be required, as described below. Special installation techniques are required for these types of hazardous areas.

Electropneumatic controls

Advantages:

- Electropneumatic controls can combine the best features of electronic and pneumatic controls. Such systems can consist of pneumatically actuated valves, electric/electronic controllers, sensors and control systems, plus electropneumatic positioners or converters.

  The combination provides the force and smooth operation of a pneumatic actuator/valve with the speed and accuracy of an electronic control system. Fail-open or fail-closed operation can be provided without cost penalty and, by using suitable barriers and/or confining the electric/electronic part of the control system to 'safe' (non-hazardous) areas, they can be used where intrinsic safety is required.

Disadvantages:

- Electrical and compressed air supplies are required, although this is not normally a problem in industrial processing environments.

There are three important factors to take into account when considering the application and the required power source:

- Changes in load.
- Whether the set value is critical or non-critical.
- Whether the set value has to be varied.

The diagrams in Figure 5.4.1 and 5.4.2 help to explain.

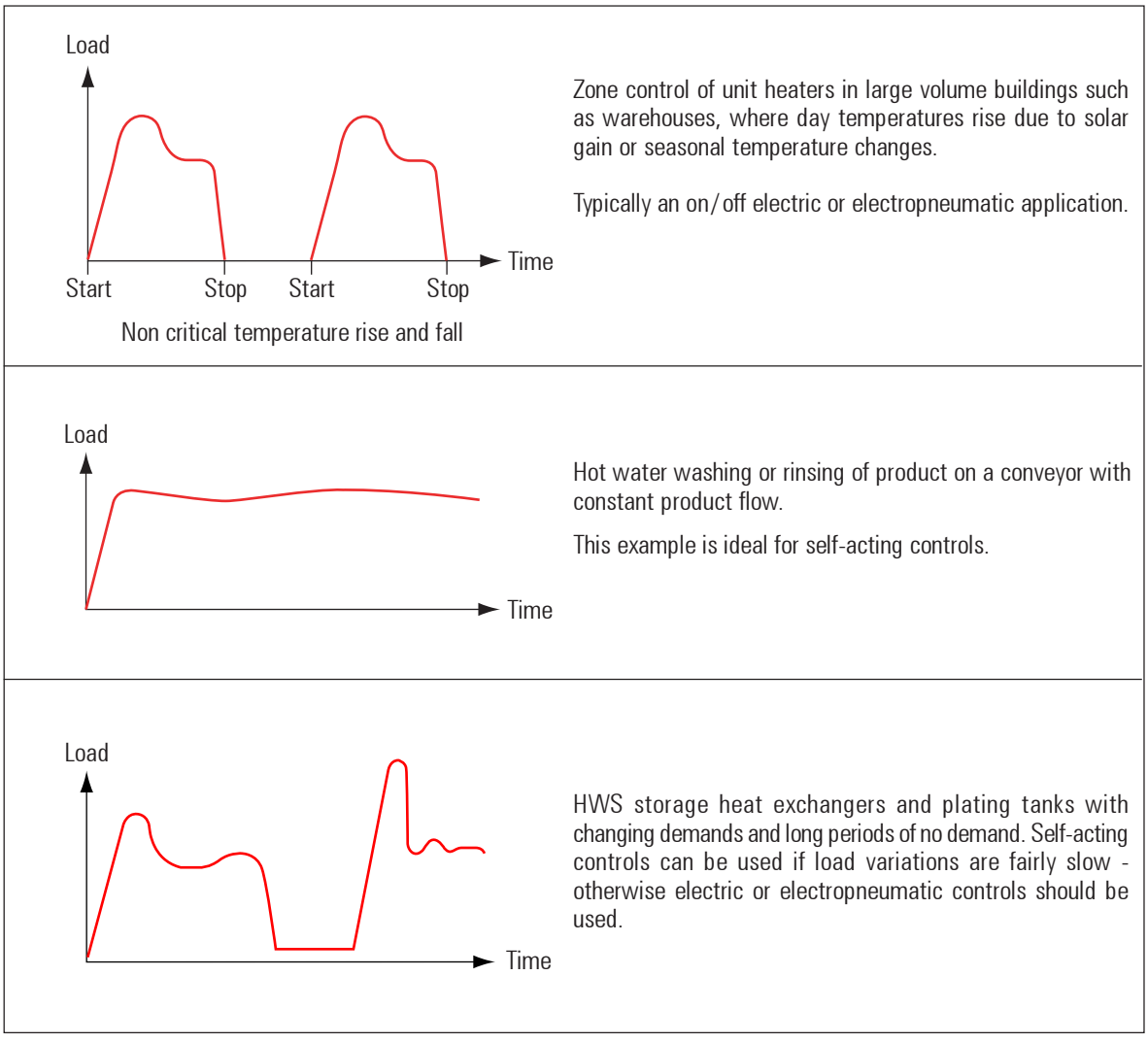
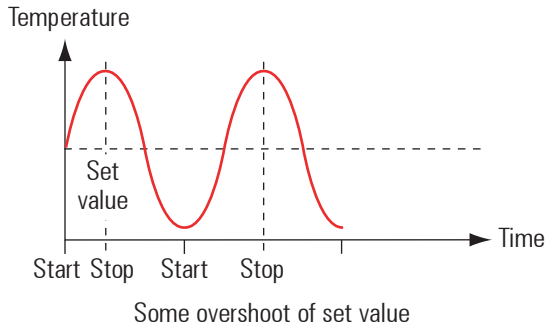
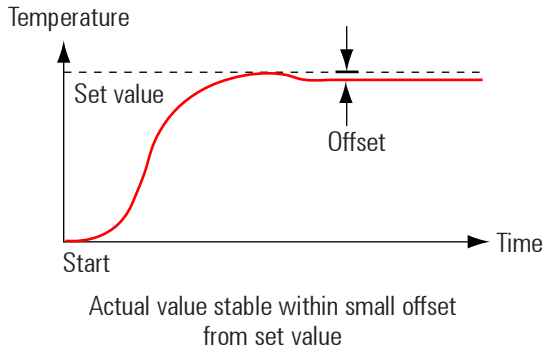


Fig. 5.4.1 Changes in load and time



Non-critical application:
Steam/water heat exchangers where the load is steady, such as jacket cooling or condenser cooling.

Actuation:
Typically electric or electropneumatic actuators used.



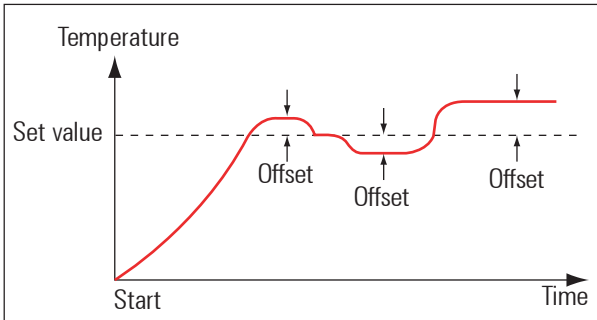
Critical application:
Steam/water heat exchangers for large central heating systems or jacket heating in processes.

Actuation:
Self-acting and pneumatic controls are used if load variations are fairly slow and if reasonable offset can be accepted - otherwise electropneumatic or electric controls should be used.

Fig. 5.4.2 Critical nature of the set value

What type of controls should be installed?

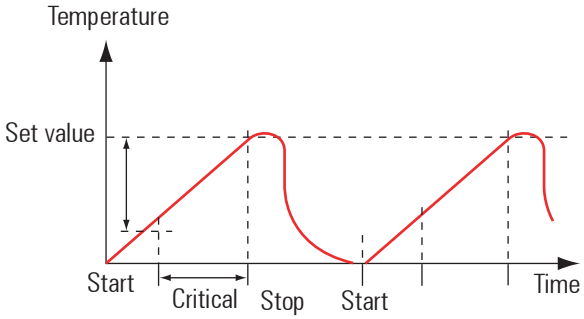
Different applications may require different types of control systems. Self-acting and pneumatic controls can be used if load variations are fairly slow and if offset can be accepted, otherwise electropneumatic or electric controls should be used. Figure 5.4.3 shows some different applications and suggestions on which method of control may be acceptable.



Temperature wants to swing around set value

Applications:
Timber curing
Platen presses
Brick baking
Paint drying

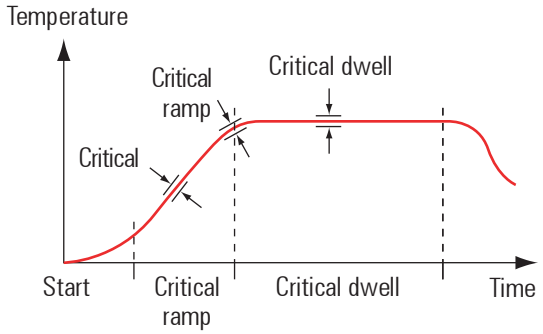
Actuation:
Typically an electric or electropneumatic actuator.



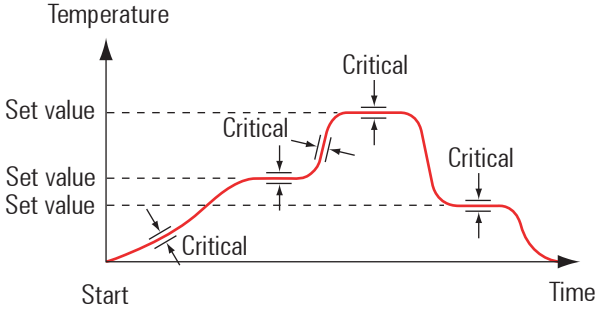
Typical ramp control calling for an accurate time versus temperature rate of rise

Applications:
Textile dyeing
Curing processes
Sterilising
De-frosting food
Paint drying

Actuation:
Electric or pneumatic actuators usually with electronic programmable controllers



In each phase temperature and time must be harmonised and close tolerance is required



Temperature wants to swing around set value

Applications:
Multi-step textile dyeing, sterilising, platen presses, canning and baking.

Actuation:
Electric or pneumatic actuators usually with electronic programmable controllers

Fig. 5.4.3 Variable set value and its critical nature

Types of valves and actuators

The actuator type is determined by the motive power which has been selected: self-acting, electrical, pneumatic or electropneumatic, together with the accuracy of control and actuator speed required.

As far as valve selection is concerned, with steam as the flowing medium, choice is restricted to a two port valve. However, if the medium is water or another liquid, there is a choice of two port or three port valves. Their basic effects on the dynamics of the piping system have already been discussed.

A water application will usually determine whether a three port valve is used to mix or divert liquid flow. If changes in system pressure with two port valves are acceptable, their advantages compared with three port valves include lower cost, simplicity and a less expensive installation. The choice of two port valves may also allow the inherent system pressure change to be used to switch on sequential pumps, or to reduce or increase the pumping rate of a variable speed pump according to the load demand.

When selecting the actual valve, all the factors considered earlier must be taken into account which include; body material, body pressure/temperature limits, connections required and the use of the correct sizing method. It is also necessary to ensure that the selection of valve/actuator combination can operate against the differential pressure experienced at all load states. (Differential pressure in steam systems is generally considered to be the maximum upstream steam absolute pressure. This allows for the possibility of steam at sub-atmospheric pressure on the downstream side of the valve).

Controllers

Safety is always of great importance. In the event of a power failure, should the valve fail-safe in the open or closed position?

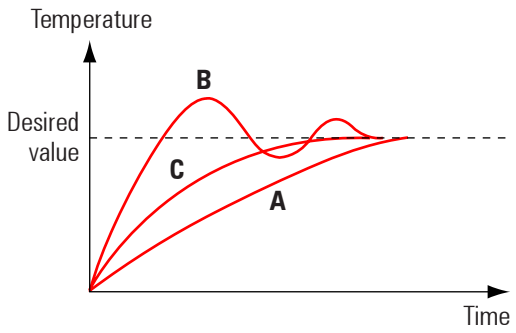
Is the control to be direct-acting (controller output signal rises with increase in measured variable) or reverse-acting (controller output signal falls with increase in measured variable)?

If the application only requires on/off control, a controller may not be needed at all. A two-position actuator may be operated from a switching device such as a relay or a thermostat. Where an application requires versatility, the multi-function ability of an electronic controller is required; perhaps with temperature and time control, multi-loop, multi-input/output.

Having determined that a controller is required, it is necessary to determine which control action is necessary, for instance on/off, P, P I, or P I D.

The choice made depends on the dynamics of the process and the types of response considered earlier, plus the accuracy of control required.

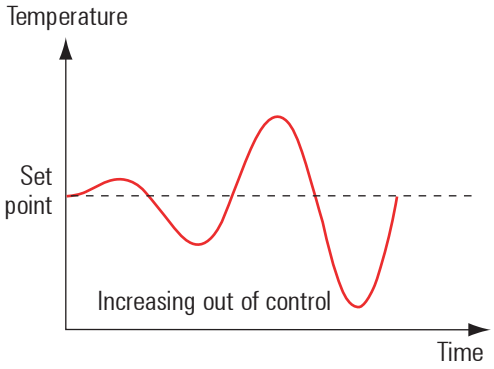
Before going any further, it is useful to define what is meant by ‘good control’. There is no simple answer to this question. Consider the different responses to changes in load as shown in Figure 5.4.4.



If a slow, steady heat up is required, the control provided by **A** would be acceptable.

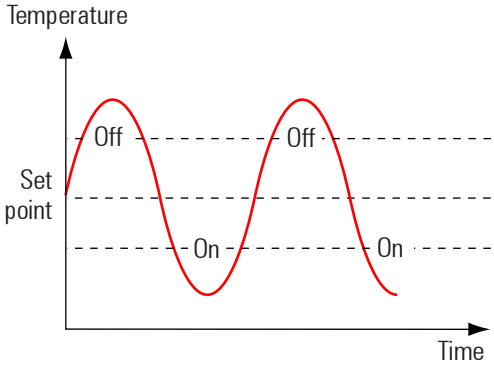
However, if a very rapid heat up is required and overshoot and undershoot of the desired value are acceptable, control **B** would provide the answer.

However, if relatively rapid heat up (in relation to **A**) is needed but no overshoot can be tolerated, then control **C** provides the solution. This shows that the definition of 'good control' will vary from application to application.



One thing that is not generally acceptable is oscillation around the set point or desired value. There may be some applications where oscillation is not a problem but it should usually be avoided. Unstable oscillations such as those shown here cause most concern. Such oscillations are due to one or all of the following:

- Incorrect choice of controller, sensor or actuator, or size of valve.
- Incorrect control settings.
- Incorrect position of sensor creating a long dead time.



Oscillation should not be confused with the response pattern we could expect from an on/off action. This will result in a wave response curve about the desired value, as shown here. When oscillation is mentioned, it is normally with reference to continuous control action.

Fig. 5.4.4 Examples of different responses to changes in load

Self-acting control is normally suitable for applications where there is a very large 'secondary-side' thermal capacity compared to the 'primary-side' capacity.

Consider a hot water storage calorifier as shown in Figure 5.4.5 where the large volume of stored water is heated by a steam coil.

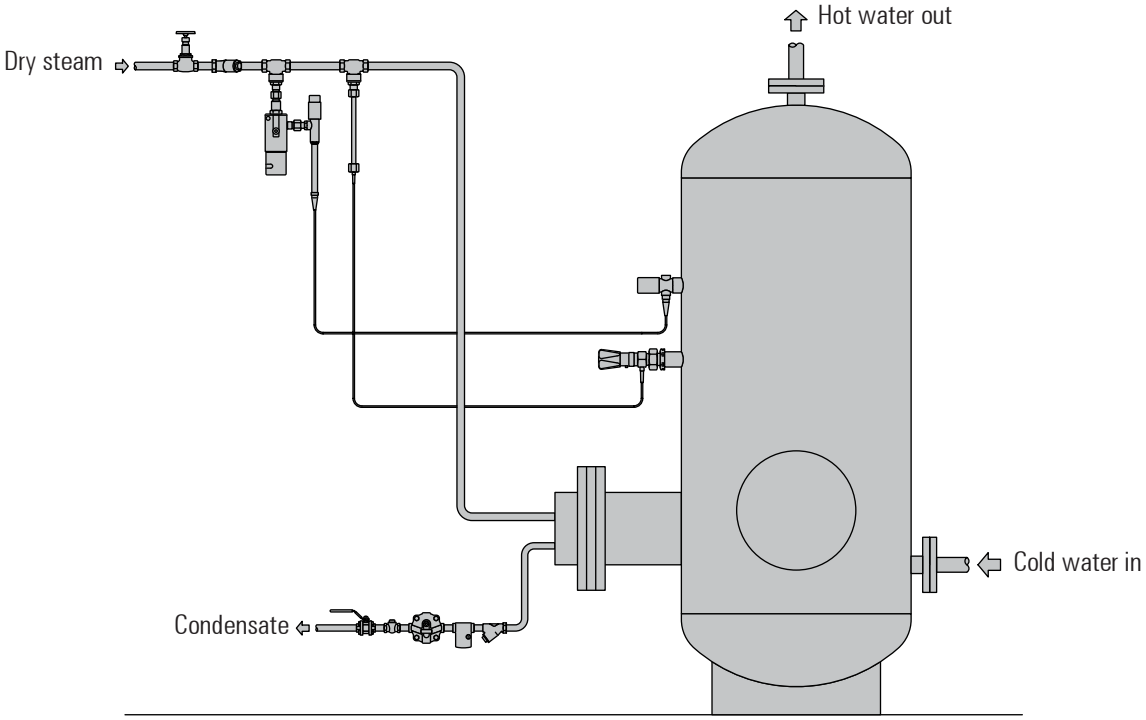


Fig. 5.4.5 Hot water storage calorifier

When the water in the vessel is cold, the valve will be wide open, allowing steam to enter the coil, until the stored water is heated to the desired temperature. When hot water is drawn from the vessel, the cold water which enters the vessel to take its place will reduce the water temperature in the vessel. Self-acting controls will have a relatively large proportional band and as soon as the temperature drops, the valve will start to open. The colder the water, the more open the steam valve.

Figure 5.4.6 shows a non-storage plate type heat exchanger with little thermal storage capacity on either the primary or the secondary side, and with a fast reaction time. If the load changes rapidly, it may not be possible for a self-acting control system to operate successfully. A better solution would be to use a control system that will react quickly to load changes, and provide accuracy at the same time.

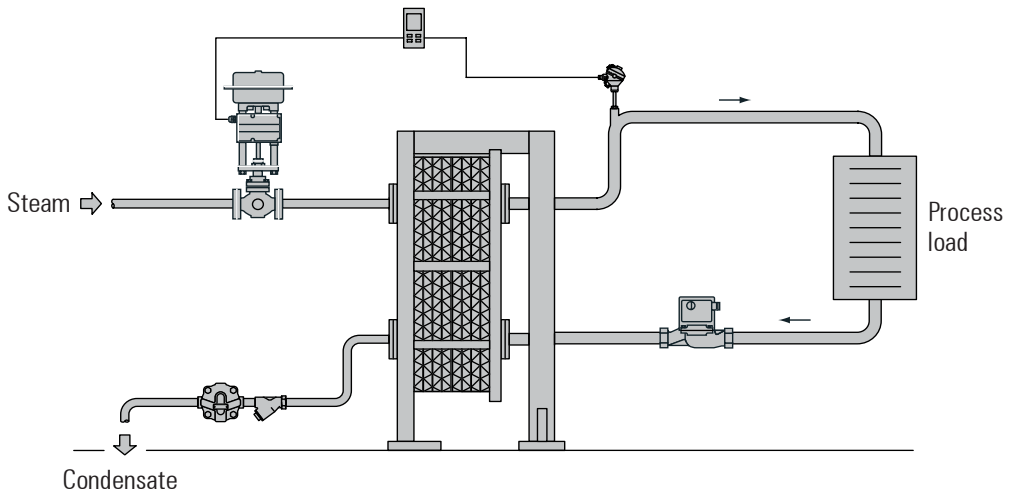


Fig. 5.4.6 Heat exchanger with little storage capacity

Questions

1. What is probably the first consideration when selecting a control system?

a| What degree of accuracy is required? ☐

b| Is the control for heating or cooling? ☐

c| Is a two or three port valve required? ☐

d| In the event of power failure, must the valve fail-open or fail-closed? ☐

2. Which of the following is NOT true of self-acting controls?

a| They are very expensive ☐

b| They are relatively slow to react to process changes ☐

c| Controls can be selected to fail-open or fail-closed in the event of an unacceptable overrun in temperature ☐

d| They are virtually maintenance free and suitable for use in hazardous areas ☐

3. Which of the following is NOT true of an electric control?

a| Controls can be selected to fail-open or fail-closed on power failure ☐

b| They are available with on/off or P I D functions of control mode ☐

c| They can provide multi-function outputs ☐

d| They operate faster than pneumatic controls ☐

**4. A plate heat exchanger uses steam as the primary medium to heat water for a small water ring main serving taps and showers.
Which type of control would be the first choice, and why?**

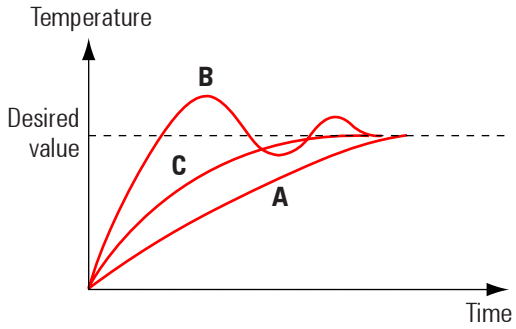
a| Self-acting because they are easy to commission, the relatively low speed of operation will match the slow changes in temperature of the water system; and very accurate control of temperature is not critical, so offset would be acceptable ☐

b| An electric control because PID functions can be adjusted to suit the system response, they give very accurate control and they are very fast acting which will suit the response of the heat exchanger ☐

c| A pneumatic control, because they are very fast acting so will suit the response of the heat exchanger, no expensive electrics are required, the sensor is small so can be easily accommodated in the water flow pipework and they can be arranged to fail-open or fail-closed in the event of loss of power ☐

d| An electropneumatic system because, the electronic controller will provide speed of operation to meet the fast response of the heat exchanger and accuracy of control, PID functions can be set to provide effective control, the control can be arranged to fail-open or fail-closed in the event of loss of power, the sensor is small and the controller can activate alarms. ☐

5. The figure below shows three responses to a sudden switch on from cold. If the plant requires a relatively fast heat-up with no overshoot, which response would be recommended?



a| A ☐

b| B ☐

c| C ☐

d| None, any control providing a fast heat-up will result in some overshoot ☐

6. Steam is supplied to a plate heat exchanger heating an acidic metal treatment solution for a large tank into which cold components are dipped. There is a possibility that the solution could be splashed over the control. What would be your recommended control and why?

a| On/off because it is simple and inexpensive ☐

b| An electropneumatic control because accurate control will be maintained, there will be no fear of a high limit control shutting off the steam due to a temperature overshoot, the control settings can be adjusted to suit the system, the rate of heat up can be programmed, alarms can be incorporated if required ☐

c| Self-acting control because it is simple, inexpensive, easy to commission, overshoot and undershoot can be accepted, no external power source is required, and the equipment will tolerate a degree of splashing with chemicals ☐

d| Pneumatic control because it provides accurate repeatable control, the equipment is inherently protected from splashing, different control modes are available, commissioning is straightforward, it can be arranged to fail-closed in the event of air failure, and speed of response is not important in this application ☐

Answers

1: d, 2: a, 3: d, 4: d, 5: c, 6: c

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Module 5.5

Installation and Commissioning of Controls

Installation and Commissioning of Controls

Installation

Valves

Before installing a control valve it is necessary to ensure that the size, pressure rating, materials and end connections are all suitable for the conditions under which the valve is expected to work.

All reputable manufacturers of automatic control equipment will provide detailed instructions covering the correct installation procedure for their equipment. Data will also be provided on how to set up the equipment, plus any routine and regular maintenance to be undertaken. In most cases, the manufacturer will also offer an on-site commissioning service. In some cases, a regular after-sales maintenance contract can be agreed. Module 5.5 covers the major points to be considered before installation.

Piping upstream and downstream of the control valve should be clear and unobstructed. The correct operation of a valve will be impaired if it is subject to line distortion stresses. It is important to ensure that all flanged joints are square and true and that pipework is adequately supported. Control valves should generally be installed in horizontal pipelines with the spindles vertical.

Pipework systems will often be subjected to pressure testing prior to use. This test may be carried out at a pressure above the normal working conditions. It is necessary to ensure that the control valve and its internals are designed to withstand this higher test pressure.

Control valves are essentially instruments and will be damaged if dirt or other abrasive or obstructive materials are allowed to enter them. It is essential in most applications to prevent this by fitting pipeline strainers upstream of any control valve.

Valves must also be accessible for routine maintenance, such as re-packing of glands and the replacement of internals. To facilitate this sort of work, isolating valves of a full bore pattern either side of the valve will keep plant downtime to a minimum while the work is carried out.

If a plant must be kept in operation at all times, even when a control valve is being inspected or maintained, it may be necessary to fit a valved bypass. However, the valve used in the bypass must be of good quality and should either be a characterised throttling valve or another control valve of the correct K_{VS}. Any leakage through it during normal operation will affect the action of the control system. It is not recommended that manual bypasses be fitted under any circumstances. The control valve must be installed to ensure the correct direction of flow of the medium passing through the valve. Usually a 'direction of flow' arrow is cast into the body of the control valve. The valve must have a suitable flow capacity and incur an acceptable pressure drop.

In steam lines, it is important to provide a steam separator and/or a trapping point upstream of the valve, as shown in Figure 5.5.1. This will prevent the carryover of condensate through the control valve, which would otherwise reduce its service life. This drain point is also important if the control valve is likely to remain closed for any length of time. If a condensate drain is not fitted, waterhammer and potentially serious damage can result when the valve opens. The provision of a steam separator and strainer ensures good steam conditioning.

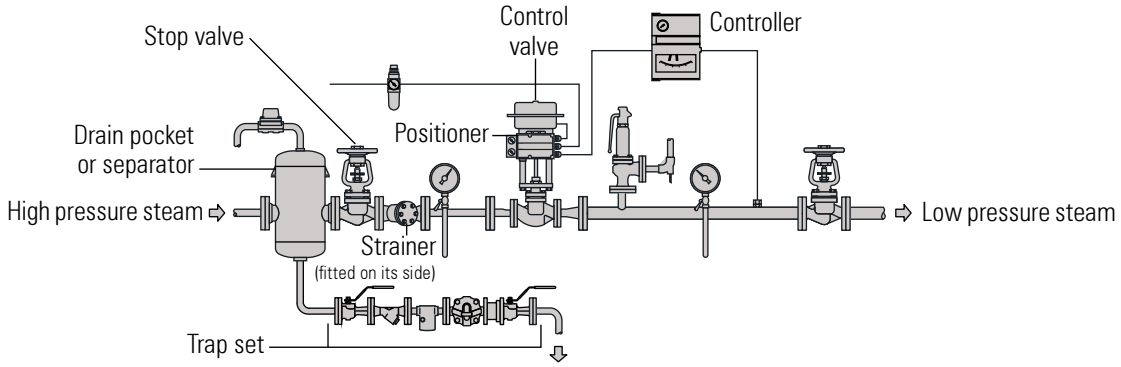


Fig. 5.5.1 A pneumatic pressure reducing station with steam conditioning

Actuators/sensors

Again, the manufacturer's instructions must be observed. Actuators are normally mounted vertically above the control valve, although different arrangements may be recommended if an electric actuator is mounted to a valve handling a high temperature medium, such as steam.

Generally, actuators should be located away from conditions such as excess heat, high humidity or corrosive fumes. These are likely to cause premature failure in components such as diaphragms or electric/electronic items. Manufacturers should state the recommended maximum ambient temperature conditions for their equipment. With some electric actuators, if condensation is likely to occur within the actuator, models with a built-in heater are available. Where such conditions cannot be avoided, actuators should be purchased which are suited to the installed conditions.

Enclosures for actuators, positioners, and so on, will usually carry an enclosure rating conforming to a national electrical code. This should specify the degree of immunity of the box to the ingress of dust and water. It is worthless using an electric actuator whose enclosure has a low rating to the ingress of water, if it is likely to be hosed down!

Care must be taken to ensure that sensors are fully and correctly immersed if they are to carry out their sensing function effectively. The use of pockets will enable inspection or replacement to take place without the need to drain the piping system, vessel or process plant. In contrast, pockets will delay response times. The use of heat conducting paste in the pocket will minimise any delay in response.

Power and signal lines

With a pneumatic system, compressed air and pneumatic signal lines must be dry, free from oil and dirt, and leak tight. Locating the pneumatic controller near the valve and actuator will minimize any delay due to the capacity and resistance of the signal line.

Usually, the valve, actuator and any positioners or converters, will be supplied as a complete pre-assembled unit. If they are not, the actuator will need to be mounted to the valve, and the positioner (for a pneumatic control) to the actuator. The assembly will then have to be set up properly, to ensure that the correct valve stroke, etc. is achieved, all in accordance with the manufacturer's instructions.

Electrical wiring for electric/electronic and electropneumatic controls

All too often, many apparent 'control' problems are traced back to incorrect wiring. To quote an obvious problem encountered as an extreme example, connecting a 110 V supply to a 24 V rated motor, will result in damage! Care must be taken with the wiring system, in accordance with the manufacturer's instructions, and subject to any local regulations.

'Noise' or electrical interference in electrical systems is often encountered, resulting in operational problems which are difficult to diagnose. The use of screened cable, separately earthed conduit or a self-acting or analogue controller may be necessary. Cables should be protected from mechanical damage.

Controllers

As mentioned earlier, the application will generally produce changes that are slower than the response time of the control system. This is why the parameters of the controller, the proportional band or gain, integral time and derivative time, must be tuned to suit each specific application/task.

There are a number of methods for adjusting controller parameters, most of which involve the use of mathematics. The behaviour of a control loop can be predicted mathematically but the process or application characteristics are usually determined by empirical measurement, which can be difficult. Methods based on design heat transfer ratios can be found, but these are outside the scope of this Module.

Before setting the control parameters, it is useful to review each of the control terms (P, I and D), and the three options regarding settings, for instance, too wide, too narrow, and correct.

P-band (Figure 5.5.2)

If P-band is too wide, large offset occurs but system is very stable (curve **A**).
Narrowing the P-band will reduce the offset.
Too narrow a P-band will cause instability and oscillation, (curve **B**).
The optimum P-band, curve **C**, is achieved at a setting just slightly wider than that causing permanent oscillation.

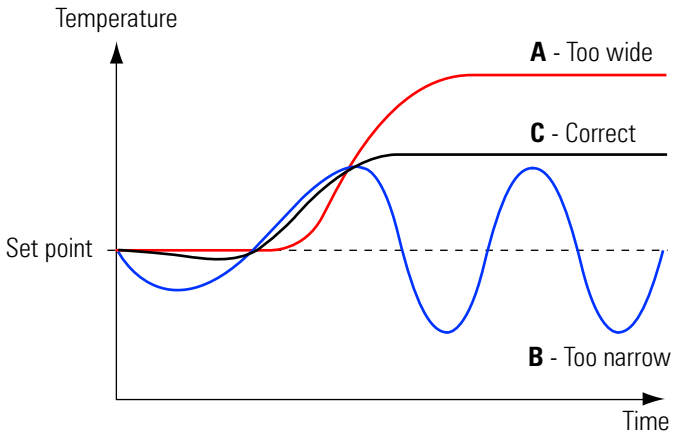


Fig. 5.5.2 P-band setting reaction to change in load

Summary of P-band (proportional action)		
Correct P-band =	Good stability, good response	Some offset
Larger P-band =	Better stability, slower response	Larger offset
Smaller P-band =	Instability, quicker response	Smaller offset with oscillation

Integral action (Figure 5.5.3)

With too short an integral time, temperature (curve **A**) will cross the set point and some oscillation will occur.
An excessive integral time will result in the temperature taking too long to return to set point (curve **B**).
Curve **C** shows a correct integral time setting where the temperature returns to set point as rapidly as possible without any overshoot or oscillation.

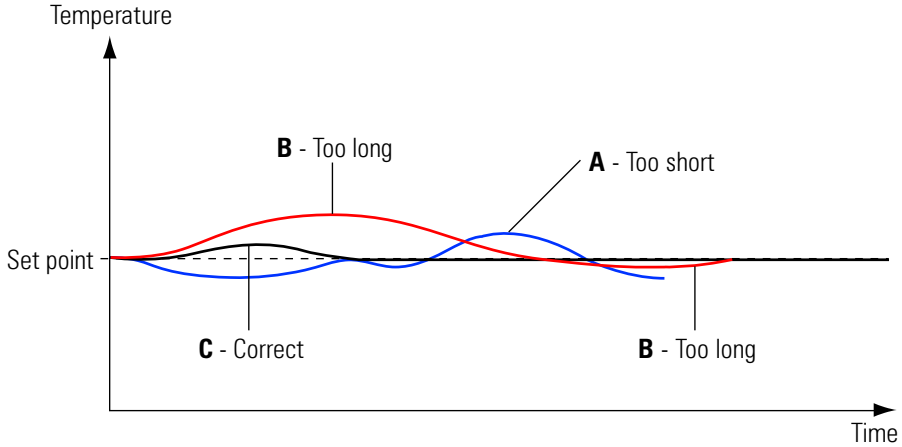


Fig. 5.5.3 Integral time reaction to change in load

Summary of integral action		
Correct IAT =	Elimination of offset	Stable - no overshoot
Too short IAT =	Elimination of offset	Response too fast, causing instability and overshoot
Too long IAT =	Elimination of offset	Slow response, stable, no overshoot

Derivative action (Figure 5.5.4)

An excessive derivative time will cause an over-rapid change in temperature, overshoot and oscillation (curve **B**).
Too short a derivative time allows the temperature to deviate from the set point for too long (curve **A**).
The optimum setting returns the temperature to the set point as quickly as possible and is consistent with good stability (curve **C**).

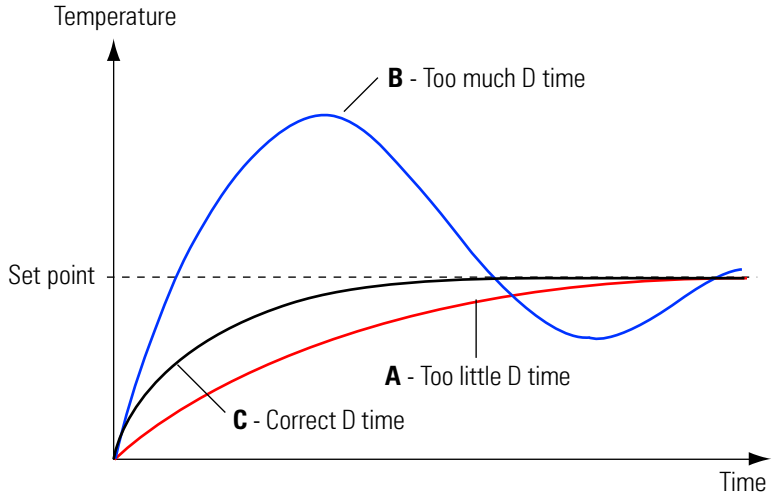


Fig. 5.5.4 Derivative time reaction to change in load

	Summary of derivative action
Correct derivative time =	Quick response, stable
Too much D time =	Faster response leading to overshoot and instability
Too little D time =	Slower response

Commissioning

Practical methods of setting up a controller

Each controller has to be set up individually to match the characteristics of a particular system. Although there are a number of different techniques by which stable and fast control can be achieved, the Ziegler-Nicholls method has proven to be very effective.

The Ziegler-Nicholls method

The Ziegler-Nicholls frequency response method (sometimes called the critical oscillation method) is very effective in establishing controller settings for the actual load. The method uses the controller as an amplifier to reach the point of instability. At this point the whole system is operating in such a way that the temperature is fluctuating around the set point with a constant amplitude, (see Figure 5.5.5). A small increase in gain, or a reduced proportional band, will make the system unstable, and the control valve will start hunting with increasing amplitude.

Conversely, an increased proportional band will make the process more stable and the amplitude will successively be reduced. At the point of instability, the system characteristic is obtained for the actual operating conditions, including the heat exchanger, control valve, actuator, piping, and temperature sensor.

The controller settings can be determined via the Ziegler-Nicholls method by reading the time period (T_n), of the temperature cycles; and the actual proportional band setting at the point of instability.

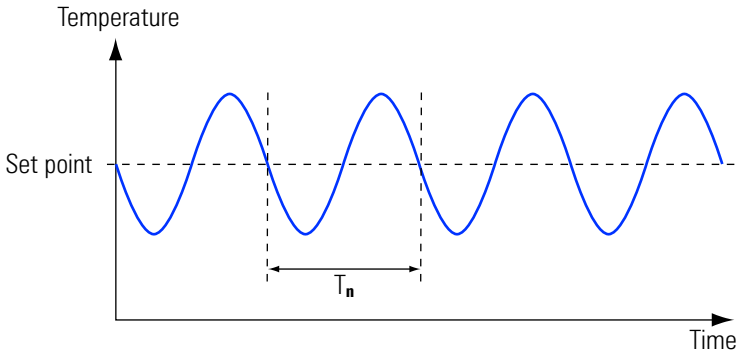


Fig. 5.5.5 Instability caused by increasing the controller gain, with no I or D action

The procedure for selecting the settings for PID parameters, using the Ziegler-Nicholls method, is as follows:

1. Remove integral action on the controller by increasing the integral time (T_i) to its maximum.
2. Remove the controller's derivative action by setting the derivation time (T_D) to 0.
3. Wait until the process reaches a stable condition.
4. Reduce the proportional band (increase gain) until the instability point is reached.
5. Measure the time for one period (T_n) and register the actual P-band (proportional band) setting on the controller at this point.
6. Using this setting as the start point, calculate the appropriate controller settings according to the values in Figure 5.5.6.

	Proportional band	Integral time	Derivative time
P I D control	P-band x 1.7	$T_n/2$	$T_n/8$
P I control	P-band x 2.2	$T_n/1.2$	
P control	P-band x 2.0		

Fig. 5.5.6 Ziegler-Nicholls calculation

The controller settings may be adjusted further to increase stability or response. The impact of changing the setting of the PID parameters on stability, and the response of the control, is shown in Figure 5.5.7.

	Stability	Response
Increase P Band	Increased	Slower
Increase T_i	Increased	Slower
Increase T_D	Decreased	Faster

Fig. 5.5.7 Effect of changing PID settings

Bumpless transfer

The technical specifications for controllers include many other terms and one that is frequently encountered is 'bumpless transfer'.

Most controllers incorporate a 'Manual' – 'Auto' switch and there can be times when certain control situations require manual control. This makes interruption of the automatic control loop necessary. Without bumpless transfer, the transfer from Auto to Manual and vice versa would mean that the control levels would be lost, unless the manual output were matched to the automatic output.

Bumpless transfer ensures that the outputs - either Manual to Auto or Auto to Manual - match, and it is only necessary to move the switch as appropriate.

Self-tuning controllers

Contemporary microprocessors provide the ability for some functions, which previously required a computer, to be packed into the confined space of a controller. Amongst these, was the ability to 'self-tune'. Controllers that no longer require a commissioning engineer to go through the process of setting the P I D terms have been available for many years. The self-tune controller switches to on/off control for a certain period of time. During this period it analyses the results of its responses, and calculates and sets its own P I D terms.

It used to be the case that the self-tune function could only apply itself during system start-up; once set by the controller, the P I D terms remained constant, regardless of any later changes in the process.

The modern controller can now operate what is termed an adaptive function, which not only sets the required initial P I D terms, but monitors and re-sets these terms if necessary, according to changes in the process during normal running conditions.

Such controllers are readily available and relatively inexpensive. Their use is becoming increasingly widespread, even for relatively unsophisticated control tasks.

Questions

1. A pneumatically actuated pressure control is fitted on the steam supply line to an air heater battery, which runs for about 5 minutes every 30 minutes. Each time the valve opens, a banging noise in the pipework occurs and the life of the valve is shortened. What might be the first thing to investigate?

a| There may be no strainer before the control valve ☐

b| The valve is fitted with the flow arrow pointing in the wrong direction ☐

c| Unsuitable PID values may have been used ☐

d| There may be no separator or steam trap set before the control valve ☐

2. A replacement sensor and pocket is installed to work with an electronic controller. The response of the system is now slower than with the original sensor. What might be the first thing to investigate?

a| The controller may not have been reconfigured when the replacement sensor was fitted ☐

b| The air space around the sensor may not have been filled with a heat conductor ☐

c| The sensor may have been fitted upside-down ☐

d| The replacement signal wiring between the sensor and controller may now be a lot longer ☐

3. On a controller with adjustable P-band, the optimum P-band is achieved at a setting:?

a| With no offset ☐

b| When the oscillation around the set point is regular ☐

c| Not more than 5% ☐

d| Just slightly wider than that which will cause oscillation ☐

4. What is the correct integral action time (IAT)?

a| Where the process returns to the set point as rapidly as possible, without any overshoot or oscillation ☐

b| Where the process temperature returns as rapidly as possible to the set point, ignoring oscillation at this stage of the setting up process ☐

c| Where the offset is 0.5 x the proportional band ☐

d| When the actual temperature oscillates equally around the set temperature ☐

5. What is the correct derivative time setting?

a| P-band x 0.85 ☐

b| The time taken for the temperature overshoot to return to the set point as quickly as possible, consistent with good stability ☐

c| The time taken for the temperature overshoot to return to the set point as quickly as possible with even periodic oscillation times ☐

d| As long as possible in order to bring the temperature overshoot as quickly as possible back to the set point. Any oscillations can be minimised by subsequent adjustments to P and I ☐

6. What is an adaptive controller?

a | A controller which 'self-tunes', thus avoiding manual commissioning ☐

b | A controller which calculates and displays the most suitable PID terms for the process which can then be programmed into the controller ☐

c | A controller which automatically sets the required initial PID terms, but resets them if necessary according to changes in the process system or changing application situations ☐

d | A controller which automatically sets the required PID terms, but then intermittently shuts itself off to save energy when no change in load has been detected for a certain time ☐

Answers

1: d, 2: b, 3: d, 4: a, 5: b, 6: c

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Module 5.6

Computers in Control

Computers in Control

It may be appropriate to end Block 5 with a broad look at the involvement of computers in control systems.

A dictionary definition of the term 'computer' is 'a programmable electronic device that can store, retrieve, and process data'.

This definition includes the basic, single- and multi-loop controllers commonly found in process industries where a condition is read by a sensor, compared to a set point in the controller via some mathematical routines performed to determine the corrective action required, followed by an output of an appropriate signal.

The development rate of the computer chip and its impact on all aspects of life is well known. The rate of advancement in controls technology surely means that some of the following comments will be redundant when read.

History

Stand-alone, single loop controllers date back to pneumatic controllers, which, through the ingenious use of flaps and nozzles, could approximate the basic PID functions. These complex and expensive controllers were often found in large petrochemical plants where accurate control of the process, as well as intrinsic safety (the absence of sparks which could initiate a fire) was essential.

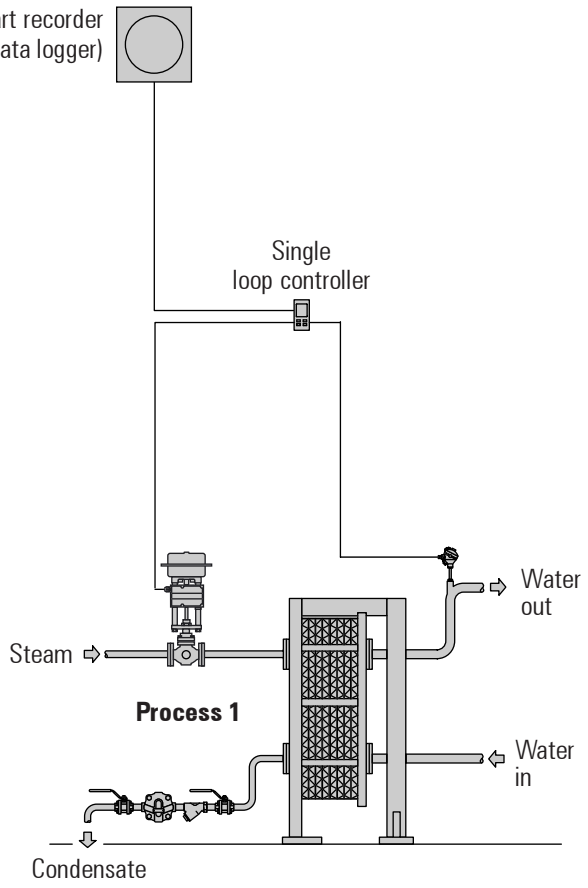


Fig. 5.6.1 Single loop controller with chart recorder

Often, these processes were individually connected to local circular chart recorders (Figure 5.6.1); alternatively, a number of processes were connected to multi-pen recorders in control rooms (Figure 5.6.2). While the multi-pen recorders enabled a number of parameters to be reviewed together, the mechanisms in the instrument and the number of lines on one chart effectively limited their use to approximately twelve inputs.

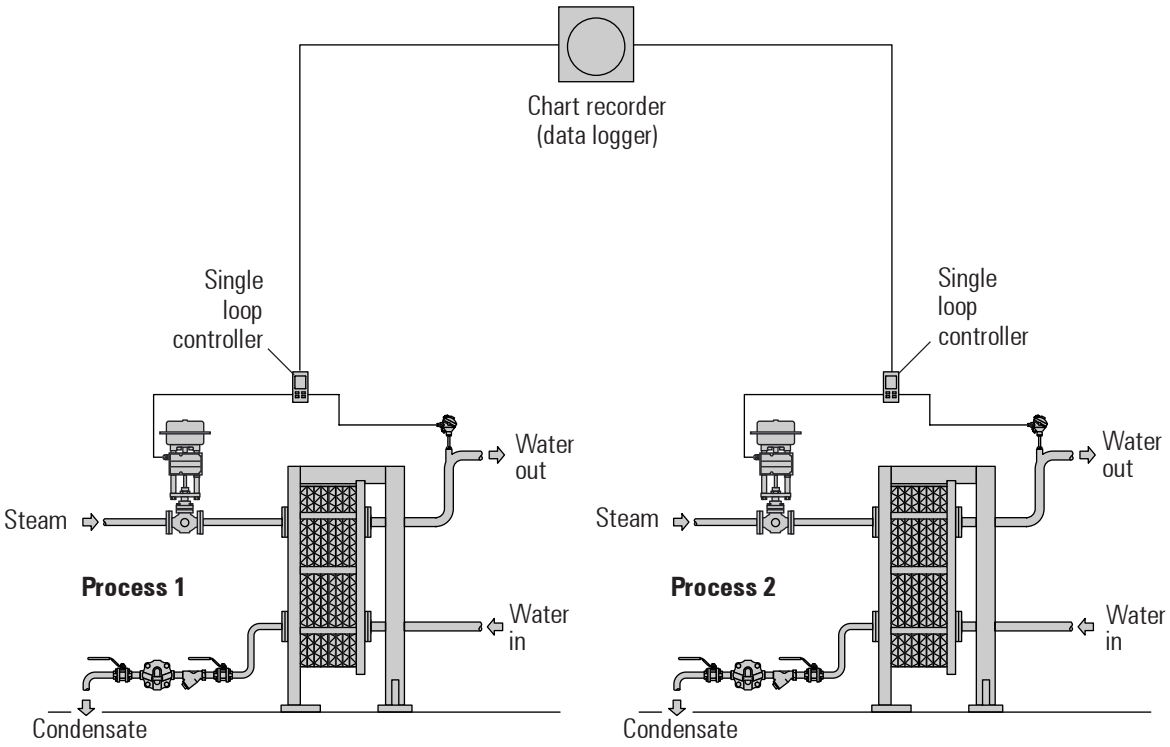


Fig. 5.6.2 Single loop controller with chart recorder

The first computers used in control systems replaced the main control room chart recorders. They gathered information (or data) from a much greater number of points around the plant. They were generally referred to as 'data loggers' (Figure 5.6.3), and had no input to the plant operation.

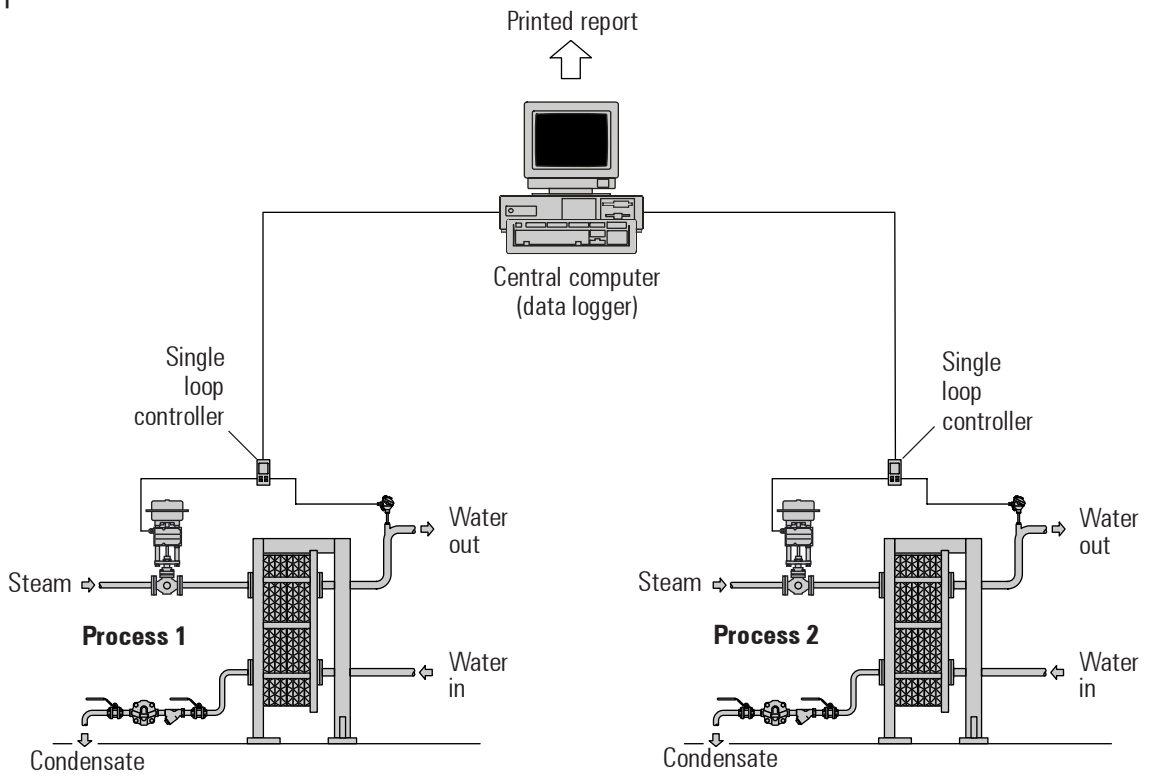


Fig. 5.6.3 A number of single loop controllers with a central data logging computer

These early computers were usually programmed to print out reports at specific time intervals on continuous computer listing paper. By manually extracting the data from the computer print-outs, the plant manager was able to review the operation of his plant as a whole, comparing the performance of different parts of the plant, looking for deterioration in performance, which would indicate the need for a shutdown, etc.

In the mid 1970's, a number of well-known instrument companies began marketing digital control systems. These systems utilised a central computer unit, which took inputs from sensors, performed mathematical routines, and provided an output to various relevant controlling devices. They also maintained a record of events for review (see Figure 5.6.4).

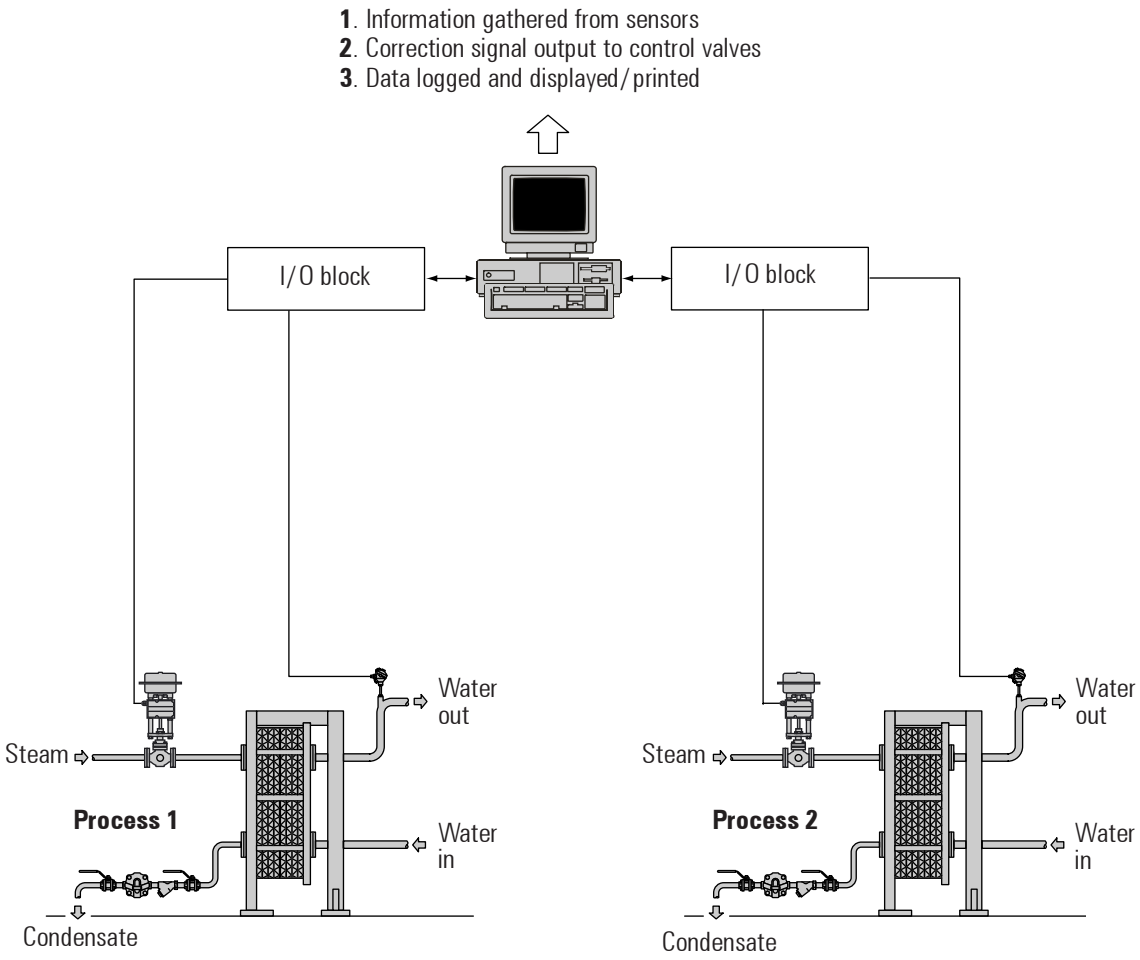


Fig. 5.6.4 A central computer gathering data and controlling the plant

Important notes:

- A personal computer (PC) cannot accept the raw instrument signals (4 - 20 mA, 0 - 10 V) from a control device. An Input/Output (I/O) device was required to 'translate' between the two. Each of the I/O manufacturers had a unique means of achieving this, which meant that the systems were not quite as compatible as had been intended.
- In the beginning, the I/O devices were in the plant's main control room, and each individual piece of equipment was connected to the main control room by its own individual signal cable. This meant that on a large plant, the cable installation and management was an important issue, in terms of its physical volume and corresponding cost.
- As technology progressed, the I/O device moved out to the plant, and the amount of cabling to the control room was reduced, but was still significant.

These Digital Control Systems led to the development of:

- Distributed Control Systems (DCS)
- Supervisory Control And Data Acquisition (SCADA) systems, and
- Building Management Systems (BMS)

. . . all of which are in prolific use today (see Figure 5.6.5).

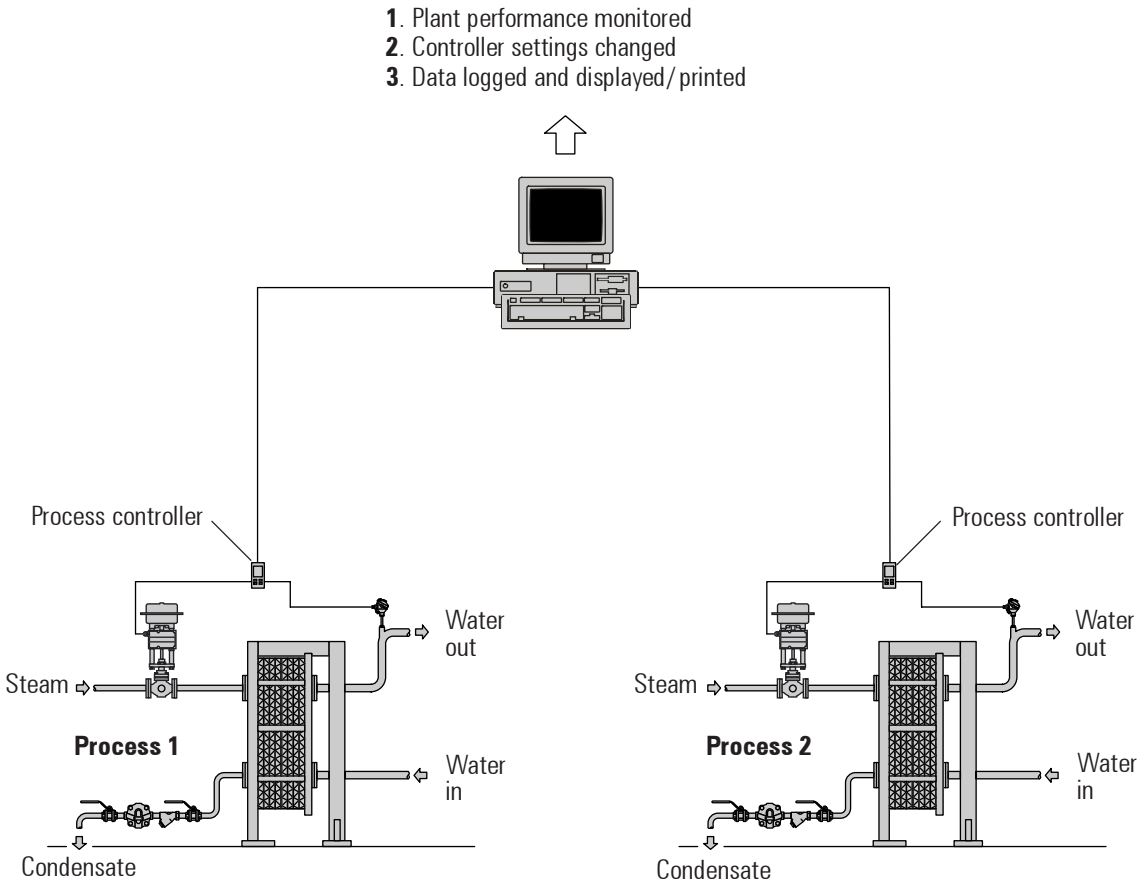


Fig. 5.6.5 A distributed control system

A giant leap forward occurred in the late 1980's with the introduction of the PC and the Windows screen environment and computer operating system. This provided a standard platform for the earlier digital control systems, as all the instrument companies needed to work in a common format. The advantage of the 'Windows' based systems was that information was exchangeable in the same way that today's personal computer user can freely exchange data between Word, 'Excel' and 'PowerPoint'. This data exchange 'language' was termed Dynamic Data Exchange (DDE), and subsequently developed into Object Linking and Embedding (OLE). This was further modified for process control to become OLE for Process Control (OPC), which is still used at the time of writing.

The use of PCs also meant that the options for viewing history were considerably easier. Instead of being confined to print-outs and manual transfer data, the plant manager could use powerful graphing programs, analyse trends, add colours, adjust scales and use symbols; different variables could be plotted against each other, and the performance of different plants compared.

Modern automation systems utilise the computer as a 'Window' on the process. The operator uses the computer to monitor what is happening on the plant as a whole, and revise set-points and control parameters, such as PID, of individual plant based controllers, thus leaving the individual controllers to run the PID algorithms and control logic.

Consequently stand-alone controllers still have a place in modern automation systems as they are in final control, but the controller usually takes the form of a PLC (Programmable Logic Controller) or a multi-loop rack mounted device. These are quite different in appearance to single loop PID controllers. Rather than an operator using a keypad to change the set point and other control parameters at the controller, they are changed by an operator at a computer, which electronically downloads the required parameter to the controller. In the event of a central computer failure, the stand-alone controller would continue with its current parameters or go to a safe condition, thus ensuring that the plant continued to operate safely.

The next major step forward was a system known as 'Fieldbus'.

Fieldbus uses a single digital cable system, which connects every item (see Figure 5.6.6).

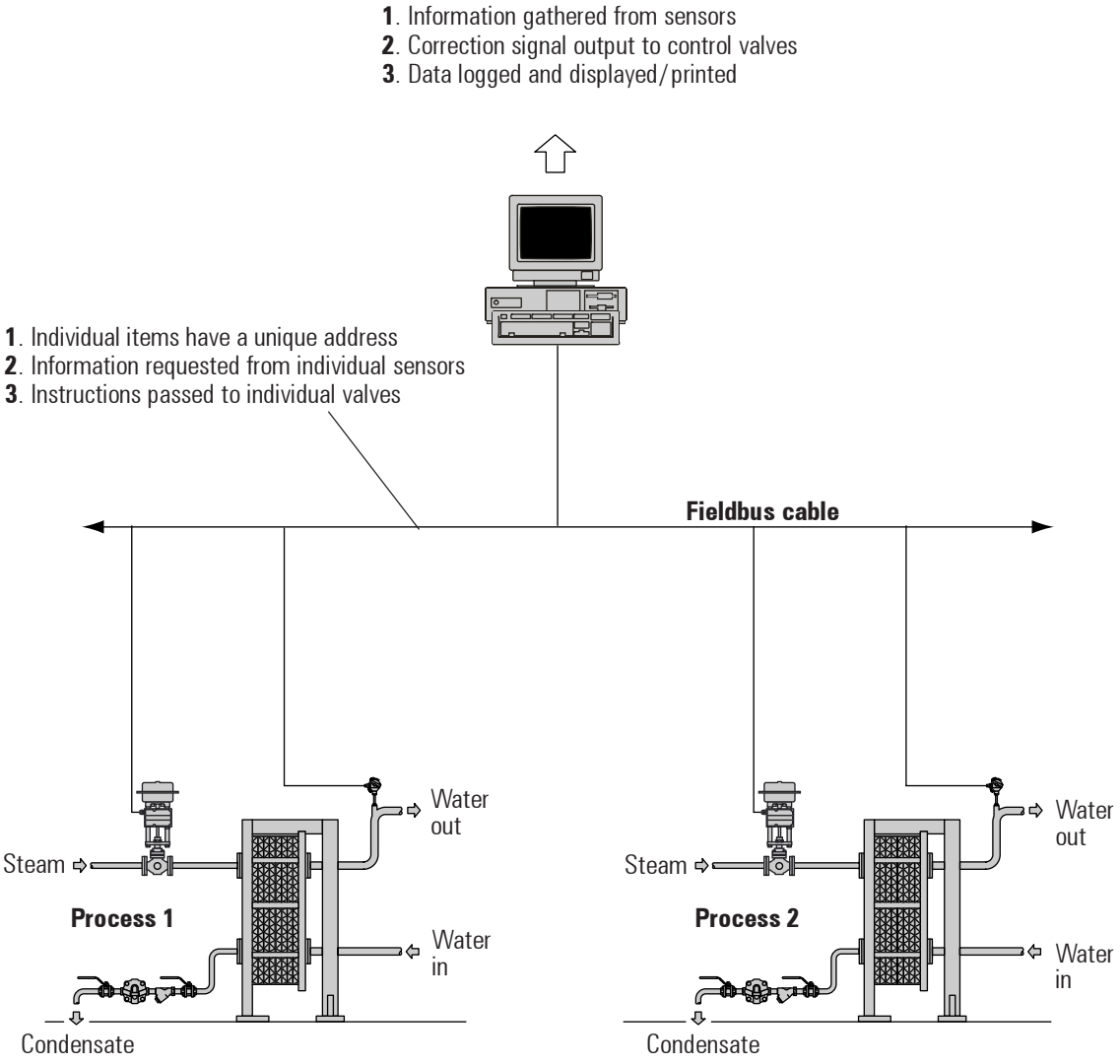


Fig. 5.6.6 A central computer with Fieldbus accepts information and transmits correction signals via Fieldbus

Each item (sensor, controller and controlled device) is given a unique address, which is used to either request information (perhaps from a sensor) or to take some action (perhaps close a control valve).

However, these systems are complex and can be expensive. A Fieldbus network needs a master controller to organise the communications and control logic on the Fieldbus. It also needs a way of interfacing the Fieldbus to computer networks so information can be shared (see Figure 5.6.8). A device that combines the role of Fieldbus controller and provides the bridge to a PC network is called a 'bridge' or 'master controller', (see Figure 5.6.7).



Fig. 5.6.7 A bridge

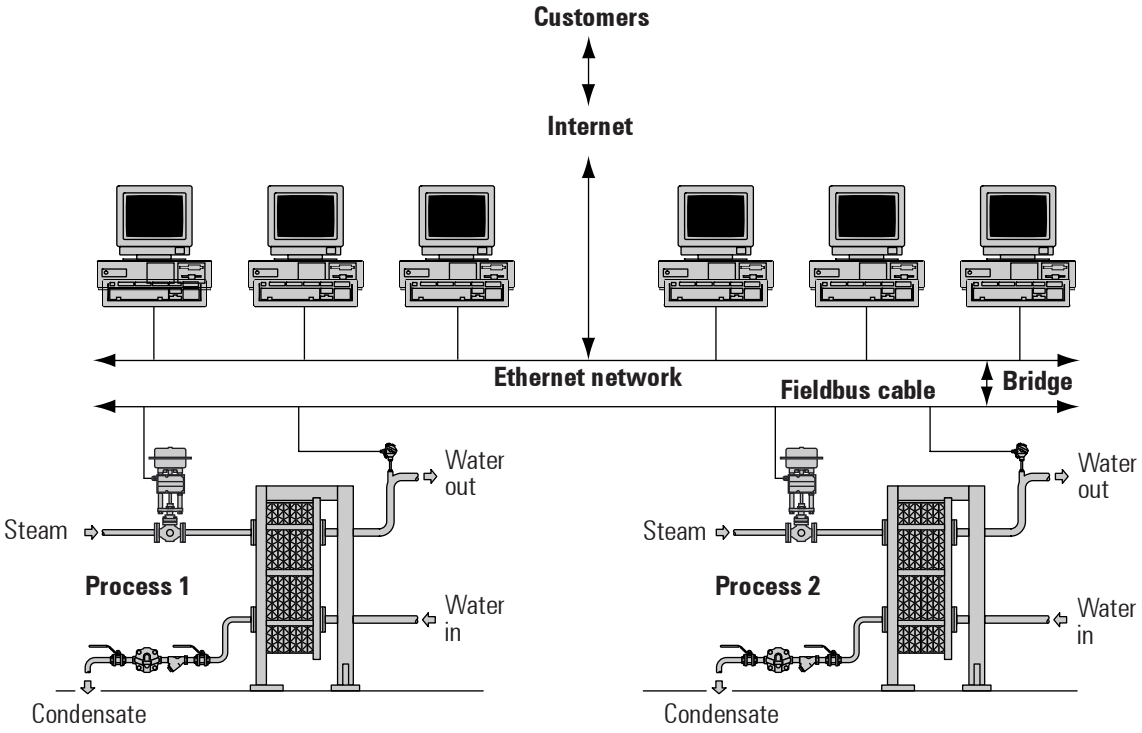


Fig. 5.6.8 Process control computer communicates with other computers over a network and the internet

On the process side the bridge can:

- Request and receive data from a number of sensors.
- Use this information in complex mathematical routines to determine and transmit the required corrective action to control devices such as valves.
- Can request the equipment to initiate a diagnostic routine, and report.

On the computer network side it can provide:

- Historical data of equipment, such as date and result of recent diagnostic routines.
- Alarms when the process or equipment exceeds set parameters.
- Detailed historical and current data on plant performance.
- Safety interlocks.

Important notes:

- Bridges vary in complexity but may control 50+ processes; the equivalent of 50 single loop PID controllers.
- If more processes are to be controlled, then more than one bridge may be used.
- The bridge(s) may be located at convenient points around a plant.
- The bridge does not usually display information, nor have any buttons to press. It is simply an electronic gateway; all interaction with it is made via the PC.

Although Fieldbus is theoretically a common technology, there are differences between the products and protocols used by different manufacturers.

Names commonly encountered in Fieldbus include:

- HART®
- CAN
- PROFIBUS®
- Interbus

Important notes:

- Fieldbus protocols and products are not directly compatible with each other. There are ways of integrating different Fieldbus' but this can be expensive. This means that users will generally adopt one system exclusively.
- Fieldbus systems can integrate older signal based instruments (4 - 20 mA, 0 - 10 V etc.). However, signals have to be interfaced to the Fieldbus by I/O units and in doing so many (but not all) of the benefits of Fieldbus are lost.
- This means that once a particular Fieldbus system has been adopted on a plant, it is unusual for the user to even consider an alternative protocol.

As control technology advances, so does the PC. Computers are able to communicate with each other over networks (LAN – Local Area Network): Finance, Stores, Production, Marketing and Sales departments within an organisation could easily share data, and have different levels of authority to perform various tasks. Inevitably, the process control computer has been connected to the network, allowing authorised personnel to view and amend the operation of the plant from a PC in an office.

As manufacturing has become global, Wide Area Networks (WAN) have developed. Consequently, an engineer located in London could, for example, interrogate a plant computer at his company's plant in New York.

The impact of this control and communications technology is enormous. The knowledge, expertise and equipment now exists where:

- A customer's stores computer, responding to a 'minimum stock' command or a production plan, can place an order over the Internet.
- The order is received by the supplier's computer which:
  - Interrogates the stores holding for the product and despatches it, or
  - Modifies the production schedule to include the order, perhaps even amending the process instructions to produce a particular product.
- The computer arranges despatch of the product and invoices the customer.
- No human intervention is required.

Benefits of Fieldbus technology

Installation:

- **Reduction in system hardware -** Fewer controllers and less wiring are required to control the process
- **Reduction in installation costs -** Not only is there less equipment to install, the installation is simpler and quicker, consequently this means a very significant reduction in material and labour costs for installing wire, cable tray, conduit, marshalling cabinets, junction boxes, and terminal blocks.
- **Less space required -** Because there is less equipment and less wiring in the control room more space is available for other uses. It equally follows that there will be more space for production equipment in the plant.
- **Engineering drawings -** The computer automatically produces the process logic drawings, so they are always accurate and up-to-date.

Operation:

- **Safety -** Fault state actions are embedded in the software with specific actions defined. In the event of a failure of the main computer, control falls back to the 'local' bridges which have independent power supplies and are programmed to default to a 'safe mode' relevant to the process.
- **Increased process information -** The amount of information available to operators and management is increased many times compared to a Distributed Control System (DCS), see Figure 5.6.9. Individual devices (such as sensors and valves) are easily interrogated, viewed and analysed. The complete process, or individual parts of the process, may be viewed and analysed to identify restrictions, capacity for improvement and so on.

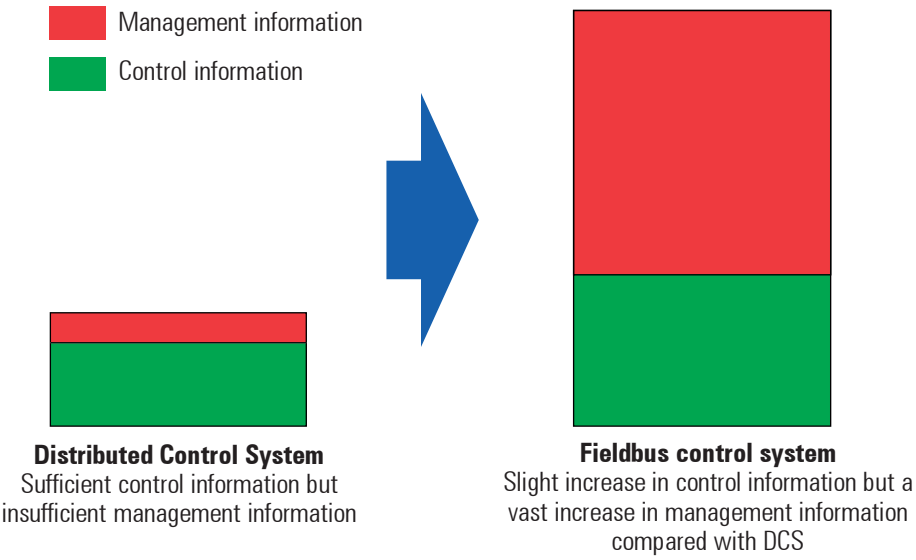


Fig. 5.6.9 Comparison of control and management information available using DCS and Fieldbus systems

- **Proactive maintenance -** The main computer can carry out detailed diagnostic routines, testing for sensor failure, output failure, memory failure, configuration error, communication error, valve position and valve travel time used, stick-slip action, and so on. Consequently, maintenance and calibration are based on the actual condition of the device rather than a time period, so maintenance is reduced to only that which is necessary.

  Several devices can perform maintenance and calibration routines at the same time. This means fewer or shorter shutdowns, giving increased plant availability.

  Time, materials and labour wasted on unnecessary maintenance is avoided, this means that the cost of maintenance is minimised.
- **System reliability -** Proactive maintenance means that equipment is well maintained.
- **Quality control -** Centralised control and the ability to view the process in parts or in total, improves quality control.
- **Stock holding -** Improved response and flexibility from the plant means that the product inventory can often be reduced.
- **Spares -** Because of the compatibility and interchangeability of components, the user is not tied to one component supplier, so prices are competitive. It also means that the spares inventory can be minimised, again saving costs.
- **Communications -** The control system or any of its components may be accessed from virtually anywhere, either over computer networks, or the Internet .

Development of a Fieldbus system

Flexibility:

- The system can easily be updated to operate with revised process requirements.
- The system can easily be expanded to take on plant expansions or new processes.
- Compatibility with other systems means that equipment can be procured at competitive prices.

Questions

1. Which of the following is NOT a Fieldbus protocol?

a| HART® ☐

b| Commbus ☐

c| CAN ☐

d| Interbus ☐

2. Which of the following applies to a modern Fieldbus system?

a| Eliminates the need for a separate controller for each process, and communicates directly with sensors ☐

b| Can control up to fifteen processes simultaneously ☐

c| Incorporates devices at each process for local display of parameters, but not for programming ☐

d| Has excellent flexibility and allows any computer operator connected to the system to read and change process parameters and saves commissioning time ☐

3. Which of the following is required to integrate older signal based instruments such as those with an output of 4 - 20 mA to a Fieldbus system?

a| Interbus protocol ☐

b| A bridge for each signal to convert it to a digital signal ☐

c| PROFIBUS® protocol which is based on an analogue system ☐

d| Signal Input/Output units ☐

4. Which of the following is UNTRUE of a Fieldbus system?

a| It will save time on plant commissioning ☐

b| It is a system designed for communication to and from a plant ☐

c| It will reduce the energy requirements of a plant ☐

d| Reliability of the process control valve is improved ☐

5. Which one of the following is an operational benefit of using Fieldbus?

a| It reduces the maintenance requirements of a plant ☐

b| It automatically guarantees consistency of product ☐

c| With regards to safety fault state, actions are embedded in the computer software ☐

d| Reliability of the process control valve is improved ☐

6. In automation terms, what is a bridge?

a| A device which permits communication between modern controllers and older PCs ☐

b| A device that interfaces between Fieldbus protocol and computers on a network ☐

c| A device that, in the event of a network failure, ensures the process controllers continue operating with their programmed parameters ☐

d| A Fieldbus arrangement to allow each process controller to interface directly with a central computer system ☐

Answers

1: b, 2: a, 3: d, 4: c, 5: c, 6: b

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Module 6.1

Control Valves

Introduction to Electric/Pneumatic Controls

Block 6 of *The Steam and Condensate Loop* considers the practical aspects of control, putting the basic control theory discussed in Block 5 into practice.

A basic control system would normally consist of the following components:
- Control valves.
- Actuators.
- Controllers.
- Sensors.

All of these terms are generic and each can include many variations and characteristics. With the advance of technology, the dividing line between individual items of equipment and their definitions are becoming less clear. For example, the positioner, which traditionally adjusted the valve to a particular position within its range of travel, can now:

- Take input directly from a sensor and provide a control function.
- Interface with a computer to alter the control functions, and perform diagnostic routines.
- Modify the valve movements to alter the characteristics of the control valve.
- Interface with plant digital communication systems.

However, for the sake of clarity at this point, each item of equipment will be considered separately.

Control Valves

Whilst a wide variety of valve types exist, this document will concentrate on those which are most widely used in the automatic control of steam and other industrial fluids. These include valve types which have linear and rotary spindle movement.

Linear types include globe valves and slide valves.

Rotary types include ball valves, butterfly valves, plug valves and their variants.

The first choice to be made is between two-port and three-port valves.
- Two-port valves 'throttle' (restrict) the fluid passing through them.
- Three-port valves can be used to 'mix' or 'divert' liquid passing through them.

Two-port valves

Globe valves

Globe valves are frequently used for control applications because of their suitability for throttling flow and the ease with which they can be given a specific 'characteristic', relating valve opening to flow.

Two typical globe valve types are shown in Figure 6.1.1. An actuator coupled to the valve spindle would provide valve movement.

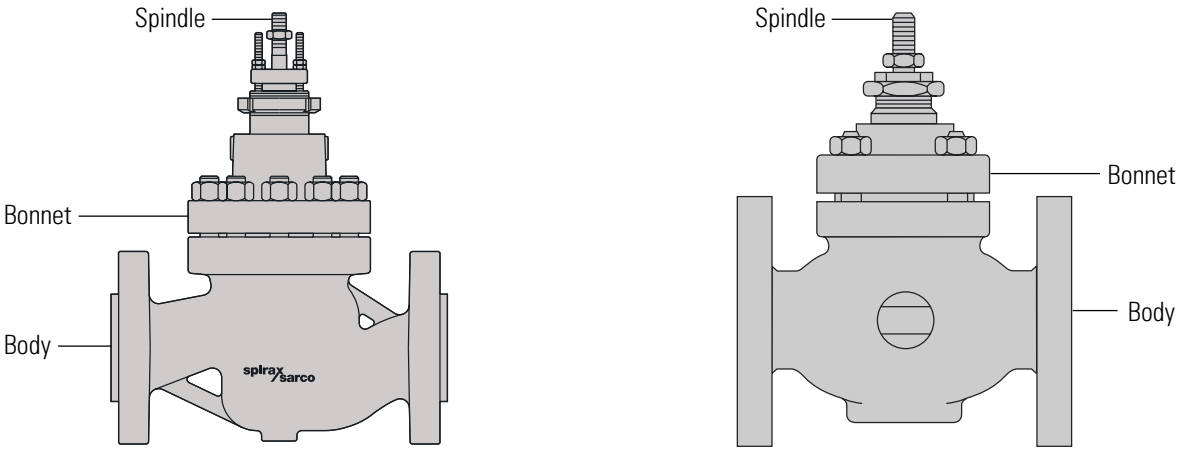


Fig. 6.1.1 Two differently shaped globe valves

The major constituent parts of globe valves are:

- The body.
- The bonnet.
- The valve seat and valve plug, or trim.
- The valve spindle (which connects to the actuator).
- The sealing arrangement between the valve stem and the bonnet.

Figure 6.1.2 is a diagrammatic representation of a single seat two-port globe valve. In this case the fluid flow is pushing against the valve plug and tending to keep the plug off the valve seat.

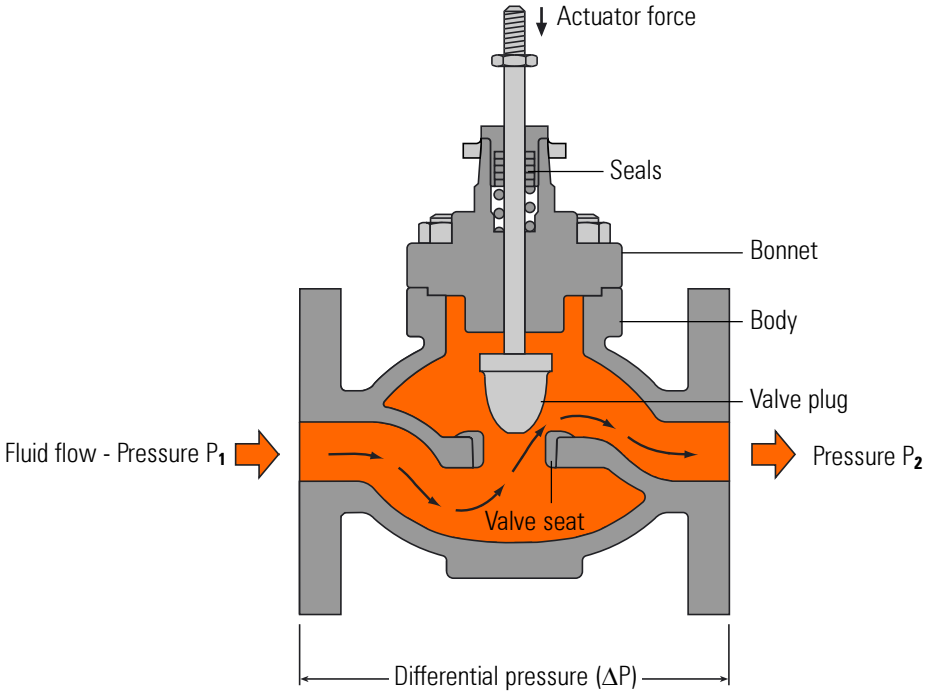


Fig. 6.1.2 Flow through a single seat, two-port globe valve

The difference in pressure upstream (P_1) and downstream (P_2) of the valve, against which the valve must close, is known as the differential pressure (ΔP). The maximum differential pressure against which a valve can close will depend upon the size and type of valve and the actuator operating it.

In broad terms, the force required from the actuator may be determined using Equation 6.1.1.

$$(A \times \Delta P) + \text{Friction allowance} = F \qquad \textbf{Equation 6.1.1}$$

Where:
A = Valve seating area (m^2)
ΔP = Differential pressure (kPa)
F = Closing force required (kN)

In a steam system, the maximum differential pressure is usually assumed to be the same as the upstream absolute pressure. This allows for possible vacuum conditions downstream of the valve when the valve closes. The differential pressure in a closed water system is the maximum pump differential head.

If a larger valve, having a larger orifice, is used to pass greater volumes of the medium, then the force that the actuator must develop in order to close the valve will also increase. Where very large capacities must be passed using large valves, or where very high differential pressures exist, the point will be reached where it becomes impractical to provide sufficient force to close a conventional single seat valve. In such circumstances, the traditional solution to this problem is the double seat two-port valve.

As the name implies, the double seat valve has two valve plugs on a common spindle, with two valve seats. Not only can the valve seats be kept smaller (since there are two of them) but also, as can be seen in Figure 6.1.3, the forces are partially balanced. This means that although the differential pressure is trying to keep the top valve plug off its seat (as with a single seat valve) it is also trying to push down and close the lower valve plug.

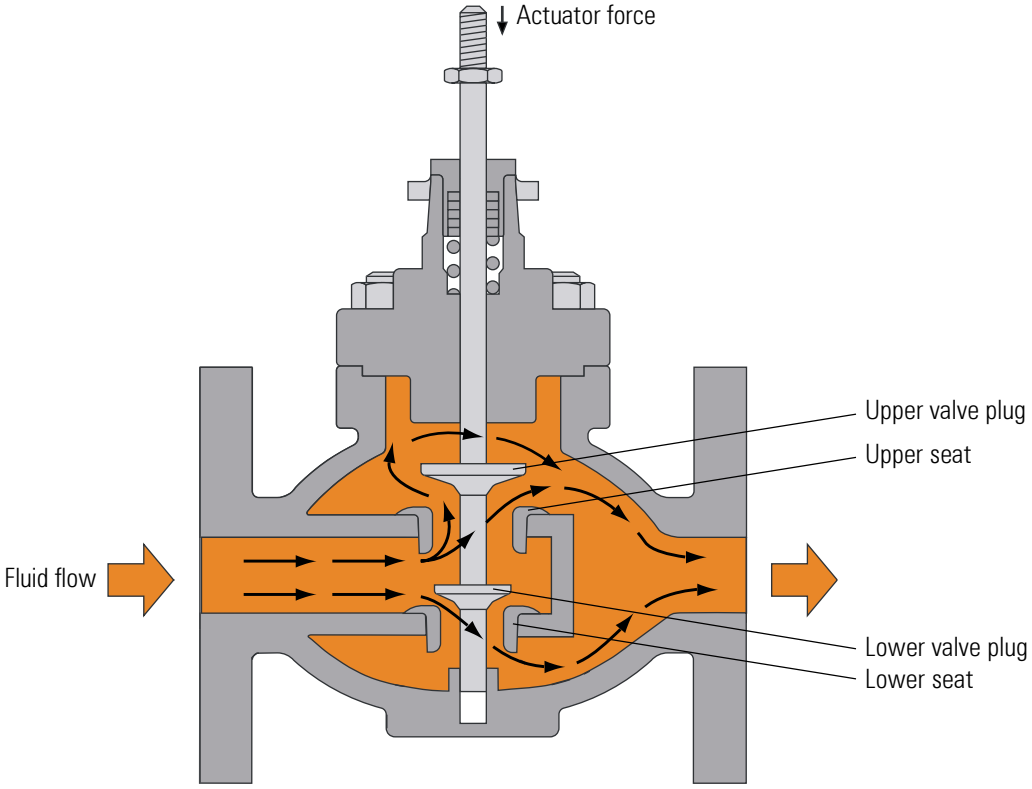


Fig. 6.1.3 Flow through a double seat, two-port valve

However, a potential problem exists with any double seat valve. Because of manufacturing tolerances and differing coefficients of expansion, few double seat valves can be guaranteed to give good shut-off tightness.

Shut-off tightness

Control valve leakage is classified with respect to how much the valve will leak when fully closed. The leakage rate across a standard double seat valve is at best Class III, (a leakage of 0.1% of full flow) which may be too much to make it suitable for certain applications. Consequently, because the flow paths through the two-ports are different, the forces may not remain in balance when the valve opens.

Various international standards exist that formalise leakage rates in control valves. The following leakage rates are taken from the British Standard BS 5793 Part 4 (IEC 60534-4). For an unbalanced standard single seat valve, the leakage rate will normally be Class IV, (0.01% of full flow), although it is possible to obtain Class V, (1.8×10^{-5} x differential pressure (bar) x seat diameter (mm). Generally, the lower the leakage rate the more the cost.

Balanced single seat valves

Because of the leakage problem associated with double seat valves, when a tight shut-off is required a single seat valve should be specified. The forces required to shut a single seat globe valve increase considerably with valve size. Some valves are designed with a balancing mechanism to reduce the closing force necessary, especially on valves operating with large differential pressures. In a piston-balanced valve, some of the upstream fluid pressure is transmitted via internal pathways into a space above the valve plug, which acts as a pressure balancing chamber. The pressure contained in this chamber provides a downforce on the valve plug as shown in Figure 6.1.4, balancing the upstream pressure and assisting the normal force exerted by the actuator, to close the valve.

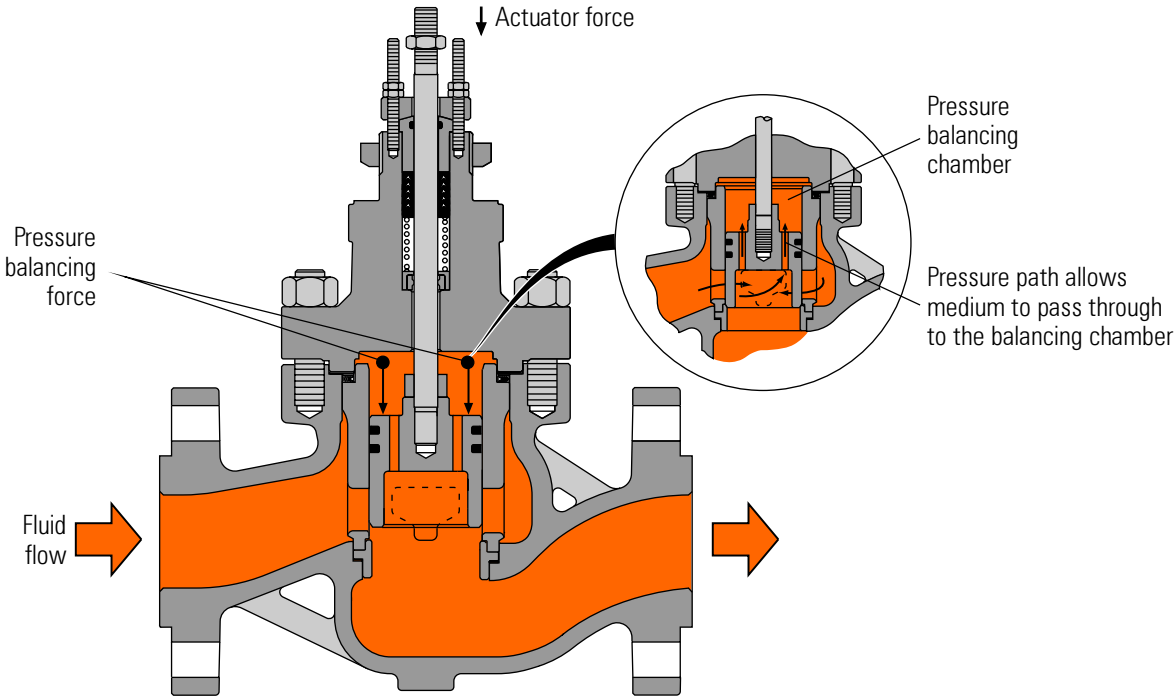


Fig. 6.1.4 A steam control valve with piston balancing

Slide valves, spindle operated

Slide valves tend to come in two different designs; wedge gate type and parallel slide type. Both types are well suited for isolating fluid flow, as they give a tight shut-off and, when open, the pressure drop across them is very small. Both types are used as manually operated valves, but if automatic actuation is required, the parallel slide valve is usually chosen, whether for isolation or control. Typical valves are shown in Figure 6.1.5.

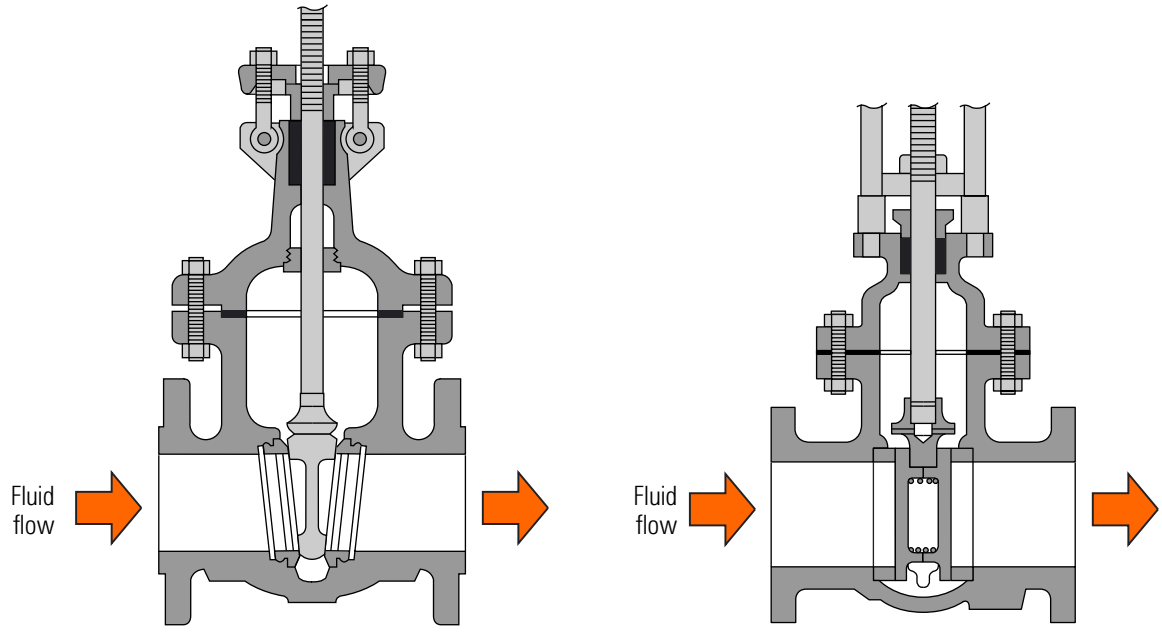


Fig. 6.1.5 Wedge gate valve and parallel slide valve (manual operation)

The parallel slide valve closes by means of two spring loaded sliding disks (springs not shown), which pass across the flow-path of the fluid, the fluid pressure ensuring a tight joint between the downstream disk and its seat. Large size parallel slide valves are used in main steam and feedlines in the power and process industries to isolate sections of the plant. Small-bore parallel slides are also used for the control of ancillary steam and water services although, mainly due to cost, these tasks are often carried out using actuated ball valves and piston type valves.

Rotary type valves

Rotary type valves, often called quarter-turn valves, include plug valves, ball valves and butterfly valves. All require a rotary motion to open and close, and can easily be fitted with actuators.

Eccentric plug valves

Figure 6.1.6 shows a typical eccentric plug valve. These valves are normally installed with the plug spindle horizontal as shown, and the attached actuator situated alongside the valve.

Plug valves may include linkages between the plug and actuator to improve the leverage and closing force, and special positioners that modify the inherent valve characteristic to a more useful equal percentage characteristic (valve characteristics are discussed in Module 6.5).

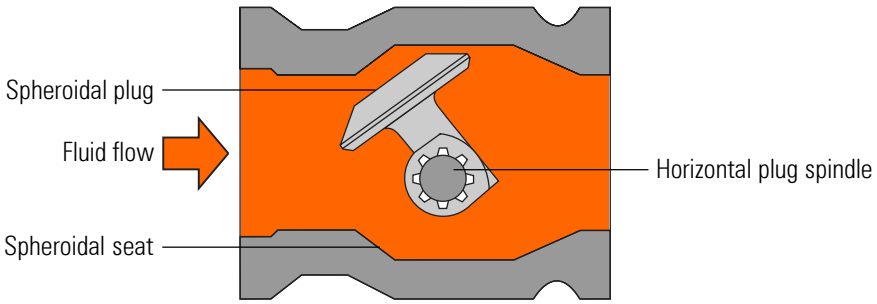
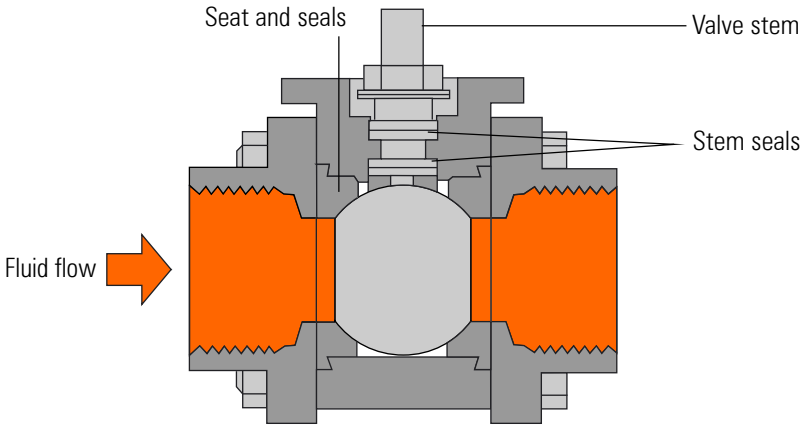


Fig. 6.1.6 Side view of an eccentric plug valve (shown in a partially open position)

Ball valves

Figure 6.1.7 shows a ball valve consisting of a spherical ball located between two sealing rings in a simple body form. The ball has a hole allowing fluid to pass through. When aligned with the pipe ends, this gives either full bore or nearly full bore flow with very little pressure drop. Rotating the ball through 90° opens and closes the flow passage. Ball valves designed specifically for control purposes will have characterized balls or seats, to give a predictable flow pattern.



End view of the ball within the ball valve at different stages of rotation

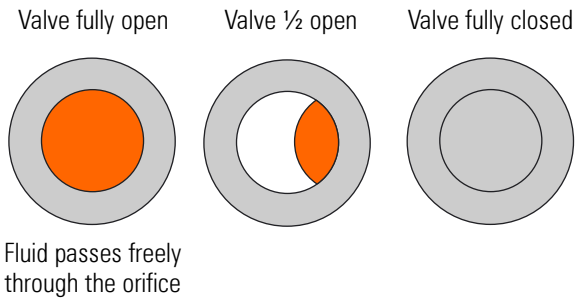
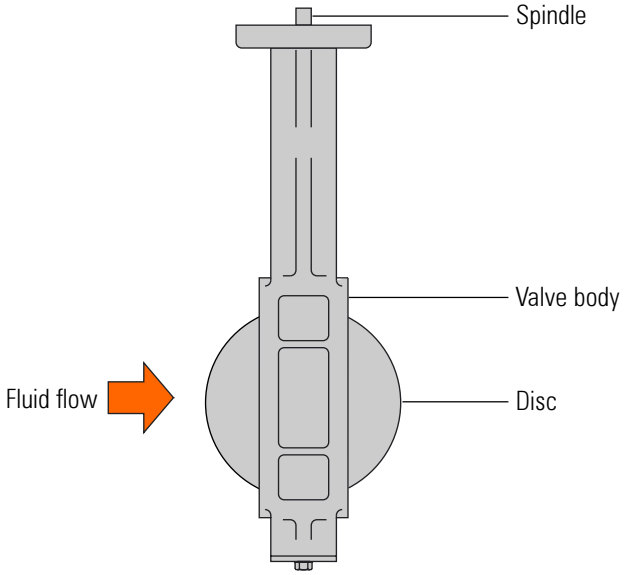


Fig. 6.1.7 Ball valve (shown in a fully open position)

Ball valves are an economic means of providing control with tight shut-off for many fluids including steam at temperatures up to 250°C (38 bar g, saturated steam). Above this temperature, special seat materials or metal-to-metal seatings are necessary, which can be expensive. Ball valves are easily actuated and often used for remote isolation and control. For critical control applications, segmented balls and balls with specially shaped holes are available to provide different flow characteristics.

Butterfly valves

Figure 6.1.8 is a simple schematic diagram of a butterfly valve, which consists of a disc rotating in trunnion bearings. In the open position the disc is parallel to the pipe wall, allowing full flow through the valve. In the closed position it is rotated against a seat, and perpendicular to the pipe wall.



End view of the disc within the butterfly valve at different stages of rotation

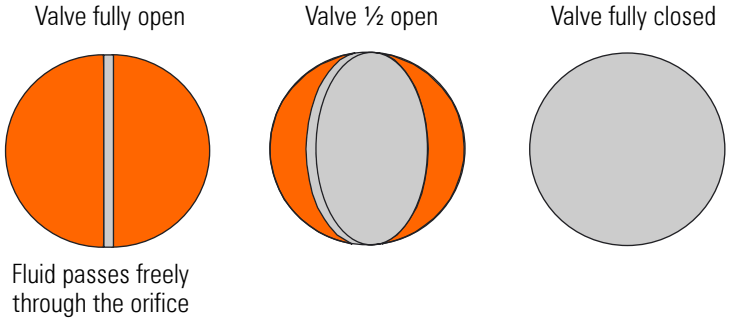


Fig. 6.1.8 Butterfly valve (shown in its open position)

Traditionally, butterfly valves were limited to low pressures and temperatures, due to the inherent limitations of the soft seats used. Currently, valves with higher temperature seats or high quality and specially machined metal-to-metal seats are available to overcome these drawbacks. Standard butterfly valves are now used in simple control applications, particularly in larger sizes and where limited turndown is required.

Special butterfly valves are available for more demanding duties.

A fluid flowing through a butterfly valve creates a low pressure drop, in that the valve presents little resistance to flow when open. In general however, their differential pressure limits are lower than those for globe valves. Ball valves are similar except that, due to their different sealing arrangements, they can operate against higher differential pressures than equivalent butterfly valves.

Options

There are always a number of options to consider when choosing a control valve. For globe valves, these include a choice of spindle gland packing material and gland packing configurations, which are designed to make the valve suitable for use on higher temperatures or for different fluids. Some examples of these can be seen in the simple schematic diagrams in Figure 6.1.9. It is worth noting that certain types of gland packing produce a greater friction with the valve spindle than others. For example, the traditional stuffing box type of packing will create greater friction than the PTFE spring-loaded chevron type or bellows sealed type. Greater friction requires a higher actuator force and will have an increased propensity for haphazard movement.

Spring-loaded packing re-adjusts itself as it wears. This reduces the need for regular manual maintenance. Bellows sealed valves are the most expensive of these three types, but provide minimal friction with the best stem sealing mechanism. As can be seen in Figure 6.1.9, bellows sealed valves usually have another set of traditional packing at the top of the valve spindle housing. This will act as a final defence against any chance of leaking through the spindle to atmosphere.

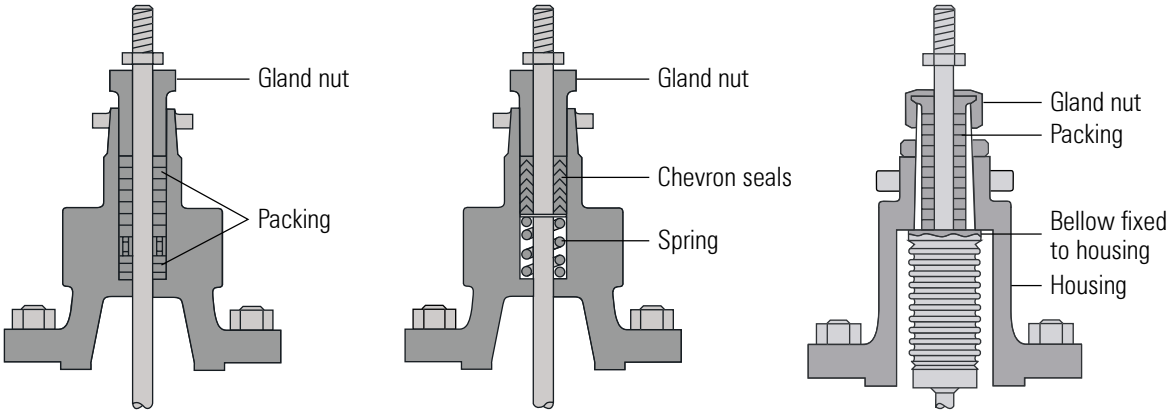


Fig. 6.1.9 Alternative gland packings

Valves also have different ways of guiding the valve plug inside the body. One common guidance method, as depicted in Figure 6.1.10, is the 'double guided' method, where the spindle is guided at both the top and the bottom of its length. Another type is the 'guided plug' method where the plug may be guided by a cage or a frame. Some valves can employ perforated plugs, which combine plug guidance and noise reduction.

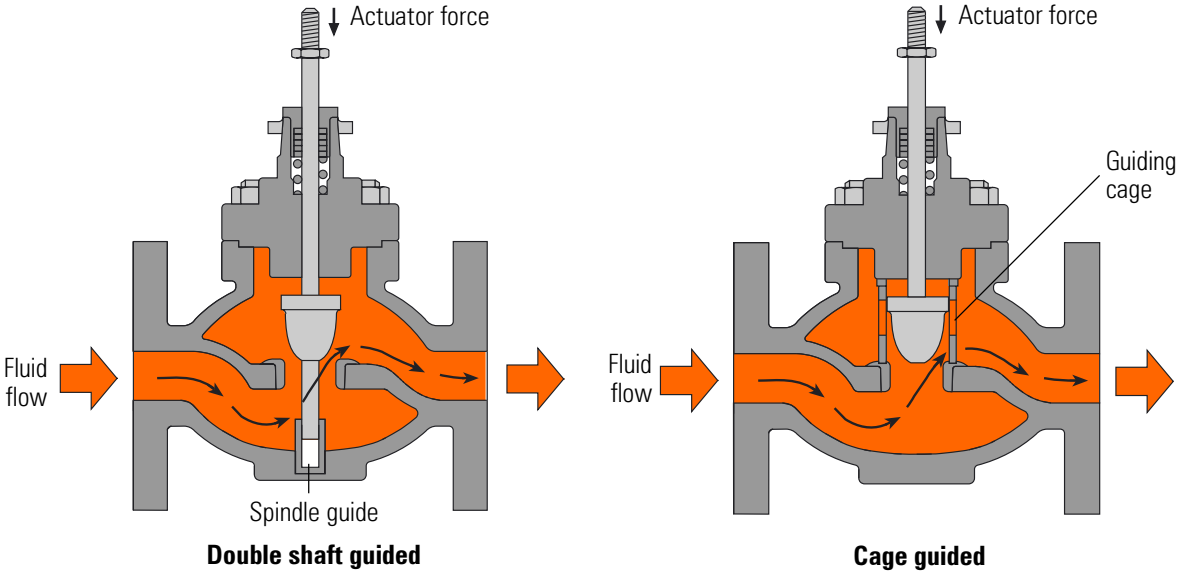


Fig. 6.1.10 Guiding arrangements

Summary of two-port valves used for automatic control

By far the most widely used valve type for the automatic control of steam processes and applications is the globe valve. It is relatively easy to actuate, it is versatile, and has inherent characteristics well suited to the automatic control needs of steam.

It should also be said that two-port automatic control valves are also used within liquid systems, such as low, medium and high temperature hot water systems, and thermal oil systems. Liquid systems carry an inherent need to be balanced with regard to mass flows. In many instances, systems are designed where two-port valves can be used without destroying the balance of distribution networks.

However, when two-port valves cannot be used on a liquid system, three-port valves are installed, which inherently maintain a balance across the distribution system, by acting in a diverting or mixing fashion.

Three-port valves

Three-port valves can be used for either mixing or diverting service depending upon the plug and seat arrangement inside the valve. A simple definition of each function is shown in Figure 6.1.11.

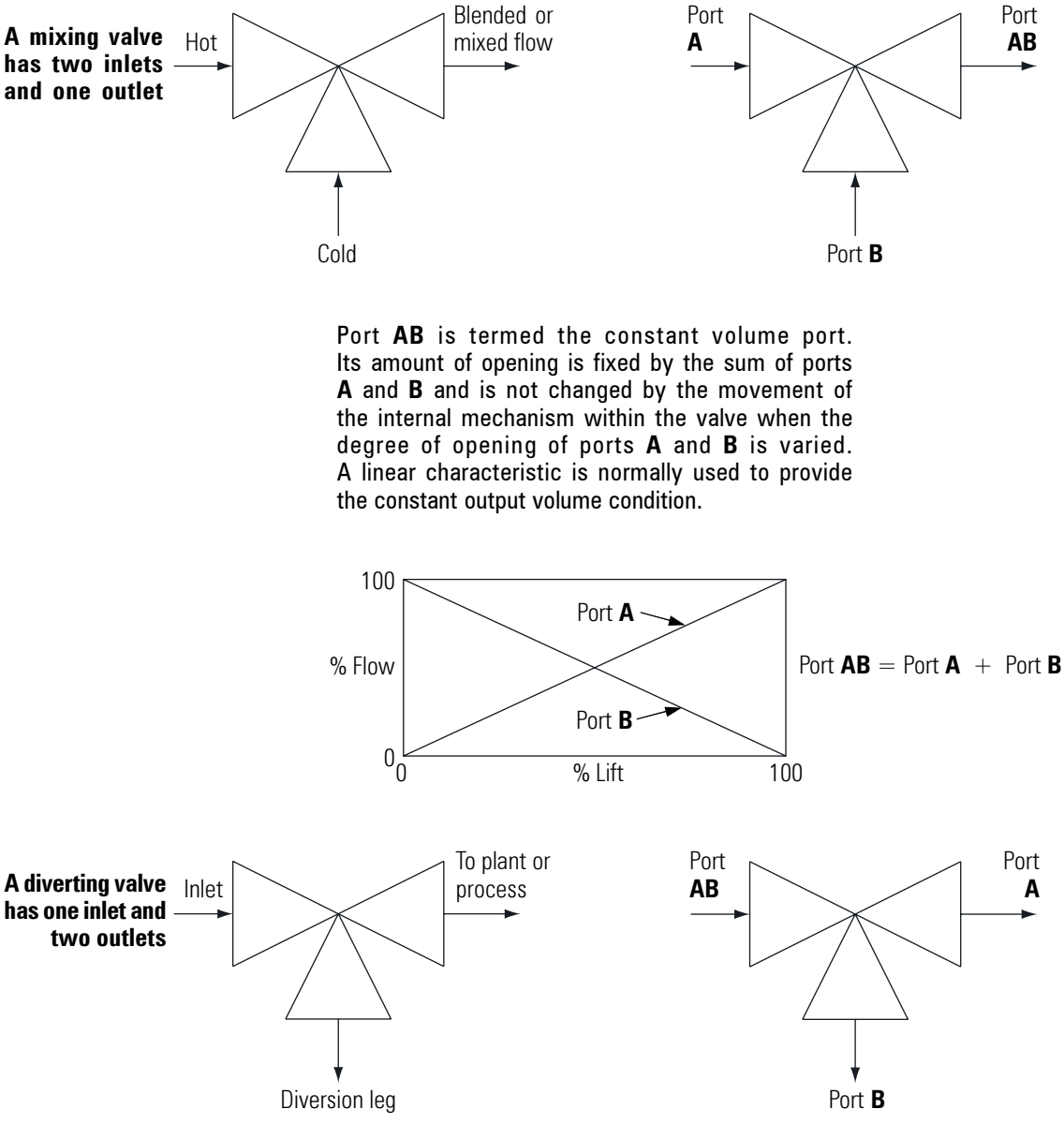
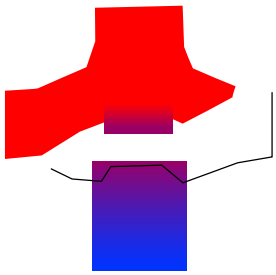


Fig. 6.1.11 Three-port valve definition



Rotating shoe three-port valve

This type of valve employs a rotating shoe, which shuttles across the port faces. The schematic arrangement in Figure 6.1.15 illustrates a mixing application with approximately 80% flowing through port **A** and 20% through port **B,** 100% to exit through port **AB**.

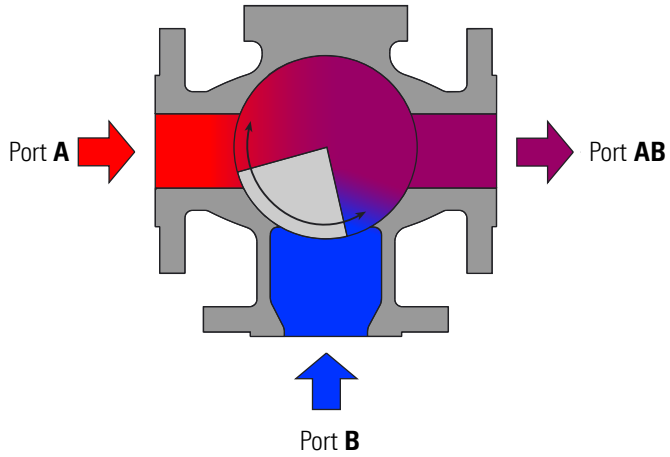


Fig. 6.1.15 Rotating shoe on a mixing application

Using three-port valves

Not all types can be used for both mixing and diverting service. Figure 6.1.16 shows the incorrect application of a globe valve manufactured as a mixing valve but used as a diverting valve.

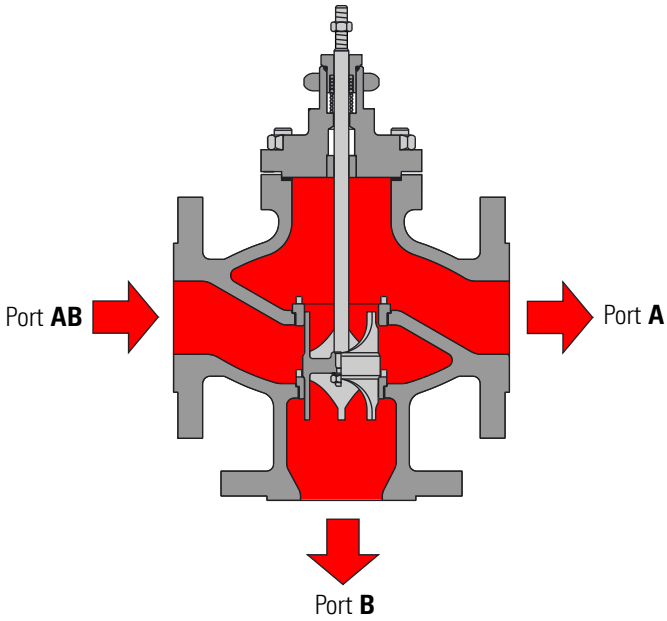


Fig. 6.1.16 Three-port mixing valve used incorrectly as a diverting valve

The flow entering the valve through port **AB** can leave from either of the two outlet ports **A** or **B**, or a proportion may leave from each. With port **A** open and port **B** closed, the differential pressure of the system will be applied to one side of the plug.

When port **A** is closed, port **B** is open, and differential pressure will be applied across the other side of the plug. At some intermediate plug position, the differential pressure will reverse. This reversal of pressure can cause the plug to move out of position, giving poor control and possible noise as the plug 'chatters' against its seat.

To overcome this problem on a plug type valve designed for diverting, a different seat configuration is used, as shown in Fig. 6.1.17. Here, the differential pressure is equally applied to the same sides of both valve plugs at all times.

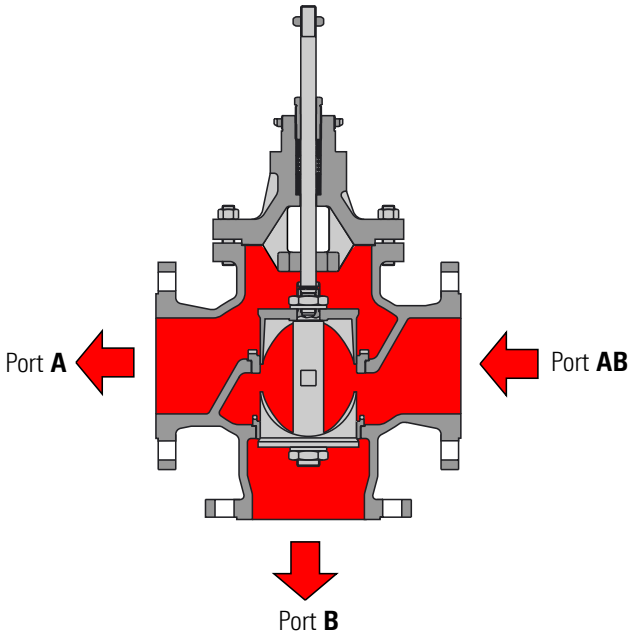


Fig. 6.1.17 Plug type diverting valve

In closed circuits, it is possible to use mixing valves or diverting valves, depending upon the system design, as depicted in Figures 6.1.18 and 6.1.19.

In Figure 6.1.18, the valve is designed as a mixing valve as it has two inlets and one outlet. However, when placed in the return pipework from the load, it actually performs a diverting function, as it diverts hot water away from the heat exchanger.

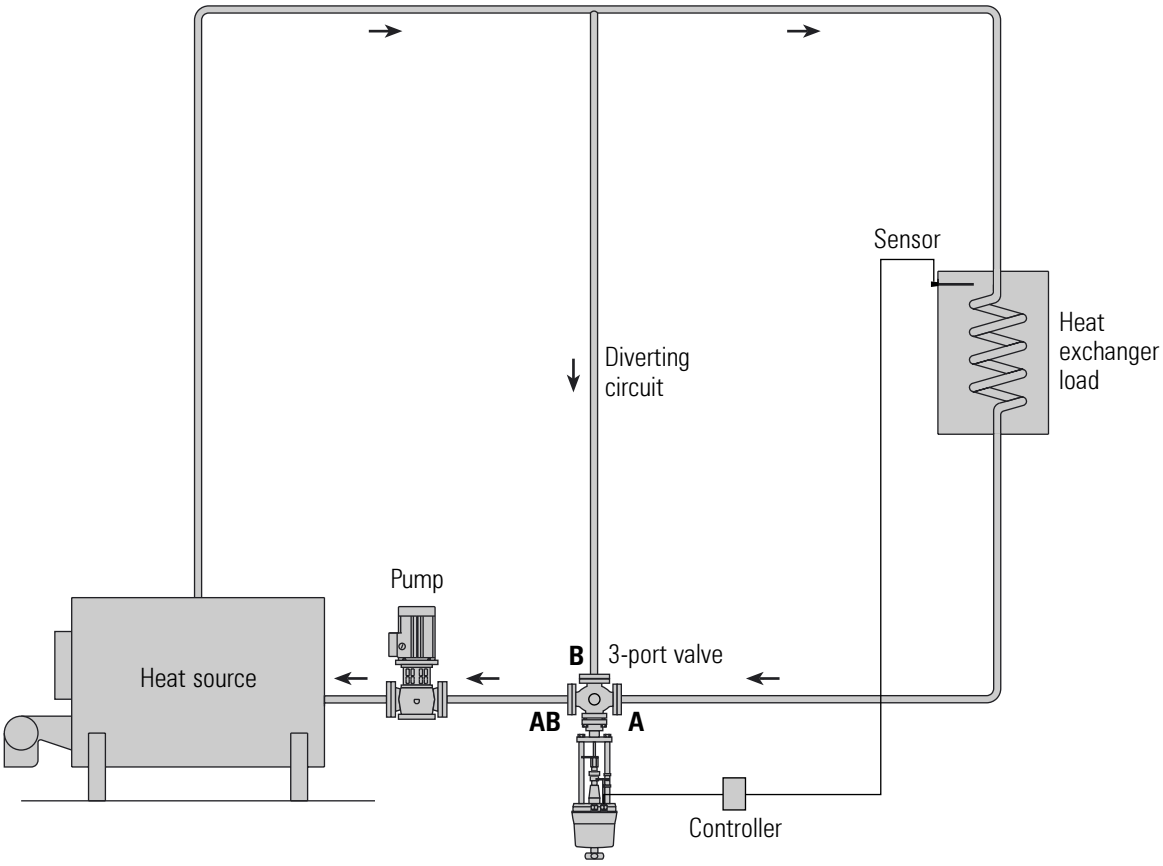


Fig. 6.1.18 Mixing Valve installed on the return pipework

Consider the mixing valve used in Figure 6.1.18, when the heat exchanger is calling for maximum heat, perhaps at start-up, port **A** will be fully open, and port **B** fully closed. The whole of the water passing from the boiler is passed through the heat exchanger and passes through the valve via ports **AB** and **A**. When the heat load is satisfied, port **A** will be fully closed and port **B** fully open, and the whole of the water passing from the boiler bypasses the load and passes through the valve via ports **AB** and **B**. In this sense, the water is being diverted from the heat exchanger in relation to the requirements of the heat load.

The same effect can be achieved by installing a diverting valve in the flow pipework, as depicted by Figure 6.1.19.

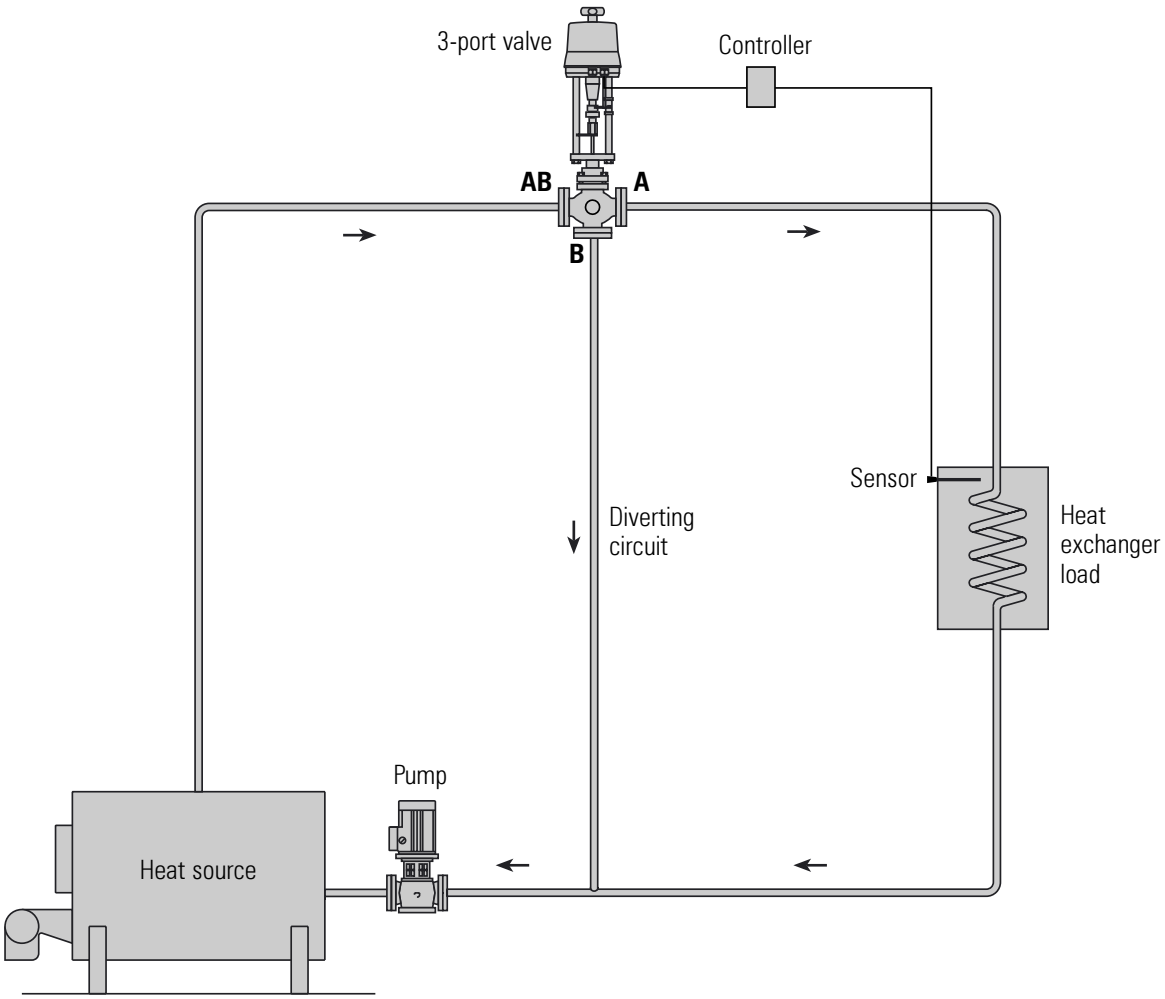


Fig. 6.1.19 Diverting valve installed on the flow pipework

Questions

1. What would an operating control system normally consist of?

a| Valve ☐

b| Valve and actuator ☐

c| Valve, actuator and controller ☐

d| Valve, actuator, controller and sensor ☐

2. What is the basic difference between 2-port and 3-port control valves?

a| 2-port valves restrict the fluid flow, 3-port valves mix or divert ☐

b| 2-port valves are only for gases, 3-port valves are only for liquids ☐

c| 2-port valves use electrical actuators, 3-port valves use pneumatic ☐

d| 2-port valves are steel, 3-port valves are bronze ☐

3. What is the basic difference between a spindle valve and a rotary valve?

a| Spindle valves have higher capacity for the same physical size ☐

b| Plug movement is in/out for spindle, side/side for rotary ☐

c| Spindle valves can only operate in a vertical plane ☐

d| Only spindle valves need valve packing ☐

4. A valve has a plug area of 500 mm², a differential pressure of 1 000 kPa, and a friction allowance of 10%. What is the minimum actuator closing force?

a| 55 kN ☐

b| 550 kN ☐

c| 0.55 kN ☐

d| 5.5 kN ☐

5. What is the main disadvantage of a double seat valve?

a| It costs more than a single seat valve ☐

b| The valve body is larger than a single seat valve of the same capacity ☐

c| It is more difficult to maintain ☐

d| It does not give a tight shut-off when fully closed ☐

6. What benefit does the bellows seal arrangement have over a traditional type of stuffing box valve packing?

a| The spindle movement produces less friction ☐

b| Fluid is less likely to leak through the spindle bonnet ☐

c| The valve operation is smoother ☐

d| All of the above ☐

Answers

1: d, 2: a, 3: b, 4: d, 5: d, 6: d

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Module 6.2

Control Valve Capacity

Introduction to Valve Capacity

A control valve must, as its name suggests, have a controlling influence on the process. Whilst details such as connection sizes and materials of construction are vitally important, they do not give any indication of the control exerted by the valve.

Control valves adjust processes by altering:

- **Flowrate -** For example, the amount of steam or water that enters the process equipment.

  With a two-port valve for example, as the valve moves to the closed position, less steam flows, and less heat is added to the process.

  With a three-port valve for example, as the valve plug moves to a new position, it diverts hot water away from the process.

And/or

- **Differential pressure -** This is defined as the difference between the pressure at the valve inlet and the pressure at the valve outlet (see Figure 6.2.1).

  For any given valve orifice size, the greater the differential pressure the greater the flowrate, within certain limitations.

  With saturated steam, the lower its pressure, the lower its temperature, and less heat transfer will occur in the heat exchanger.

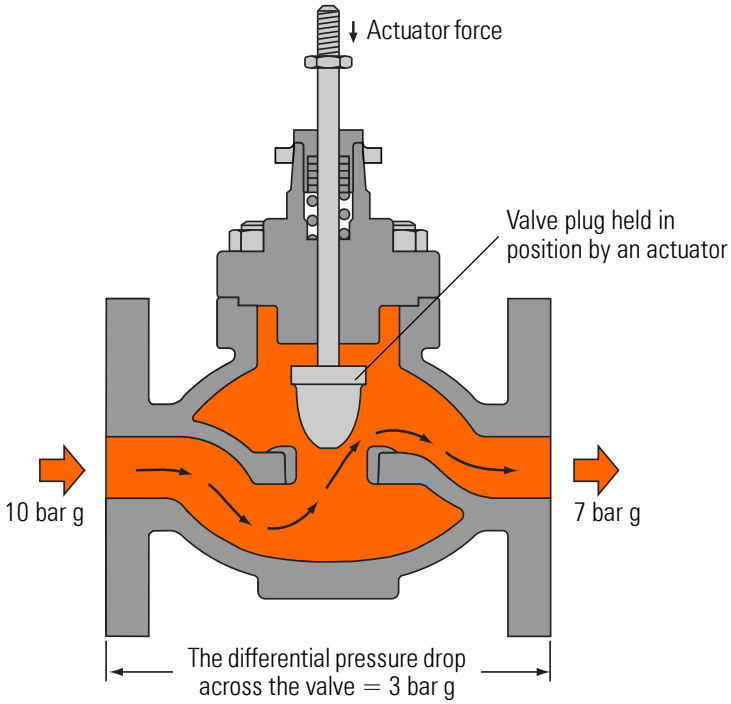


Fig. 6.2.1 Differential pressure across a valve

These two factors (**a**) **Flowrate** and (**b**) **Differential pressure** are brought together as a **flow coefficient** or 'capacity index' as it is sometimes termed.

The flow coefficient allows:

- The performance of valves to be compared.
- The differential pressure across a valve to be determined from any flowrate.
- The flowrate through a control valve to be determined for a given differential pressure.

Because many different units of measurement are used around the world, a number of flow coefficients are available, and it is worthwhile understanding their definitions. Table 6.2.1 identifies and defines the most commonly encountered capacity indices.

Table 6.2.1 Symbols and definitions used to identify and quantify flow through a control valve

K_v	Flowrate in m^3/h of water at a defined temperature, typically between 5°C and 40°C, that will create a pressure drop of one bar across a valve orifice. (Widely used in Europe)
K_{vs}	The actual or stated K_v value of a particular valve when fully open, constituting the valve flow coefficient, or capacity index.
K_{vr}	The K_{vr} is the flow coefficient required by the application.
C_v	The flowrate in gallons per minute of water at a defined temperature, typically between 40°F and 100°F that will create a pressure drop of one pound per square inch. (Widely used in the US, and certain other parts of the world). Care needs to be taken with this term, as both C_v Imperial and C_v US exist. Whilst the basic definition is the same, the actual values are slightly different because of the difference between Imperial and US gallons.
A_v	Flowrate in m^3/s of water that will create a pressure drop of one Pascal.

For conversion:

C_v (Imperial) = K_v x 0.962 658
C_v (US) = K_v x 1.156 099
A_v = 2.88 x 10^{-5} C_v (Imperial)

The flow coefficient, K_{vs} for a control valve is essential information, and is usually stated, along with its other data, on the manufacturer's technical data sheets.

Control valve manufacturers will usually offer a number of trim sizes (combination of valve seat and valve plug) for a particular valve size. This may be to simplify the pipework by eliminating the need for reducers, or to reduce noise.

A typical range of K_{vs} flow coefficients available for a selection of valves is shown in Table 6.2.2

Table 6.2.2 K_{vs} values for a typical range of valves

Sizes	DN15	DN20	DN25	DN32	DN40	DN50	DN65	DN80	DN100
K_{vs}	4.0	6.3	10.0	16.0	25.0	36.0	63.0	100.0	160.0
	2.5	4.0	6.3	10.0	16.0	25.0	36.0	63.0	100.0
	1.6	2.5	4.0	6.3	10.0	16.0	25.0	36.0	63.0
	1.0	1.6	2.5	4.0	6.3	10.0	16.0	25.0	36.0

The relationship between flowrates, differential pressures, and the flow coefficients will vary depending upon the type of fluid flowing through the valve. These relationships are predictable and satisfied by equations, and are discussed in further detail in:

- Module 6.3 - Control Valve Sizing for Water Systems.
- Module 6.4 - Control Valve Sizing for Steam Systems.

Questions

1. What two basic properties enable control valves to 'control'?

a| Temperature and pressure ☐

b| Pressure and valve movement ☐

c| Pressure and flowrate ☐

d| Temperature and flowrate ☐

2. For a given orifice size, which of the following is true?

a| The greater the pressure drop, the less the flow ☐

b| The greater the flow, the less the pressure drop ☐

c| The greater the pressure drop, the greater the flow ☐

d| The less the flow, the greater the pressure drop ☐

3. Which of the following is recognised as a valve flow coefficient for a fully open valve?

a| K_v ☐

b| C_v ☐

c| A_v ☐

d| K_{vs} ☐

Answers

1: c, 2: c, 3: d,

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Module 6.3

Control Valve Sizing for Water System

Control Valve Sizing For Water Systems

Sizing valves for water service

In order to size a valve for a water application, the following must be known:

- The volumetric flowrate through the valve.
- The differential pressure across the valve.

The control valve can be sized to operate at a certain differential pressure by using a graph relating flowrate, pressure drop, and valve flow coefficients.

Alternatively, the flow coefficient may be calculated using a formula. Once determined, the flow coefficient is used to select the correct sized valve from the manufacturer's technical data.

Historically, the formula for flow coefficient was derived using Imperial units, offering measurement in terms of gallons/minute with a differential pressure of one pound per square inch. There are two versions of the Imperial coefficient, a British version and an American version, and care must be taken when using them because each one is different, even though the adopted symbol for both versions is 'C_v'. The British version uses Imperial gallons, whilst the American version uses American gallons, which is 0.833 the volume of an Imperial gallon. The adopted symbol for both versions is C_v.

The metric version of flow coefficient was originally derived in terms of cubic metres an hour (m^3/h) of flow for a differential pressure measured in kilogram force per square metre (kgf/m^2). This definition had been derived before an agreed European standard existed that defined K_v in terms of SI units (bar). However, an SI standard has existed since 1987 in the form of IEC 534 -1 (Now EN 60534 -1). The standard definition now relates flowrate in terms of m^3/h for a differential pressure of 1 bar. Both metric versions are still used with the adopted symbol K_v, and although the difference between them is quite small, it is important to be certain or to make clear which one is being used. Some manufacturers mistakenly quote K_v conversion values without qualifying the unit of pressure differential.

Table 6.3.1 converts the different types of flow coefficient mentioned above:

Table 6.3.1 Multiplication factors for flow coefficient conversion between K_v and C_v

Multiply	K_v (bar)	K_v (kgf)	C_v (UK)	C_v (US)
K_v (bar)	1.00	1.01	0.96	1.16
K_v (kgf)	0.99	1.00	0.97	1.17
C_v (UK)	1.04	1.05	1.00	1.20
C_v (US)	0.87	0.88	0.83	1.00

For example, multiply K_v (bar) by 1.16 to convert to C_v (US).

The K_v version quoted in these Modules is always measured in terms of K_v (bar), that is units of m^3/h bar, unless otherwise stated.

For liquid flow generally, the formula for K_v is shown in Equation 6.3.1.

$$K_v = \dot{V}\sqrt{\frac{G}{\Delta P}} \qquad \textbf{Equation 6.3.1}$$

Where:

K_v = Flow of liquid that will create a pressure drop of 1 bar (m^3/ h bar)
$\dot{V}$ = Flowrate (m^3/h)
G = Relative density/specific gravity of the liquid (dimensionless). Note: Relative density is a ratio of the mass of a liquid to the mass of an equal volume of water at 4°C
ΔP = Pressure drop across the valve (bar)

Sometimes, the volumetric flowrate needs to be determined, using the valve flow coefficient and differential pressure.

Rearranging Equation 6.3.1 gives: $$\dot{V} = K_v\sqrt{\frac{\Delta P}{G}}$$

For water, G = 1, consequently the equation for water may be simplified to that shown in Equation 6.3.2.

$$\dot{V} = K_v\sqrt{\Delta P} \qquad \textbf{Equation 6.3.2}$$

Example 6.3.1

10 m³/h of water is pumped around a circuit; determine the pressure drop across a valve with a K_v of 16 by using Equation 6.3.2:

$$\dot{V} = K_v\sqrt{\Delta P} \qquad \textbf{Equation 6.3.2}$$

Where:

$\dot{V}$ = 10 m³/h

K_v = 16

$$10 = 16\sqrt{\Delta P}$$

$$\Delta P = \left(\frac{10}{16}\right)^2$$

$$\mathbf{\Delta P = 0.39\ bar}$$

Alternatively, for this example the chart shown in Figure 6.3.1, may be used. (Note: a more comprehensive water K_v chart is shown in Figure 6.3.2):

1. Enter the chart on the left hand side at 10 m³/h.
2. Project a line horizontally to the right until it intersects the K_v = 16 (estimated).
3. Project a line vertically downwards and read the pressure drop from the 'X' axis (approximately 40 kPa or 0.4 bar).

Note: Before sizing valves for liquid systems, it is necessary to be aware of the characteristics of the system and its constituent apparatus such as pumps.

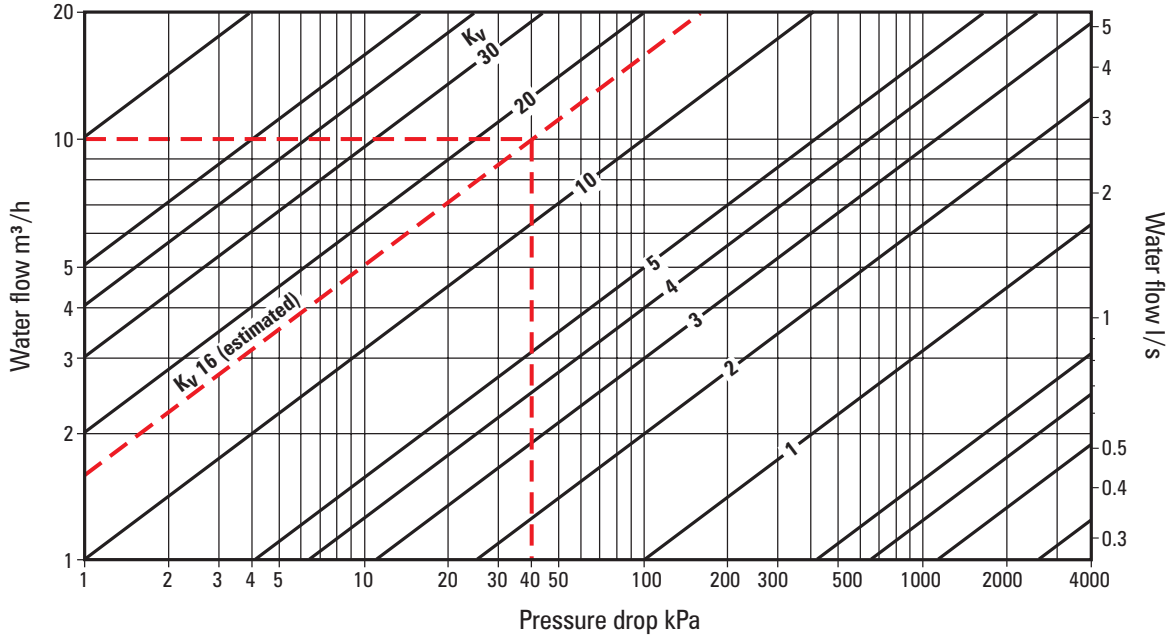


Fig. 6.3.1 Extract from the water K_v chart Figure 6.3.2

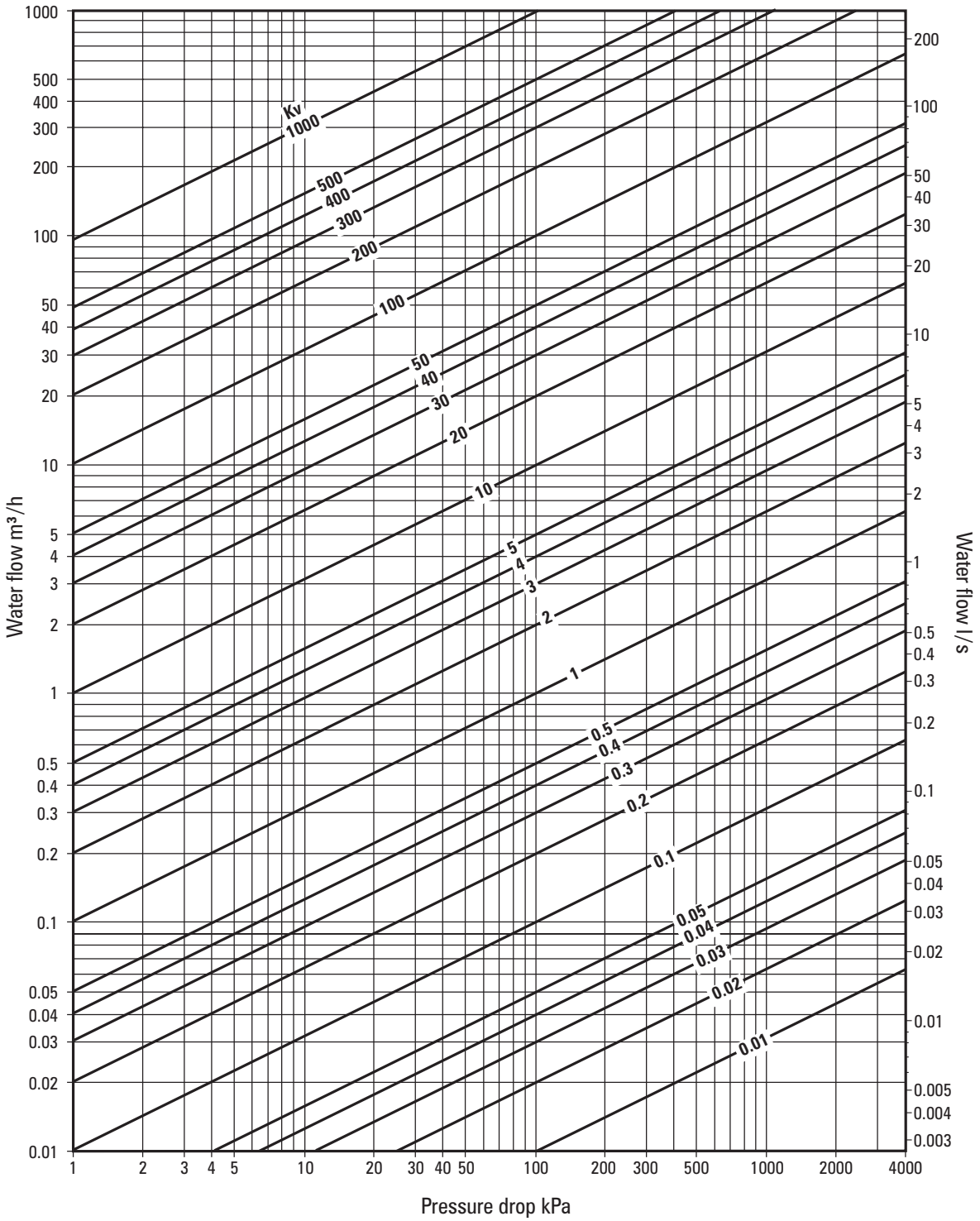


Fig. 6.3.2 Water K_v chart

Pumps

Unlike steam systems, liquid systems require a pump to circulate the liquid. Centrifugal pumps are often used, which have a characteristic curve similar to the one shown in Figure 6.3.3. Note that as the flowrate increases, the pump discharge pressure falls.

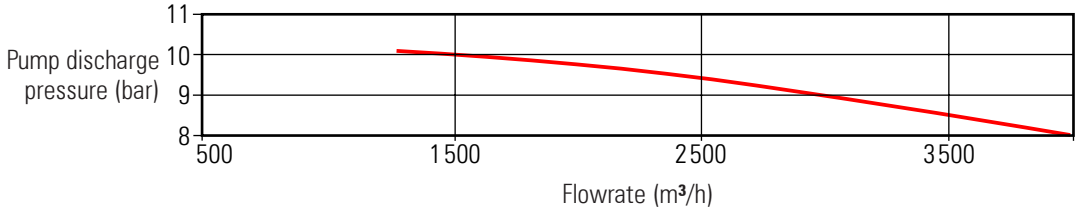


Fig. 6.3.3 Typical pump performance curve

Circulation system characteristics

It is important not only to consider the size of a water control valve, but also the system in which the water circulates; this can have a bearing on which type and size of valve is used, and where it should be positioned within the circuit.

As water is circulated through a system, it will incur frictional losses. These frictional losses may be expressed as pressure loss, and will increase in proportion to the square of the velocity. The flowrate can be calculated through a pipe of constant bore at any other pressure loss by using Equation 6.3.3, where $\dot{V}_1$ and $\dot{V}_2$ must be in the same units, and P_1 and P_2 must be in the same units. $\dot{V}_1$, $\dot{V}_2$, P_1 and P_2 are defined below.

$$\frac{\dot{V}_1^2}{\dot{V}_2^2} = \frac{P_1}{P_2} \qquad \textbf{Equation 6.3.3}$$

Where:
$\dot{V}_1$ = Flowrate at pressure loss P_1
$\dot{V}_2$ = Flowrate at pressure loss P_2

Example 6.3.2
It is observed that the flowrate ($\dot{V}_1$) through a certain sized pipe is 2 500 m³/h when the pressure loss (P_1) is 4 bar. Determine the pressure loss through the same size pipe (P_2) if the flowrate ($\dot{V}_2$) were 3 500 m³/h, using Equation 6.3.3.

$$\frac{\dot{V}_1^2}{\dot{V}_2^2} = \frac{P_2}{P_1}$$

$$P_2 = P_1 \times \frac{\dot{V}_1^2}{\dot{V}_2^2}$$

$$P_2 = 4 \times \frac{3500^2}{2500^2}$$

$$P_2 = 4 \times \frac{1225 \times 10^4}{625 \times 10^4}$$

$$\mathbf{P_2 = 7.84\ bar}$$

It can be seen that as more liquid is pumped through the same size pipe, the flowrate will increase. On this basis, a system characteristic curve, like the one shown in Figure 6.3.4, can be created using Equation 6.3.3, where the flowrate increases in accordance to the square law.

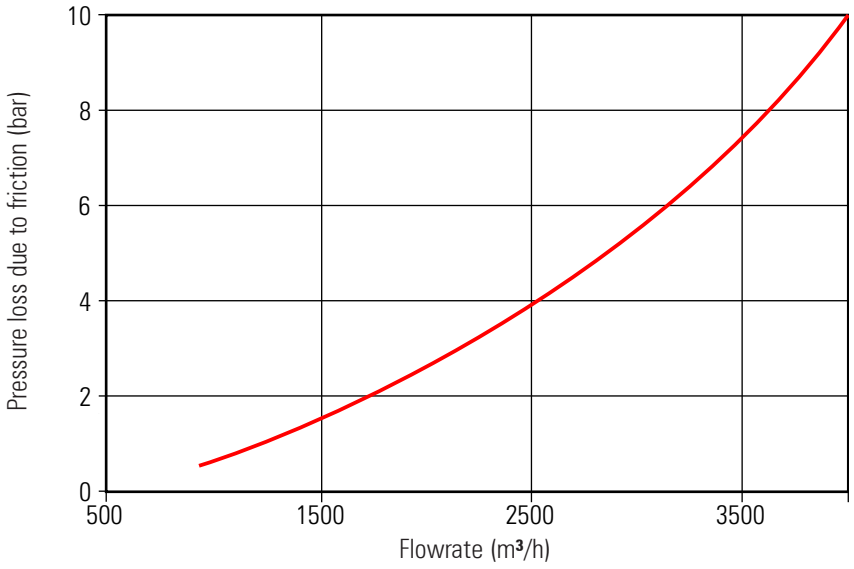


Fig. 6.3.4 Typical system curve

Actual performance

It can be observed from the pump and system characteristics, that as the flowrate and friction increase, the pump provides less pressure. A situation is eventually reached where the pump pressure equals the friction around the circuit, and the flowrate can increase no further.

If the pump curve and the system characteristic curve are plotted on the same chart - Figure 6.3.5, the point at which the pump curve and the system characteristic curve intersect will be the actual performance of the pump/circuit combination.

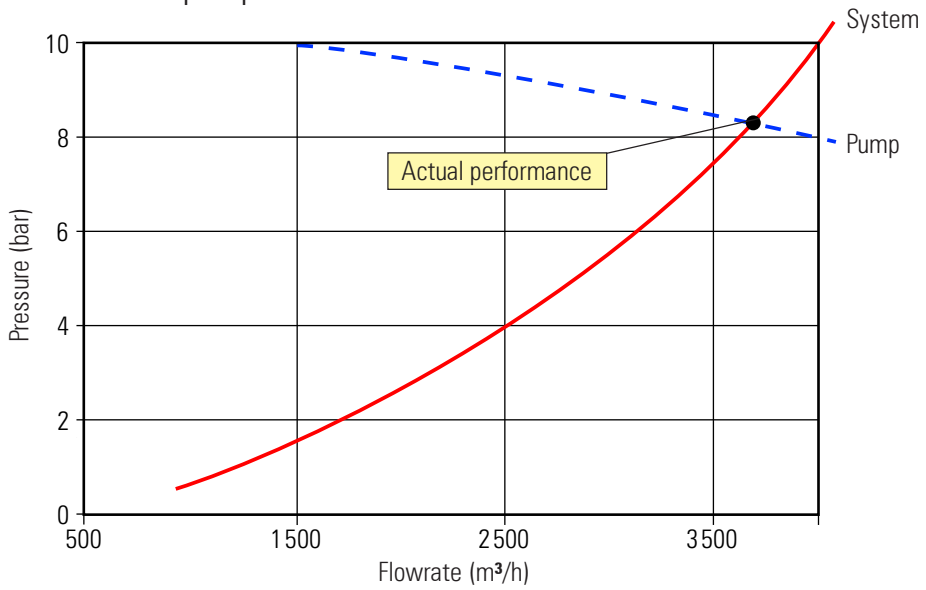
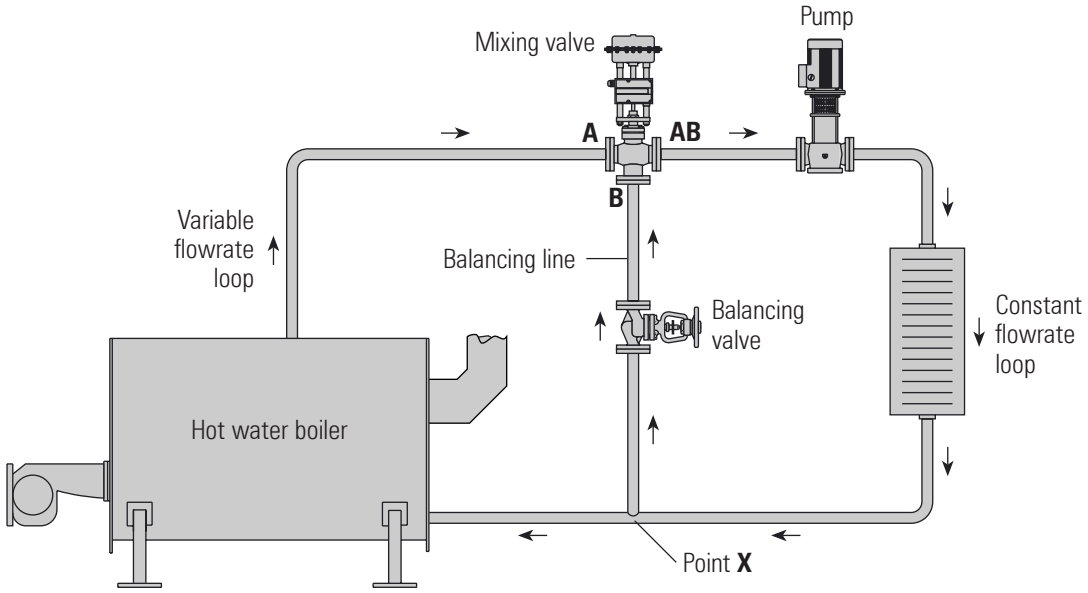


Fig. 6.3.5 Typical system performance curve

Three-port valve

A three-port valve can be considered as a constant flowrate valve, because, whether it is used to mix or divert, the total flow through the valve remains constant. In applications where such valves are employed, the water circuit will naturally split into two separate loops, constant flowrate and variable flowrate.

The simple system shown in Figure 6.3.6 depicts a mixing valve maintaining a constant flowrate of water through the 'load' circuit. In a heating system, the load circuit refers to the circuit containing the heat emitters, such as radiators in a building.



Resistance from Point **X** to Point **B** = Resistance from Point **X** to Point **A**

Fig. 6.3.6 Mixing valve (constant flowrate, variable temperature)

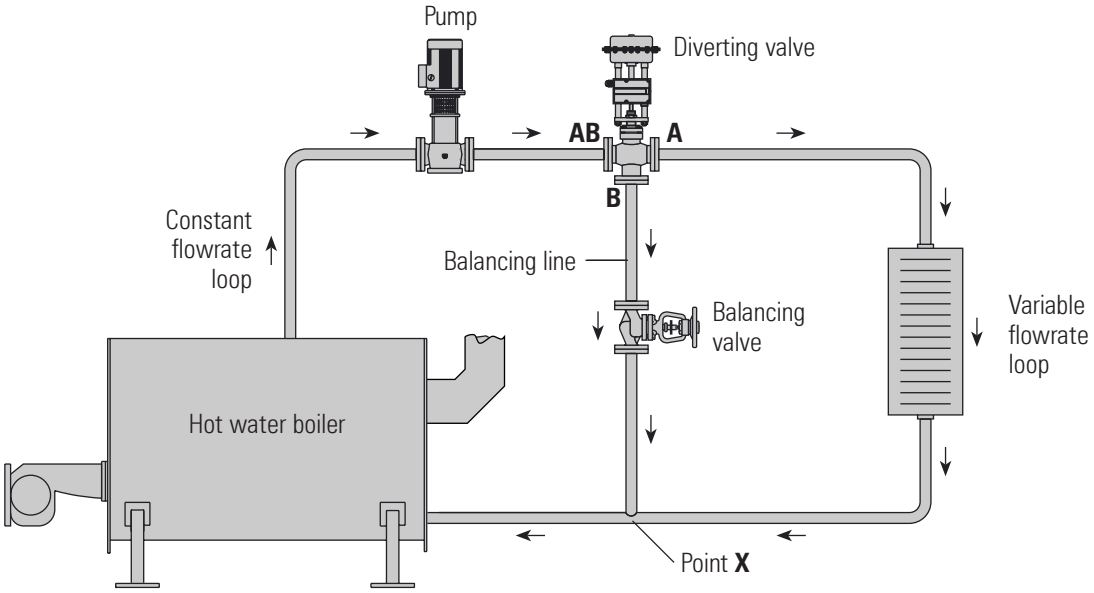
The amount of heat emitted from the radiators depends on the temperature of the water flowing through the load circuit, which in turn, depends upon how much water flows into the mixing valve from the boiler, and how much is returned to the mixing valve via the balancing line.

It is necessary to fit a balance valve in the balance line. The balance valve is set to maintain the same resistance to flow in the variable flowrate part of the piping network, as illustrated in Figures 6.3.6 and 6.3.7. This helps to maintain smooth regulation by the valve as it changes position.

In practice, the mixing valve is sometimes designed not to shut port **A** completely; this ensures that a minimum flowrate will pass through the boiler at all times under the influence of the pump. Alternatively, the boiler may employ a primary circuit, which is also pumped to allow a constant flow of water through the boiler, preventing the boiler from overheating.

The simple system shown in Figure 6.3.7 shows a diverting valve maintaining a constant flowrate of water through the constant flowrate loop. In this system, the load circuit receives a varying flowrate of water depending on the valve position.

The temperature of water in the load circuit will be constant, as it receives water from the boiler circuit whatever the valve position. The amount of heat available to the radiators depends on the amount of water flowing through the load circuit, which in turn, depends on the degree of opening of the diverting valve.



Resistance from Point **B** to Point **X** = Resistance from Point **A** to Point **X**

Fig. 6.3.7 Diverting valve (constant temperature in load circuit with variable flow)

The effect of not fitting and setting a balance valve can be seen in Figure 6.3.8. This shows the pump curve and system curve changing with valve position. The two system curves illustrate the difference in pump pressure required between the load circuit P_1 and the bypass circuit P_2, as a result of the lower resistance offered by the balancing circuit, if no balance valve is fitted. If the circuit is not correctly balanced then short-circuiting and starvation of any other sub-circuits (not shown) can result, and the load circuit may be deprived of water.

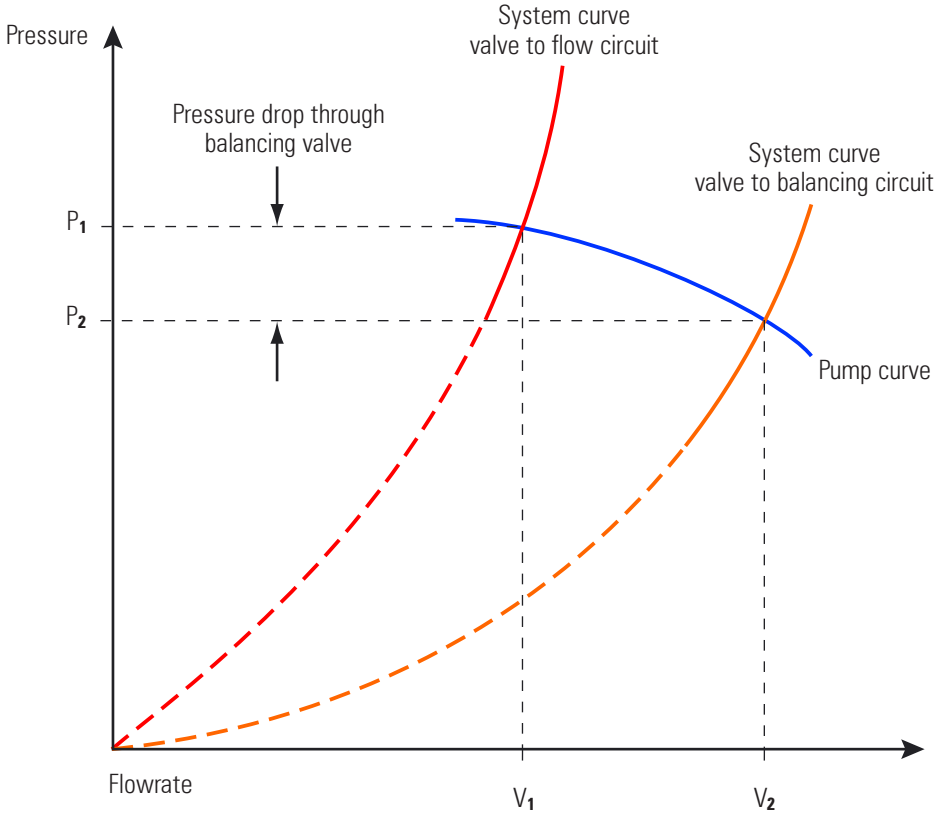


Fig. 6.3.8 Effect of not fitting a balance valve

Two-port Valves

When a two-port valve is used on a water system, as the valve closes, flow will decrease and the pressure upstream of the valve will increase. Changes in pump head will occur as the control valve throttles towards a closed position. The effects are illustrated in Figure 6.3.9.

A fall in flowrate not only increases the pump pressure but may also increase the power consumed by the pump. The change in pump pressure may be used as a signal to operate two or more pumps of varying duties, or to provide a signal to variable speed pump drive(s). This enables pumping rates to be matched to demand, saving pumping power costs.

Two port control valves are used to control water flow to a process, for example, for steam boiler level control, or to maintain the water level in a feedtank.

They may also be used on heat exchange processes, however, when the two-port valve is closed, the flow of water in the section of pipe preceding the control valve is stopped, creating a 'dead-leg'. The water in the dead-leg may lose temperature to the environment. When the control valve is opened again, the cooler water will enter the heat exchange coils, and disturb the process temperature. To avoid this situation, the control system may include an arrangement to maintain a minimum flow via a small bore pipe and adjustable globe valve, which bypass the control valve and load circuit.

Two-port valves are used successfully on large heating circuits, where a multitude of valves are incorporated into the overall system. On large systems it is highly unlikely that all the two-port valves are closed at the same time, resulting in an inherent 'self-balancing' characteristic. These types of systems also tend to use variable speed pumps that alter their flow characteristics relative to the system load requirements; this assists the self-balancing operation.

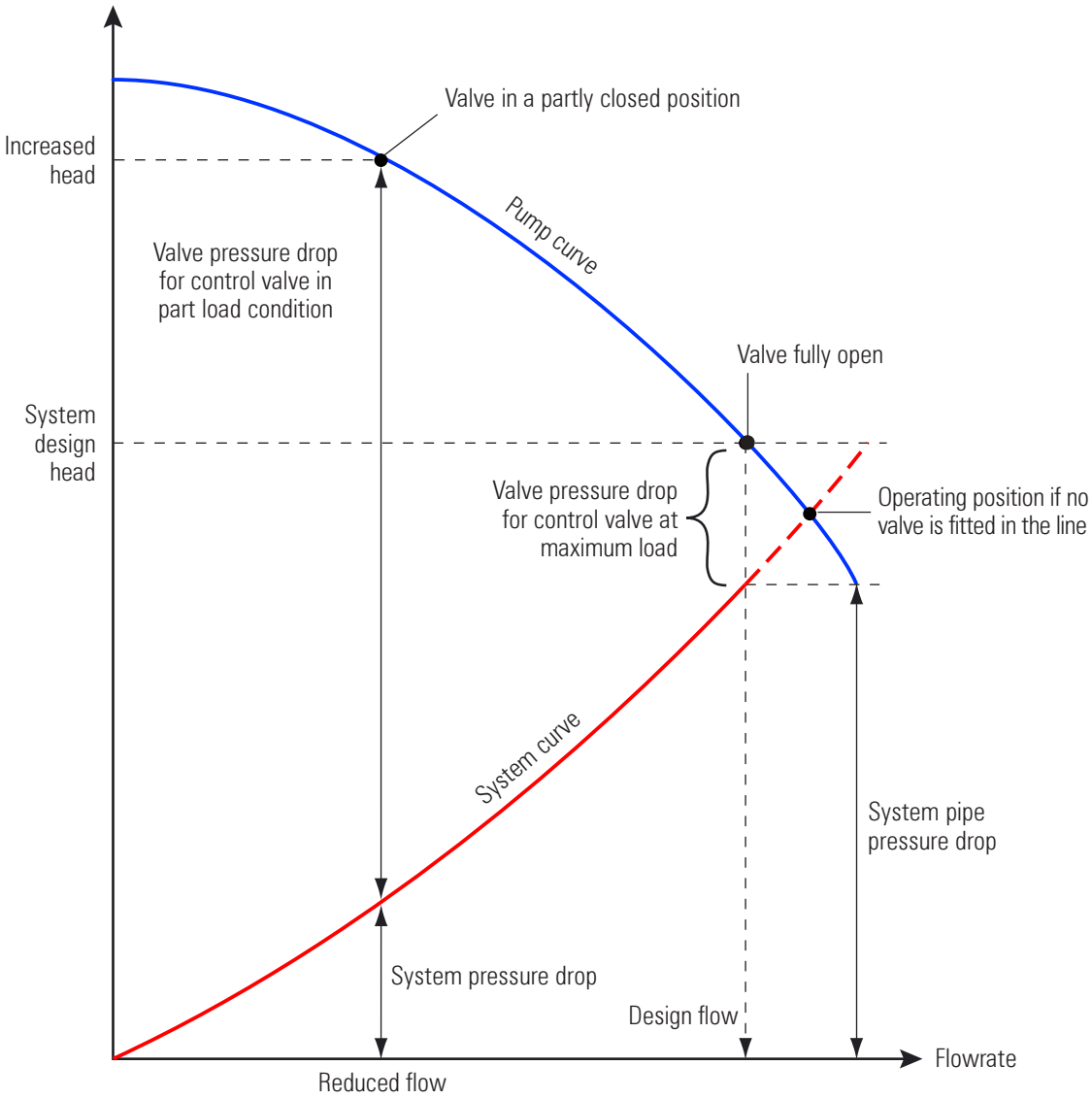


Fig. 6.3.9 Effect of two-port valve on pump head and pressure

When selecting a two-port control valve for an application:

- If a hugely undersized two-port control valve were installed in a system, the pump would use a large amount of energy simply to pass sufficient water through the valve.

  Assuming sufficient water could be forced through the valve, control would be accurate because even small increments of valve movement would result in changes in flowrate. This means that the entire travel of the valve might be utilised to achieve control.

- If a hugely oversized two-port control valve were installed in the same system, the energy required from the pump would be reduced, with little pressure drop across the valve in the fully open position.

However, the initial valve travel from fully open towards the closed position would have little effect on the flowrate to the process. When the point was reached where control was achieved, the large valve orifice would mean that very small increments of valve travel would have a large effect on flowrate. This could result in erratic control with poor stability and accuracy.

A compromise is required, which balances the good control achieved with a small valve against the reduced energy loss from a large valve. The choice of valve will influence the size of pump, and the capital and running costs. It is good practice to consider these parameters, as they will have a bearing on the overall lifetime cost of the system.

These balances can be realised by calculating the 'valve authority' relative to the system in which it is installed.

Valve authority

Valve authority may be determined using Equation 6.3.4.

$$N = \frac{\Delta P_1}{\Delta P_1 + \Delta P_2} \qquad \textbf{Equation 6.3.4}$$

Where:

N = Valve authority
ΔP_1 = Pressure drop across a fully open control valve
ΔP_2 = Pressure drop across the remainder of the circuit
$\Delta P_1 + \Delta P_2$ = Pressure drop across the whole circuit

The value of N should be near to 0.5 (but not greater than), and certainly not lower than 0.2.

This will ensure that each increment of valve movement will have an effect on the flowrate without excessively increasing the cost of pumping power.

Example 6.3.3

A circuit has a total pressure drop ($\Delta P_1 + \Delta P_2$) of 125 kPa, which includes the control valve.

a) If the control valve must have a valve authority (N) of 0.4, what pressure drop is used to size the valve?

b) If the circuit/system flowrate ($\dot{V}$) is 3.61 l/s, what is the required valve K_v?

Part a) Determine the ΔP

$$N = \frac{\Delta P_1}{\Delta P_1 + \Delta P_2} \qquad \textbf{Equation 6.3.4}$$

$$N = 0.4$$

$$\Delta P_1 + \Delta P_2 = 125 \text{ kPa}$$

$$N = \frac{\Delta P_1}{\Delta P_1 + \Delta P_2}$$

$$\Delta P_1 = N\,(\Delta P_1 + \Delta P_2)$$

$$\Delta P_1 = 0.4 \times 125 \text{ kPa}$$

$$\mathbf{\Delta P_1 = 50 \text{ kPa}}$$

Consequently, a valve ΔP of 50 kPa is used to size the valve, leaving 75 kPa (125 kPa - 50 kPa) for the remainder of the circuit.

Part b) Determine the required K_v

$$\dot{V} = K_v\sqrt{\Delta P} \qquad \textbf{Equation 6.3.2}$$

Where:

$\dot{V}$ = 3.61 l/s (13m³/h)
ΔP = 50 kPa (0.5 bar)

$$13 = K_v\sqrt{0.5}$$

$$K_v = \frac{13}{\sqrt{0.5}}$$

$$\mathbf{K_v = 18.38}$$

Alternatively, the water K_v chart (Figure 6.3.2) may be used.

Three-port control valves and valve authority

Three-port control valves are used in either mixing or diverting applications, as explained previously in this Module. When selecting a valve for a diverting application:

- A hugely undersized three-port control valve will incur high pumping costs, and small increments of movement will have an effect on the quantity of liquid directed through each of the discharge ports.
- A hugely oversized valve will reduce the pumping costs, but valve movement at the beginning, and end, of the valve travel will have minimal effect on the distribution of the liquid. This could result in inaccurate control with large sudden changes in load. An unnecessarily oversized valve will also be more expensive than one adequately sized.

The same logic can be applied to mixing applications.

Again, the valve authority will provide a compromise between these two extremes.

With three-port valves, valve authority is always calculated using P_2 in relation to the circuit with the variable flowrate. Figure 6.3.10 shows this schematically.

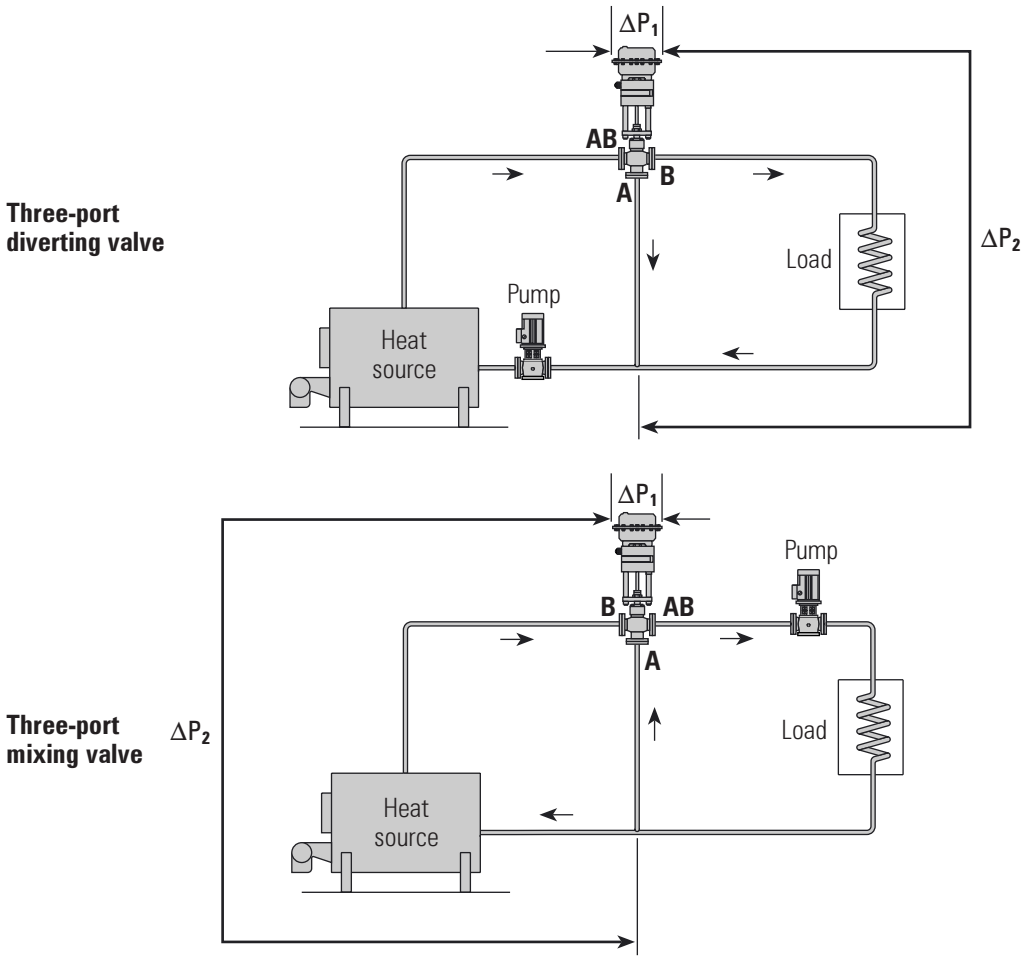


Fig. 6.3.10 Valve authority diagrams showing three-port valves

Note: Because mixing and diverting applications use three-port valves in a 'balanced' circuit, the pressure drop expected over a three-port valve is usually significantly less than with a two-port valve.

As a rough guide:

- A three-port valve will be 'line sized' when based on water travelling at recommended velocities (Typically ranging from 1 m/s at DN25 to 2 m/s at DN150).
- 10 kPa may be regarded as typical pressure drop across a three-port control valve.
- Aim for valve authority (N) to be between 0.2 and 0.5, the closer to 0.5 the better.

Cavitation and flashing

Other symptoms sometimes associated with water flowing through two-port valves are due to 'cavitation' and 'flashing'.

Cavitation in liquids

Cavitation can occur in valves controlling the flow of liquid if the pressure drop and hence the velocity of the flow is sufficient to cause the local pressure after the valve seat to drop below the vapour pressure of the liquid. This causes vapour bubbles to form. Pressure may then recover further downstream causing vapour bubbles to rapidly collapse. As the bubbles collapse very high local pressures are generated which, if adjacent to metal surfaces can cause damage to the valve trim, the valve body or downstream pipework. This damage typically has a very rough, porous or sponge-like appearance which is easily recognised. Other effects which may be noticed include noise, vibration and accelerated corrosion due to the repeated removal of protective oxide layers.

Cavitation will tend to occur in control valves:

- On high pressure drop applications, due to the high velocity in the valve seat area causing a local reduction in pressure.
- Where the downstream pressure is not much higher than the vapour pressure of the liquid. This means that cavitation is more likely with hot liquids and/or low downstream pressure.

Cavitation damage is likely to be more severe with larger valves sizes due to the increased power in the flow.

Flashing in liquids

Flashing is a similar symptom to cavitation, but occurs when the valve outlet pressure is lower than the vapour pressure condition. Under these conditions, the pressure does not recover in the valve body, and the vapour will continue to flow into the connecting pipe. The vapour pressure will eventually recover in the pipe and the collapsing vapour will cause noise similar to that experienced with cavitation. Flashing will reduce the capacity of the valve due to the throttling effect of the vapour having a larger volume than the water. Figure 6.3.11 illustrates typical pressure profiles through valves due to the phenomenon of cavitation and flashing.

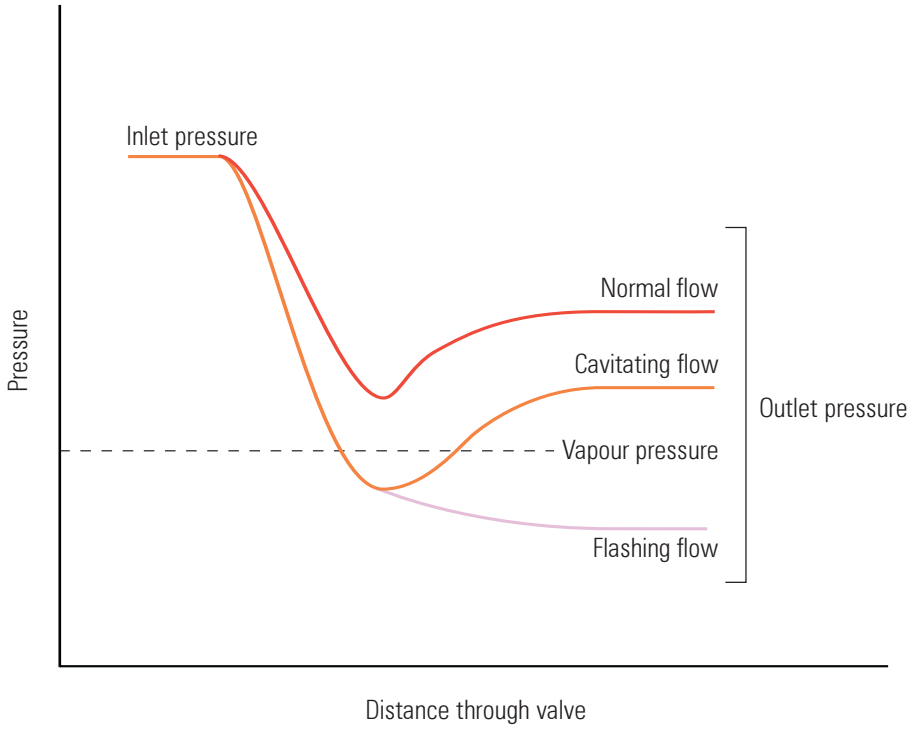


Fig. 6.3.11 Cavitation and flashing through a water control valve

Avoiding cavitation

It is not always possible to ensure that the pressure drop across a valve and the temperature of the water is such that cavitation will not occur. Under these circumstances, one possible solution is to install a valve with a valve plug and seat especially designed to overcome the problem. Such a set of internals would be classified as an 'anti-cavitation' trim.

The anti-cavitation trim consists of the standard equal percentage valve plug operating inside a valve seat fitted with a perforated cage. Normal flow direction is used. The pressure drop is split between the characterised plug and the cage which limits the pressure drop in each stage and hence the lowest pressures occur. The multiple flow paths in the perforated cage also increase turbulence and reduce the pressure recovery in the valve. These effects both act to prevent cavitation occuring in case of minor cavitation, or to reduce the intensity of cavitation in slightly more severe conditions. A typical characterised plug and cage are shown in Figure 6.3.12.

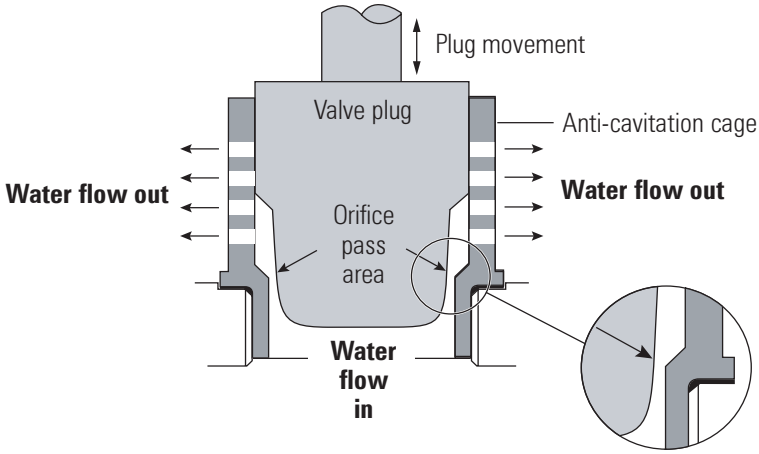


Fig. 6.3.12 A typical two-port valve anti-cavitation trim

The pressure drop is split between the orifice pass area and the cage. In many applications the pressure does not drop below the vapour pressure of the liquid and cavitation is avoided. Figure 6.3.12 shows how the situation is improved.

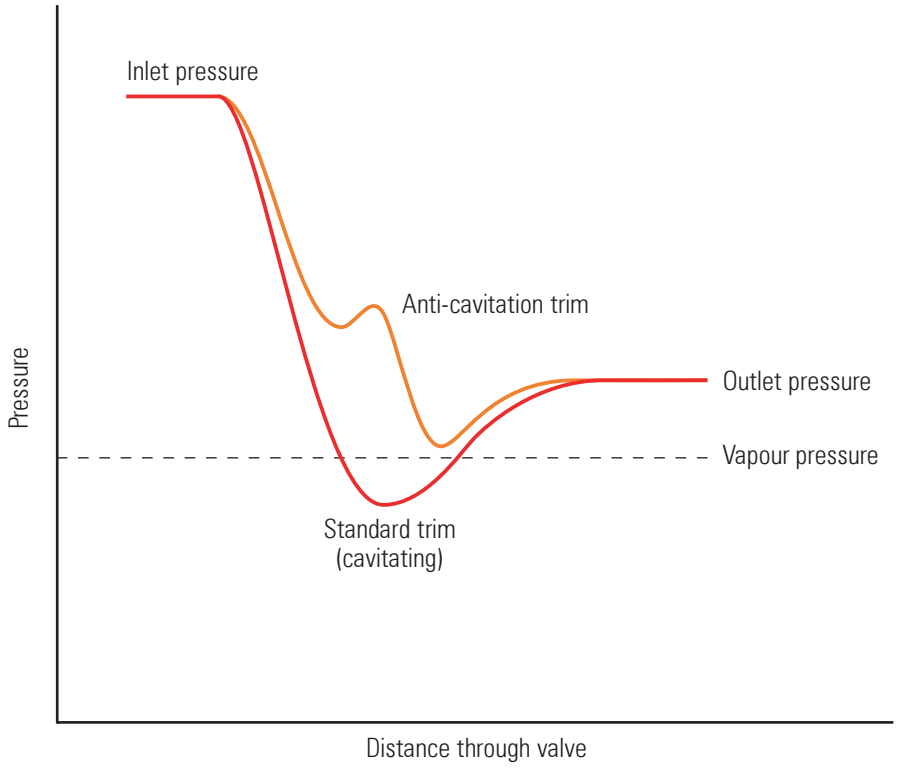
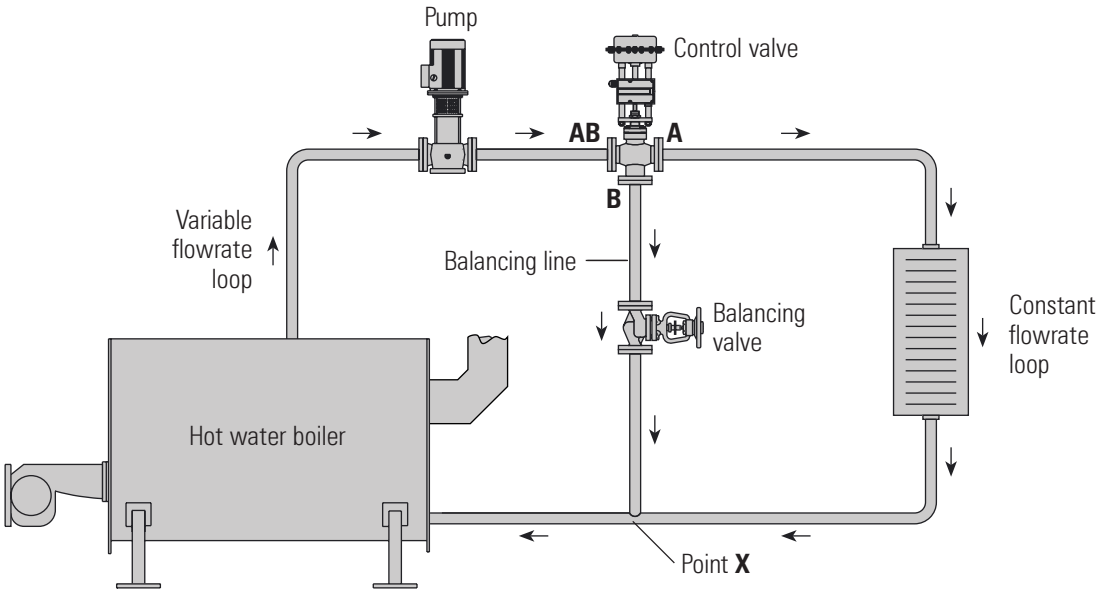


Fig. 6.3.13 Cavitation is alleviated by anti-cavitation valve trim

Questions

1. In the arrangement shown below, what will be the effect of omitting the balance valve?



a| The pump curve will change as the control valve diverts more of the flow through the balancing pipe ☐

b| Short circuiting and starvation of water to the process ☐

c| The pump must be repositioned to the process outlet ☐

d| None ☐

2. What is the optimum range of valve authority?

a| 0 – 0.2 ☐

b| 0.2 – 1.0 ☐

c| 0.5 – 1.0 ☐

d| 0.2 – 0.5 ☐

3. Calculate the valve authority if $\Delta P_1 = 15$ kPa and $\Delta P_2 = 45$ kPa

a| 0.75 ☐

b| 0.25 ☐

c| 0.33 ☐

d| 3.0 ☐

4. Water flowing through a fully open valve at a rate of 5 m^3/h creates a differential pressure of 0.25 bar across the valve. What is the valve K_{vs}?

a| 20 ☐

b| 1.25 ☐

c| 10 ☐

d| 80 ☐

5. It is noticed that the pressure loss along a certain sized pipe is 1.0 bar when the flowrate of water is 1 L/s. Using Equation 6.3.3, determine the flowrate of water along the same pipe if the pressure loss falls to 0.75 bar.

a| 1.155 L/s ☐

b| 0.500 L/s ☐

c| 1.333 L/s ☐

d| 0.866 L/s ☐

6. What are the two basic configurations for which a three-port valve is used?

a| Hot and cold ☐

b| Flow and return ☐

c| Series and parallel ☐

d| Mixing and diverting ☐

Answers

1: a, 2: d, 3: b, 4: c, 5: d, 6: d

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Module 6.4

Control Valve Sizing for Steam Systems

Control Valve Sizing for Steam Systems

Before discussing the sizing of control valves for steam systems, it is useful to review the characteristics of steam in a heat transfer application.

- Steam is supplied at a specific pressure to the upstream side of the control valve through which it passes to a heat exchanger, also operating at a specific pressure.
- Steam passes through the control valve and into the steam space of the equipment where it comes into contact with the heat transfer surfaces.
- Steam condenses on the heat transfer surfaces, creating condensate.
- The volume of condensate is very much less than steam. This means that when steam condenses, the pressure in the steam space is reduced.
- The reduced pressure in the steam space means that a pressure difference exists across the control valve, and steam will flow from the high-pressure zone (upstream of the control valve) to the lower pressure zone (the steam space in the equipment) in some proportion to the pressure difference and, ideally, balancing the rate at which steam is condensing.
- The rate of steam flow into the equipment is governed by this pressure difference and the valve orifice size. Should, at any time, the flowrate of steam through the valve be less than the condensing rate (perhaps the valve is too small), the steam pressure and the heat transfer rate in the heat exchanger will fall below that which is required; the heat exchanger will not be able to satisfy the heat load.
- If a modulating control system is used, as the temperature of the process approaches the controller set point, the controller will close the valve by a related amount, thereby reducing the steam flowrate to maintain the lower pressure required to sustain a lower heat load. (The action of opening and closing the valve is often referred to as increasing or decreasing the 'valve lift'; this is explained in more detail in Module 6.5, 'Control Valve Characteristics').
- Closing the valve reduces the mass flow. The steam pressure falls in the steam space and so too the steam temperature. This means that a smaller difference in temperature exists between the steam and the process, so the rate of heat transfer is reduced, in accordance with Equation 2.5.3.

$$\dot{Q} = U\ A\ \Delta T_M \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W (J/s))
U = Overall heat transfer coefficient (W/m²°C)
A = Heat transfer area (m²)
ΔT_M = Mean temperature difference between the steam and secondary fluid (°C)

The overall heat transfer coefficient (U) does not change very much during the process, and the area (A) is fixed, so if the mean temperature difference (ΔT_M) is reduced, then the heat transfer from the steam to the secondary fluid is also reduced.

Saturated steam flow through a control valve

A heat exchanger manufacturer will design equipment to give a certain heat output. To achieve this heat output, a certain saturated steam temperature will be required at the heat transfer surface (such as the inside of a heating coil in a shell and tube heat exchanger). With saturated steam, temperature and pressure are strictly related; therefore controlling the steam pressure easily regulates the temperature.

Consider an application where steam at 10 bar g is supplied to a control valve, and a given mass flow of steam passes through the valve to a heat exchanger. The valve is held fully open (see Figure 6.4.1).

- If a DN50 valve is fitted and the valve is fully open, the pressure drop is relatively small across the valve, and the steam supplied to the heat exchanger is at a fairly high pressure (and temperature). Because of this, the heating coil required to achieve the design load is relatively small.
- Consider now, a fully open DN40 valve in the steam supply line passing the same flowrate as the DN50 valve. As the valve orifice is smaller the pressure drop across the valve must be greater, leading to a lower pressure (and temperature) in the heat exchanger. Because of this, the heat transfer area required to achieve the same heat load must be increased. In other words, a larger heating coil or heat exchanger will be required.
- Further reduction of the valve size will require more pressure drop across the control valve for the same mass flow, and the need for an increased heat transfer surface area to maintain the same heat output.

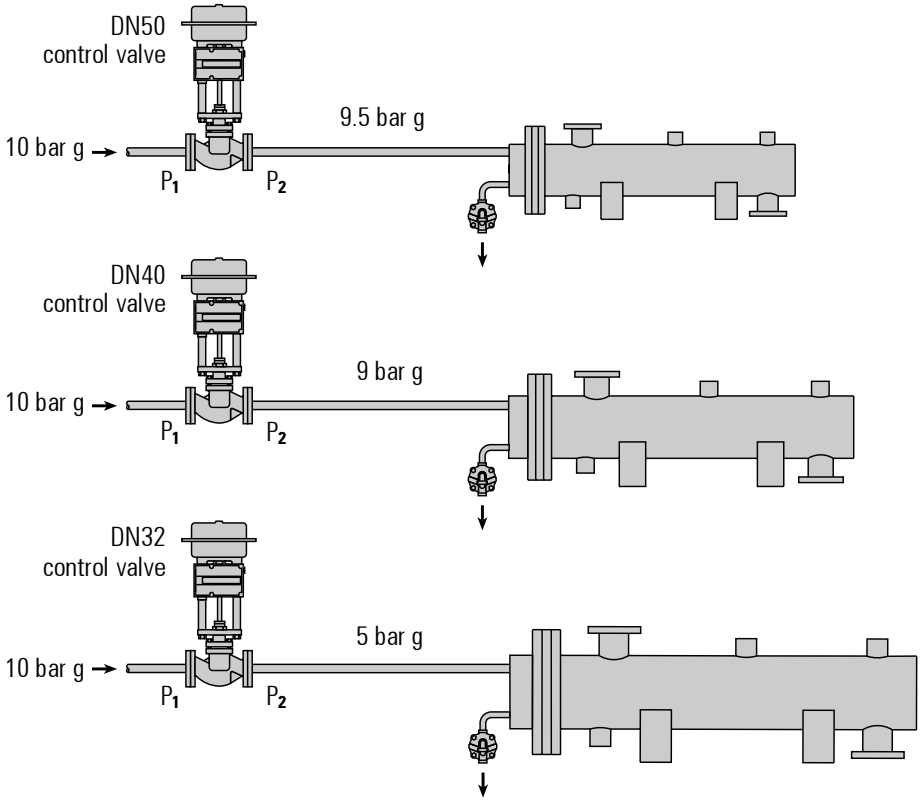


Fig. 6.4.1 Flow through a fully open control valve

Whatever the size of the control valve, if the process demand is reduced, the valve must modulate from the fully open position towards closed. However, the first part of the travel has only a small regulating effect, with any percentage change in valve lift producing a lesser percentage change in flowrate. Typically, a 10% change in lift might produce only a 5% change in flowrate. With further travel, as the valve plug approaches the seat, this effect reverses such that perhaps a 5% change in lift might produce a 10% change in flowrate, and better regulation is achieved.

The initial part of the control valve travel, during which this lowered control effect is seen, is greater with the selection of the larger control valves and the accompanying small pressure drop at full load. When the control valve chosen is small enough to require a 'critical pressure drop' at full load the effect disappears. Critical pressure is explained in the Section below.

Further, if a larger control valve is selected, the greater size of the valve orifice means that a given change in flowrate is achieved with a smaller percentage change in lift than is needed with a smaller control valve.

This can often make the control unstable, increasing the possibility of 'hunting', especially on reduced loads.

Critical pressure

The mass flow of steam passing through the valve will increase in line with differential pressure until a condition known as 'critical pressure' is reached. The principle can be explained by looking at how nozzles work and how they compare to control valves.

Consider an almost perfect orifice, such as a convergent-divergent nozzle shown in Figure 6.4.2. Its shape, if designed correctly to match the upstream and downstream pressure conditions and the condition of the supplied steam, will allow it to operate at high efficiency.

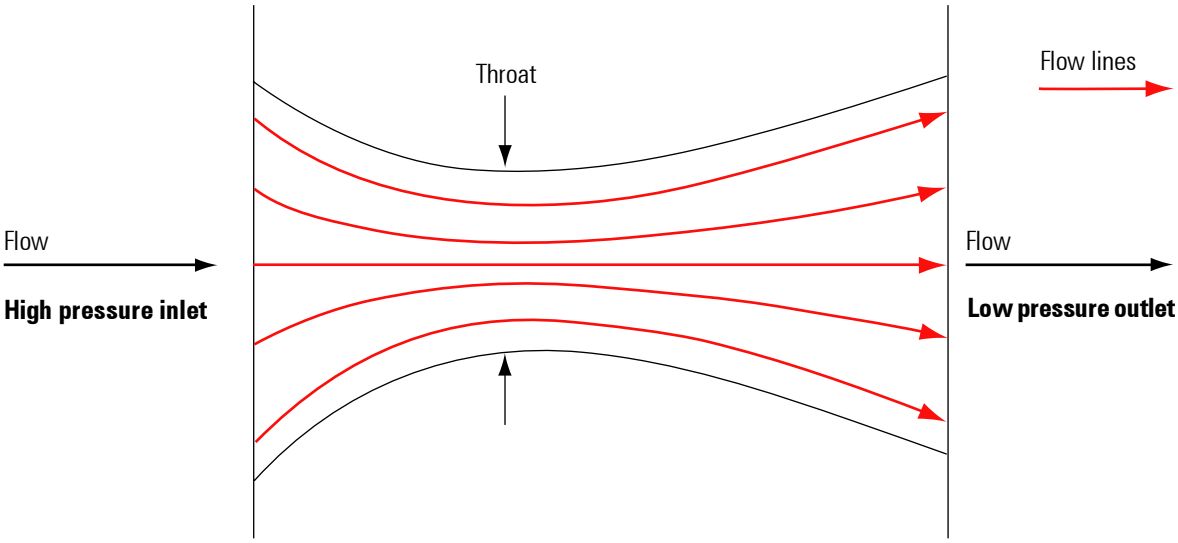


Fig. 6.4.2 A convergent-divergent nozzle

Such a nozzle can be thought of as a type of heat engine, changing heat energy into mechanical (kinetic) energy. It is designed to discharge the required weight of steam with a given pressure drop, and with minimum turbulence and friction losses.

In the convergent section, the steam velocity increases as the pressure falls, though the specific volume of the steam also increases with the lowered pressures. At first, the velocity increases more quickly than the specific volume, and the required flow area through this part of the nozzle becomes less. At a certain point, the specific volume begins to increase more rapidly than does the velocity and the flow area must become greater. At this point, the steam velocity will be sonic and the flow area is at a minimum. The steam pressure at this minimum flow area or 'throat' is described as the 'critical pressure', and the ratio of this pressure to the initial (absolute) pressure is found to be close to 0.58 when saturated steam is passing.

Critical pressure varies slightly according to the fluid properties, specifically in relation to the ratio of the specific heats c_p/c_v of the steam (or other gaseous fluid), which is termed the adiabatic index or isentropic exponent of the fluid, often depicted by the symbols 'n', 'k' or 'γ'. With superheated steam the ratio is about 0.55, and for air about 0.53.

Point of interest:
Critical pressure ratio can also be determined by Equation 6.4.1.

$$\text{Critical pressure ratio} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}} \qquad \textbf{Equation 6.4.1}$$

γ can be taken from the Spirax Sarco website steam table. If this is not available, an approximation can be made as follows:

Wet steam: $\gamma = 1.035 + 0.1(x)$
where 'x' is dryness fraction, $0.8 > x > 1$.

Dry saturated steam: $\gamma = 1.135$

Superheated steam: $\gamma = 1.3$

For dry saturated steam, using Equation 6.4.1:

$$\text{Critical pressure ratio} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135 - 1}}$$

$$= \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}}$$

$$= (0.937)^{8.41}$$

$$= 0.58$$

Clearly, the mass flow through the throat of a given size is at a maximum at this 'critical pressure drop'. To achieve a greater flow, either:

a. The velocity would have to be greater, which could only be reached with a greater pressure drop – but this would also increase the specific volume by an even greater amount, or:

b. The specific volume would have to be less, which could only be the case with a lesser pressure drop – but this would reduce the velocity by an even greater amount.

Thus, once the critical pressure drop is reached at the throat of the nozzle, or at the 'vena contracta' when an orifice is used, further lowering of the downstream pressure cannot increase the mass flow through the device.

If the pressure drop across the whole nozzle is greater than the critical pressure drop, critical pressure will always occur at the throat. The steam will expand after passing the throat such that, if the outlet area has been correctly sized, the required downstream pressure is achieved at the nozzle outlet, and little turbulence is produced as the steam exits the nozzle at high velocity.

Should the nozzle outlet be too big or too small, turbulence will occur at the nozzle outlet, reducing capacity and increasing noise:

- If the nozzle outlet is too small, the steam has not expanded enough, and has to continue expanding outside the nozzle until it reaches the required downstream pressure in the low pressure region.
- If the nozzle outlet is too large, the steam will expand too far in the nozzle and the steam pressure in the nozzle outlet will be lower than the required pressure, causing the steam to recompress outside the outlet in the low pressure region.

The shape of the nozzle (Figure 6.4.3) is gently contoured such that the vena contracta occurs at the nozzle throat. (This is in contrast to a sharp-edged orifice, where a vena contracta occurs downstream of the orifice. The vena contracta effect is discussed in more detail in Module 4.2 'Principles of Flowmetering').

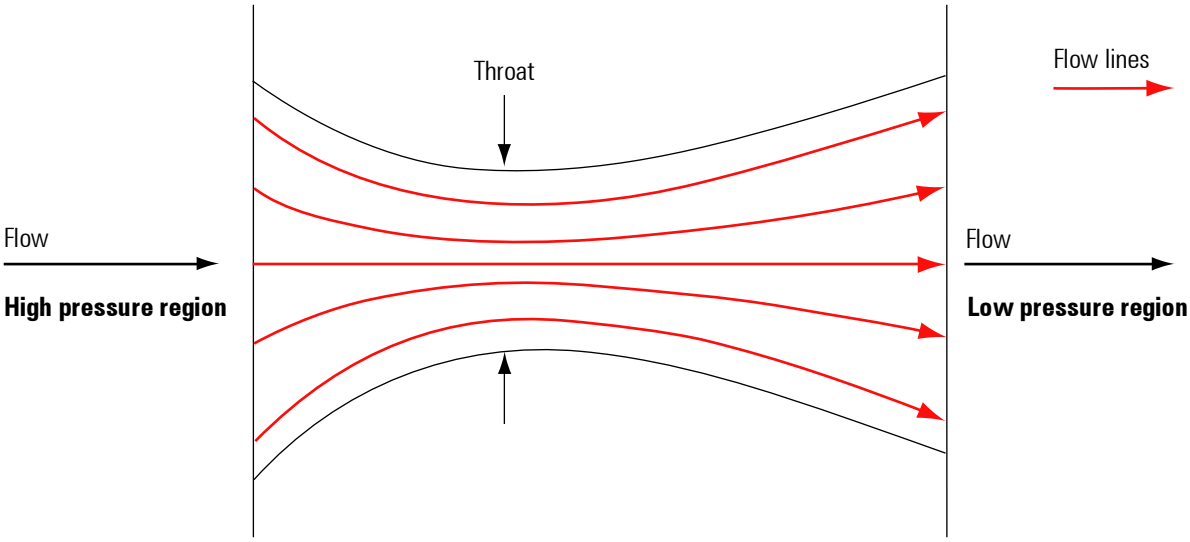


Fig. 6.4.3 The convergent-divergent nozzle

Control valves can be compared to convergent-divergent nozzles, in that each has a high-pressure region (the valve inlet), a convergent area (the inlet between the valve plug and its seat), a throat (the narrowest gap between the valve plug and its seat), a divergent area (the outlet from the valve plug and its seat, and a low-pressure region (the downstream valve body). See Figure 6.4.4.

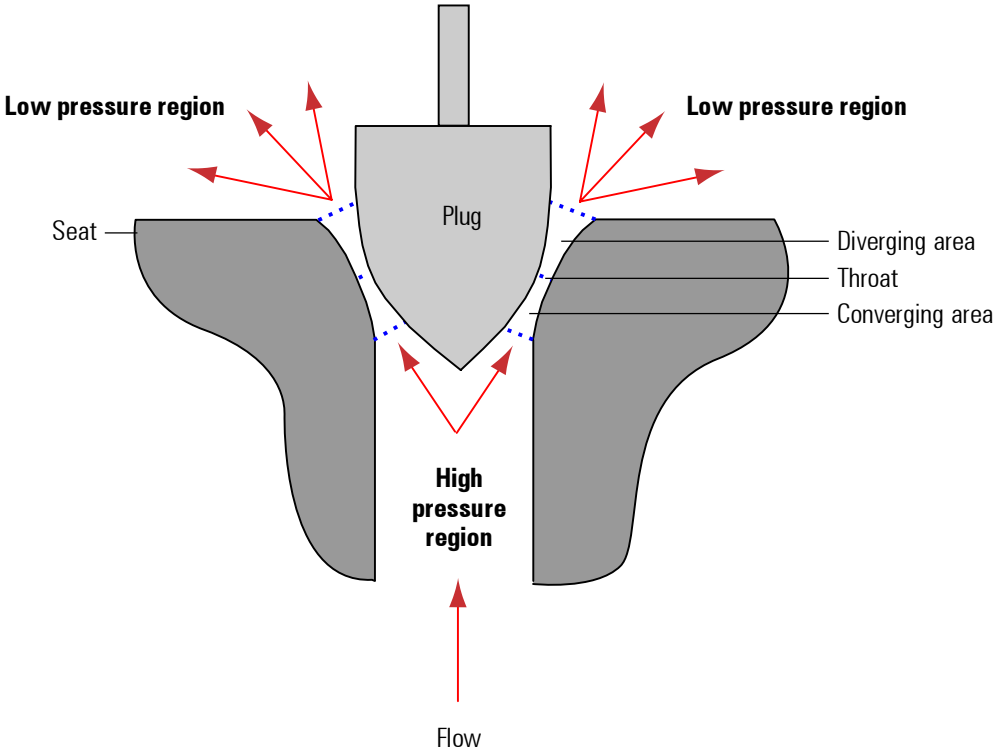


Fig. 6.4.4 The convergent-divergent principle in a control valve

Nozzles and control valves have different purposes. The nozzle is primarily designed to increase steam velocity in order to produce work (perhaps to turn a turbine blade), so the velocity of steam leaving the nozzle is required to remain high.

In contrast, the control valve is a flow restricting or 'throttling' device designed to produce a significant pressure drop in the steam. The velocity of steam passing out of a control valve throat will behave in a similar fashion to that of the steam passing out of the throat of a convergent-divergent nozzle; in that it will increase as the steam expands in the diverging area between the plug and seat immediately after the throat. If the pressure drop across the valve is greater than critical pressure drop, the steam velocity will increase to supersonic in this area, as the pressure here is less than that at the throat.

Past this point, the steam passes into the relatively large chamber encased by the valve body (the low pressure region), which is at a higher pressure due to the backpressure imposed by the connecting pipework, causing the velocity and kinetic energy to fall rapidly. In accordance with the steady flow energy equation (SFEE), this increases the steam enthalpy to almost that at the valve entrance port. A slight difference is due to energy lost to friction in passing through the valve.

From this point, the valve body converges to port the steam flow to the valve outlet, and the pressure (and density) approach the pressure (and density) in the downstream pipe. As this pressure stabilises, so does the velocity, relative to the cross sectional area of the valve outlet port.

The relative change in volume through the valve is represented by the dotted lines in the schematic diagram shown in Figure 6.4.5.

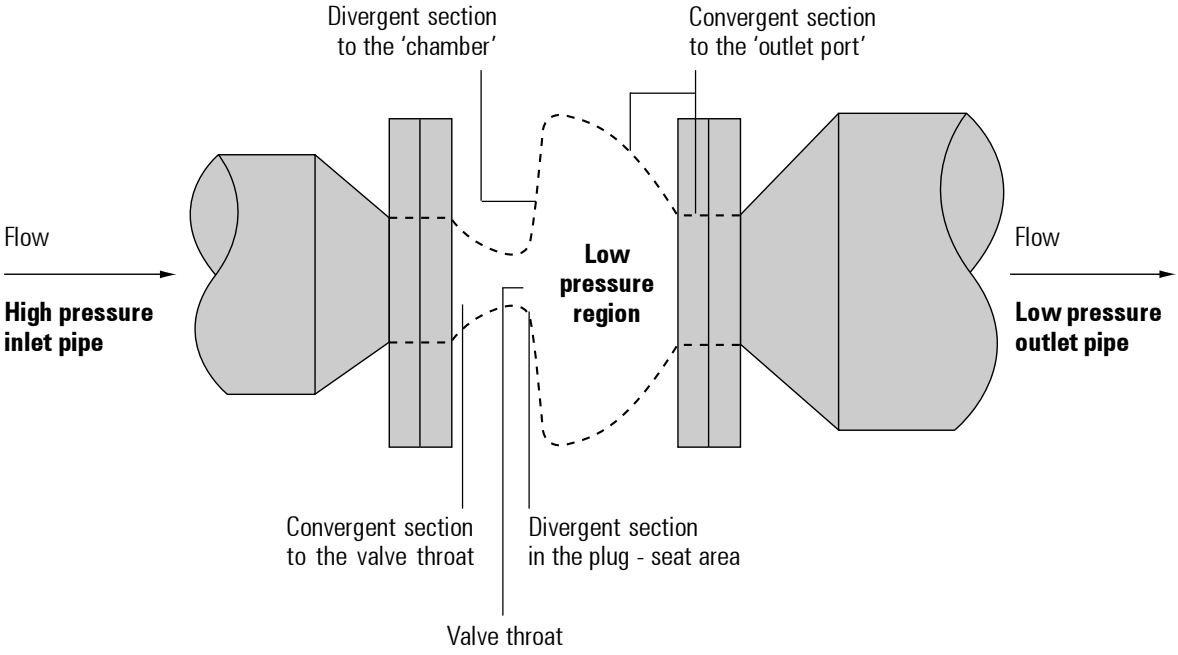


Fig. 6.4.5 The convergent-divergent-convergent valve body

When the pressure drop across a valve is greater than critical, noise can be generated by the large instantaneous exchange from kinetic energy to heat energy in the low pressure region, sometimes exacerbated by the presence of supersonic steam.

Valve outlet velocity, noise, erosion, drying and superheating effect

Noise can be an important consideration when sizing control valves, not only because it creates increased sound levels but because its associated vibration can damage valve internals. Special noise-reducing valve trims are available but, sometimes, a less expensive solution is to fit a larger valve body than required. Complicated equations are required to calculate noise emitted from control valves and these are difficult to use manually. It is usually considered that the control valve will produce unacceptable noise if the velocity of dry saturated steam in the control valve outlet is greater than 0.3 Mach. The speed of sound in steam will depend upon the steam temperature and the quality of the steam, but can be calculated from Equation 6.4.2 if the conditions are known (Mach 1 = speed of sound).

$$C = 31.6 \sqrt{\gamma \, R \, T}$$ **Equation 6.4.2**

Where:
C = Speed of sound in steam (m/s)
31.6 = Constant of proportionality
γ = Steam isentropic exponent (1.135 : saturated, 1.3 : superheated)
R = 0.4615 the gas constant for steam (kJ/kg)
T = Absolute steam temperature (K)

A less accurate but useful method to estimate whether noise will be a problem is by calculating the velocity in the valve outlet port. In simplistic terms and for dry saturated steam, if this is greater than 150 m/s, there is a chance that the valve body is too small (even though the valve trim size suits the required capacity). Higher velocities also cause erosion in the downstream valve body, especially if the steam is wet at this point. It is recommended that the maximum exit velocity for wet steam is 40 m/s in the outlet port.

Another result of dropping steam pressure across a control valve is to dry or superheat the steam, depending upon its condition as it enters the valve. Large degrees of superheat are usually unwanted in heating processes, and so it is useful to be able to determine if this will occur. Superheated steam (and dry gas) velocities, however, may be allowed to reach 0.5 Mach in the outlet port; whereas, at the other end of the scale, liquids might be restricted to a maximum outlet velocity of 10 m/s.

Example 6.4.1 The valve outlet velocity and drying/superheating effect

A control valve is supplied with dry saturated steam from a separator at 12 bar g and used to drop steam pressure to 4 bar g at full load. The full load flowrate is 1300 kg/h requiring a K_{vr} of 8.3. A DN25 (1″) valve is initially considered for selection, which has a K_{vs} of 10 and a valve outlet area of 0.000 49 m². What is the steam velocity in the valve outlet?

Determine the state of the steam in the valve outlet at 4 bar g.

The degree of drying and superheating can be calculated from the following procedure:

From steam tables, total heat (h_g) in the upsteam dry saturated steam at 12 bar g = 2 787 kJ/kg

As the supply steam is in a dry saturated state, the steam will certainly be superheated after it passes through the valve; therefore the superheated steam table should be used to quantify its properties.

Using the Spirax Sarco website steam tables, it is possible to calculate the condition of the downstream steam at 4 bar g by selecting 'Superheated steam' and entering a pressure of '4 bar g' and a total heat (h) of 2 787 kJ/kg.

By entering these values, the steam table returns the result of superheated steam at 4 bar g with 16.9 degrees of superheat (442 K). (Further details on how to determine the downstream state are given in Module 2.3 'Superheated steam'.

Specific volume of superheated steam, 4 bar g, 442 K is 0.391 8 m³/kg (from the steam table).

$$\text{The volumetric flow} = 1\,300 \text{ kg/h} \times 0.391\,8 \text{ m}^3/\text{kg}$$
$$= 509.3 \text{ m}^3/\text{h}$$
$$= 0.141\,5 \text{ m}^3/\text{s}$$
$$\text{Valve outlet velocity} = \frac{\text{Volumetric flowrate}}{\text{Outlet area}}$$
$$= \frac{0.141\,5 \text{ m}^3/\text{s}}{0.000\,49 \text{ m}^2}$$
$$= 289 \text{ m/s}$$

It is necessary to see if this velocity is less than 0.5 Mach, the limit placed on valve outlet velocities for superheated steam.

The speed of sound (Mach 1) can be calculated from Equation 6.4.2.

$$C = 31.6 \sqrt{\gamma R T} \qquad \textbf{Equation 6.4.2}$$

A value of 1.3 is chosen for the isentropic exponent 'γ' due to the steam in the valve outlet being superheated.

R is the gas constant for steam 0.461 5 kJ/kg

T is the absolute temperature of 442 K

Therefore the speed of sound in the valve outlet:

$$C = 31.6 \sqrt{\gamma R T}$$
$$C = 31.6 \sqrt{1.3 \times 0.4615 \times 442}$$
$$C = 31.6 \times 16.28$$
$$C = 515 \text{ m/s}$$

As the steam is superheated in the valve outlet, the criterion of 0.5 Mach is used to determine whether the valve will be noisy.

$$0.5 \times 515 = 257.5 \text{ m/s}$$

As the expected velocity is 289 m/s and above the limit of 257.5 m/s, the DN25 valve would not be suitable for this application if noise is an issue.

Consider the next largest valve, a DN32 (but with a 25 mm trim). The outlet area of this valve is 0.000 8 m² (see Table 6.4.1).

$$\text{Valve outlet velocity} = \frac{0.141\,5 \text{ m}^3/\text{s}}{0.000\,8 \text{ m}^2} = 177 \text{ m/s}$$

The DN32 bodied valve will be suitable because the outlet velocity is less than 0.5 Mach allowed for superheated steam.

The same procedure can be used to determine the conditions of the downstream steam for other upstream conditions. For instance, if the upstream steam is known to be wet, the downstream condition might be wet, dry saturated or superheated, depending on the pressure drop. The allowable outlet velocity will depend on the downstream steam condition as previously outlined in this section, and observed in Example 6.4.2.

Erosion

Another problem is the possibility of erosion in the valve body caused by excessive velocity in the valve outlet. In Example 6.4.1, due to the drying and superheating effect of the pressure drop from 12 bar g to 4 bar g, the steam is in a dry gaseous state containing absolutely no moisture, and erosion should not be an issue.

Simplistically, if it can be guaranteed that the steam leaving a control valve is superheated, then 250 m/s is an appropriate limit to place on the outlet velocity.

Sometimes, when saturated steam is supplied to a control valve, it will be carrying a certain amount of water and the steam may be, for example, 97% or 98% dry. If it has just passed through a properly designed separator it will be close to 100% dry, as in Example 6.4.1.

With anything more than a small pressure drop and wet steam, the steam will probably be dried to saturation point or even slightly superheated.

If the supply steam is dry and/or the valve encounters quite a large pressure drop, (as in Example 6.4.1), the steam will be more superheated.

Equations for sizing control valves

Control valves are not as efficient as nozzles in changing heat into kinetic energy. The path taken by steam through the valve inlet, the throat and into the valve outlet is relatively tortuous.

In a control valve a great deal more energy is lost to friction than in a nozzle, and, because...

- The outlet area of the valve body is unlikely to match the downstream pressure condition.
- The relationship between the plug position and the seat is continually changing.

. . . turbulence is always likely to be present in the valve outlet.

It seems that control valves of differing types may appear to reach critical flow conditions at pressure drops other than those quoted above for nozzles. Restricted flow passages through the seat of a valve and on the downstream side of the throat may mean that maximum flowrates may only be reached with somewhat greater pressure drops. A ball valve or butterfly valve may be so shaped that some pressure recovery is achieved downstream of the throat, so that maximum flow conditions are reached with an overall pressure drop rather less than expected.

Complicated valve sizing equations can be used to take these and other criteria into consideration, and more than one standard exists incorporating such equations.

One such standard is IEC 60534. Unfortunately, the calculations are so complicated, they can only be used by computer software; manual calculation would be tedious and slow.

Nevertheless, when sizing a control valve for a critical process application, such software is indispensable. For example, IEC 60534 is designed to calculate other symptoms such as the noise levels generated by control valves, which are subjected to high pressure drops. Control valve manufacturers will usually have computer sizing and selection software complementing their own range of valves.

However, a simple steam valve sizing equation, such as that shown in Equation 3.21.2 for saturated steam, is perfectly adequate for the vast majority of steam applications with globe valves.

Also, if consideration is given to critical pressure occurring at 58% of the upstream absolute pressure, a globe valve is unlikely to be undersized.

For simplicity, the rest of this Module assumes critical pressure for saturated steam occurs at 58% of the upstream absolute pressure.

For example, if the pressure upstream of a control valve is 10 bar a, the maximum flowrate through the valve occurs when the downstream pressure is:

$$10 \text{ bar a} \times 58\% = 5.8 \text{ bar a}$$

Equally, critical pressure drop is 42% of the upstream pressure, that is, a pressure drop ratio of 0.42. As shown in the previous text, once this downstream pressure is reached, any further increase in pressure drop does not cause an increase in mass flowrate.

This effect can be observed in Figure 6.4.6 showing how, in the case of a globe valve, the flowrate increases with falling downstream pressure until critical pressure drop is achieved.

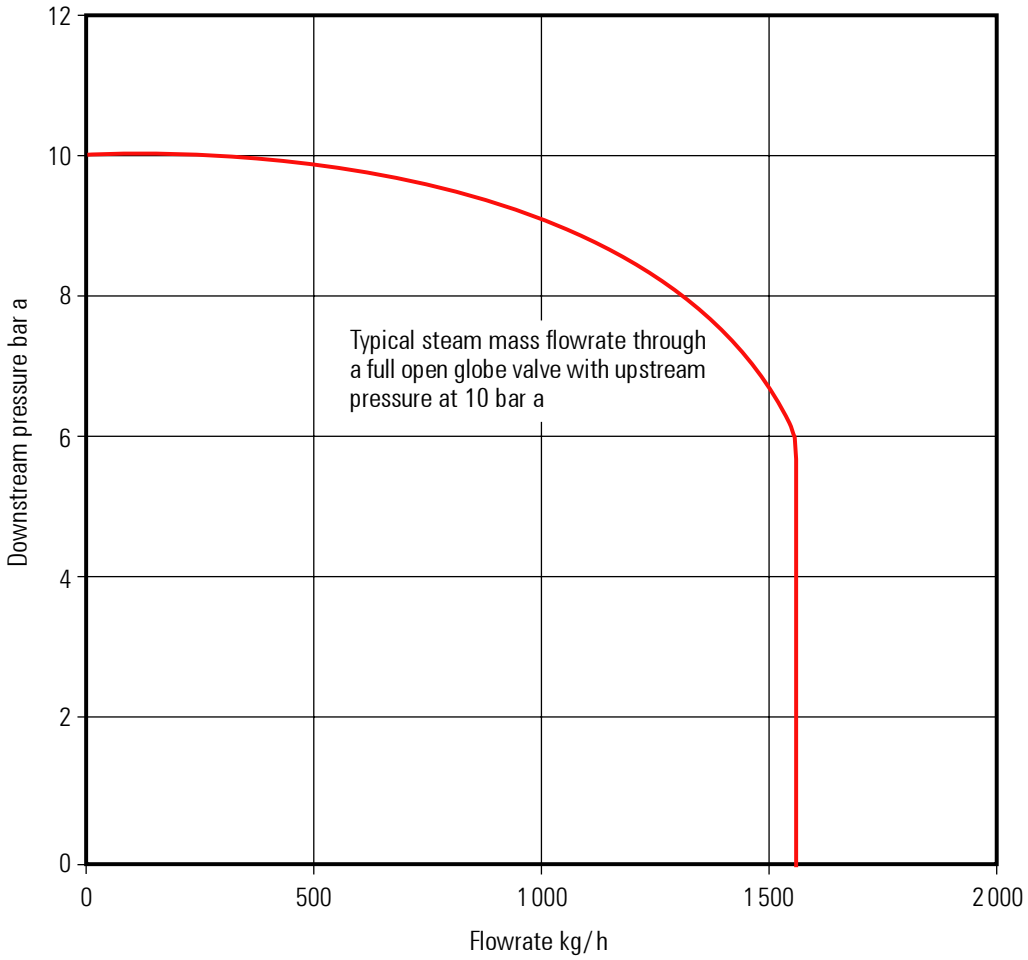


Fig. 6.4.6 The mass flowrate through a steam valve increases until critical pressure is reached

Sizing a control valve for a steam heat exchanger is a compromise between:

1. A smaller pressure drop that will minimise the size (and perhaps the cost) of the heat exchanger.
2. A larger pressure drop that allows the valve to apply effective and accurate control over the pressure and flowrate for most of its travel.

If the pressure drop is less than 10% at full load, three problems can occur:

- Depending upon the controller settings and secondary temperature, and system time lags, 'hunting' of the temperature around the set value may occur because the valve is effectively oversized; small changes in lift will cause large changes in flowrate, especially in the case of a valve with a linear characteristic.
- Running loads are often much less than the full load, and the valve may operate for very long periods with the valve plug close to its seat. This creates a risk of wiredrawing, (erosion caused by high velocity water droplets squeezing through the narrow orifice). Wiredrawing will result in a reduced valve service life.
- The system will not control well at low heat loads, effectively reducing the 'turndown' capability of the valve.

Simple sizing routine for globe valves in steam service

The flow and expansion of steam through a control valve is a complex process. There are a variety of very complex sizing formulae available, but a pragmatic approach, based on the 'best fit' of a mathematical curve to empirical results, is shown in Equation 3.21.2 for globe valves throttling saturated steam. The advantage of this relatively simple formula is that it can be used with the aid of a simple calculator. It assumes that critical pressure drop occurs at 58% of the upstream pressure.

$$\dot{m}_s = 12\ K_v\ P_1 \sqrt{1 - 5.67\ (0.42 - \varkappa)^2} \qquad \textbf{Equation 3.21.2}$$

Where:

$\dot{m}_s$ = Mass flowrate (kg/h)

K_v = Valve flow coefficient (m^3/h bar)

P_1 = Upstream pressure (bar a)

$\varkappa$ = Pressure drop ratio = $\dfrac{P_1 - P_2}{P_1}$

P_2 = Downstream pressure (bar a)

Note: If Equation 3.21.2 is used when P_2 is less than the critical pressure, then the term within the bracket (0.42 - $\varkappa$) becomes negative. This is then taken as zero and the function within the square root sign becomes unity, and the equation is simplified as shown in Equation 6.4.3.

$$\dot{m}_s = 12\ K_v\ P_1 \qquad \textbf{Equation 6.4.3}$$

Alternatively, valve-sizing or K_v charts can be used.

Terminology

Normally the full lift value of the valve will be stated using the term K_{vs}, thus:

K_{vr} = Actual value required for an application

K_{vs} = Full lift capacity stated for a particular valve

Manufacturers give the maximum lift K_{vs} values for their range of valves. Hence the K_v value is not only used for sizing valves but also as a means of comparing the capacity of alternative valve types and makes. Comparing two DN15 valves from different sources shows that valve 'A' has a K_{vs} of 10 and valve 'B' a K_{vs} of 8. Valve 'A' will give a higher flowrate for the same pressure drop.

Bringing together the information for steam valve sizing

Certain minimum information is required to determine the correct valve size:

- The pressure of the steam supply must be known.
- The steam pressure in the heat exchanger to meet the maximum heat load must be known.

The difference between the above criteria defines the differential pressure across the valve at its full load condition.

- The heat output of the equipment must be known, along with the enthalpy of evaporation (h_{fg}) at the working pressure in the heat exchanger. These factors are required to determine the steam mass flowrate.

Example 6.4.2
A control valve is required for the application shown in Figure 6.4.7.

The shell and tube heat exchanger manufacturer specifies that a steam pressure of 5 bar absolute is required in the tube bundle to satisfy a process demand of 500 kW.

Wet steam, at dryness 0.96 and 10 bar a, is available upstream of the control valve. Enthalpy of evaporation (h_{fg}) at 5 bar a is 2108.23 kJ/kg.

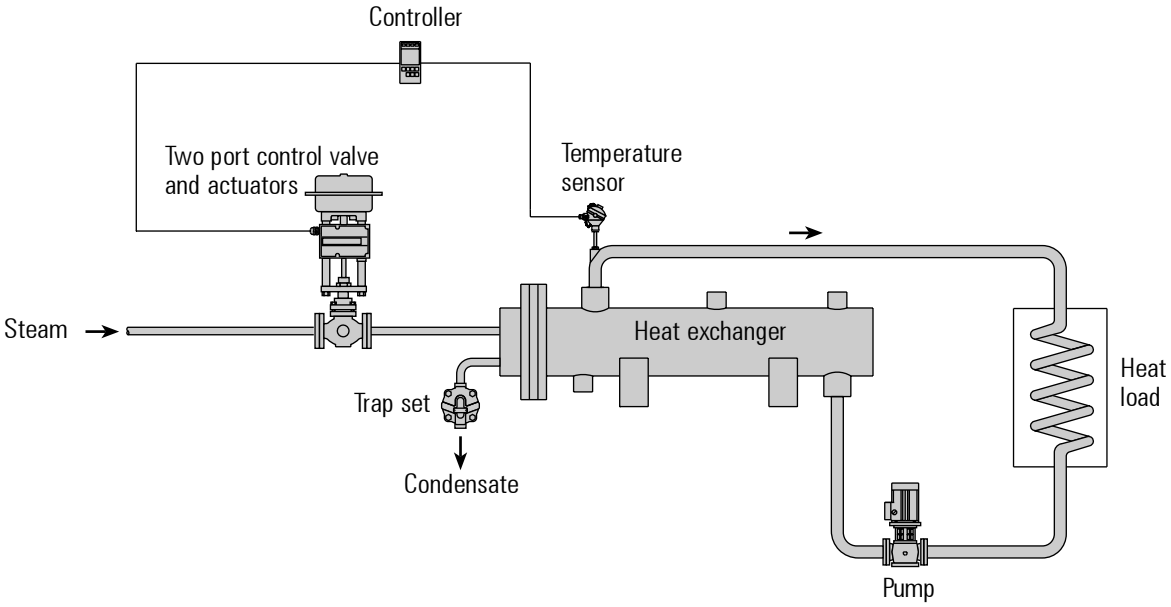


Fig. 6.4.7 Control valve on steam supply to a shell and tube heat exchanger

Determine the steam flowrate

First, it is necessary to determine the steam state for the downstream condition of 5 bar a. By entering wet steam at 10 bar a, and 0.96 dryness into the Spirax Sarco website wet steam table, it can be seen that the total heat (h_g) held in the 10 bar wet steam is 2697.15 kJ/kg.

The heat exchanger design pressure is 5 bar a, and the total heat in dry saturated steam at this pressure is 2748.65 kJ/kg (from the steam table).

The total heat in the 10 bar steam (due to its 'wetness'), is less than the total heat in saturated steam at 5 bar, and so the lower pressure steam will not contain enough heat to be totally dry. The dryness fraction of the lower pressure steam is the quotient of the two total heat figures.

Dryness fraction of the 5 bar a steam $= 2697.15 / 2748.65$

$= 0.98$

The energy available for heat transfer at 5 bar a is $0.98 \times h_{fg}$ at 5 bar a

$= 0.98 \times 2108.23$ kJ/kg

$= 2066$ kJ/kg

The steam flowrate can now be determined from Equation 2.8.1, where h_{fg} is the enthalpy of evaporation available after accounting for wet steam.

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW} \times 3600}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\text{Steam flowrate} = \frac{500 \text{ kW}}{2066 \text{ kJ/kg}} \left(\frac{\text{kJ/s}}{\text{kW}} \right) \times 3600 \text{ s/h}$$

$$\text{Steam flowrate} = 871 \text{ kg/h of wet steam}$$

Determine the pressure drop ratio (x) at full load

$$\text{Pressure drop ratio } (x) = \frac{10 \text{ bar a} - 5 \text{ bar a}}{10 \text{ bar a}} = 0.5$$

Determine the required K_{vr}

The pressure drop ratio at full load is larger than 0.42, so critical conditions apply and Equation 6.4.3 may be used to find the required K_{vr}.

$$\dot{m}_s = 12 K_v P_1 \qquad \textbf{Equation 6.4.3}$$

$$\dot{m}_s = 12 K_v P_1$$
$$871 \text{ kg/h} = 12 K_{vr} 10 \text{ bar a}$$
$$K_{vr} = \frac{871}{12 \times 10}$$
$$K_{vr} = 7.26$$

A DN25 control valve with a K_{vs} of 10 is initially selected. A calculation can now be carried out to determine if noise is an issue with this sized valve passing wet steam in the valve outlet.

The speed of sound in the valve outlet:

$$C = 31.6 \sqrt{\gamma R T}$$
$$\text{As the steam is wet, } \gamma = 1.035 + 0.1(x) \text{ where 'x' is the dryness fraction}$$
$$\gamma = 1.035 + 0.1(0.98)$$
$$\gamma = 1.133$$

R = 0.461 5 kJ/kg (the gas constant for steam)

The temperature of wet steam at 5 bar a is the same as dry saturated steam at the same pressure;

T = 425 K.

The speed of sound in the wet steam in the value outlet:

$$\text{Speed of sound, } C = 31.6 \sqrt{\gamma R T}$$
$$C = 31.6 \sqrt{1.133 \times 0.4615 \times 425}$$
$$C = 31.6 \times 14.91$$
$$C = 471 \text{ m/s}$$

A DN25 valve has an outlet area of 0.000 49 m²

The specific volume of wet steam at 5 bar a, and 0.98 dry = 0.367 4 m³/kg

$$\text{The volumetric flow} = 871 \text{ kg/h} \times 0.367\,4 \text{ m}^3\text{/kg}$$
$$= 320 \text{ m}^3\text{/h}$$
$$\text{The volumetric flow} = 0.088\,8 \text{ m}^3\text{/s}$$
$$\text{Valve outlet velocity} = \frac{\text{Volumetric flowrate}}{\text{Outlet area}}$$
$$= \frac{0.088\,8 \text{ m}^3\text{/s}}{0.000\,49 \text{ m}^2}$$
$$\text{Valve outlet velocity} = 181 \text{ m/s}$$

The noise criterion for wet steam in the valve outlet = 40 m/s

As this outlet velocity is higher than 40 m/s, the DN25 control valve might:

1. Create an unacceptable noise.

2. Cause unreasonable erosion in the valve outlet.

The DN25 control valve will therefore be unsuitable for this application where wet steam passes through the valve outlet.

One solution to this problem is to fit a larger bodied valve with the same K_{vs} of 10 to reduce the wet steam outlet velocity.

$$\text{As,valve outlet velocity} = \frac{\text{Volumetric flowrate}}{\text{Outlet area}}$$

$$\text{Minimum outlet area} = \frac{\text{Volumetric flowrate}}{\text{Valve outlet velocity}}$$

$$\text{Minimum outlet area} = \frac{0.0888\ \text{m}^3/\text{s}}{40\ \text{m/s}}$$

$$\text{Minimum outlet area} = 0.00222\ \text{m}^2$$

Consider Table 6.4.1 to determine the minimum sized control valve with an outlet area greater than 0.00222 m².

Table 6.4.1 Typical valve outlet areas DN15 - DN200 control valves

Control valve size	Outlet areas (m²)
DN15	0.00018
DN20	0.00031
DN25	0.00049
DN32	0.00080
DN40	0.00126
DN50	0.00196
DN65	0.00332
DN80	0.00500
DN100	0.00785
DN125	0.01227
DN150	0.01767
DN200	0.03142

It can be seen from Table 6.4.1 that the smallest valve required to satisfy the maximum outlet velocity of 40 m/s for wet steam is a DN65 valve, having an outlet area of 0.00332 m².

Therefore, due to wet steam passing through the valve outlet, the size of the control valve would increase from, in this instance a DN25 (1") to DN65 (2½").

A better solution might be to fit a separator before the control valve. This will allow the smaller DN25 control valve to be used, and is preferred because:

- It will give better regulation as it is more appropriately sized to handle changes in the steam load.
- It will ensure dry steam passes through the control valve, thereby reducing the propensity for erosion at the valve seat and valve outlet.
- It will ensure optimal performance of the heat exchanger, as the heating surface is not thermally insulated by moisture from wet steam.
- The cost of the smaller valve and its actuator plus separator will probably be the same as the larger valve with a larger actuator.

Sizing on an arbitrary pressure drop

If the apparatus working pressure is not known, it is sometimes possible to compromise.

It should be stressed that this method should only be used as a last resort, and that every effort should be made to determine the working pressures and flowrate.

Under these circumstances, it is suggested that the control valve be selected using a pressure drop of 10% to 20% of the upstream pressure. In this way, the selected control valve will more than likely be oversized.

To help this situation, an equal percentage valve will give better operational performance than a linear valve (this is discussed in more detail in Module 6.5 'Control valve characteristics'.

Sizing on an arbitrary pressure drop is not recommended for critical applications.

The higher the pressure drop the better?

It is usually better to size a steam valve with critical pressure drop occurring across the control valve at maximum load. This helps to reduce the size and cost of the control valve.

However, the application conditions may not allow this.

For example, if the heat exchanger working pressure is 4.5 bar a, and the maximum available steam pressure is only 5 bar a, the valve can only be sized on a 10% pressure drop ([5 – 4.5] / 5) = 0.1. In this situation, sizing on critical pressure drop would have unduly reduced the size of the control valve, and the heat exchanger would be starved of steam.

If it is impossible to increase the steam supply pressure, one solution is to install a larger heat exchanger operating at a lower pressure. In this way, the pressure drop will increase across the control valve.

This could result in a smaller valve but, unfortunately, a larger heat exchanger, because the heat exchanger operating pressure (and temperature) is now lower.

However, a larger heat exchanger working at a lower pressure brings some advantages:

- There is less tendency for the heating surfaces to scale and foul as the required steam temperature is lower.
- Less flash steam is produced in the condensate system leading to less backpressure in the condensate return pipework.

It is important to balance the cost of the valve and heat exchanger, the ability of the valve to control properly, and the effects on the rest of the system, as explained previously.

On steam systems, equal percentage valves will usually be a better choice than linear valves, as low pressure drops will have less effect on their operating performance.

Types of steam heated heat exchangers

This subject is outside the scope of this Module, but it is useful to have a brief look at the two main types of heat exchanger used for steam heating and process applications.

The shell and tube heat exchanger

Traditionally, the shell-and-tube heat exchanger has been used for many steam heating and process applications across a broad spectrum of industries. It is robust and often 'over-engineered' for the job. It tends to have an inherently high mass and large thermal hysteresis, which can make it unwieldy for certain critical applications.

Shell-and-tube heat exchangers are often greatly oversized on initial installation, mainly because of large fouling factors applied to the calculation. They tend to have low steam velocity in the steam tube, which reduces:

- Turbulence.
- The sheer stress between the flowing steam and the tube wall.
- Heat transfer.

Low sheer stress also tends not to clean the tube surfaces; hence high fouling factors are usually applied at the design stage leading to oversizing. Due to oversizing, the actual steam pressure after installation is often much less than predicted. If this is not anticipated, the steam trap might not be correctly sized and the steam tubes might flood with condensate, causing erratic control and poor performance.

The plate (and frame) heat exchanger

Plate heat exchangers are a useful alternative; being relatively small and light, they have a small mass and are extremely quick to respond to changes in heat load.

When properly designed, they tend not to foul, but if they do, they are easily disassembled, cleaned and recommissioned. Compared to shell-and-tube exchangers, they can operate at lower pressures for the same duty, but because of their high heat transfer characteristics, and a lower requirement for oversizing, they are still smaller and less expensive than a comparable shell-and-tube exchanger.

Plate heat exchangers (when properly engineered to use steam) are therefore more economically suited to high pressure drops across control valves than their shell-and-tube counterparts. This can give the advantage of smaller and less expensive control valves, whilst minimising the cost of the heat exchanger itself. Generally, it is better to design the system so that the plate exchanger operates with critical pressure drop (or the highest possible pressure drop) across the control valve at full load.

It must be stressed that not all plate heat exchangers are suitable for steam use. It is very easy to buy a heat exchanger designed for liquid use and wrongly assume that it will perform perfectly when heated with steam. Correct selection for steam is not just a matter of pressure/temperature compatibility. Proper expertise is available from bona fide manufacturers, and this should always be sought when steam is the prime energy source.

Steam sizing examples using charts

The required 'flow coefficient' (K_{vr}) may be determined in a number of ways, including calculation using Equation 3.21.2 or Equation 6.4.3 or via computer software. An alternative method of simple valve sizing is to use a K_v chart, Figure 6.4.8. A few examples of how these may be used are shown below:

Saturated steam

Example 6.4.3 Critical pressure drop application

Steam demand of heat exchanger = 800 kg/h

Steam pressure upstream of valve = 9 bar a

Steam pressure required in heat exchanger = 4 bar a

Reference steam K_v chart (Figure 6.4.8)

1. Draw a line from 800 kg/h on the steam flow ordinate.

2. Draw a horizontal line from 9 bar on the inlet pressure ordinate.

3. At the point where this crosses the critical pressure drop line (top right diagonal) draw a vertical line downwards until it intersects the horizontal 800 kg/h line.

4. Read the K_v at this crossing point, i.e. $K_{vr} \approx 7.5$

Example 6.4.4 A non critical-pressure-drop application

Steam demand of heat exchanger = 200 kg/h

Steam pressure upstream of valve = 6 bar a

Steam pressure required in heat exchanger = 5 bar a

Reference steam K_v chart (Appendix 1)

As in example 6.4.3, draw a line across from the 200 kg/h steam flow ordinate, and then draw another line from the 6 bar inlet pressure ordinate to the 1 bar pressure drop line.

Drop a vertical line from the resulting intersection point, to meet the 200 kg/h horizontal and read the K_v at this crossing point i.e. $K_{vr} \approx 3.8$

Example 6.4.5 Find the pressure drop (ΔP) across the valve having a known K_{vs} value

Steam demand of heat exchanger = 3 000 kg/h

Steam pressure upstream of valve = 10 bar a

K_{vs} of valve to be used = 36

Reference steam K_v chart (Appendix 1)

Draw a horizontal line from 3 000 kg/h to meet at the K_v 36 line. Draw a vertical line upward from this intersection to meet the 10 bar horizontal line.

Read the pressure drop at this crossing point, $\Delta P \approx 1.6$ bar.

Note: In the examples, to convert gauge pressure (bar g) to absolute pressure (bar a) simply add '1' to the gauge pressure, for example, 10 bar g = 11 bar a.

Saturated steam sizing chart

This sizing chart is empirical and should not be used for critical applications

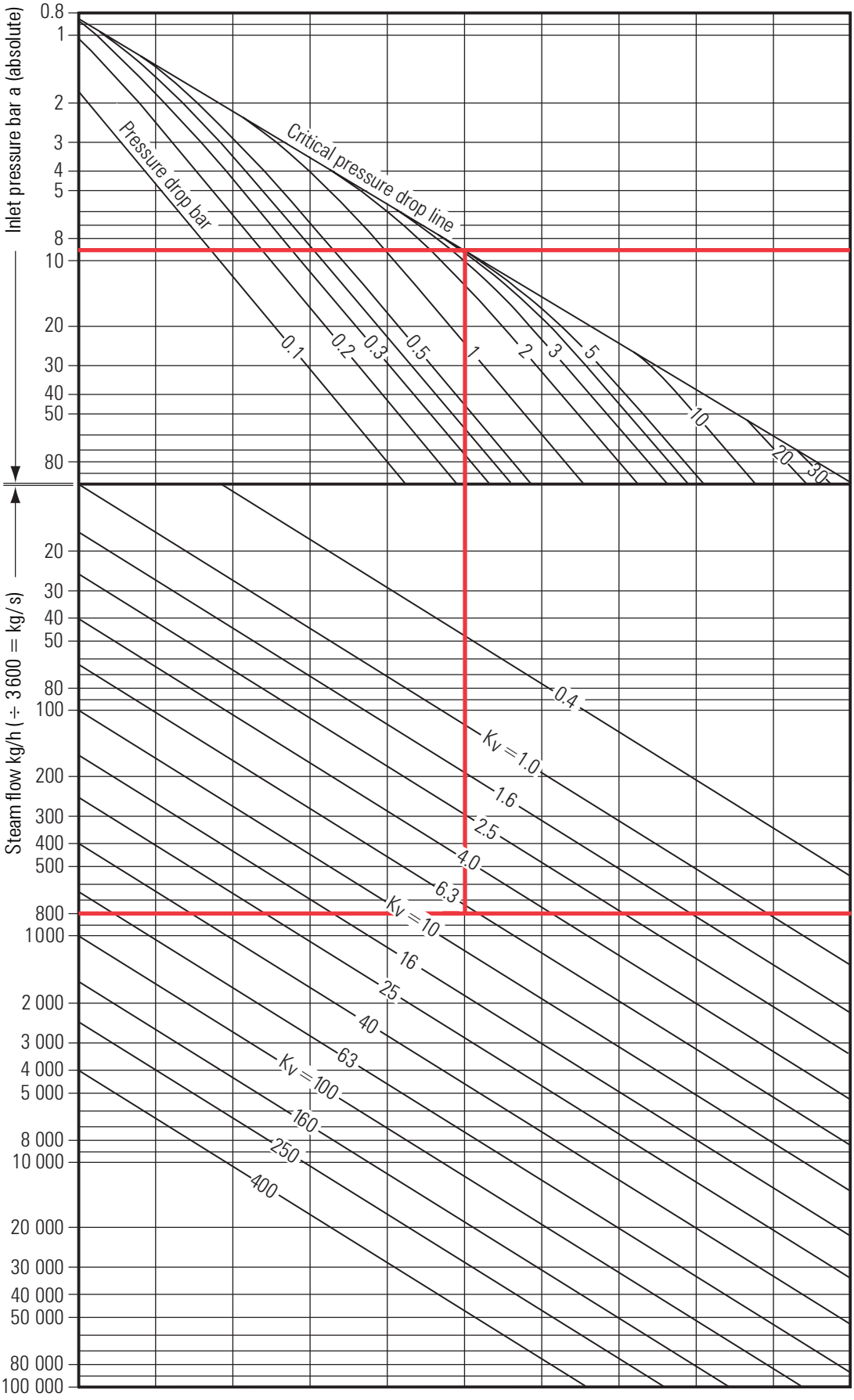


Fig. 6.4.8 Steam K_v chart

Superheated steam

To size a valve for use with superheated steam refer to Example 6.4.6 and the superheated steam chart, Figure 6.4.9.

Example 6.4.6

The following example shows how to use the chart for 100°C of superheat: follow the respective steam flow line on the left to the vertical line which represents 100°C of superheat, then draw a horizontal line across as normal from the resulting intersection. By doing this, the graph introduces a correction factor for the superheat and corrects the K_v value.

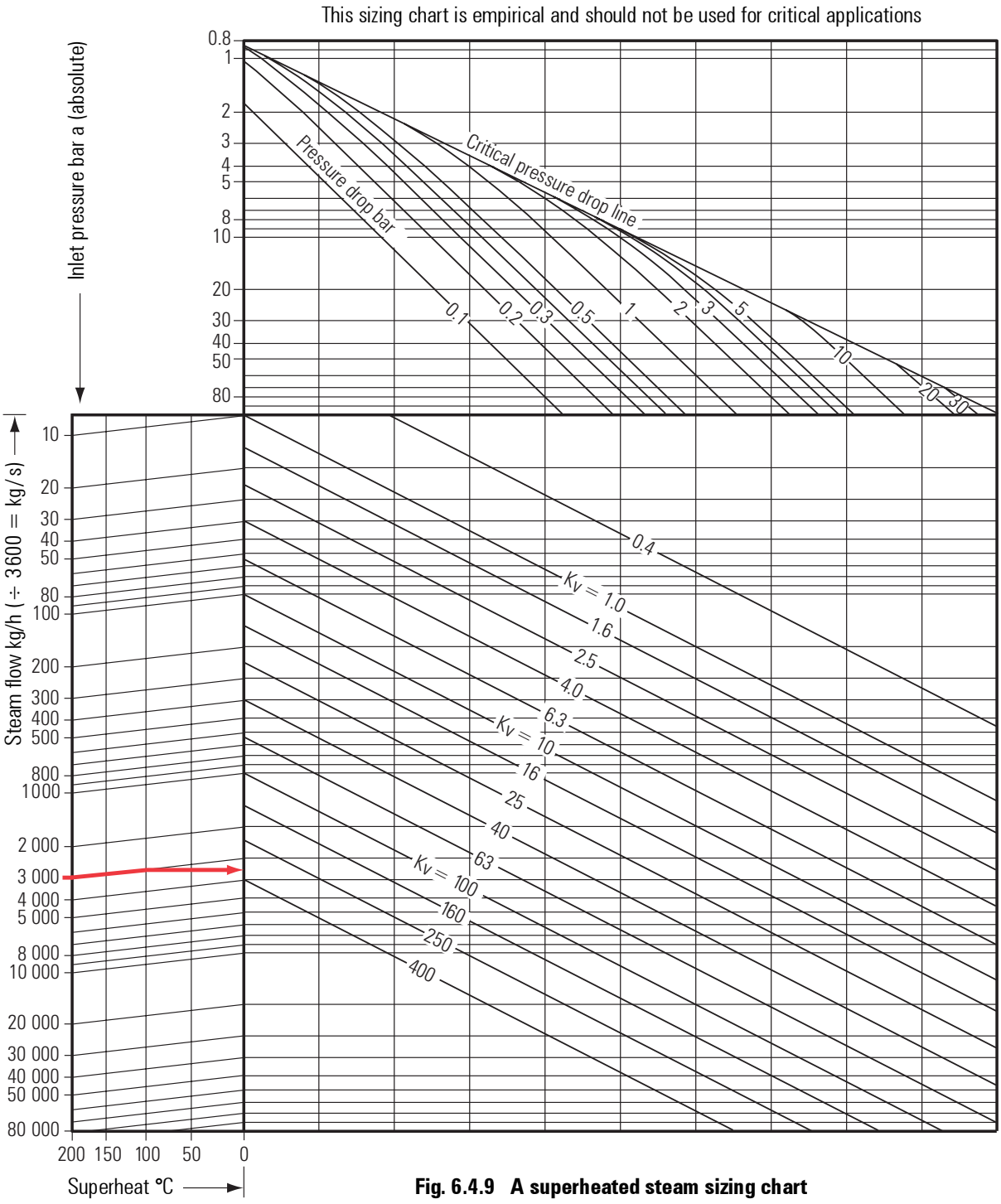


Fig. 6.4.9 A superheated steam sizing chart

Selecting a control valve for steam service

The previous Section covered the procedure for sizing a control valve based on the flowrate it needs to pass, and the pressure drop across the valve. From this data, the K_{vs} value of the control valve can be obtained. Reference to the appropriate product literature will provide the information needed to select the required valve size.

Control valve selection requires several other factors to be taken into account. The body material must be selected to suit the application. Valves are available in cast iron, SG iron, bronze, steel, stainless steel, and exotic materials for very special applications, for example titanium steel.

The design and material of the control valve must be suitable for the pressure of the system in which it will be fitted. In Europe, most valves have a nominal pressure body rating, stipulated by the letters 'PN' which actually means 'Pression Nominale'. This relates to the maximum pressure (bar gauge) the valve can withstand at a temperature of 120°C. The higher the temperature, the lower the allowable pressure, resulting in a typical pressure / temperature graph as shown in Figure 6.4.10.

It should be noted that the type of material used in manufacturing the control valve plays an important part in the pressure/temperature chart. Typical limiting conditions are:

PN16 - Cast iron	PN25 - SG iron	PN40 - Cast steel

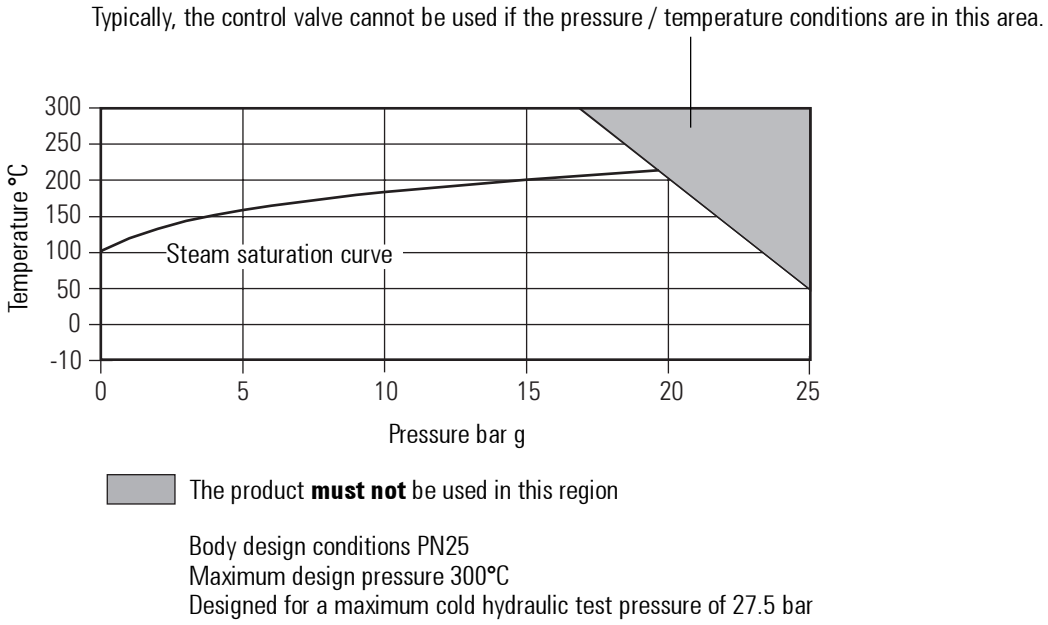


Fig. 6.4.10 An example of PN25 temperature/pressure limiting conditions

The design thickness and body jointing methods also have an effect. For example, an SG iron valve could have a PN16 rating and may also be available with a slightly different design, with a PN25 rating. Local or national regulations may affect the limits, as may the type of connection which is used.

A checklist of the major factors to be taken into account when selecting a control valve for steam service include:

1. Mass flow or volumetric flow to be considered (typically maximum, normal or minimum).

2. Flow medium (this may affect the type of material used for the valve body and internals).

3. Upstream pressure available at maximum, normal and minimum loads.

4. Downstream pressure for maximum, normal and minimum loads.

5. K_v value required.

6. Pressure drop across the valve at maximum, normal and minimum loads.

7. Body size of valve.

8. Body material and nominal pressure rating.

9. Maximum differential pressure for shut-off.

10. Connection required. Which pipe connections are required on the inlet and outlet of the valve? Screwed or flanged connections, and which type of flange, for example, ANSI, EN 1092 or DIN?

11. Maximum temperature of the medium flowing through the valve.

12. Any special requirements, for example, special gland packing variations; hardened valve seat and plug, soft seats for absolutely tight shut-off; and others.

Note: Manufacturers restrict the leakage rates of control valves to agreed limits and/or they are sometimes the subject of national standards. Also see point 17.

13. Details of the application control requirements. This is explained in more detail in Module 6.5. Briefly, an application needing on/off control (either fully-open or fully-closed) may require a valve characteristic suited to that purpose, whereas an application calling for continuous control (any degree of opening or closing), might perform better with a different type of valve characteristic.

14. Method of actuation and type of control to be used; for example, self-acting, electric, pneumatic, electropneumatic.

15. Noise levels. It is often a requirement to keep noise below 85 dBA at 1 m from the pipe if people are to work unprotected in the area. Keeping the same size internals but increasing the size of the connections may achieve this. (Many control valves have the option of reduced trim variants, alternatively special noise-reducing trims are available, and/or acoustic lagging can be applied to the valve and pipework. Valves for critical process applications should be sized using computer software utilising the IEC 60534 standard or national equivalent.

16. Pressure drops, sizes of valve body and noise level are related and should be considered. It is good practice to keep the downstream steam velocity in the valve body typically below 150 m/s for saturated steam and 250 m/s for superheated steam. This can be achieved by increasing the valve body size, which will also reduce the velocity in the valve outlet and the likelihood of excess noise. It is possible to consider a saturated steam exit velocity of 150 m/s to 200 m/s if the steam is always guaranteed to be dry saturated at the valve inlet. This is because, under these circumstances, the steam leaving the control valve will be superheated due to the superheating effect of reducing the pressure of dry saturated steam. Please note that these are general figures, different standards will quote different guidelines.

17. Leakage and isolation. Control valves are meant to control flowrate rather than isolate the supply, and are likely to leak slightly when fully shut. Control valves will be manufactured to a standard relating to shut-off tightness. Generally, the better the shut-off, the higher the cost of the valve. For steam control valves, a leakage rate of 0.01% is perfectly adequate for most applications.

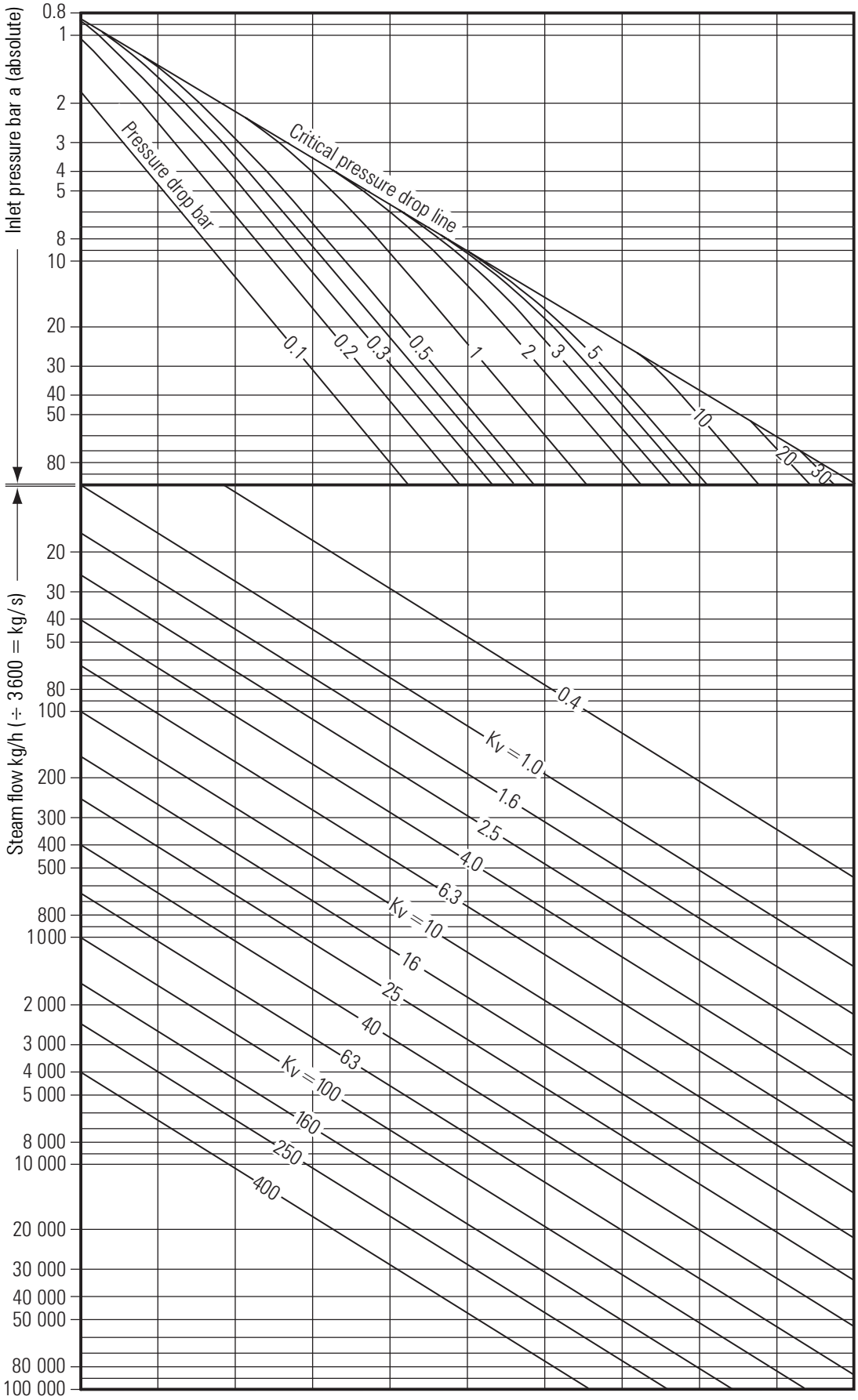
18. Turndown. Usually expressed as a ratio of the application maximum expected flow to the minimum controllable flow through a control valve.

19. Rangeability. Usually expressed as a ratio of the valve maximum controllable flow to the minimum controllable flow, between which the characteristics of the control valve are maintained. Typically, a rangeability of 50:1 is acceptable for steam applications.

20. It would be wrong to end this Module on control valves without mentioning cost. The type of valve, its materials of construction, variations in design and special requirements will inevitably result in cost variations. For optimum economy the selected valve should be correct for that application and not over-specified.

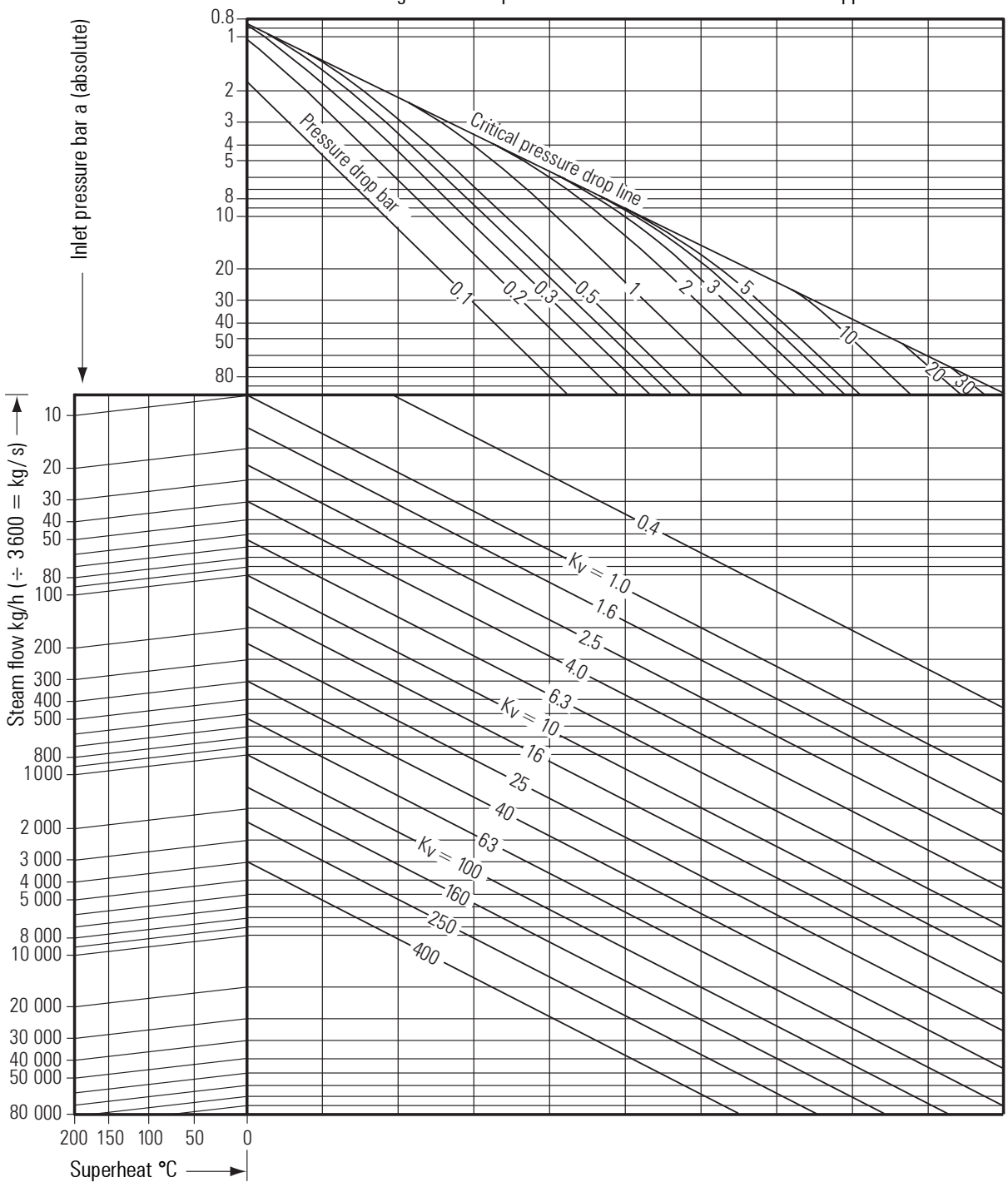
Appendix 1 Saturated steam valve sizing chart

This sizing chart is empirical and should not be used for critical applications



Appendix 2 Superheated steam valve sizing chart

This sizing chart is empirical and should not be used for critical applications



Questions

1. What factor determines the rate of heat transfer between fluids across a barrier?

a| The overall heat transfer coefficient 'U' ☐

b| The area of the heat transfer surface ☐

c| The mean temperature difference between the fluids ☐

d| All of the above ☐

2. **The upstream saturated steam pressure before a control valve is 7 bar g, the downstream pressure is 4 bar g, and the valve K_{vs} is 4. What is the pressure drop ratio?**

a| 0.429 ☐

b| 0.75 ☐

c| 0.375 ☐

d| 0.6 ☐

3. **Using Appendix 1, what is the flow of saturated steam through a valve of K_{vs} 10, when the upstream pressure is 9 bar g, and the downstream pressures are (i) 2 bar g (ii) 4.5 bar g (iii) 8 bar g.**

a| (i) 1080 kg/h (ii) 1000 kg/h (iii) 1000 kg/h ☐

b| (i) 40 kg/h (ii) 120 kg/h (iii) 120 kg/h ☐

c| (i) 1200 kg/h (ii) 695 kg/h (iii) 695 kg/h ☐

d| (i) 1200 kg/h (ii) 1200 kg/h (iii) 695 kg/h ☐

4. **A heat exchanger control valve is supplied with wet steam at 4 bar g. If the steam is dry in the heat exchanger, its flowrate is 97 kg/h and the heat exchanger is delivering 60 kW, what is the steam pressure in the heat exchanger? (Steam tables are required). Use Equation 2.8.1.**

a| 2.1 bar g ☐

b| 0.48 bar g ☐

c| 0.48 bar a ☐

d| 2.1 bar a ☐

5. In the above example, what is the K_{vr}?

a| 17 ☐

b| 1.6 ☐

c| 5.4 ☐

d| 0.7 ☐

6. For Figure 6.4.7; with an upstream pressure of 3 bar g, determine the pressure drop across a control valve with a K_{vs} of 16 passing 700 kg/h of dry saturated steam. Use Spirax Sarco on-line valve sizing calculator in the Engineering Support Centre.

a| 0.981 bar ☐

b| Critical pressure drop ☐

c| 0.5 bar ☐

d| 0.1 bar ☐

Answers

1: d, 2: c, 3: d, 4: b 5: b, 6: a

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Module 6.5

Control Valve Characteristics

Control Valve Characteristics

Flow characteristics

All control valves have an inherent flow characteristic that defines the relationship between 'valve opening' and flowrate under constant pressure conditions. Please note that 'valve opening' in this context refers to the relative position of the valve plug to its closed position against the valve seat. It does not refer to the orifice pass area. The orifice pass area is sometimes called the 'valve throat' and is the narrowest point between the valve plug and seat through which the fluid passes at any time. For any valve, however it is characterised, the relationship between flowrate and orifice pass area is always directly proportional.

Valves of any size or inherent flow characteristic which are subjected to the same volumetric flowrate and differential pressure will have exactly the same orifice pass area. However, different valve characteristics will give different 'valve openings' for the same pass area. Comparing linear and equal percentage valves, a linear valve might have a 25% valve opening for a certain pressure drop and flowrate, whilst an equal percentage valve might have a 65% valve opening for exactly the same conditions. The orifice pass areas will be the same.

The physical shape of the plug and seat arrangement, sometimes referred to as the valve 'trim', causes the difference in valve opening between these valves. Typical trim shapes for spindle operated globe valves are compared in Figure 6.5.1.

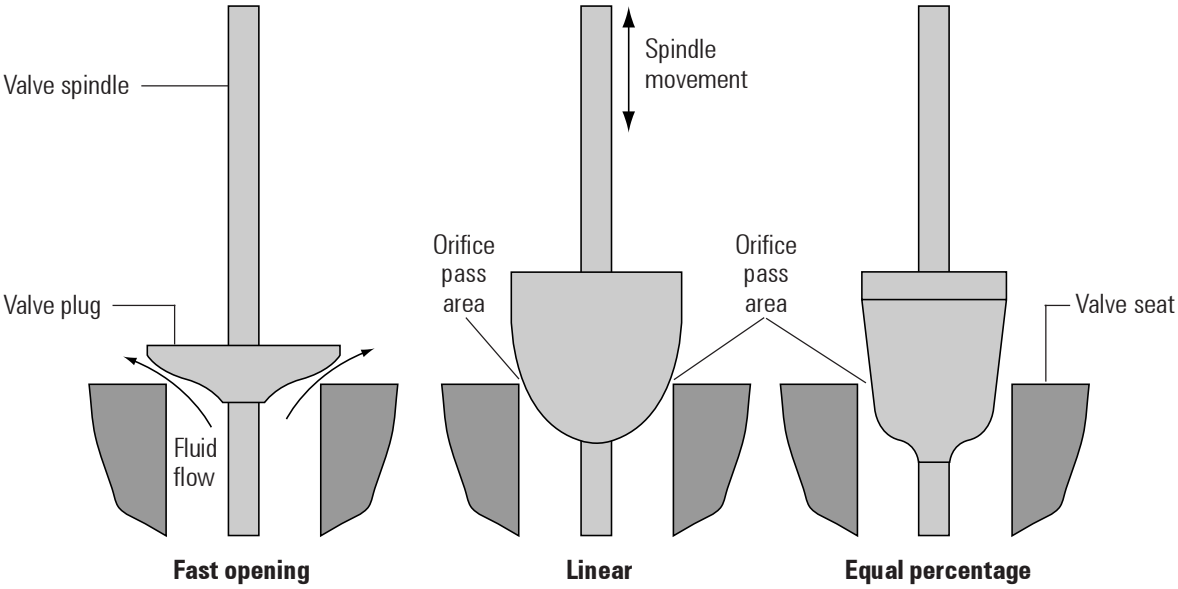


Fig. 6.5.1 The shape of the trim determines the valve characteristic

In this Module, the term 'valve lift' is used to define valve opening, whether the valve is a globe valve (up and down movement of the plug relative to the seat) or a rotary valve (lateral movement of the plug relative to the seat).

Rotary valves (for example, ball and butterfly) each have a basic characteristic curve, but altering the details of the ball or butterfly plug may modify this. The inherent flow characteristics of typical globe valves and rotary valves are compared in Figure 6.5.2.

Globe valves may be fitted with plugs of differing shapes, each of which has its own inherent flow/opening characteristic. The three main types available are usually designated:

- Fast opening.
- Linear.
- Equal percentage.

Examples of these and their inherent characteristics are shown in Figures 6.5.1 and 6.5.2.

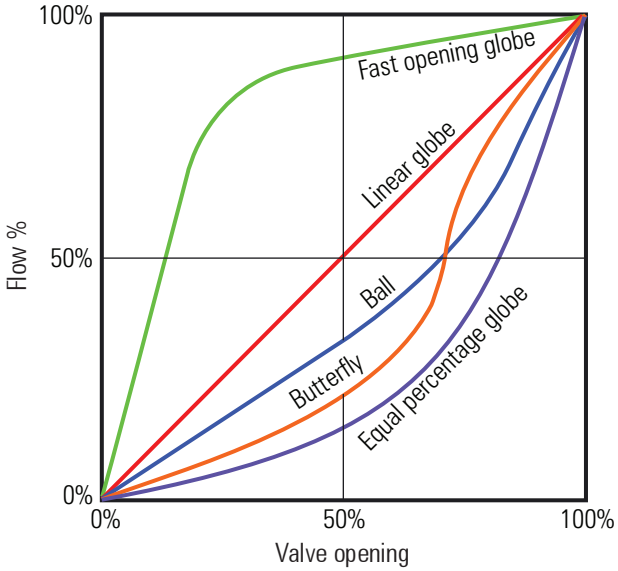


Fig. 6.5.2 Inherent flow characteristics of typical globe valves and rotary valves

Fast opening characteristic

The fast opening characteristic valve plug will give a large change in flowrate for a small valve lift from the closed position. For example, a valve lift of 50% may result in an orifice pass area and flowrate up to 90% of its maximum potential.

A valve using this type of plug is sometimes referred to as having an 'on/off' characteristic.

Unlike linear and equal percentage characteristics, the exact shape of the fast opening curve is not defined in standards. Therefore, two valves, one giving a 80% flow for 50% lift, the other 90% flow for 60% lift, may both be regarded as having a fast opening characteristic.

Fast opening valves tend to be electrically or pneumatically actuated and used for 'on/off' control.

The self-acting type of control valve tends to have a plug shape similar to the fast opening plug in Figure 6.5.1. The plug position responds to changes in liquid or vapour pressure in the control system. The movement of this type of valve plug can be extremely small relative to small changes in the controlled condition, and consequently the valve has an inherently high rangeability. The valve plug is therefore able to reproduce small changes in flowrate, and should not be regarded as a fast opening control valve.

Linear characteristic

The linear characteristic valve plug is shaped so that the flowrate is directly proportional to the valve lift (H), at a constant differential pressure. A linear valve achieves this by having a linear relationship between the valve lift and the orifice pass area (see Figure 6.5.3).

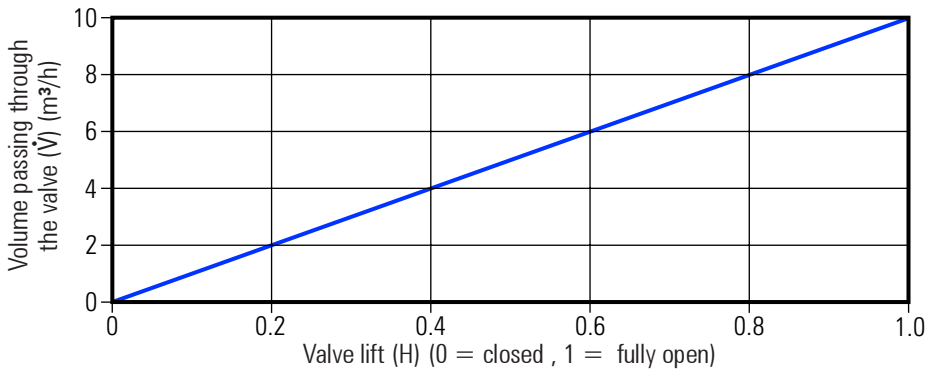


Fig. 6.5.3 Flow / lift curve for a linear valve

For example, at 40% valve lift, a 40% orifice size allows 40% of the full flow to pass.

Equal percentage characteristic (or logarithmic characteristic)

These valves have a valve plug shaped so that each increment in valve lift increases the flowrate by a certain percentage of the previous flow. The relationship between valve lift and orifice size (and therefore flowrate) is not linear but logarithmic, and is expressed mathematically in Equation 6.5.1:

$$\dot{V} = \frac{e^x}{\tau} \dot{V}_{max} \qquad \textbf{Equation 6.5.1}$$

Where:

$\dot{V}$ = Volumetric flow through the valve at lift H.

x = (ln τ) H

Note: 'In' is a mathematical function known as 'natural logarithm'.

τ = Valve rangeability (ratio of the maximum to minimum controllable flowrate, typically 50 for a globe type control valve)

H = Valve lift (0 = closed, 1 = fully open)

$\dot{V}_{max}$ = Maximum volumetric flow through the valve

Example 6.5.1

The maximum flowrate through a control valve with an equal percentage characteristic is 10 m³/h. If the valve has a turndown of 50:1, and is subjected to a constant differential pressure, by using Equation 6.5.1 what quantity will pass through the valve with lifts of 40%, 50%, and 60% respectively?

$\dot{V}_{max}$ = Maximum volumetric flow through the valve = 10 m³/h

H = Valve lift (0 closed to 1 fully open) = 0.4; 0.5; 0.6

τ = Valve rangeability = 50

$$\dot{V} = \frac{e^x}{\tau} \dot{V}_{max} \qquad \textbf{Equation 6.5.1}$$

40% open, H = 0.4	**50% open, H = 0.5**	**60% open, H = 0.6**
x = (ln τ) x H	x = (ln τ) x H	x = (ln τ) x H
x = (ln 50) x 0.4	x = (ln 50) x 0.5	x = (ln 50) x 0.6
x = 3.912 x 0.4	x = 3.912 x 0.5	x = 3.912 x 0.6
x = 1.5648	**x = 1.956**	**x = 2.347**
$\dot{V} = \frac{e^{1.5648}}{\tau} \times 10$	$\dot{V} = \frac{e^{1.956}}{\tau} \times 10$	$\dot{V} = \frac{e^{2.347}}{\tau} \times 10$
$\dot{V} = \frac{4.7817}{50} \times 10$	$\dot{V} = \frac{7.071}{50} \times 10$	$\dot{V} = \frac{10.45}{50} \times 10$
$\dot{V} = 0.0956 \times 10$	$\dot{V} = 0.1414 \times 10$	$\dot{V} = 0.2091 \times 10$
$\dot{V}$ = 0.956 m³/h	**$\dot{V}$ = 1.414 m³/h**	**$\dot{V}$ = 2.091 m³/h**

The increase in volumetric flowrate through this type of control valve increases by an equal percentage per equal increment of valve movement:

- When the valve is 50% open, it will pass 1.414 m³/h, an increase of 48% over the flow of 0.956 m³/h when the valve is 40% open.
- When the valve is 60% open, it will pass 2.091 m³/h, an increase of 48% over the flow of 1.414 m³/h when the valve is 50% open.

It can be seen that (with a constant differential pressure) for any 10% increase in valve lift, there is a 48% increase in flowrate through the control valve. This will always be the case for an equal percentage valve with rangeability of 50. For interest, if a valve has a rangeability of 100, the incremental increase in flowrate for a 10% change in valve lift is 58%.

Table 6.5.1 shows how the change in flowrate alters across the range of valve lift for the equal percentage valve in Example 6.5.1 with a rangeability of 50 and with a constant differential pressure.

Table 6.5.1
Change in flowrate and valve lift for an equal percentage characteristic with constant differential pressure

Valve Lift (H)	Flowrate ($\dot{V}$ m³/h)	Increase in flow from previous increment (%)
0.0	0.20 *	-
0.1	0.30	48%
0.2	0.44	48%
0.3	0.65	48%
0.4	0.96	48%
0.5	1.41	48%
0.6	2.09	48%
0.7	3.09	48%
0.8	4.57	48%
0.9	6.76	48%
1.0	10.00	48%

* Flowrate according to theoretical characteristic due to rangeability. In practice the valve will be fully shut at zero lift.

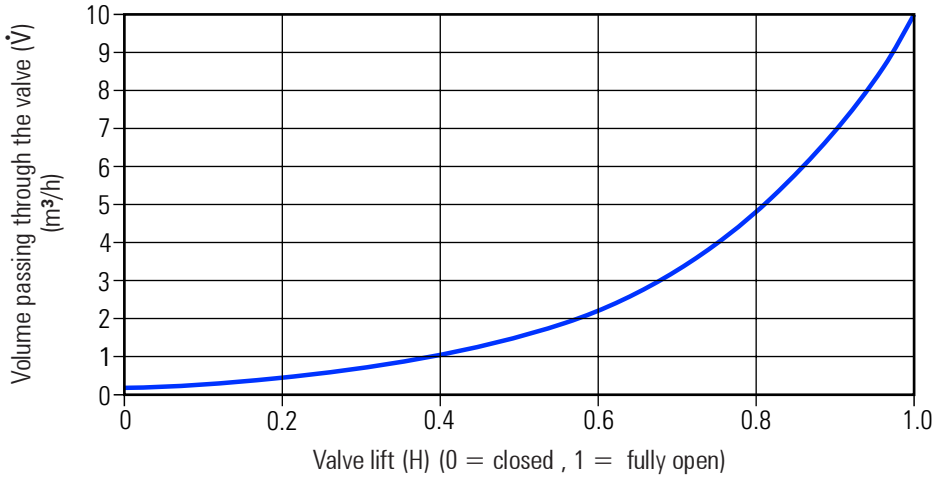


Fig. 6.5.4 Flowrate and valve lift for an equal percentage characteristic with constant differential pressure for Example 6.5.1

A few other inherent valve characteristics are sometimes used, such as parabolic, modified linear or hyperbolic, but the most common types in manufacture are fast opening, linear, and equal percentage.

Matching the valve characteristic to the installation characteristic

Each application will have a unique installation characteristic that relates fluid flow to heat demand. The pressure differential across the valve controlling the flow of the heating fluid may also vary:

- In water systems, the pump characteristic curve means that as flow is reduced, the upstream valve pressure is increased (refer to Example 6.5.2, and Module 6.3).
- In steam temperature control systems, the pressure drop over the control valve is deliberately varied to satisfy the required heat load.

The characteristic of the control valve chosen for an application should result in a direct relationship between valve opening and flow, over as much of the travel of the valve as possible.

This section will consider the various options of valve characteristics for controlling water and steam systems. In general, linear valves are used for water systems whilst steam systems tend to operate better with equal percentage valves.

1. A water circulating heating system with three-port valve

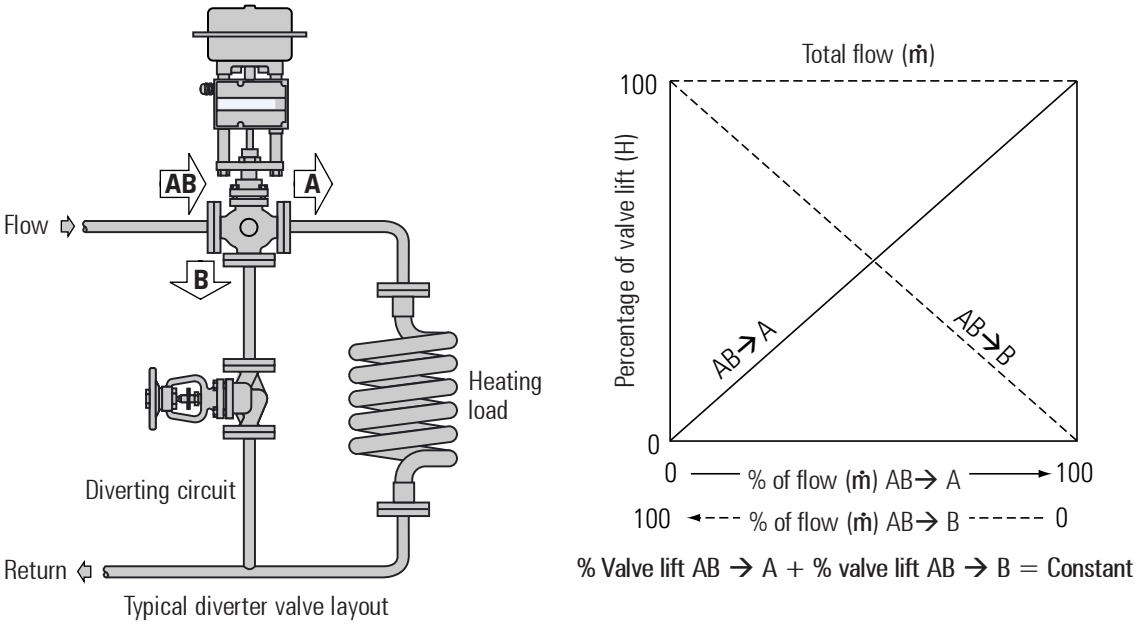


Fig. 6.5.5 A three-port diverting valve on a water heating system

In water systems where a constant flowrate of water is mixed or diverted by a three-port valve into a balanced circuit, the pressure loss over the valve is kept as stable as possible to maintain balance in the system.

Conclusion - The best choice in these applications is usually a valve with a linear characteristic. Because of this, the installed and inherent characteristics are always similar and linear, and there will be limited gain in the control loop.

2. A boiler water level control system – a water system with a two-port valve

In systems of this type (an example is shown in Figure 6.5.6), where a two-port feedwater control valve varies the flowrate of water, the pressure drop across the control valve will vary with flow. This variation is caused by:

- The pump characteristic. As flowrate is decreased, the differential pressure between the pump and boiler is increased (this phenomenon is discussed in further detail in Module 6.3).
- The frictional resistance of the pipework changes with flowrate. The head lost to friction is proportional to the square of the velocity. (This phenomenon is discussed in further detail in Module 6.3).
- The pressure within the boiler will vary as a function of the steam load, the type of burner control system and its mode of control.

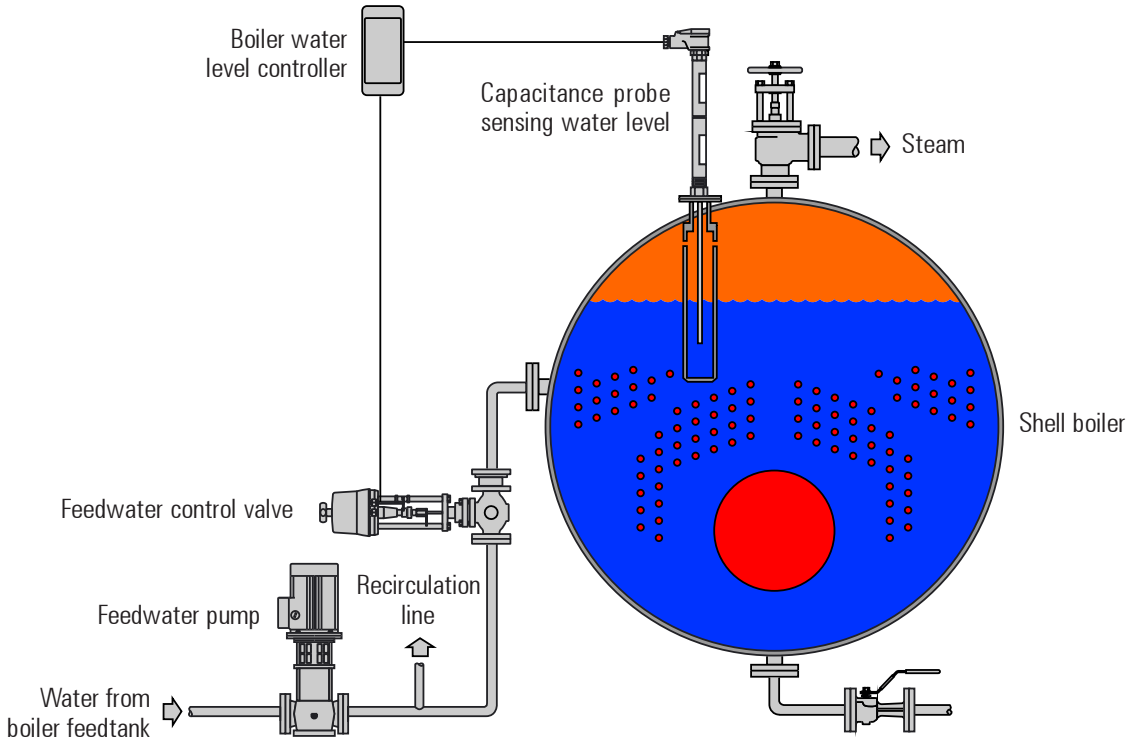


Fig. 6.5.6 A modulating boiler water level control system (not to scale)

Example 6.5.2 Select and size the feedwater valve in Figure 6.5.6

In a simplified example (which assumes a constant boiler pressure and constant friction loss in the pipework), a boiler is rated to produce 10 tonnes of steam per hour. The boiler feedpump performance characteristic is tabulated in Table 6.5.2, along with the resulting differential pressure (ΔP) across the feedwater valve at various flowrates at, and below, the maximum flow requirement of 10 m³/h of feedwater.

Note: The valve ΔP is the difference between the pump discharge pressure and a constant boiler pressure of 10 bar g. Note that the pump discharge pressure will fall as the feedwater flow increases. This means that the water pressure before the feedwater valve also falls with increased flowrate, which will affect the relationship between the pressure drop and the flowrate through the valve.

It can be determined from Table 6.5.2 that the fall in the pump discharge pressure is about 26% from no-load to full-load, but the fall in differential pressure across the feedwater valve is a lot greater at 72%. If the falling differential pressure across the valve is not taken into consideration when sizing the valve, the valve could be undersized.

Table 6.5.2 Feedwater flowrate, pump discharge pressure, and valve differential pressure (ΔP)

Flow (m³/h)	0	1	2	3	4	5	6	7	8	9	10
Pump discharge pressure (bar)	15.58	15.54	15.42	15.23	14.95	14.58	14.41	13.61	13.00	12.31	11.54
Valve ΔP (bar)	5.58	5.54	5.42	5.23	4.95	4.58	4.41	3.61	3.00	2.31	1.54

As discussed in Modules 6.2 and 6.3, valve capacities are generally measured in terms of K_v. More specifically, K_{vs} relates to the pass area of the valve when fully open, whilst K_{vr} relates to the pass area of the valve as required by the application.

Consider if the pass area of a fully open valve with a K_{vs} of 10 is 100%. If the valve closes so the pass area is 60% of the full-open pass area, the K_{vr} is also 60% of 10 = 6. This applies regardless of the inherent valve characteristic. The flowrate through the valve at each opening will depend upon the differential pressure at the time.

Using the data in Table 6.5.2, the required valve capacity, K_{vr}, can be calculated for each incremental flowrate and valve differential pressure, by using Equation 6.5.2, which is derived from Equation 6.3.2.

The K_{vr} can be thought of as being the actual valve capacity required by the installation and, if plotted against the required flowrate, the resulting graph can be referred to as the 'installation curve'.

$$\dot{V} = K_v \sqrt{\Delta P} \qquad \textbf{Equation 6.3.2}$$

Where:
$\dot{V}$ = Flowrate through the valve (m³/h)
K_v = Valve K_{vr} (m³/h bar)
ΔP = The differential pressure across the valve (bar)

Equation 6.3.2 is transposed into Equation 6.5.2 to solve for K_{vr}:

$$K_{vr} = \frac{\dot{V}}{\sqrt{\Delta P}} \qquad \textbf{Equation 6.5.2}$$

Where
K_{vr} = The actual valve capacity required by the installation (m³/h bar)
$\dot{V}$ = Flowrate through the valve (m³/h)
ΔP = The differential pressure across the valve (bar)

At the full-load condition, from Table 6.5.2:

Required flow through the valve = 10 m³/h

ΔP across the valve = 1.54 bar

From Equation 6.5.2: $$K_{vr} = \frac{10}{\sqrt{1.54}}$$

$$\mathbf{K_{vr} = 8.06\ m^3/h\ bar}$$

Taking the valve flowrate and valve ΔP from Table 6.5.2, a K_{vr} for each increment can be determined from Equation 6.5.2; and these are tabulated in Table 6.5.3.

Table 6.5.3 The relationship between flowrate, differential pressure (ΔP), and K_{vr}

Flow m³/h	0*	1	2	3	4	5	6	7	8	9	10
Valve ΔP bar	5.58*	5.54	5.42	5.23	4.95	4.58	4.14	3.61	3.00	2.31	1.54
K_{vr} m³/h bar	0*	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06

* Assumes the valve is fully shut and the pump produces maximum discharge pressure at no flow.

Constructing the installation curve

The K_{vr} of 8.06 satisfies the maximum flow condition of 10 m³/h for this example.

The installation curve could be constructed by comparing flowrate to K_{vr}, but it is usually more convenient to view the installation curve in percentage terms. This simply means the percentage of K_{vr} to K_{vs}, or in other words, the percentage of actual pass area relative to the full open pass area.

For this example: The installation curve is constructed, by taking the ratio of K_{vr} at any load relative to the K_{vs} of 8.06. A valve with a K_{vs} of 8.06 would be 'perfectly sized', and would describe the installation curve, as tabulated in Table 6.5.4, and drawn in Figure 6.5.7. This installation curve can be thought of as the valve capacity of a perfectly sized valve for this example.

Table 6.5.4 Installation curve plotted by the valve K_{vs} equalling the full-load K_{vr}

Flow m³/h	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06
Valve K_{vs}	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06	8.06
% K_{vr} / K_{vs} (Installation curve)	0	5.2	10.7	16.3	22.3	29.0	36.6	45.7	57.3	73.4	100

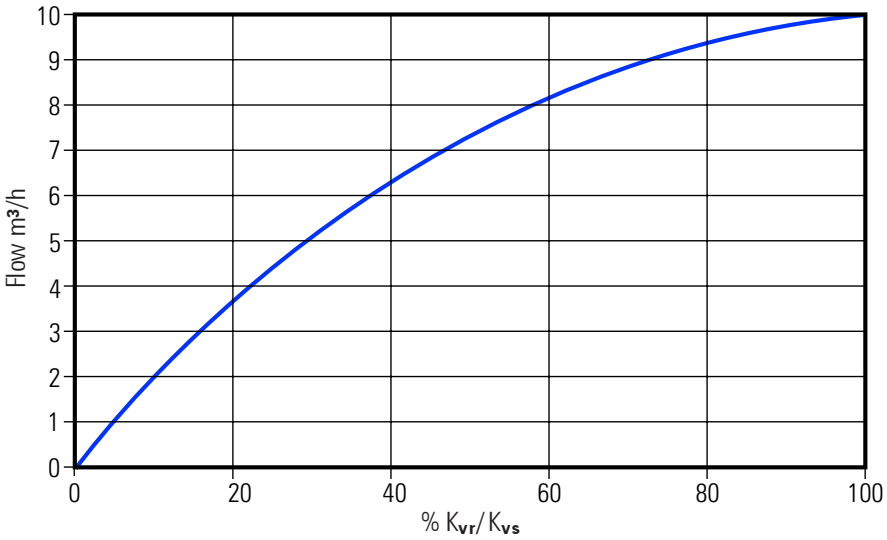


Fig. 6.5.7 The installation curve for Example 6.5.2

It can be seen that, as the valve is 'perfectly sized' for this installation, the maximum flowrate is satisfied when the valve is fully open.

However, it is unlikely and undesirable to select a perfectly sized valve. In practice, the selected valve would usually be at least one size larger, and therefore have a K_{vs} larger than the installation K_{vr}.

As a valve with a K_{vs} of 8.06 is not commercially available, the next larger standard valve would have a K_{vs} of 10 with nominal DN25 connections.

It is interesting to compare linear and equal percentage valves having a K_{vs} of 10 against the installation curve for this example.

Consider a valve with a linear inherent characteristic

A valve with a linear characteristic means that the relationship between valve lift and orifice pass area is linear. Therefore, both the pass area and valve lift at any flow condition is simply the K_{vr} expressed as a proportion of the valve K_{vs}. For example:

$$\text{Percentage valve lift} = \frac{K_{vr}}{K_{vs}} \times \frac{100}{1}$$

It can be seen from Table 6.5.4, that at the maximum flowrate of 10 m³/h, the K_{vr} is 8.06. If the linear valve has a K_{vs} of 10, for the valve to satisfy the required maximum flowrate, the valve will lift:

$$\frac{8.06}{10.0} \times \frac{100}{1} = 80.6\%$$

Using the same routine, the orifice size and valve lift required at various flowrates may be determined for the linear valve, as shown in Table 6.5.5.

Table 6.5.5 Pass area and valve lift for a linear valve with K_{vs} 10

Flow m³/h	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06
Valve K_{vs}	10	10	10	10	10	10	10	10	10	10	10
% Pass area	0	4.20	8.60	13.10	18.00	23.40	29.50	36.80	46.20	59.20	80.60
% Valve lift	0	4.20	8.60	13.10	18.00	23.40	29.50	36.80	46.20	59.20	80.60

An equal percentage valve will require exactly the same pass area to satisfy the same maximum flowrate, but its lift will be different to that of the linear valve.

Consider a valve with an equal percentage inherent characteristic

Given a valve rangeability of 50:1, $\tau = 50$, the lift (H) may be determined using Equation 6.5.1:

$$\dot{V} = \frac{e^x}{\tau}\,\dot{V}_{max} \qquad \textbf{Equation 6.5.1}$$

Where:

$\dot{V}$ = Flow through the valve at lift H.

x = (ln τ) H

Note: 'In' is a mathematical function known as 'natural logarithm'.

τ = Valve rangeability (ratio of the maximum to minimum controllable flowrate, typically 50 for a globe type control valve)

H = Valve lift (0 = closed, 1 = fully open)

$\dot{V}_{max}$ = Maximum flow through the valve

Transposing from Equation 6.5.1: $e^x = \dfrac{\dot{V}\tau}{\dot{V}_{max}}$

By taking logarithms on both sides: $x = \ln\left[\dfrac{\dot{V}\tau}{\dot{V}_{max}}\right]$

As: $x = (\ln \tau)\, H$

$$(\ln \tau)\ H = \ln\left[\frac{\dot{V}\tau}{\dot{V}_{max}}\right]$$

$$H = \frac{\ln\left[\dfrac{\dot{V}\tau}{\dot{V}_{max}}\right]}{\ln \tau}$$

Percentage valve lift is denoted by Equation 6.5.3.

$$H\,\% = \frac{\ln\left[\dfrac{\dot{V}\tau}{\dot{V}_{max}}\right]}{\ln \tau} \times 100 \qquad \textbf{Equation 6.5.3}$$

As the volumetric flowrate through any valve is proportional to the orifice pass area, Equation 6.5.3 can be modified to give the equal percentage valve lift in terms of pass area and therefore K_v.

This is shown by Equation 6.5.4.

$$H\,\% = \frac{\ln\left[\dfrac{K_{vr}\,\tau}{K_{vs}}\right]}{\ln \tau} \times 100 \qquad \textbf{Equation 6.5.4}$$

As already calculated, the K_{vr} at the maximum flowrate of 10 m³/h is 8.06, and the K_{vs} of the DN25 valve is 10. By using Equation 6.5.4 the required valve lift at full-load is therefore:

$$H\,\% = \frac{\ln\left[\dfrac{8.06 \times 50}{10}\right]}{\ln 50} \times 100$$

$$H\,\% = \frac{\ln 40.3}{\ln 50} \times 100$$

$$H\,\% = \frac{3.696}{3.912} \times 100$$

$$\mathbf{H\,\% = 94.5\%}$$

Using the same routine, the valve lift required at various flowrates can be determined from Equation 6.5.4 and is shown in Table 6.5.6.

Table 6.5.6 Pass area and valve lift for the equal % valve with K_{vs} 10.

Flow m³/h	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	0.42	0.86	1.31	1.80	2.34	2.95	3.68	4.62	5.92	8.06
Valve K_{vs}	10	10	10	10	10	10	10	10	10	10	10
% Pass area	0	4.2	8.6	13.1	18.00	23.4	29.5	36.8	46.2	59.2	80.6
% Valve lift	0	19.0	37.0	48.0	56.20	62.9	68.8	74.4	80.3	86.6	94.5

Comparing the linear and equal percentage valves for this application

The resulting application curve and valve curves for the application in Example 6.5.2 for both the linear and equal percentage inherent valve characteristics are shown in Figure 6.5.8.

Note that the equal percentage valve has a significantly higher lift than the linear valve to achieve the same flowrate. It is also interesting to see that, although each of these valves has a K_{vs} larger than a 'perfectly sized valve' (which would produce the installation curve), the equal percentage valve gives a significantly higher lift than the installation curve. In comparison, the linear valve always has a lower lift than the installation curve.

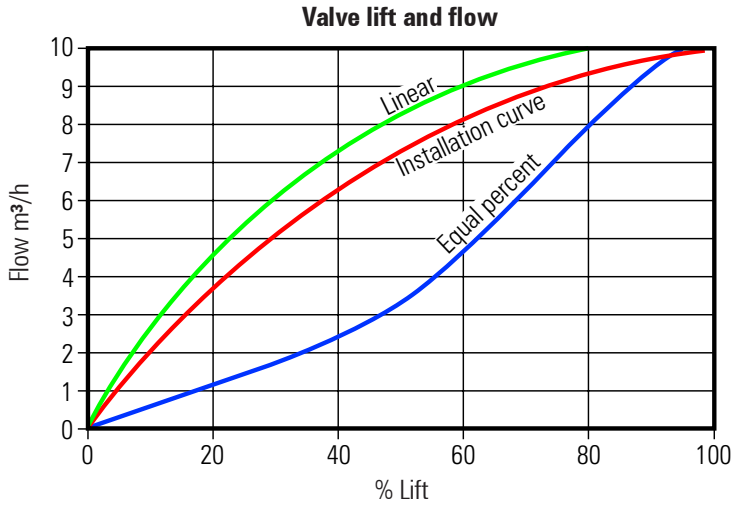


Fig. 6.5.8 Comparing linear and equal percent valve lift and the installation curve for Example 6.5.2

The rounded nature of the curve for the linear valve is due to the differential pressure falling across the valve as the flow increases. If the pump pressure had remained constant across the whole range of flowrates, the installation curve and the curve for the linear valve would both have been straight lines.

By observing the curve for the equal percentage valve, it can be seen that, although a linear relationship is not achieved throughout its whole travel, it is above 50% of the flowrate.

The equal percentage valve offers an advantage over the linear valve at low flowrates. Consider, at a 10% flowrate of 1 m³/h, the linear valve only lifts roughly 4%, whereas the equal percentage valve lifts roughly 20%. Although the orifice pass area of both valves will be exactly the same, the shape of the equal percentage valve plug means that it operates further away from its seat, reducing the risk of impact damage between the valve plug and seat due to quick reductions in load at low flowrates.

An oversized equal percentage valve will still give good control over its full range, whereas an oversized linear valve might perform less effectively by causing fast changes in flowrate for small changes in lift.

Conclusion - In most applications, an equal percentage valve will provide good results, and is very tolerant of over-sizing. It will offer a more constant gain as the load changes, helping to provide a more stable control loop at all times. However, it can be observed from Figure 6.5.8, that if the linear valve is properly sized, it will perform perfectly well in this type of water application.

3. Temperature control of a steam application with a two-port valve

In heat exchangers, which use steam as the primary heating agent, temperature control is achieved by varying the flow of steam through a two-port control valve to match the rate at which steam condenses on the heating surfaces. This varying steam flow varies the pressure (and hence temperature) of the steam in the heat exchanger and thus the rate of heat transfer.

Example 6.5.3

In a particular steam-to-water heat exchange process, it is proposed that:

- Water is heated from 10°C to a constant 60°C.
- The water flowrate varies between 0 and 10 L/s (kg/s).
- At full-load, steam is required at 4 bar a in the heat exchanger coils.
- The overall heat transfer coefficient (U) is 1 500 W/m²°C at full-load, and reduces by 4% for every 10% drop in secondary water flowrate.

Using this data, and by applying the correct equations, the following properties can be determined:

- The heat transfer area to satisfy the maximum load. Not until this is established can the following be found:
- The steam temperature at various heat loads.
- The steam pressure at various heat loads.
- The steam flowrate at various heat loads.

The heat transfer area must be capable of satisfying the maximum load.

At maximum load:

- Find the heat load.

Heat load is determined from Equation 2.6.5:

$$\dot{Q} = \dot{m}\ c_p\ \Delta T \qquad \textbf{Equation 2.6.5}$$

Where:
$\dot{Q}$ = Mean heat transfer rate (kW)
$\dot{m}$ = Mean seconday fluid flowrate (kg/s)
c_p = Specific heat capacity of water (4.19 kJ/kg°C)
ΔT = Temperature rise of the secondary fluid (°C)

$$\dot{Q} = 10 \text{ kg/s} \times 4.19 \text{ kJ/kg°C} \times (60 - 10)\text{°C} = \mathbf{2\,095\ kW}$$

- Find the corresponding steam flowrate.

The steam flowrate may be calculated from Equation 2.8.1:

$$\text{Steam flowrate (kg/h)} = \frac{\text{Heat load in kW} \times 3\,600 \text{ s/h}}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

h_{fg} for steam at 4 bar a = 2 133.6 kJ/kg, consequently:

$$\textbf{Steam flowrate} = \frac{2\,095 \text{ kW} \times 3\,600 \text{ s/h}}{2\,133.6 \text{ kJ/kg}} = \mathbf{3\,535\ kg/h}$$

- Find the heat transfer area required to satisfy the maximum load.

The heat transfer area (A) can be determined from Equation 2.5.3:

$$\dot{Q} = U\ A\ \Delta T_{LM} \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W (J/s))
U = Overall heat transfer coefficient (W/m² K or W/m²°C)
A = Heat transfer area (m²)
ΔT_{LM} = Log mean temperature difference (K or °C)

At this stage, ΔT_{LM} is unknown, but can be calculated from the primary steam and secondary water temperatures, using Equation 2.5.5.

- Find the log mean temperature difference.

ΔT_{LM} may be determined from Equation 2.5.5:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)} \qquad \textbf{Equation 2.5.5}$$

Where:
T_1 = 10°C
T_2 = 60°C
T_s = Saturation temperature at 4 bar a = 143.6°C
ln = A mathematical function known as 'natural logarithm'

$$\Delta T_{LM} = \frac{(T_2 - T_1)}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)}$$

$$\Delta T_{LM} = \frac{(60 - 10)}{\ln\left(\frac{143.6 - 10}{143.6 - 60}\right)}$$

$$\Delta T_{LM} = \frac{50}{\ln 1.598\ 1}$$

$$\Delta T_{LM} = \frac{50}{0.469}$$

$$\mathbf{\Delta T_{LM} = 106.6°C}$$

- The heat transfer area must satisfy the maximum design load, consequently from Equation 2.5.3:

$$\dot{Q} = U\ A\ \Delta T_{LM} \qquad \textbf{Equation 2.5.3}$$

$$2\,095\ \text{kW}\left[1\,000\ \frac{\text{W}}{\text{kW}}\right] = 1\,500\ \text{W/m}^2\text{°C} \times \text{Area (m}^2\text{)} \times 106.6\text{°C}$$

The heat transfer (A) = 13.1m²

Find the conditions at other heat loads at a 10% reduced water flowrate:

- Find the heat load.

If the water flowrate falls by 10% to 9 kg/s, the heat load reduces to:

$\dot{Q}$ = 9 kg/s x (60 – 10°C) x 4.19 kJ/kg °C **= 1 885.5 kW**

The initial 'U' value of 1 500 W/m²°C is reduced by 4%, so the temperature required in the steam space may be calculated from Equation 2.5.3:

$$\dot{Q} = U\ A\ \Delta T_{LM} \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = 1 885.5 kW
U = 1500 kW/m²°C x 0.96 (representing the 4% decrease in U value)
A = 13.1 m²

$$1885.5\ kW\left[1000\ \frac{W}{kW}\right] = 1\ 500\ W/m^2{}^{\circ}C \times 0.96 \times 13.1\ m^2 \times \Delta T_{LM}$$

$$\Delta T_{LM} = 100°C$$

- Find the steam temperature at this reduced load.

If ΔT_{LM} = 100°C, and T_1, T_2 are already known, then T_s may be determined from Equation 2.5.5:

$$\Delta T_{LM} = \frac{(T_2 - T_1)}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)} \qquad \textbf{Equation 2.5.5}$$

$$100 = \frac{(60-10)}{\ln\left(\frac{T_s - 10}{T_s - 60}\right)}$$

$$\ln\left(\frac{T_s - 10}{T_s - 60}\right) = 0.5$$

By taking antilogs on either side:

$$\left(\frac{T_s - 10}{T_s - 60}\right) = e^{0.5}$$

$$\left(\frac{T_s - 10}{T_s - 60}\right) = 1.65$$

$$T_s - 10 = 1.65 \times (Ts - 60)$$

$$T_s = 137°C$$

- Find the steam flowrate.

The saturated steam pressure for 137°C is 3.32 bar a (from the Spirax Sarco steam tables).

At 3.32 bar a, h_{fg} = 2 153.5 kJ/kg, consequently from Equation 2.8.1:

$$\textbf{Steam flowrate} = \frac{1\ 885.5\ kW \times 3\ 600\ s/h}{2\ 153.5\ kJ/kg} = \textbf{3 152 kg/h}$$

Using this routine, a set of values may be determined over the operating range of the heat exchanger, as shown in Table 6.5.7.

Table 6.5.7 The heat transfer, steam pressure in the coil, and steam flowrate

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
Energy (kW)	0	210	419	629	838	1 048	1 257	1 467	1 676	1 886	2 095
Steam Pressure (bar a)	0	0.22	0.27	0.37	0.54	0.81	1.19	1.71	2.42	3.35	4.0
Steam flowrate (kg/h)	0	321	644	974	1312	1 659	2 016	2 383	2 762	3 152	3 535

If the steam pressure supplying the control valve is given as 5.0 bar a, and using the steam pressure and steam flowrate information from Table 6.5.7; the K_{vr} can be calculated from Equation 6.5.6, which is derived from the steam flow formula, Equation 3.21.2.

$$\dot{m}_s = 12\ K_v\ P_1 \sqrt{1 - 5.67\ (0.42 - x)^2} \qquad \textbf{Equation 3.21.2}$$

Where:
$\dot{m}_s$ = Mass flowrate (kg/h)
K_v = Valve flow coefficient (m^3/h. bar)
P_1 = Upstream pressure (bar a)
X = Pressure drop ratio = $\dfrac{P_1 - P_2}{P_1}$
P_2 = Downstream pressure (bar a)

Equation 3.21.2 is transposed to give Equation 6.5.5.

$$K_{vr} = \frac{\dot{m}_s}{12\ P_1 \sqrt{1 - 5.67\ (0.42 - x)^2}} \qquad \textbf{Equation 6.5.5}$$

Known information at full-load includes:
$\dot{m}_s$ = 3 535 kg/h
P_1 = 5 bar a
P_2 = 4 bar a

$$x = \frac{P_1 - P_2}{P_2}$$

$$x = \frac{5 - 4}{5}$$

$$\mathbf{x = 0.2}$$

$$\text{Full-load } K_{vr} = \frac{3\,535}{12 \times 5 \times \sqrt{1 - 5.67\ (0.42 - 0.2)^2}}$$

$$K_{vr} = \frac{3\,535}{60 \times \sqrt{0.726}}$$

$$\textbf{Full - load } K_{vr} \textbf{ = 69.2}$$

Using this routine, the K_{vr} for each increment of flow can be determined, as shown in Table 6.5.8.

The installation curve can also be defined by considering the K_{vr} at all loads against the 'perfectly sized' K_{vs} of 69.2.

Table 6.5.8

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0.0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.2
Valve K_{vs}	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2	69.2
% Installation curve	0.0	7.7	15.5	23.4	31.6	39.9	48.6	57.4	66.5	77.7	100

The K_{vr} of 69.2 satisfies the maximum secondary flow of 10 kg/s.

In the same way as in Example 6.5.2, the installation curve is described by taking the ratio of K_{vr} at any load relative to a K_{vs} of 69.2.

Such a valve would be 'perfectly sized' for the example, and would describe the installation curve, as tabulated in Table 6.5.8, and drawn in Figure 6.5.9.

The installation curve can be thought of as the valve capacity of a valve perfectly sized to match the application requirement.

It can be seen that, as the valve with a K_{vs} of 69.2 is 'perfectly sized' for this application, the maximum flowrate is satisfied when the valve is fully open.

However, as in the water valve sizing Example 6.5.2, it is undesirable to select a perfectly sized valve. In practice, it would always be the case that the selected valve would be at least one size larger than that required, and therefore have a K_{vs} larger than the application K_{vr}.

A valve with a K_{vs} of 69.2 is not commercially available, and the next larger standard valve has a K_{vs} of 100 with nominal DN80 connections.

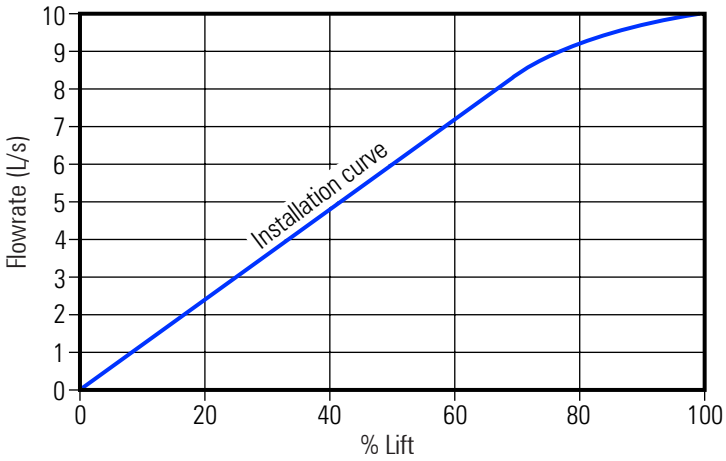


Fig. 6.5.9 The installation curve for Example 6.5.3

It is interesting to compare linear and equal percentage valves having a K_{vs} of 100 against the installation curve for this example.

Consider a valve with a linear inherent characteristic

A valve with a linear characteristic means that the relationship between valve lift and orifice pass area is linear. Therefore, both the pass area and valve lift at any flow condition is simply the K_{vr} expressed as a proportion of the valve K_{vs}. For example.

$$\text{Percentage valve lift} = \frac{K_{vr}}{K_{vs}} \times \frac{100}{1}$$

At the maximum water flowrate of 10 kg/s, the steam valve K_{vr} is 69.2. The K_{vs} of the selected valve is 100, consequently the lift is:

$$\frac{69.2}{100} \times 100 = 69.2\%$$

Using the same procedure, the linear valve lifts can be determined for a range of flows, and are tabulated in Table 6.5.9.

Table 6.5.9 Comparing valve lifts (K_{vs} 100) the K_{vr}, and the installation curve

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.2
Valve K_{vs}	100	100	100	100	100	100	100	100	100	100	100
% Lift - Linear valve	0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.2
% Lift - Equal percentage valve	0	25.1	43.0	53.5	61.1	67.1	72.1	76.4	80.2	84.2	90.6
% installation curve*	0	7.7	15.5	23.5	31.6	40.0	48.6	57.4	66.5	77.8	100.0

* The installation curve is the percentage of K_{vr} at any load to the K_{vr} at maximum load

Consider a valve with an equal percentage inherent characteristic

An equal percentage valve will require exactly the same pass area to satisfy the same maximum flowrate, but its lift will be different to that of the linear valve.

Given that the valve turndown ratio, $\tau = 50$, the lift (H) may be determined using Equation 6.5.4.

$$H\,\% = \frac{\ln\left[\frac{K_{vr}\,\tau}{K_{vs}}\right]}{\ln \tau} \times 100 \qquad \textbf{Equation 6.5.4}$$

For example, at the maximum water flowrate of 10 kg/s, the K_{vr} is 69.2. The K_{vs} of the selected valve is 100, consequently the lift is:

$$H\,\% = \frac{\ln\left[\frac{69.2 \times 50}{100}\right]}{\ln 50} \times 100$$

$$H\,\% = \frac{\ln 34.6}{\ln 50} \times 100$$

$$H\,\% = \frac{3.544}{3.912} \times 100$$

$$H\,\% = 90.6\%$$

Using the same procedure, the percentage valve lift can be determined from Equation 6.5.4 for a range of flows for this installation.

The corresponding lifts for linear and equal percentage valves are shown in Table 6.5.9 along with the installation curve.

As in Example 6.5.2, the equal percentage valve requires a much higher lift than the linear valve to achieve the same flowrate. The results are graphed in Figure 6.5.10.

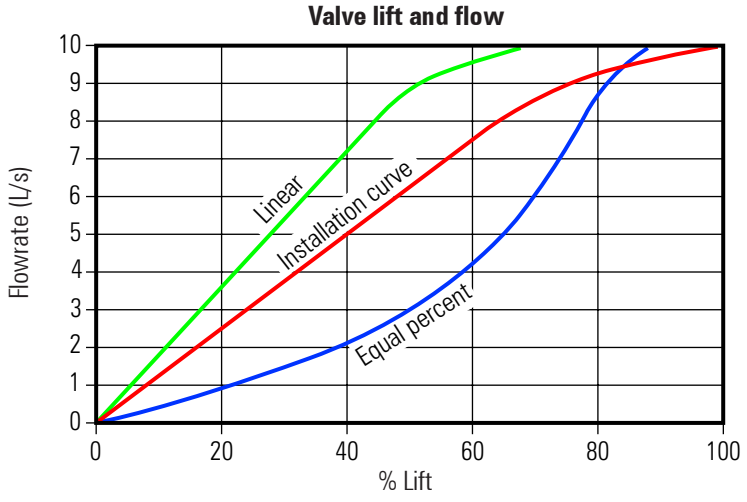


Fig. 6.5.10 Comparing linear and equal % valve lift and the installation curve for Example 6.5.3

There is a sudden change in the shape of the graphs at roughly 90% of the load; this is due to the effect of critical pressure drop across the control valve which occurs at this point.

Above 86% load in this example, it can be shown that the steam pressure in the heat exchanger is above 2.9 bar a which, with 5 bar a feeding the control valve, is the critical pressure value. (For more information on critical pressure, refer to Module 6.4, Control valve sizing for steam).

It is generally agreed that control valves find it difficult to control below 10% of their range, and in practice, it is usual for them to operate between 20% and 80% of their range.

The graphs in Figure 6.5.10 refer to linear and equal percentage valves having a K_{vs} of 100, which are the next larger standard valves with suitable capacity above the application curve (the required K_{vr} of 69.2), and would normally be chosen for this particular example.

The effect of a control valve which is larger than necessary

It is worth while considering what effect the next larger of the linear or equal percentage valves would have had if selected. To accommodate the same steam loads, each of these valves would have had lower lifts than those observed in Figure 6.5.10.

The next larger standard valves have a K_{vs} of 160. It is worth noting how these valves would perform should they have been selected, and as shown in Table 6.5.10 and Figure 6.5.11.

Table 6.5.10 Comparing valve lifts (K_{vs} 160) the K_{vr} and the installation curve

Secondary water flowrate (kg/s)	0	1	2	3	4	5	6	7	8	9	10
K_{vr}	0	5.3	10.7	16.2	21.9	27.6	33.6	39.7	46.0	53.8	69.0
Valve K_{vs}	160	160	160	160	160	160	160	160	160	160	160
% Lift - Linear valve	0	3.3	6.7	10.1	13.7	17.3	21.0	24.8	28.8	33.6	43.0
% Lift - Equal percentage valve	0	13.1	30.9	41.5	49.1	55.1	60.1	64.4	68.2	72.1	78.0
% Installation curve*	0	7.7	15.5	23.5	31.6	40.0	48.6	57.4	66.5	77.8	100

* The installation curve is the percentage of K_{vr} at any load to the K_{vr} at maximum load

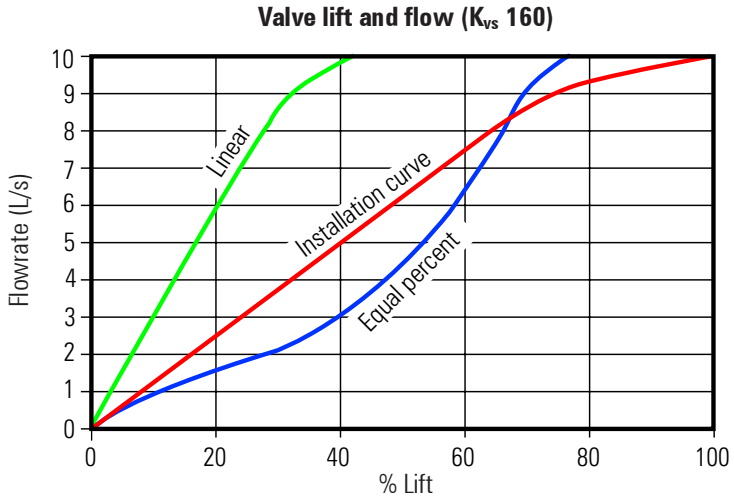


Fig. 6.5.11 Percentage valve lift required for equal percentage and linear valves in Example 6.5.3 with K_{vs} 160

It can be seen from Figure 6.5.11 that both valve curves have moved to the left when compared to the smaller (properly sized) valves in Figure 6.5.10, whilst the installation curve remains static.

The change for the linear valve is quite dramatic; it can be seen that, at 30% load, the valve is only 10% open. Even at 85% load, the valve is only 30% open. It may also be observed that the change in flowrate is large for a relatively small change in the lift. This effectively means that the valve is operating as a fast acting valve for up to 90% of its range. This is not the best type of inherent characteristic for this type of steam installation, as it is usually better for changes in steam flow to occur fairly slowly.

Although the equal percentage valve curve has moved position, it is still to the right of the installation curve and able to provide good control. The lower part of its curve is relatively shallow, offering slower opening during its initial travel, and is better for controlling steam flow than the linear valve in this case.

Circumstances that can lead to over-sizing include:

- The application data is approximate, consequently an additional 'safety factor' is included.
- Sizing routines that include operational 'factors' such as an over-zealous allowance for fouling.
- The calculated K_{vr} is only slightly higher than the K_{vs} of a standard valve, and the next larger size has to be selected.

There are also situations where:

- The available pressure drop over the control valve at full-load is low.

For example, if the steam supply pressure is 4.5 bar a and the steam pressure required in the heat exchanger at full-load is 4 bar a, this only gives an 11% pressure drop at full-load.

- The minimum load is a lot less than the maximum load.

A linear valve characteristic would mean that the valve plug operates close to the seat, with the possibility of damage.

In these common circumstances, the equal percentage valve characteristic will provide a much more flexible and practical solution.

This is why most control valve manufacturers will recommend an equal percentage characteristic for two-port control valves, especially when used on compressible fluids such as steam.

Please note: Given the opportunity, it is better to size steam valves with as high a pressure drop as possible at maximum load; even with critical pressure drop occurring across the control valve if the conditions allow. This helps to reduce the size and cost of the control valve, gives a more linear installation curve, and offers an opportunity to select a linear valve.

However, conditions may not allow this. The valve can only be sized on the application conditions. For example, should the heat exchanger working pressure be 4.5 bar a, and the maximum available steam pressure is only 5 bar a, the valve can only be sized on a 10% pressure drop ([5 – 4.5] / 5). In this situation, sizing the valve on critical pressure drop would have reduced the size of the control valve and starved the heat exchanger of steam.

If it were impossible to increase the steam supply pressure, a solution would be to install a heat exchanger that operates at a lower operating pressure. In this way, the pressure drop would increase across the control valve. This could result in a smaller valve but also a larger heat exchanger, because the heat exchanger operating temperature is now lower.

Another set of advantages accrues from larger heat exchangers operating at lower steam pressures:

- There is less propensity for scaling and fouling on the heating surfaces.
- There is less flash steam produced in the condensate system.
- There is less backpressure in the condensate system.

A balance has to be made between the cost of the control valve and heat exchanger, the ability of the valve to control properly, and the effects on the rest of the system as seen above. On steam systems, equal percentage valves will usually be a better choice than linear valves, because if low pressure drops occur, they will have less of an affect on their performance over the complete range of valve movement.

Questions

1. An equal percentage valve has a certain orifice pass area. For the same flowrate and differential pressure what would be the pass area of a linear valve?

a| More than the equal percentage valve ☐
b| Less than the equal percentage valve ☐
c| Almost the same as the equal percentage valve ☐
d| Exactly the same as the equal percentage valve ☐

2. **An equal percentage valve has a certain orifice pass area. For the same flowrate and differential pressure what would be the lift of a linear valve?**

a| More than the equal percentage valve ☐
b| Less than the equal percentage valve ☐
c| Almost the same as the equal percentage valve ☐
d| Exactly the same as the equal percentage valve ☐

3. **A linear valve with K_{vs} 4 and rangeability 50 passes 10 m³/h of water when fully open. What will be the percentage orifice pass area, the K_{vr}, and the valve lift with a flow of 5 m³/h with the same differential pressure across the same valve?**

a| Pass area 50%; K_{vr} 2; lift 50% ☐
b| Pass area 40%; K_{vr} 2; lift 40% ☐
c| Pass area 60%; K_{vr} 2; lift 60% ☐
d| Pass area 50%; K_{vr} 1; lift 50% ☐

4. **An equal percentage valve with K_{vs} 4 and rangeability 50 passes 10 m³/h of water when fully open. What will be the percentage orifice pass area, the K_{vr}, and the valve lift with a flow of 5 m³/h with the same differential pressure across the same valve?**

a| Pass area 50%; K_{vr} 2; lift 82.3% ☐
b| Pass area 40%; K_{vr} 3.29; lift 41.1% ☐
c| Pass area 60%; K_{vr} 2; lift 60% ☐
d| Pass area 82.3%; K_{vr} 2; lift 82.3% ☐

5. What is the effect on the control performance of a linear valve when it is oversized?

a| None ☐
b| The valve tends to control better ☐
c| The valve will tend to act as a fast opening valve ☐
d| The valve will tend to act as a slow opening valve ☐

6. What is the effect on the control performance of an equal percentage valve when it is oversized?

a| None ☐
b| The valve tends to control better ☐
c| The valve will tend to act as a slow opening valve ☐
d| The valve is still likely to perform with a reasonable degree of control ☐

Answers

1: d, 2: b, 3: a, 4: a, 5: c, 6: d

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Module 6.6

Control Valve Actuators and Positioners

Actuators

In Block 5, 'Controls Theory', an analogy was used to describe simple process control:

- The arm muscle and hand (the actuator) turned the valve - (the controlled device).

One form of controlling device, the control valve, has now been covered. The actuator is the next logical area of interest.

The operation of a control valve involves positioning its movable part (the plug, ball or vane) relative to the stationary seat of the valve. The purpose of the valve actuator is to accurately locate the valve plug in a position dictated by the control signal.

The actuator accepts a signal from the control system and, in response, moves the valve to a fully-open or fully-closed position, or a more open or a more closed position (depending on whether 'on/off' or 'continuous' control action is used).

There are several ways of providing this actuation. This Module will concentrate on the two major ones:

- Pneumatic
- Electric.

Other significant actuators include the hydraulic and the direct acting types. These are discussed in Block 7, 'Control Equipment: Self-Acting Controls'.

Pneumatic actuators – operation and options

Pneumatic actuators are commonly used to actuate control valves and are available in two main forms; piston actuators (Figure 6.6.1) and diaphragm actuators (Figure 6.6.2)

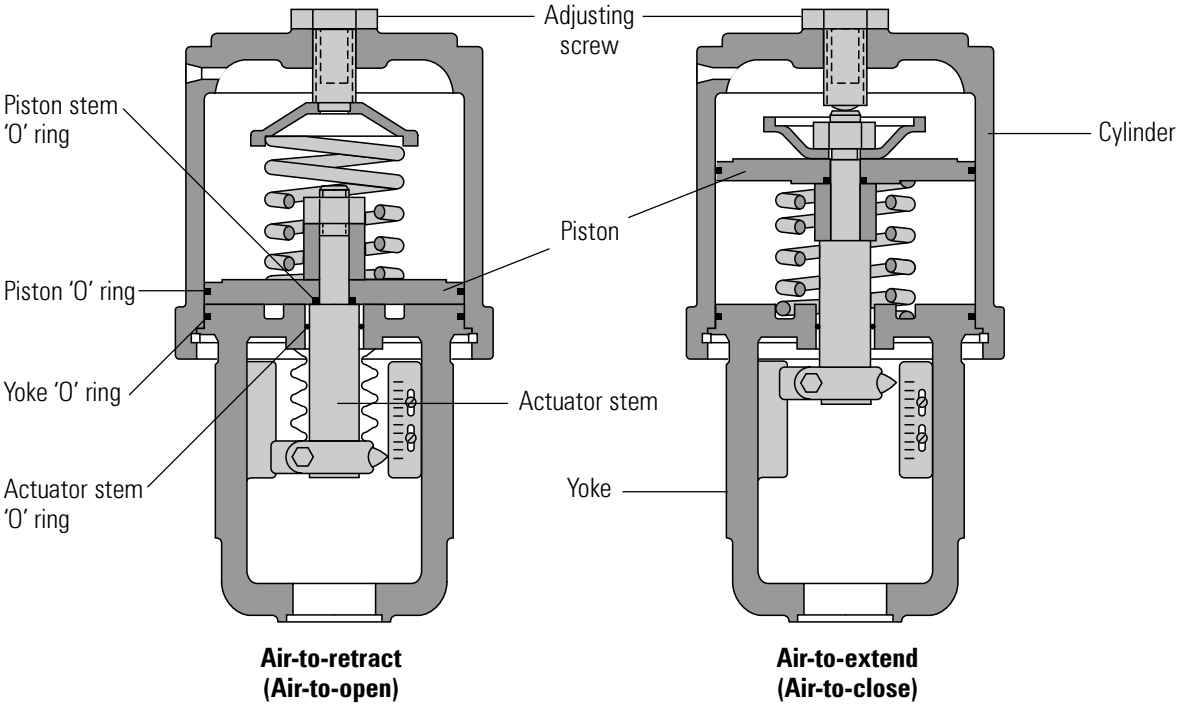


Fig. 6.6.1 Typical piston actuators

Piston actuators

Piston actuators are generally used where the stroke of a diaphragm actuator would be too short or the thrust is too small. The compressed air is applied to a solid piston contained within a solid cylinder. Piston actuators can be single acting or double acting, can withstand higher input pressures and can offer smaller cylinder volumes, which can act at high speed.

Diaphragm actuators

Diaphragm actuators have compressed air applied to a flexible membrane called the diaphragm. Figure 6.6.2 shows a rolling diaphragm where the effective diaphragm area is virtually constant throughout the actuator stroke. These types of actuators are single acting, in that air is only supplied to one side of the diaphragm, and they can be either direct acting (spring-to-retract) or reverse acting (spring-to-extend).

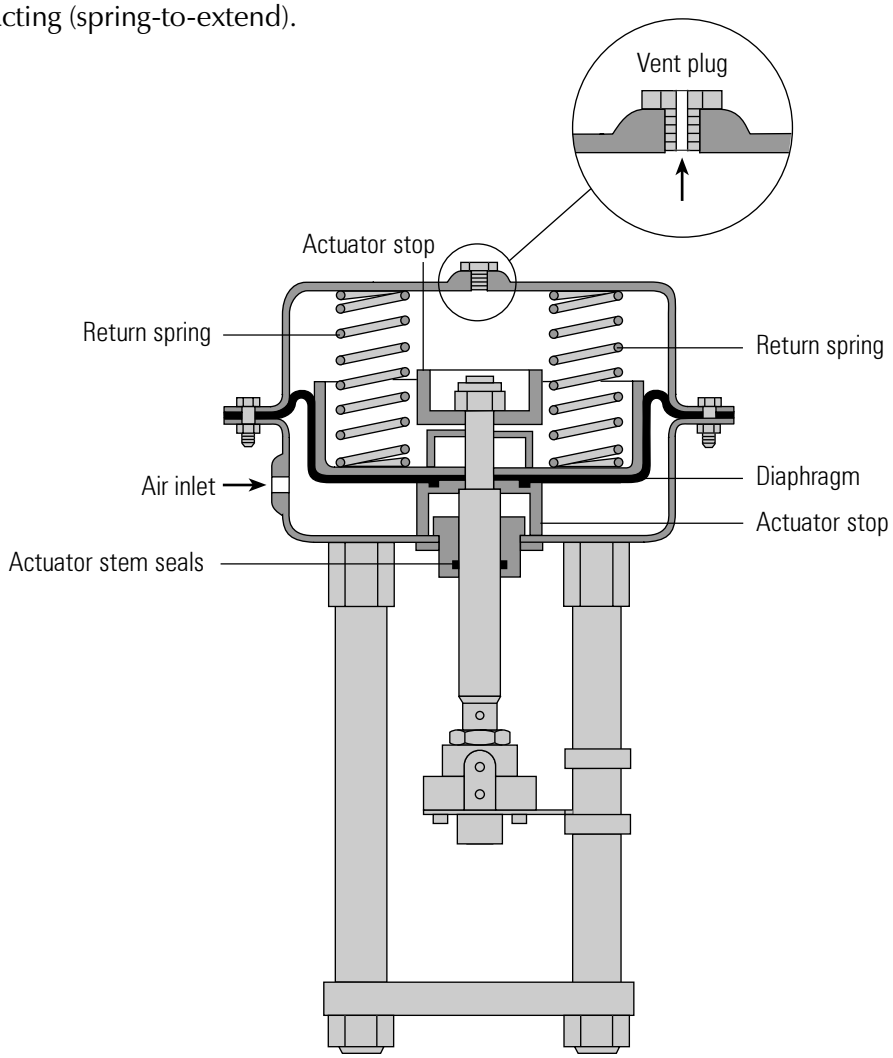


Fig. 6.6.2 A pneumatic diaphragm actuator

Reverse acting (spring-to-extend)

The operating force is derived from compressed air pressure, which is applied to a flexible diaphragm. The actuator is designed so that the force resulting from the air pressure, multiplied by the area of the diaphragm, overcomes the force exerted (in the opposite direction) by the spring(s).

The diaphragm (Figure 6.6.2) is pushed upwards, pulling the spindle up, and if the spindle is connected to a direct acting valve, the plug is opened. The actuator is designed so that with a specific change of air pressure, the spindle will move sufficiently to move the valve through its complete stroke from fully-closed to fully-open.

As the air pressure decreases, the spring(s) moves the spindle in the opposite direction. The range of air pressure is equal to the stated actuator spring rating, for example 0.2 - 1 bar.

With a larger valve and/or a higher differential pressure to work against, more force is needed to obtain full valve movement.

To create more force, a larger diaphragm area or higher spring range is needed. This is why controls manufacturers offer a range of pneumatic actuators to match a range of valves – comprising increasing diaphragm areas, and a choice of spring ranges to create different forces.

The diagrams in Figure 6.6.3 show the components of a basic pneumatic actuator and the direction of spindle movement with increasing air pressure.

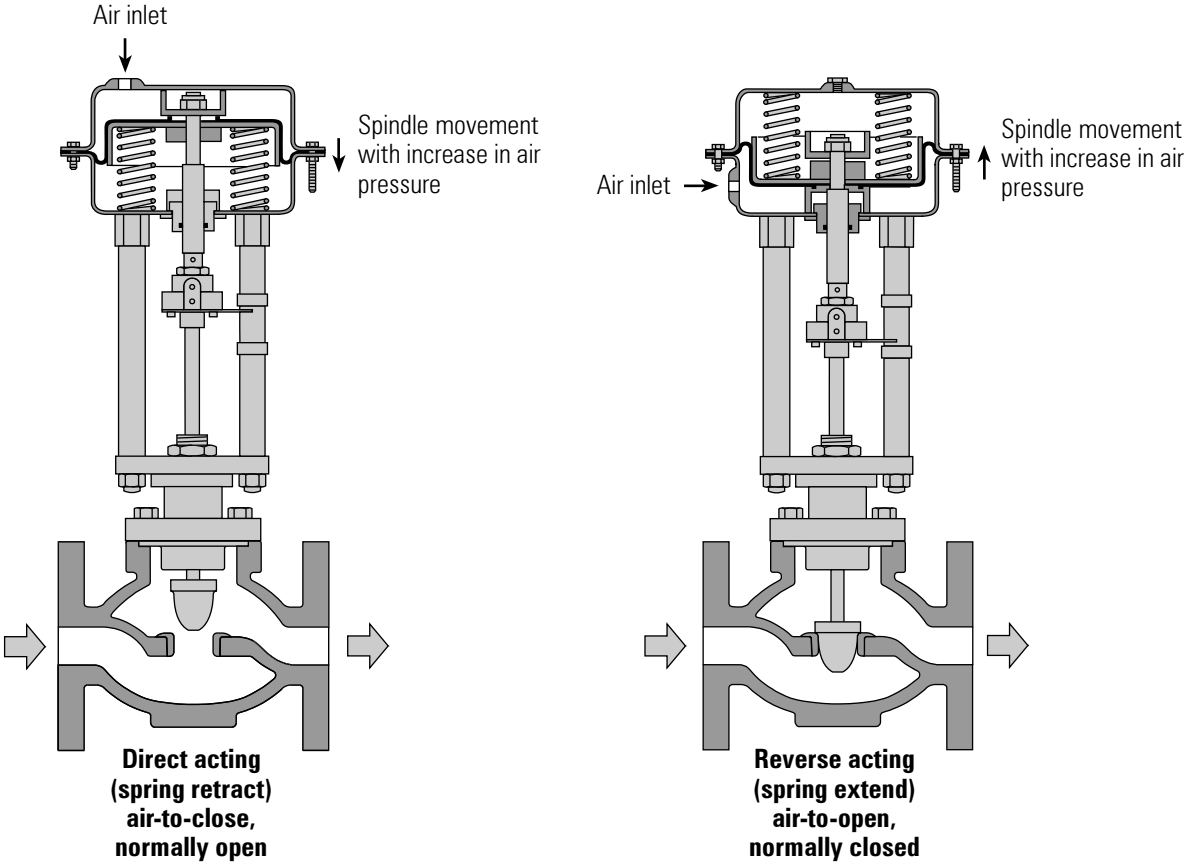


Fig. 6.6.3 Valve and actuator configurations

Direct acting actuator (spring-to-retract)

The direct acting actuator is designed with the spring below the diaphragm, having air supplied to the space above the diaphragm. The result, with increasing air pressure, is spindle movement in the opposite direction to the reverse acting actuator.

The effect of this movement on the valve opening depends on the design and type of valve used, and is illustrated in Figure 6.6.3. There is however, an alternative, which is shown in Figure 6.6.4. A direct acting pneumatic actuator is coupled to a control valve with a reverse acting plug (sometimes called a 'hanging plug').

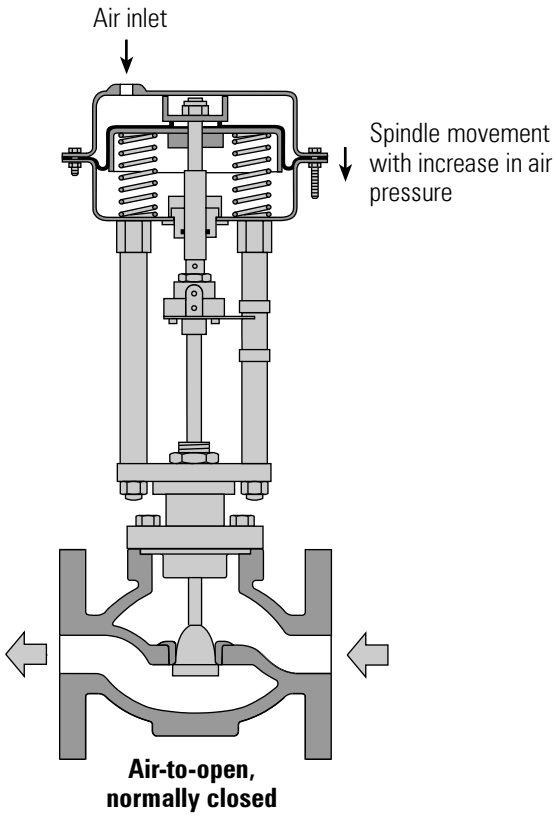


Fig. 6.6.4 Direct acting actuator and reverse acting control valve

The choice between direct acting and reverse acting pneumatic controls depends on what position the valve should revert to in the event of failure of the compressed air supply. Should the valve close or be wide-open? This choice depends upon the nature of the application and safety requirements. It makes sense for steam valves to close on air failure, and cooling valves to open on air failure. The combination of actuator and valve type must be considered. Figure 6.6.5 and Figure 6.6.6 show the net effect of the various combinations.

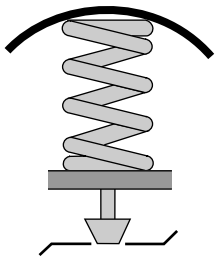
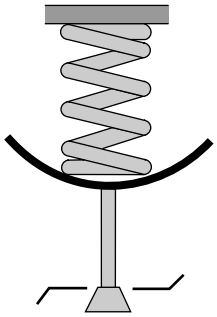
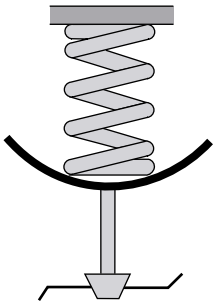
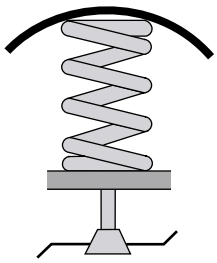
Two port valves				
Actuator action	Direct	Reverse	Reverse	Direct
Valve action	Direct	Reverse	Direct	Reverse
On air failure	Valve opens		Valve closes	

Fig. 6.6.5 Net effect of various combinations for two port valves

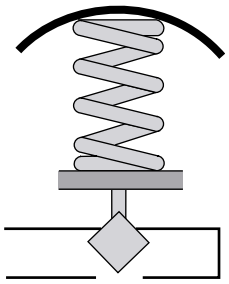
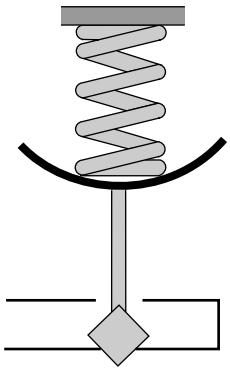
Three port valves (typical mixing valve depicted)		
Actuator action	Direct	Reverse
On air failure	Top seat closes bottom seat opens	Bottom seat closes top seat opens

Fig. 6.6.6 Net effect of the two combinations for three port valves

Effect of differential pressure on the valve lift

The air fed into the diaphragm chamber is the control signal from the pneumatic controller. The most widely used signal air pressure is 0.2 bar to 1 bar. Consider a reverse acting actuator (spring to extend) with standard 0.2 to 1.0 bar spring(s), fitted to a direct acting valve (Figure 6.6.7).

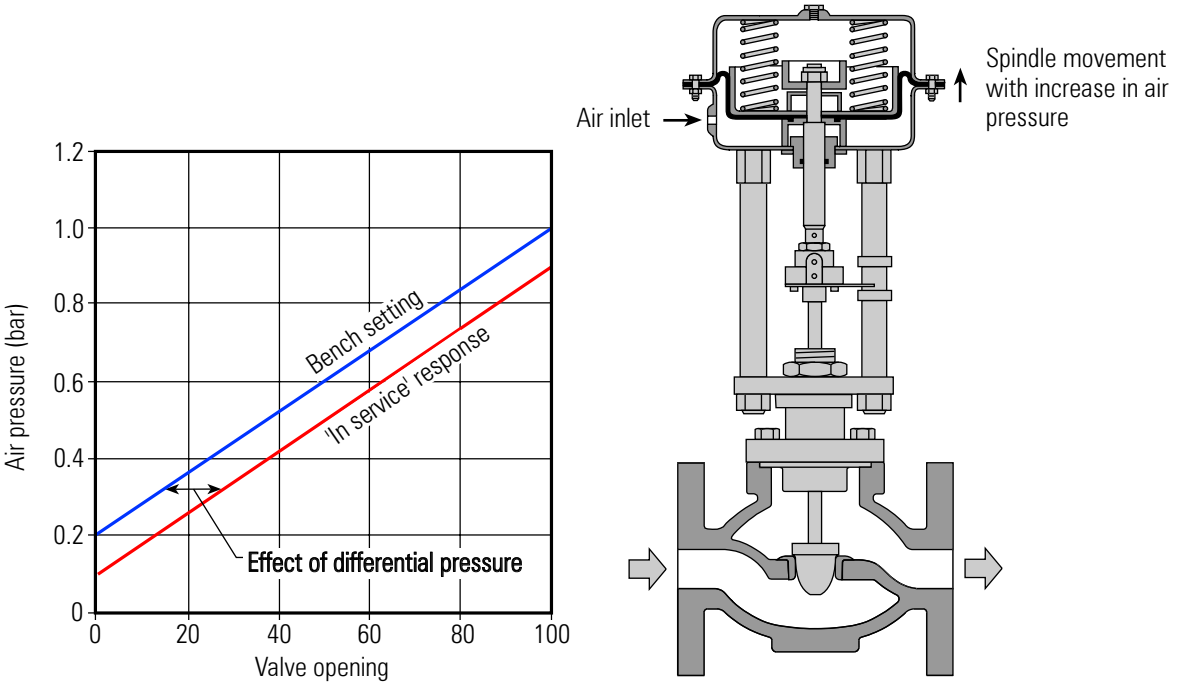


Fig. 6.6.7 Reverse acting actuator, air-to-open, direct acting valve - normally closed

When the valve and actuator assembly is calibrated (or 'bench set'), it is adjusted so that an air pressure of 0.2 bar will just begin to overcome the resistance of the springs and move the valve plug away from its seat.

As the air pressure is increased, the valve plug moves progressively further away from its seat, until finally at 1 bar air pressure, the valve is 100% open. This is shown graphically in Figure 6.6.7.

Now consider this assembly installed in a pipeline in a pressure reducing application, with 10 bar g on the upstream side and controlling the downstream pressure to 4 bar g.

The differential pressure across the valve is 10 - 4 = 6 bar. This pressure is acting on the underside of the valve plug, providing a force tending to open the valve. This force is in addition to the force provided by the air pressure in the actuator.

Therefore, if the actuator is supplied with air at 0.6 bar (halfway between 0.2 and 1 bar), for example, instead of the valve taking up the expected 50% open position, the actual opening will be greater, because of the extra force provided by the differential pressure.

Also, this additional force means that the valve is **not closed** at 0.2 bar. In order to close the valve in this example, the control signal must be reduced to approximately 0.1 bar.

The situation is slightly different with a steam valve controlling temperature in a heat exchanger, as the differential pressure across the valve will vary between:

- A minimum, when the process is calling for maximum heat, and the control valve is 100% open.
- A maximum, when the process is up to temperature and the control valve is closed.

The steam pressure in the heat exchanger increases as the heat load increases. This can be seen in Module 6.5, Example 6.5.3 and Table 6.5.7.

If the pressure upstream of the control valve remains constant, then, as the steam pressure rises in the heat exchanger, the differential pressure across the valve must decrease.

Figure 6.6.8 shows the situation with the air applied to a direct acting actuator. In this case, the force on the valve plug created by the differential pressure works against the air pressure. The effect is that if the actuator is supplied with air at 0.6 bar, for example, instead of the valve taking up the expected 50% open position, the percentage opening will be greater because of the extra force provided by the differential pressure. In this case, the control signal has to be increased to approximately 1.1. bar to fully close the valve.

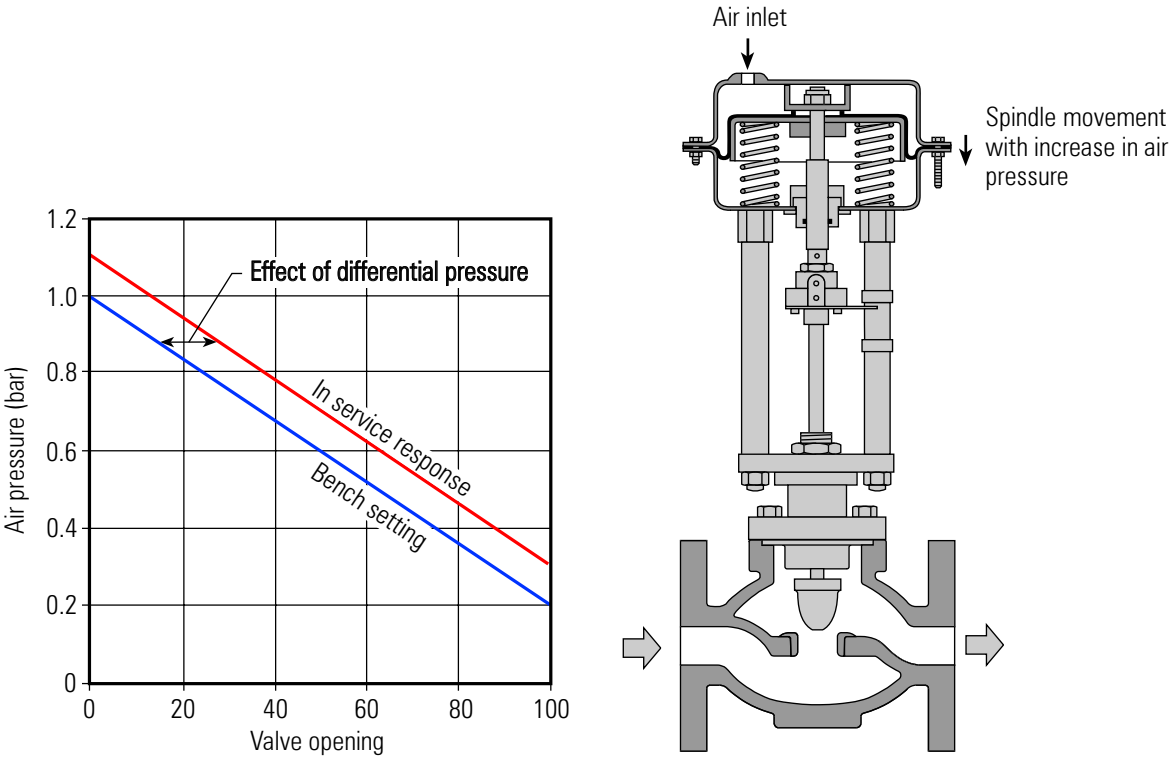


Fig. 6.6.8 Direct acting actuator, air-to-close, direct acting valve - normally open

It may be possible to recalibrate the valve and actuator to take the forces created by differential pressure into account, or perhaps using different springs, air pressure and actuator combinations. This approach can provide an economic solution on small valves, with low differential pressures and where precise control is not required. However, the practicalities are that:

- Larger valves have greater areas for the differential pressure to act over, thus increasing the forces generated, and having an increasing effect on valve position.
- Higher differential pressures mean that higher forces are generated.
- Valves and actuators create friction, causing hysteresis. Smaller valves are likely to have greater friction relative to the total forces involved.

The solution is to fit a positioner to the valve/actuator assembly. (More information is given on positioners later in this Module).

Note: For simplicity, the above examples assume a positioner is not used, and hysteresis is zero. The formulae used to determine the thrust available to hold a valve on its seat for various valve and actuator combinations are shown in Figure 6.6.9.

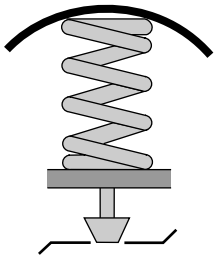
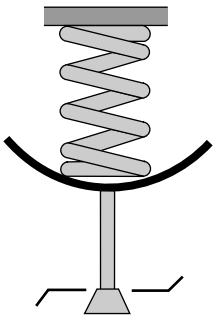
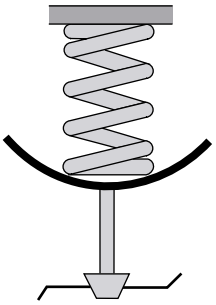
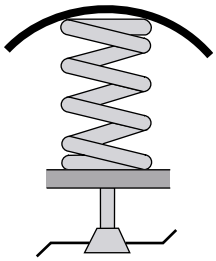
Where:
A = Effective area of diaphragm
P_{max} = Maximum pressure to actuator (normally 1.2 bar)
S_{max} = Maximum bench setting of spring
P_{min} = Minimum pressure to actuator (normally 0 bar)
S_{min} = Minimum bench setting of spring

The thrust available to close the valve has to provide three functions:

1. To overcome the fluid differential pressure at the closed position.
2. To overcome friction in the valve and actuator, primarily at the valve and actuator stem seals.
3. To provide a sealing load between the valve plug and valve seat to ensure the required degree of tightness.

Control valve manufacturers will normally provide full details of the maximum differential pressures against which their various valve and actuator/spring combinations will operate; the Table in Figure 6.6.10 is an example of this data.

Note: When using a positioner, it is necessary to refer to the manufacturer's literature for the minimum and maximum air pressures.

Two port valves				
Actuator action	Direct	Reverse	Reverse	Direct
Valve action	Direct	Reverse	Direct	Reverse
Thrust available to close valve	A (P_{max} - S_{max})		A (P_{min} - S_{min})	

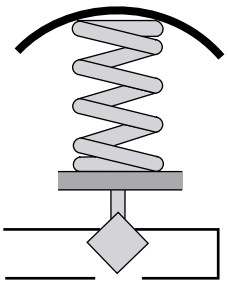
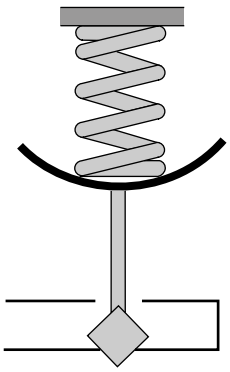
Three port valves (typical mixing valve depicted)		
Actuator action	Direct	Reverse
Thrust available against top seat	A (P_{min} - S_{min})	A (P_{min} - S_{min})
Thrust available against bottom seat	A (P_{max} - S_{max})	A (P_{min} - S_{min})

Fig. 6.6.9 Two and three port formulae

KE and LE valves

Valve size		DN15	DN20	DN25	DN32	DN40	DN50	DN65	DN80	DN100
Actuator	Spring range	Maximum differential pressure (bar)								
PN5123	2.0 to 4.0	40.0	40.0	30.5	14.9	10.3	5.5	-	-	-
PN5126	1.0 to 2.0	34.2	16.1	8.2	3.2	1.1	-	-	-	-
PN5120	0.2 to 1.0	7.7	4.9	-	-	-	-	-	-	-
	0.4 to 1.2	17.6	10.1	4.4	-	-	-	-	-	-
PN5220	0.2 to 1.0	21.3	12.1	5.6	2.2	1.8	0.7	-	-	-
	0.4 to 1.2	40.0	24.6	13.4	6.1	4.5	2.2	-	-	-
PN5226	1.0 to 2.0	40.0	40.0	31.1	14.7	8.0	4.4	-	-	-
PN5223	2.0 to 4.0	40.0	40.0	40.0	38.0	25.6	14.1	-	-	-
PN5320	0.2 to 1.0	34.4	19.1	10.0	4.4	3.3	1.6	-	-	
	0.4 to 1.2	40.0	32.6	22.1	10.6	7.5	3.9	-	-	-
PN5326	1.0 to 2.0	40.0	40.0	40.0	24.0	13.6	7.9	-	-	-
PN5323	2.0 to 4.0	40.0	40.0	40.0	40.0	30.0	22.3	-	-	-
PN5330	0.4 to 1.2	-	-	-	-	-	-	0.7	-	-
PN5336	1.0 to 2.0	-	-	-	-	-	-	4.0	2.3	1.2
PN5333	2.0 to 4.0	-	-	-	-	-	-	11.7	7.4	4.6
PN5420	0.2 to 1.0	40.0	31.3	17.5	8.3	5.9	3.0	-	-	-
	0.4 to 1.2	40.0	40.0	37.2	18.4	12.6	6.8	-	-	-
PN5426	1.0 to 2.0	40.0	40.0	40.0	38.5	22.4	13.3	-	-	-
PN5423	2.0 to 4.0	40.0	40.0	40.0	40.0	30.0	30.0	-	-	-
PN5430	0.4 to 1.2	-	-	-	-	-	-	2.5	1.3	0.6
PN5436	1.0 to 2.0	-	-	-	-	-	-	7.3	4.5	2.6
PN5433	2.0 to 4.0	-	-	-	-	-	-	20.2	13.1	8.3
PN5520	0.2 to 1.0	40.0	40.0	34.0	16.0	11.5	5.6	-	-	-
	0.4 to 1.2	40.0	40.0	40.0	36.0	24.2	13.0	-	-	-
PN5524	0.8 to 1.5	40.0	40.0	40.0	40.0	30.0	27.0	-	-	-
PN5530	0.2 to 1.0	-	-	-	-	-	-	3.8	2.6	1.6
	0.4 to 1.2	-	-	-	-	-	-	7.9	5.2	3.3
PN5534	0.8 to 1.5	-	-	-	-	-	-	15.8	10.4	6.6
PN5620	0.2 to 1.0	40.0	40.0	40.0	22.3	16.0	7.8	-	-	-
	0.4 to 1.2	40.0	40.0	40.0	40.0	30.0	18.1	-	-	-
PN5624	0.8 to 1.5	40.0	40.0	40.0	40.0	30.0	30.0	-	-	-
PN5630	0.2 to 1.0	-	-	-	-	-	-	5.4	3.6	2.3
	0.4 to 1.2*	-	-	-	-	-	-	11.0	7.3	4.6
PN5634	0.8 to 1.5	-	-	-	-	-	-	22.0	14.5	9.2

Fig. 6.6.10 Typical manufacturer's valve/actuator selection chart

Positioners

For many applications, the 0.2 to 1 bar pressure in the diaphragm chamber may not be enough to cope with friction and high differential pressures. A higher control pressure and stronger springs could be used, but the practical solution is to use a positioner.

This is an additional item (see Figure 6.6.11), which is usually fitted to the yoke or pillars of the actuator, and it is linked to the spindle of the actuator by a feedback arm in order to monitor the valve position. It requires its own higher-pressure air supply, which it uses to position the valve.

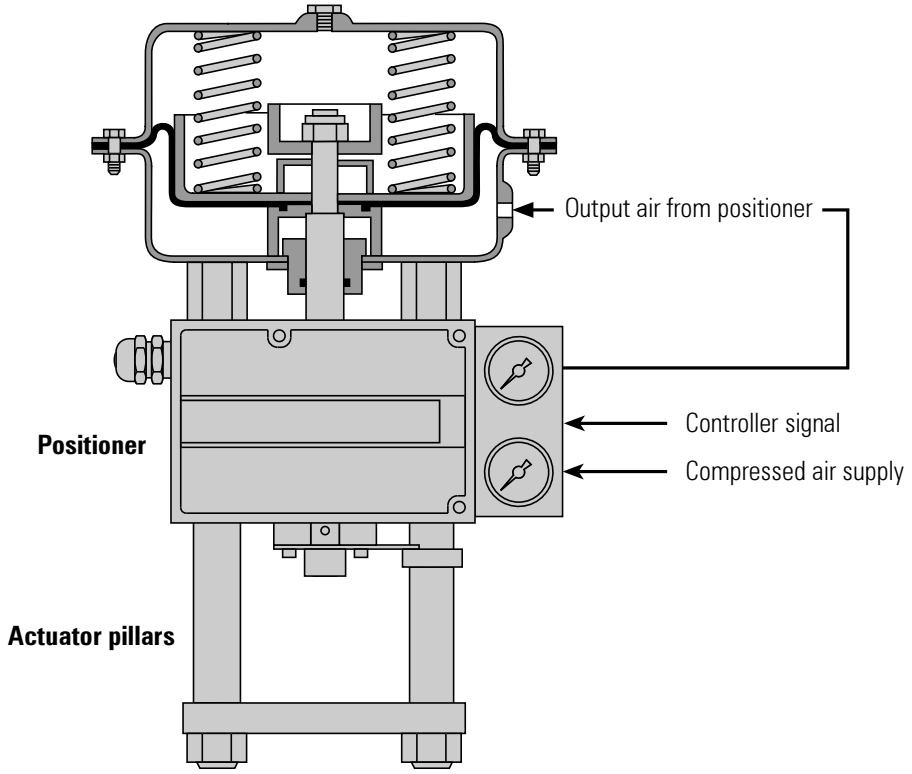


Fig. 6.6.11 Basic pneumatic positioner fitted to actuator pillars (valve not shown)

A valve positioner relates the input signal and the valve position, and will provide any output pressure to the actuator to satisfy this relationship, according to the requirements of the valve, and within the limitations of the maximum supply pressure.

When a positioner is fitted to an 'air-to-open' valve and actuator arrangement, the spring range may be increased to increase the closing force, and hence increase the maximum differential pressure a particular valve can tolerate. The air pressure will also be adjusted as required to overcome friction, therby reducing hysteresis effects.

Example: Taking a PN5400 series actuator fitted to a DN50 valve (see Table in Figure 6.6.10)

1. With a standard 0.2 to 1.0 bar spring range (PN5420), the maximum allowable differential pressure is 3.0 bar.
2. With a 1.0 to 2.0 bar spring set (PN5426), the maximum allowable differential pressure is increased to 13.3 bar.

With the second option, the 0.2 to 1.0 bar signal air pressure applied to the actuator diaphragm cannot provide sufficient force to move an actuator against the force provided by the 1.0 to 2.0 bar springs, and even less able to control it over its full operating range. In these circumstances the positioner acts as an amplifier to the control signal, and modulates the supply air pressure, to move the actuator to a position appropriate to the control signal pressure.

For example, if the control signal was 0.6 bar (50% valve lift), the positioner would need to allow approximately 1.5 bar into the actuator diaphragm chamber. Figure 6.6.12 illustrates this relationship.

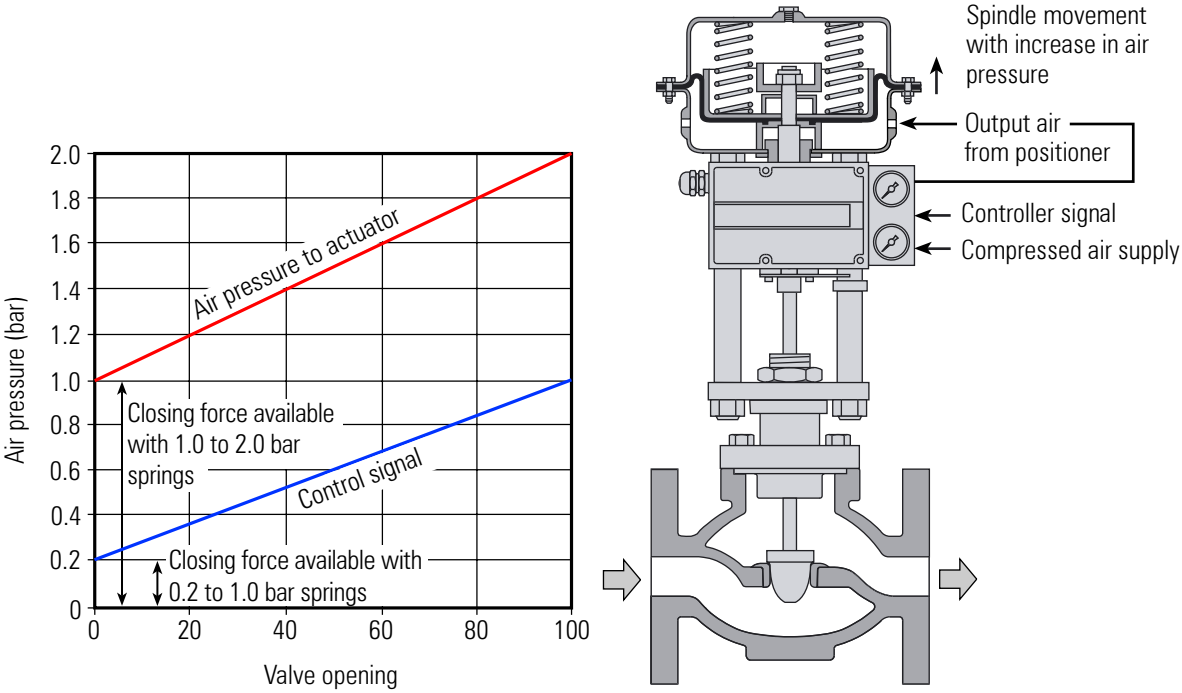


Fig. 6.6.12 The positioner as a signal amplifier

It should be noted that a positioner is a proportional device, and in the same way that a proportional controller will always give an offset, so does a positioner.

On a typical positioner, the proportional band may be between 3 and 6%. The positioner sensitivity can usually be adjusted. It is essential that the installation and maintenance instructions be read prior to the commissioning stage.

Summary - Positioners

1. A positioner ensures that there is a linear relationship between the signal input pressure from the control system and the position of the control valve. This means that for a given input signal, the valve will always attempt to maintain the same position regardless of changes in valve differential pressure, stem friction, diaphragm hysteresis and so on.
2. A positioner may be used as a signal amplifier or booster. It accepts a low pressure air control signal and, by using its own higher pressure input, multiplies this to provide a higher pressure output air signal to the actuator diaphragm, if required, to ensure that the valve reaches the desired position.
3. Some positioners incorporate an electropneumatic converter so that an electrical input (typically 4 - 20 mA) can be used to control a pneumatic valve.
4. Some positioners can also act as basic controllers, accepting input from sensors.

A frequently asked question is, 'When should a positioner be fitted?'

A positioner should be considered in the following circumstances:

1. When accurate valve positioning is required.
2. To speed up the valve response. The positioner uses higher pressure and greater air flow to adjust the valve position.
3. To increase the pressure that a particular actuator and valve can close against. (To act as an amplifier).
4. Where friction in the valve (especially the packing) would cause unacceptable hysteresis.
5. To linearise a non-linear actuator.
6. Where varying differential pressures within the fluid would cause the plug position to vary.

To ensure that the full valve differential pressure can be accepted, it is important to adjust the positioner zero setting so that no air pressure opposes the spring force when the valve is seating.

Figure 6.6.13 shows a typical positioner. Commonly, this would be known as a P to P positioner since it takes a pneumatic signal (P) from the control system and provides a resultant pneumatic output signal (P) to move the actuator.

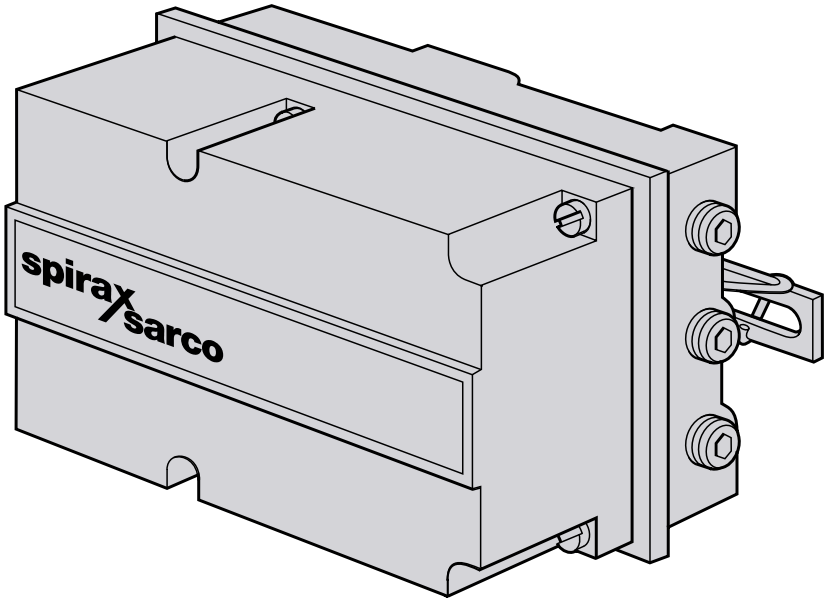


Fig. 6.6.13 Typical P to P positioner (gauges omitted for clarity)

One advantage of a pneumatic control is that it is intrinsically safe, i.e. there is no risk of explosion in a dangerous atmosphere, and it can provide a large amount of force to close a valve against high differential pressure. However, pneumatic control systems themselves have a number of limitations compared with their electronic counterparts.

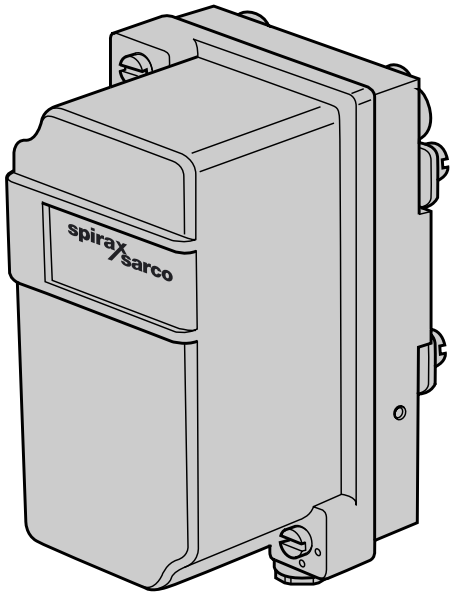


Fig. 6.6.14 Typical I to P converter

To alleviate this, additional components are available to enable the advantages of a pneumatic valve and actuator to be used with an electronic control system.

The basic unit is the I to P converter. This unit takes in an electrical control signal, typically 4 - 20 mA, and converts it to a pneumatic control signal, typically 0.2 - 1 bar, which is then fed into the actuator, or to the P to P positioner, as shown in Figure 6.6.15.

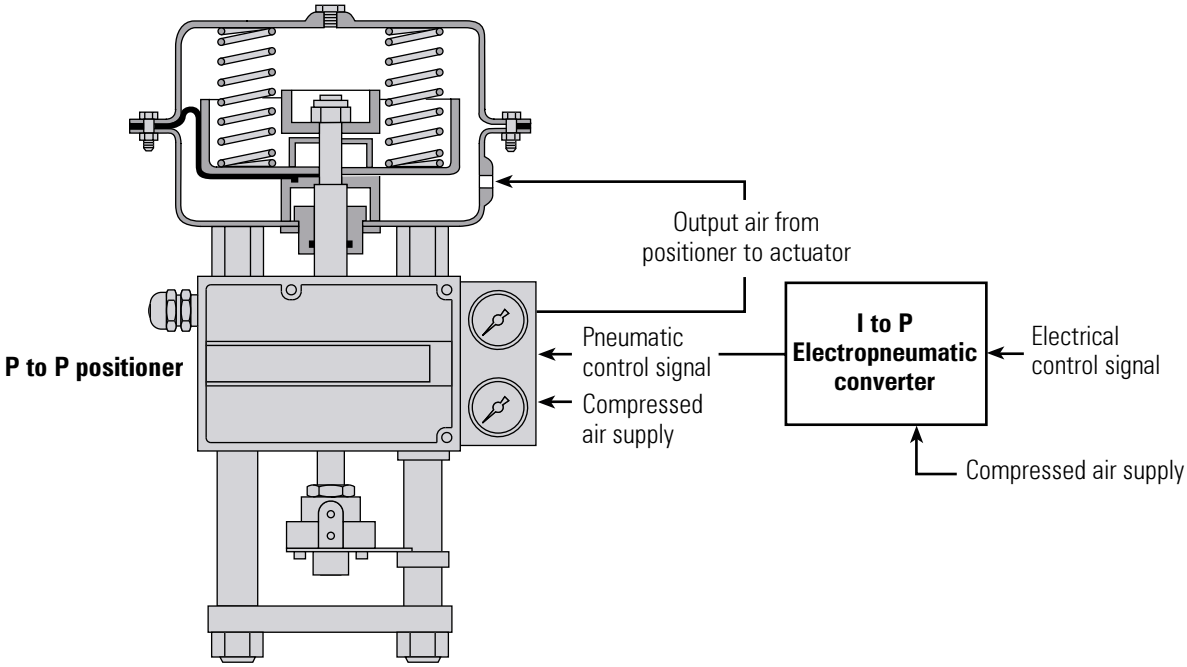


Fig. 6.6.15 Pneumatic valve/actuator operated by a control signal using I to P converter and P to P positioner

With this arrangement, an I to P (electrical to pneumatic) conversion can be carried out outside any hazardous area, or away from any excessive ambient temperatures, which may occur near the valve and pipeline.

However, where the conditions do not present such problems, a much neater solution is to use a single component electropneumatic converter / positioner, which combines the functions of an I to P converter and a P to P positioner, that is a combined valve positioner and electropneumatic converter. Figure 6.6.16 shows a typical I to P converter/positioner.



Fig. 6.6.16 A typical I to P converter/positioner fitted to a pneumatic valve (gauges omitted for clarity)

Most sensors still have analogue outputs (for example 4 - 20 mA or 0 - 10 V), which can be converted to digital form. Usually the controller will perform this analogue-to-digital (A / D) conversion, although technology is now enabling sensors to perform this A/D function themselves. A digital sensor can be directly connected into a communications system, such as Fieldbus, and the digitised data transmitted to the controller over a long distance. Compared to an analogue signal, digital systems are much less susceptible to electrical interference.

Analogue control systems are limited to local transmission over relatively short distances due to the resistive properties of the cabling.

Most electrical actuators still require an analogue control signal input (for example 4 - 20 mA or 0 - 10 V), which further inhibits the completion of a digital communications network between sensors, actuators, and controllers.

Digital positioners

Sometimes referred to as a SMART positioner, the digital positioner monitors valve position, and converts this information into a digital form. With this information, an integrated microprocessor offers advanced user features such as:

- High valve position accuracy.
- Adaptability to changes in control valve condition.
- Many digital positioners use much less air than analogue types.
- An auto stroking routine for easy setting-up and calibration.
- On-line digital diagnostics*
- Centralised monitoring*

*Using digital communications protocols such as HART® ; Fieldbus, or Profibus.

The current industrial trend is to provide equipment with the capability to communicate digitally with networked systems in a Fieldbus environment. It is widely thought that digital communications of this type offer great advantages over traditional analogue systems.



Fig. 6.6.17 Digital positioner

Selecting a pneumatic valve and actuator

In summary, the following is a list of the major factors that must be considered when selecting a pneumatic valve and actuator:

1. Select a valve using the application data.
2. Determine the valve action required in the event of power failure, fail-open or fail-closed.
3. Select the valve actuator and spring combination required to ensure that the valve will open or close against the differential pressure.
4. Determine if a positioner is required.
5. Determine if a pneumatic or electric control signal is to be provided. This will determine whether an I to P converter or, alternatively a combined I to P converter/positioner, is required.

Rotary pneumatic actuators and positioners

Actuators are available to drive rotary action valves, such as ball and butterfly valves. The commonest is the piston type, which comprises a central shaft, two pistons and a central chamber all contained within a casing. The pistons and shaft have a rack and pinion drive system.

In the simplest types, air is fed into the central chamber (Figure 6.6.18a), which forces the pistons outwards.

The rack and pinion arrangement turns the shaft and, because the latter is coupled to the valve stem, the valve opens or closes.

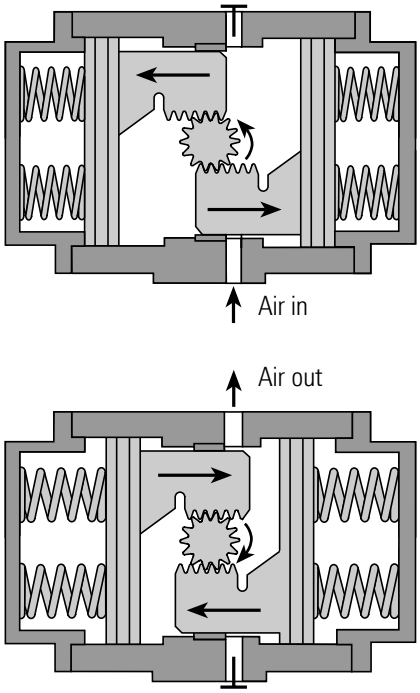
When the air pressure is relieved, movement of the shaft in the opposite direction occurs due to the force of the return springs (Figure 6.6.18b).

It is also possible to obtain double acting versions, which have no return springs. Air can be fed into either side of the pistons to cause movement in either direction. As with diaphragm type actuators, they can also be fitted with positioners.

a

Anticlockwise

Air is supplied forcing the pistons away from each other (towards the ends), rotating the drive pinion anticlockwise.



b

Clockwise

Air failure (loss of pressure) allows compressed springs to force pistons towards each other (toward centre), rotating the drive pinion clockwise and exhausting the air.

Fig. 6.6.18 Spring return rotary pneumatic actuator

Air supply

An adequate compressed air supply system is essential to provide clean and dry air at the right quantity and pressure. It is advantageous to install an individual coalescing filter/regulator unit ahead of the final supply connection to each piece of equipment. Air quality is particularly important for pneumatic instrumentation such as controllers, I to P convertors and positioners.

The decision to opt for a pneumatically operated system may be influenced by the availability and/or the costs to install such a system. An existing air supply would obviously encourage the use of pneumatically powered controls.

Electrical actuators

Where a pneumatic supply is not available or desirable it is possible to use an electric actuator to control the valve. Electric actuators use an electric motor with voltage requirements in the following range: 230 Vac, 110 Vac, 24 Vac and 24 Vdc.

There are two types of electrical actuator; VMD (Valve Motor Drive) and Modulating.

VMD (Valve Motor Drive)
This basic version of the electric actuator has three states:

1. Driving the valve open.
2. Driving the valve closed.
3. No movement.

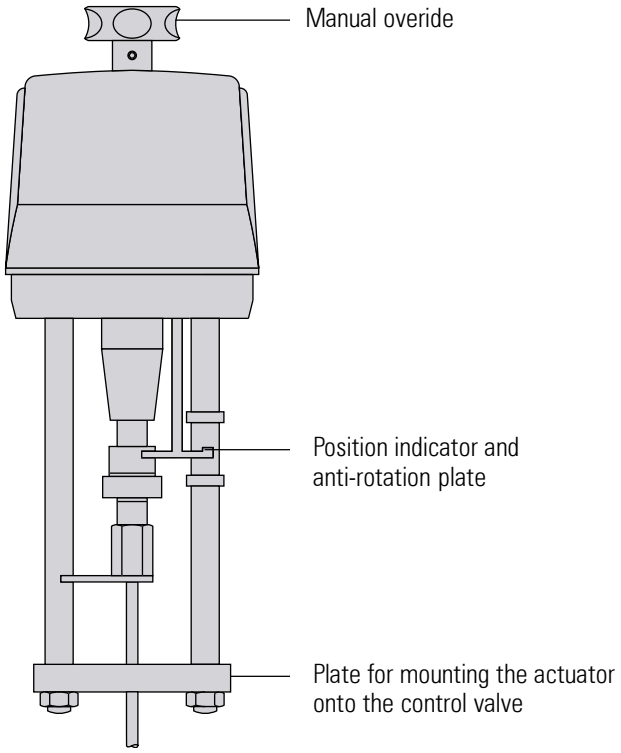


Fig. 6.6.19 Typical electric valve actuator

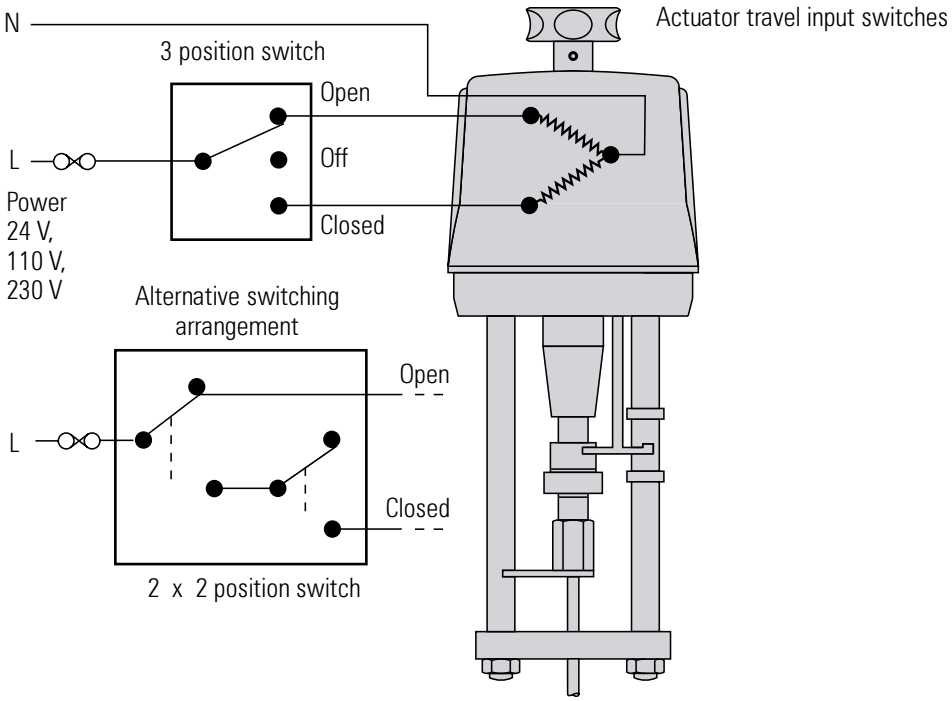


Fig. 6.6.20 Valve motor drive actuator system

Figure 6.6.20 shows the VMD system where the forward and reverse travel of the actuator is controlled directly from any external 3-position or two 2-position switch units. The switches are rated at the actuator voltage and may be replaced by suitable relays.

Limiting devices are fitted within the VMD actuators to protect the motors from over-travel damage. These devices are based on either the maximum motor torque or physical position limit switches. Both devices stop the motor driving by interrupting the motor power supply.

- Position limit switches have the advantage that they can be adjusted to limit valve strokes in oversized valves.
- Torque switches have the advantage of giving a defined closing force on the valve seat, protecting the actuator in the case of valve stem seizure.
- If only position limit switches are used, they may be combined with a spring-loaded coupling to ensure tight valve shut-off.

A VMD actuator may be used for on/off actuation or for modulating control. The controller positions the valve by driving the valve open or closed for a certain time, to ensure that it reaches the desired position. Valve position feedback may be used with some controllers.

Modulating

In order to position the control valve in response to the system requirements a modulating actuator can be used. These units may have higher rated motors (typically 1 200 starts/hour) and may have built-in electronics.

A positioning circuit may be included in the modulating actuator, which accepts an analogue control signal (typically 0-10 V or 4-20 mA). The actuator then interprets this control signal, as the valve position between the limit switches.

To achieve this, the actuator has a position sensor (usually a potentiometer), which feeds the actual valve position back to the positioning circuit. In this way the actuator can be positioned along its stroke in proportion to the control signal. A schematic of the modulating actuator is shown in Figure 6.6.21.

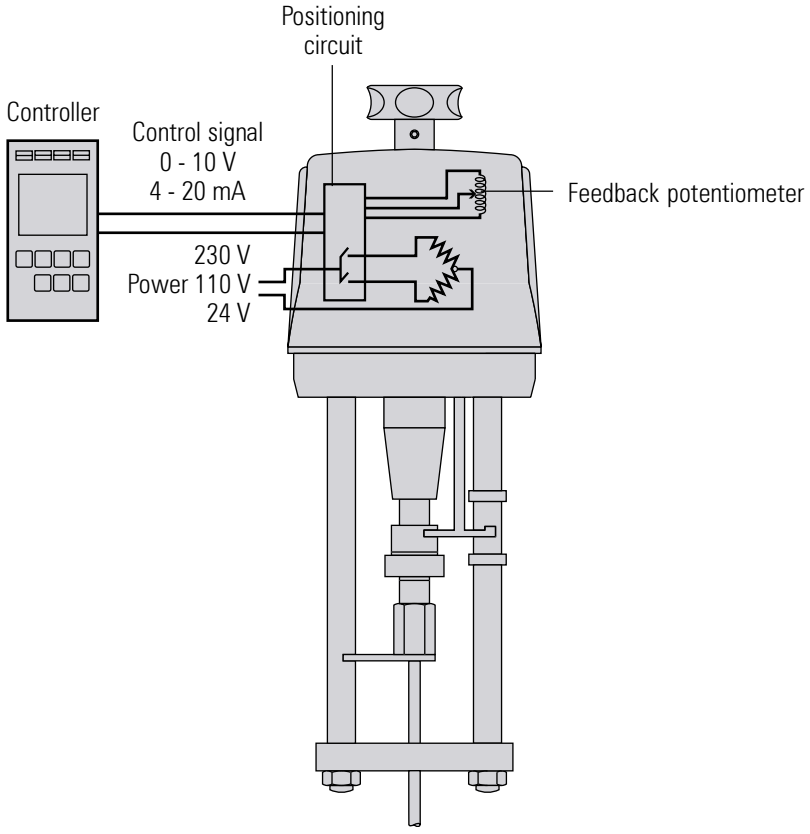


Fig. 6.6.21 Integral positioning circuit for modulating electric actuators

Pneumatic actuators have an inherent fail-safe feature; should the air supply or control signal fail the valve will close. To provide this function in electric actuators, 'spring reserve' versions are available which will open or close the valve on power or control signal failure. Alternatively, fail-safe can be provided with battery power.

Electric actuators offer specified forces, which may be limited on spring reserve versions. The manufacturer's charts should always be consulted during selection.

When sizing an actuator, it is wise to refer to the manufacturer's technical data sheets for maximum differential pressure across the valve (see Figure 6.6.22).

Another limitation of an electric actuator is the speed of valve movement, which can be as low as 4 seconds/mm, which in rapidly varying systems may be too slow.

EL series actuators

Valve size		DN15	DN20	DN25	DN32	DN40	DN50	DN65	DN80	DN100
Actuator	**Voltage**	**Maximum differential pressure (bar)**								
EL5601	230	40.0	30.3	18.3	9.3	5.4	2.9	1.2	0.6	0.3
EL5602	110	40.0	30.3	18.3	9.3	5.4	2.9	1.2	0.6	0.3
EL5603	24	40.0	30.3	18.3	9.3	5.4	2.9	1.2	0.6	0.3
EL5611	230	40.0	40.0	38.3	19.8	12.0	6.7	3.5	2.2	1.3
EL5612	110	40.0	40.0	38.3	19.8	12.0	6.7	3.5	2.2	1.3
EL5613	24	40.0	40.0	38.3	19.8	12.0	6.7	3.5	2.2	1.3
EL5621	230			40.0	40.0	28.5	16.3	9.3	6.1	3.8
EL5622	110			40.0	40.0	28.5	16.3	9.3	6.1	3.8
EL5623	24			40.0	40.0	28.5	16.3	9.3	6.1	3.8
EL5631	230					40.0	29.7	17.5	11.5	7.4
EL5632	110				40.0	29.7	17.5	11.5	7.4	
EL5633	24					40.0	29.7	17.5	11.5	7.4
EL5641	230						40.0	26.7	17.8	11.4
EL5642	110						40.0	26.7	17.8	11.4
EL5643	24						40.0	26.7	17.8	11.4
EL5651	230							40.0	38.0	24.6
EL5652	110							40.0	38.0	24.6
EL5653	24							40.0	38.0	24.6

Fig. 6.6.22 Typical manufacturer's electric actuator selection chart

Questions

1. In a reverse acting actuator what happens upon air failure?

a | The valve spindle does not move ☐
b | The valve spindle retracts ☐
c | The valve spindle extends ☐
d | The valve will always close ☐

2. **In a direct acting actuator what happens upon air failure?**

a | The valve spindle does not move ☐
b | The valve spindle retracts ☐
c | The valve spindle extends ☐
d | The valve will always open ☐

3. **With a direct acting actuator on a reverse acting valve, what happens upon air failure?**

a | The valve spindle does not move ☐
b | The valve closes ☐
c | The valve opens ☐
d | It is not possible to fit this combination of actuator and valve ☐

4. **With a reverse acting actuator on a direct acting 2-port valve, what is required due to the effect of differential pressure?**

a | The closing force must decrease ☐
b | The air pressure must decrease ☐
c | The air pressure must increase ☐
d | It is not possible to fit this combination of actuator and valve ☐

5. What is the difference between an I to P positioner and I to P converter?

a | The positioner is fitted off the valve, the converter on the valve ☐
b | The positioner and converter are both fitted on the valve ☐
c | The positioner and converter are both fitted off the valve ☐
d | The positioner is fitted on the valve, the converter off the valve ☐

6. A VMD electric actuator can only be used for on/off control – true or false?

a | True ☐
b | False ☐

Answers

1: c, 2: b, 3: b, 4: b, 5: b, 6: b

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Module 6.7

Controllers and Sensors

Controllers

It is important to state at the outset that not all control applications need a sophisticated controller.

An on/off valve and actuator, for example, can be operated directly from a thermostat. Another example is the operation of high limit safety controls, which have a 'snap' action to close valves or to switch off fuel supplies.

However, when the control requirements become more sophisticated, a controller is needed to match these requirements.

The controller receives a signal, decides what action is needed and then sends a signal to the actuator to make it move.

In the age of the microchip, integrated circuits and computers, the functions performed by the controller can be very complex indeed.

However, since an analogy between the human brain and controllers/computers has been made in previous Modules, the renowned IBM motto can be paraphrased:

Computer - Fast, accurate and stupid

Human being - Slow, slovenly and brilliant

To summarise, the controller will not solve all problems. It must be properly selected and commissioned, subjects which will be dealt with later.

Although most controllers are now electronic digital/microprocessor based, a range of pneumatic controllers is commercially available. These might be used in hazardous areas where the risk of explosion precludes the use of electrics/electronics. It is possible to make electrical equipment 'intrinsically safe' or explosion-proof or flameproof, however, there is usually a substantial cost implication.

As previously mentioned, the functions carried out by the controller can be very complex and it is beyond the scope of this publication to list them in detail, or to explain how they operate.

The major variations that require consideration are as follows:

Single loop controller
Operates one valve/actuator from a single sensor.

Multi-loop controller
May operate more than one valve/actuator from more than one sensor.

Single input/output
Can accept only one signal from the sensor and send only one to the actuator.

Multi-input/output (multi-channel)
Can accept several signals and send out several signals.

Real time
May include a time clock to switch at pre-determined, pre-set times.

Elapsed time
May switch at some predetermined, pre-set length of time before or after other items of plant have been switched on or off.

Ramp and dwell
Using temperature as an example, the capability to raise the temperature of a controlled medium over a specified time period and then to hold it at a pre-set value. Such controllers frequently incorporate a series of ramps and dwells.

Figure 6.7.1, shows a typical electronic, single loop controller. This has P + I + D action (discussed in Modules 5.2 and 5.4), suitable for 110 or 230 volt supply.

Figure 6.7.2 shows a pneumatic single loop controller with P action.

Different models can be selected to control either temperature or pressure.



Fig. 6.7.1 Electronic single loop controller



Fig. 6.7.2 Pneumatic single loop temperature controller

A single loop controller, which has the ability to perform ramp and dwell functions, may have a typical sequence pattern like the one shown in Figure 6.7.3. This shows a series of ramps (temperature change) and dwell (maintaining temperature) functions, carried out over a period of time.

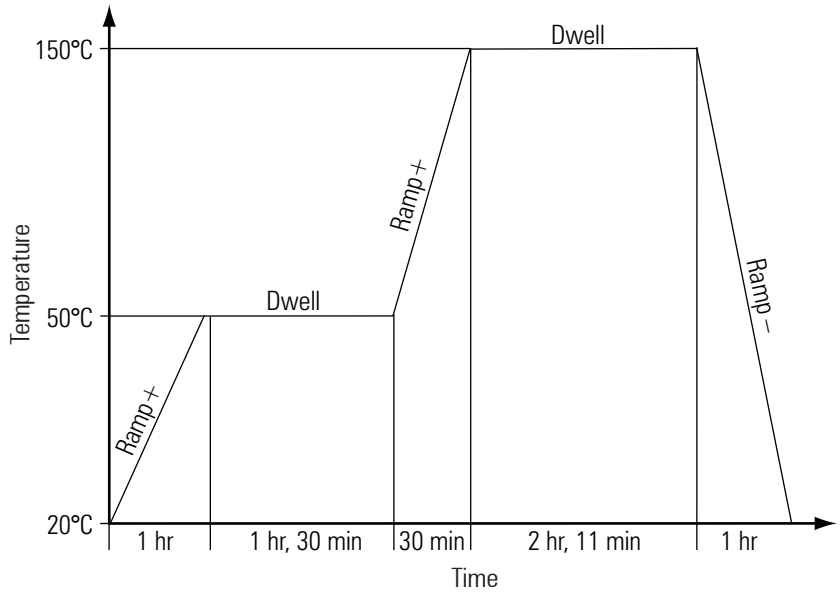


Fig. 6.7.3 Typical multi-sequence ramp and dwell pattern

One term frequently found in control literature is 'Programmable Logic Controller (PLC)'. In a batch process, the controller must trigger a sequence of actions, for example, turning valves or pumps on or off. In some cases the whole sequence is on a timed basis, but often the various steps may be triggered by a specific condition being reached and maintained for a certain time period; for example a certain temperature being reached or a vessel filled. These sequences can be controlled by a PLC, a microcomputer-based device that utilises standard interfaces for sensors and actuators to control the process.

Another type of complex controller is the plant room controller, which might be used to control the boiler, pump, heating control valve, HWS valve, as well as providing a number of other features.

Sensors

In this Section the subject of temperature measurement will be covered more broadly. There are a wide variety of sensors and transducers available for measuring pressure, level, humidity, and other physical properties. The sensor is the part of the control system, which experiences the change in the controlled variable.

The sensor may be of a type where a change in temperature results in a change of voltage or perhaps a change in resistance.

The signal from the sensor may be very small, creating the need for local signal conditioning and amplification to read it effectively. A small change in resistance signalled by a sensor in response to a change in temperature, may, for example, be converted to an electrical voltage or current for onward transmission to the controller.

The transmission system itself is a potential source of error.

Wiring incurs electrical resistance (measured in ohms), as well as being subject to electrical interference (noise). In a comparable pneumatic system, there may also be minute leaks in the piping system.

The term 'thermostat' is generally used to describe a temperature sensor with on/off switching. 'Transducer' is another common term, and refers to a device that converts one physical characteristic into another; for example, temperature into voltage (millivolts).

An example of a transducer is a device that converts a change in temperature to a change in electrical resistance.

With pneumatic devices, the word 'transmitter' is frequently encountered. It is simply another description of transducer or sensor, but usually with some additional signal conditioning.

However, the actual measuring device is usually termed as the sensor, and the more common types will be outlined in the following Section.

Filled system sensors

With pneumatic controllers, filled system sensors are employed. Figure 6.7.4 illustrates the principles of such a system.

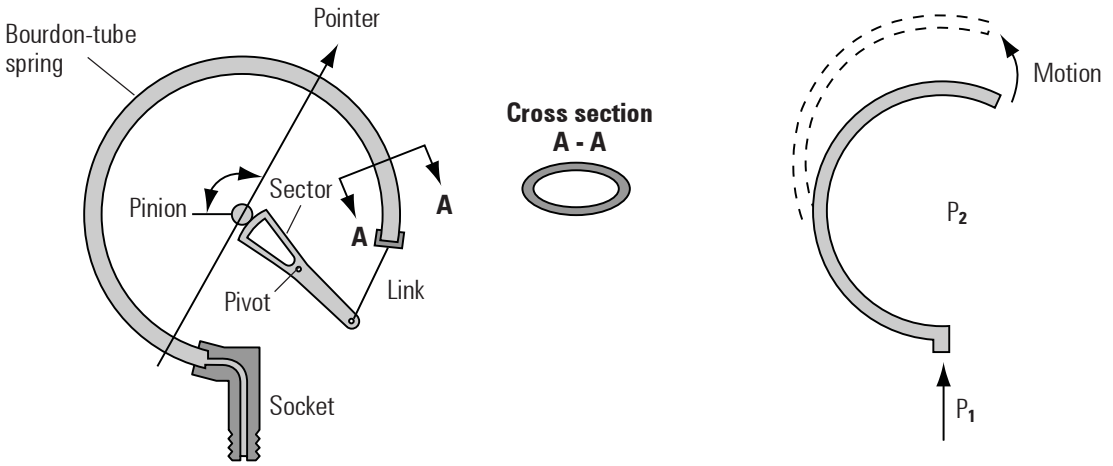


Fig. 6.7.4 Liquid filled system sensor and gas filled or vapour pressure system

When the temperature changes, the fluid expands or contracts, causing the Bourdon tube to tend to straighten out. Sometimes a bellows is used instead of a Bourdon tube.

In the past, the filling has often been mercury. When heated, it expands, causing the Bourdon tube to uncoil; cooling causes contraction and forces the Bourdon tube to coil more tightly. This coil movement is used to operate levers within the pneumatic controller enabling it to perform its task. A pressure sensing version will simply utilise a pressure pipe connected to the Bourdon tube. Note: for health and safety reasons, mercury is now used less often. Instead, an inert gas such as nitrogen is often employed.

Resistance temperature detectors (RTDs)

RTDs (Figure 6.7.5) employ the fact that the electrical resistance of certain metals change as the temperature alters. They act as electrical transducers, converting temperature changes to changes in electrical resistance. Platinum, copper, and nickel are three metals that meet RTD requirements and Figure 6.7.6 shows the relationship between resistance and temperature.

A resistance temperature detector is specified in terms of its resistance at 0°C and the change in resistance from 0°C to 100°C. The most widely used RTD for the typical applications covered in these Modules are platinum RTDs. These are constructed with a resistance of 100 ohms at 0°C and are often referred to as Pt100 sensors. They can be used over a temperature range of -200°C to +800°C with high accuracy (±0.5%) between 0°C and 100°C.

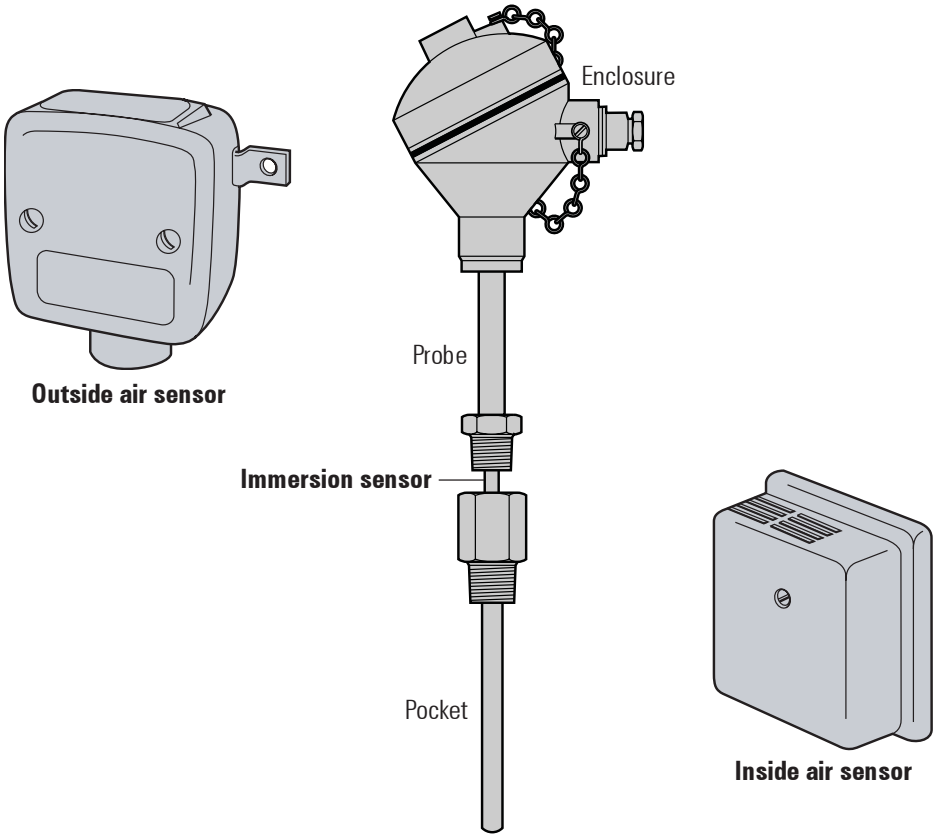


Fig. 6.7.5 Typical resistance temperature sensors

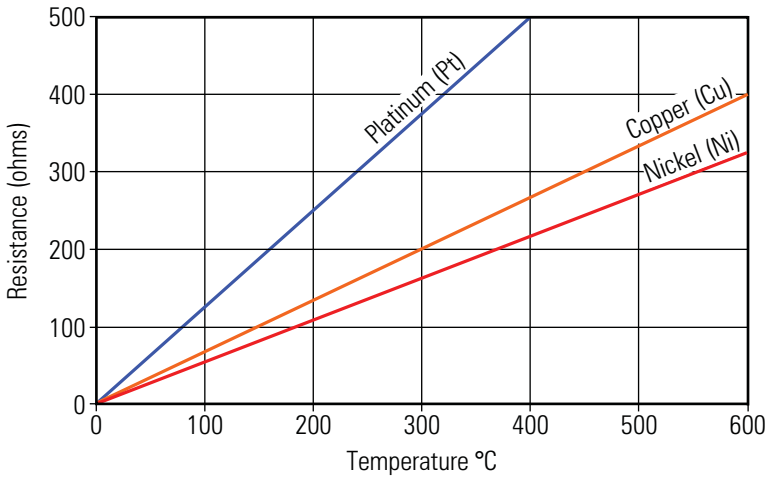


Fig. 6.7.6 RTD element typical resistance/temperature graphs

As can be seen from Figure 6.7.6, the increase of resistance with temperature is virtually linear. RTDs have a relatively small change in resistance, which requires careful measurement. Resistance in the connecting cables needs to be properly compensated for.

Thermistors

Thermistors use semi-conductor materials, which have a large change in resistance with increasing temperature, but are non-linear. The resistance decreases in response to rising temperatures (negative coefficient thermistor), as shown in Figure 6.7.7.

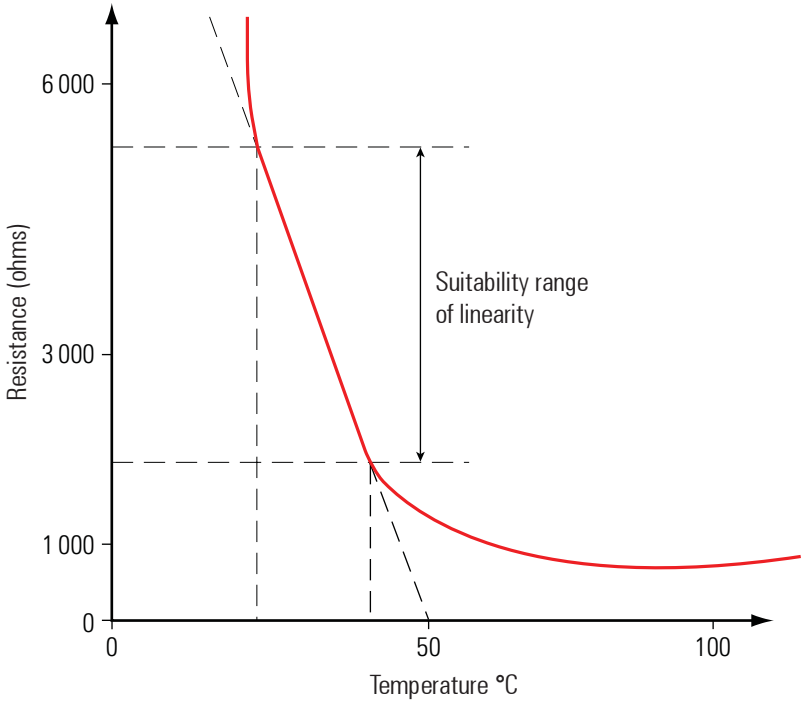


Fig. 6.7.7 Negative coefficient thermistor

Positive coefficient thermistors can be manufactured where the resistance increases with rising temperature (Figure 6.7.8) but their response curve makes them generally unsuitable for temperature sensing.

Thermistors are less complex and less expensive than RTDs but do not have the same high accuracy and repeatability. Their high resistance means that the resistance of the connecting cable is less important.

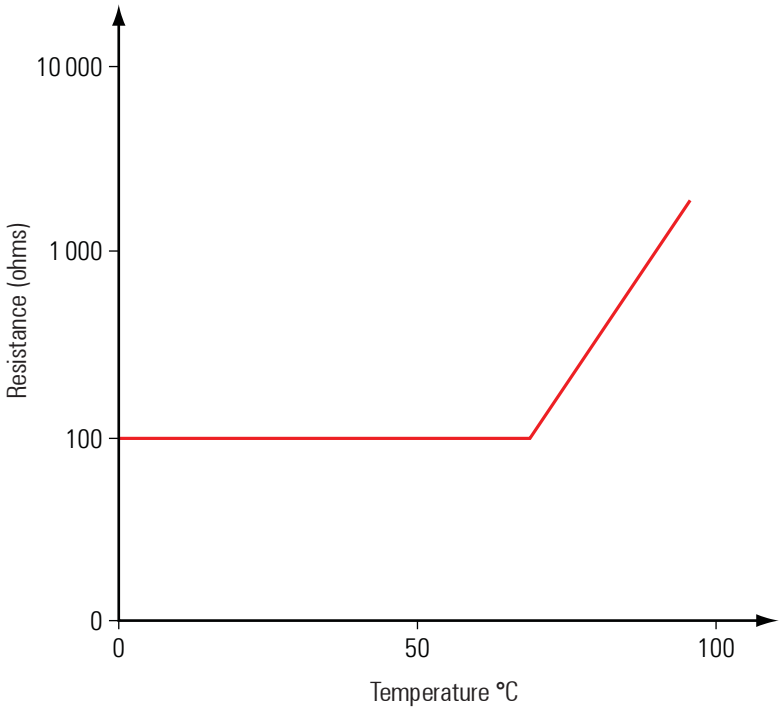


Fig. 6.7.8 Positive coefficient thermistor

Thermocouples

If two dissimilar metals are joined at two points and heat is applied to one junction (as shown in Figure 6.7.9), an electric current will flow around the circuit. Thermocouples produce a voltage corresponding to the temperature difference between the measuring junction (hot) and the reference junction (cold).

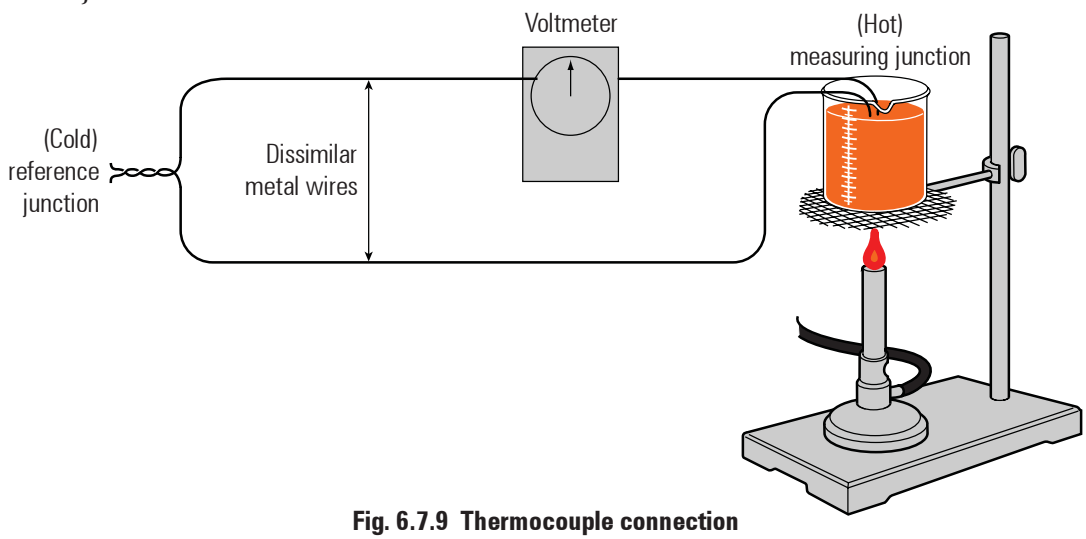


Fig. 6.7.9 Thermocouple connection

The cold reference junction temperature must be accurately known if the thermocouple itself is to provide accurate sensing.

Traditionally, the cold junction was immersed in melting ice (0°C), but the temperature of the cold junction is now measured by a thermistor or an RTD and, from this, the indicated temperature, generally at the measuring junction, is corrected. This is known as cold junction compensation.

Any pair of dissimilar metals could be used to make a thermocouple. But over the years, a number of standard types have evolved which have a documented voltage and temperature relationship. The standard types are referred to by the use of letters, that is, Type J, K, T and others.

Table 6.7.1 Standard range of thermocouples and their range (°C)

Thermocouple ISA Type designation	J	K	T	R	S	N	B	L
Temperature Range (°C)	-200 to +1 000	0 to 1 260	-200 to +400	0 to 1 760	0 to 1 760	0 to 1 760	0 to 1 760	0 to 500

The most widely used general-purpose thermocouple is Type K.

The dissimilar metals used in this type are Chrome (90% nickel, 10% chromium) and Alumel (94% nickel, 3% manganese, 2% aluminium and 1% silicon) and can be used between the range 0°C to 1 260°C. Figure 6.7.10 illustrates the sensitivity of Type K thermocouples, and it can be seen that the output voltage is linear across the complete range.

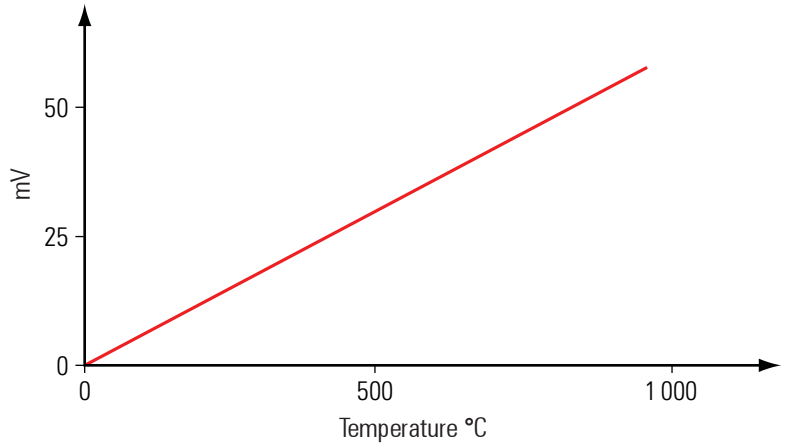


Fig. 6.7.10 Sensitivity of Type K thermocouple

Extension tail wires are used to connect the measuring junction to the reference junction in the instrument case. These extension tails may be of the same material as the wires in the thermocouple itself, or may be a compensating cable made of copper and copper-nickel alloy. Both extension tails must be of the same material.

Thermocouples are available in a wide variety of sizes and shapes. They are inexpensive and rugged and reasonably accurate, with wide temperature ranges. However, the reference junction temperature must be held at a constant value otherwise deviations must be compensated for. The low junction voltages mean that special screened cable and careful installation must be used to prevent electrical interference or 'noise' from distorting signals.

Electrical communication signals

The output signals from most control systems are low power analogue signals but there is a growing use of digital systems such as 'Fieldbus®' or 'PROFIBUS®'.

An analogue system provides a continuous but modulating signal whereas a digital system provides a stream of binary numeric values represented by a change between two specific voltage levels or frequencies.

A comparison between digital and analogue systems can be made using Example 6.7.1 and Example 6.7.2:

Example 6.7.1

Imagine two people, person **A** and person **B**, each on opposite hilltops and each with a flag and a flag-pole. The aim is for person **A** to communicate to person **B** by raising his flag to a certain height. Person **A** raises his flag half way up his pole. Person **B** sees this and also raises his flag halfway. As person **A** moves his flag up or down so does person **B** to match. This would be similar to an analogue system.

Example 6.7.2

Now assume that person **A** does not have a pole but instead has two boards, one with the figure '0' and the other with the figure '1', and again wants person **B** to raise his flag half way, that is to a height of 50% of his flag-pole. The binary number for 50 is 110010, so he displays his boards, two at a time, in the corresponding order. Person **B** reads these boards, translates them to mean 50 and raises his flag exactly half way. This would be similar to a digital system.

It can be seen that the digital system is more precise as the information is either a '1' or a '0' and the position can be accurately defined. The analogue example is not so precise because person **B** cannot determine if person **A**'s flag is at exactly 50%. It could be at 49% or 51%. It is for this reason, together with higher integration of microprocessor circuitry that digital signals are becoming more widely used.

Digital addressing

Digital addressing allows a controller to send information over a set of wires onto which several receivers are connected and yet be able to communicate with only one of those receivers if required. This is done by allocating an address to each receiver, which the controller must broadcast first.

To explain this, consider the digital example above but now assume that there is another person, person **C** on a third hill. Person **B** and person **C** can both see person **A**, so person **A** must first indicate to whom he is communicating.

This is done with the first board. If the first board is a '0' then all subsequent data is intended for person **B** who adjusts his flag accordingly. Conversely, if the first board is a '1' then all subsequent data is intended for person **C**. Hence person **B** has a digital address of '0' and person **C** has a digital address of '1'; each person knows that the first number to be seen by them refers to the address not the message.

HART®, PROFIBUS® and Foundation™ Fieldbus.

What is HART®?

HART® stands for 'Highway Addressable Remote Transducer' and is a standard originally developed as a communications protocol for control field devices operating on a 4-20 mA control signal.

The HART® protocol uses 1200 baud Frequency Shift Keying (FSK) based on the Bell 202 standard to superimpose digital information on the conventional 4-20 mA analogue signal. Maintained by an independent organisation, the HART® Communication Foundation, the HART® protocol is an industry standard developed to define the communications protocol between intelligent field devices and a control system.

HART® is probably the most widely used digital communication protocol in the process industries, and:

- Is supported by all of the major suppliers of process field instruments.
- Preserves existing control strategies by allowing 4-20 mA signals to co-exist with digital communication on existing 2-wire loops.
- Is compatible with analogue devices.
- Provides important information for installation and maintenance, such as Tag-IDs, measured values, range and span data, product information and diagnostics.
- Can support cabling savings through use of multidrop networks.
- Reduces operating costs via improved management and utilisation of smart instrument networks.

What is PROFIBUS®?

PROFIBUS® is an open fieldbus standard for a wide range of applications in manufacturing and process automation independent of manufacturers. Manufacture independence and transparency are ensured by the international standards EN 50170, EN 50254 and IEC 61158.

It allows communication between devices of different manufacturers without any special interface adjustment. PROFIBUS® can be used for both high-speed time critical applications and complex communication tasks. PROFIBUS® offers functionally graduated communication protocols DP and FMS. Depending on the application, the transmission technologies RS-485, IEC 1158-2 or fibre optics can be used.

It defines the technical characteristics of a serial Fieldbus® system with which distributed digital programmable controllers can be networked, from field level to cell level. PROFIBUS® is a multi-master system and thus allows the joint operation of several automation, engineering or visualization systems with their distributed peripherals on one bus.

At sensor/actuator level, signals of the binary sensors and actuators are transmitted via a sensor/actuator bus. Data are transmitted purely cyclically.

At field level, the distributed peripherals, such as I/O modules, measuring transducers, drive units, valves and operator terminals communicate with the automation systems via an efficient, real-time communication system. As with data, alarms, parameters and diagnostic data can also be transmitted cyclically if necessary.

At cell level, programmable controllers such as PLC and IPC can communicate with each other. The information flow requires large data packets and a large number of powerful communication functions, such as smooth integration into company-wide communication systems, such as Intranet and Internet via TCP/IP and Ethernet.

What is Foundation™ Fieldbus?

Foundation™ Fieldbus is an all-digital, serial, two-way communications system that serves as a Local Area Network (LAN) for factory/plant instrumentation and control devices. The Fieldbus® environment is the base level group of the digital networks in the hierarchy of plant networks. Foundation™ Fieldbus is used in both process and manufacturing automation applications and has a built-in capability to distribute the control application across the network.

Unlike proprietary network protocols, Foundation™ Fieldbus is neither owned by any individual company, nor regulated by a single nation or standards body. The Foundation™ Fieldbus, a not-for-profit organization consisting of more than 100 of the world's leading controls and instrumentation suppliers and end users, controls the technology.

While Foundation™ Fieldbus retains many of the desirable features of the 4-20 mA analogue system, such as a standardized physical interface to the wire, bus-powered devices on a single wire, and intrinsic safety options, it also offers many other benefits.

Device interoperability
Foundation™ Fieldbus offers interoperability; one Fieldbus® device can be replaced by a similar device with added functionality from a different supplier on the same Fieldbus® network while maintaining specified operations. This permits users to 'mix and match' field devices and host systems from various suppliers. Individual Fieldbus® devices can also transmit and receive multivariable information, and communicate directly with each other over a common Fieldbus®, allowing new devices to be added to the Fieldbus® without disrupting services.

Enhanced process data
With Foundation™ Fieldbus, multiple variables from each device can be brought into the plant control system to analyse trends, optimise processes, and generate reports. Access to accurate, high-resolution data enables processes to be fine-tuned for better productivity, less downtime, and higher plant performance.

Overall view of the process
Modern Fieldbus® devices, with powerful microprocessor-based communications capabilities, permit process errors to be recognized faster and with greater certainty. As a result, plant operators are notified of abnormal conditions or the need for preventive maintenance, allowing personnel to consider pro-active decisions. Lower operating efficiencies are corrected more quickly, enabling production to rise while raw material costs and regulatory problems fall.

Improved in plant safety
Fieldbus technology helps manufacturing plants keep up with stringent safety requirements. It can provide operators with earlier warning of potential hazardous conditions, thereby allowing corrective action to be taken to reduce unplanned shutdowns. Enhanced plant diagnostic capabilities also offer less frequent access to hazardous areas, thus minimizing the risks to personnel.

Easier predictive maintenance
Enhanced device diagnostics capabilities make it possible to monitor and track insidious conditions such as valve wear and transmitter fouling. Plant personnel are able to perform predictive maintenance without waiting for a scheduled shutdown, thus reducing or even avoiding downtime.

Reduced wiring and maintenance costs
The use of existing wiring and multi-drop connections provides significant savings in network installation costs. This includes reductions in intrinsic safety barriers and cabling costs, particularly in areas where wiring is already in situ.

Additional cost savings can be achieved through the decreased time required for construction and start-up, as well as simplified programming of control and logic functions using software control blocks built into Fieldbus® devices.

Questions

1. If the temperature of a RTD sensor increases by 150°C, what happens to its electrical resistance?

a | The resistance falls ☐

b | The resistance remains the same ☐

c | The resistance rises ☐

2. What main advantage does a thermistor have over a RTD sensor?

a | It is more accurate ☐

b | It has a higher repeatability ☐

c | It is cheaper to buy ☐

d | It is linear over its complete range ☐

3. What main advantage does a thermocouple have over a RTD sensor?

a | It is more accurate ☐

b | It has a higher repeatability ☐

c | It is cheaper to buy ☐

d | It is linear over its complete range ☐

Answers

1: c, 2: c, 3: c

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Module 7.1

Self-acting Temperature Controls

Self-acting Temperature Controls

What are self-acting temperature controls and how do they operate?

There are two main forms of self-acting temperature control available on the market: Liquid filled systems and vapour tension systems.

Self-acting temperature controls are self-powered, without the need for electricity or compressed air.

The control system is a single-piece unit comprising a sensor, capillary tubing and an actuator. This is then connected to the appropriate control valve, as shown in Figure 7.1.1.

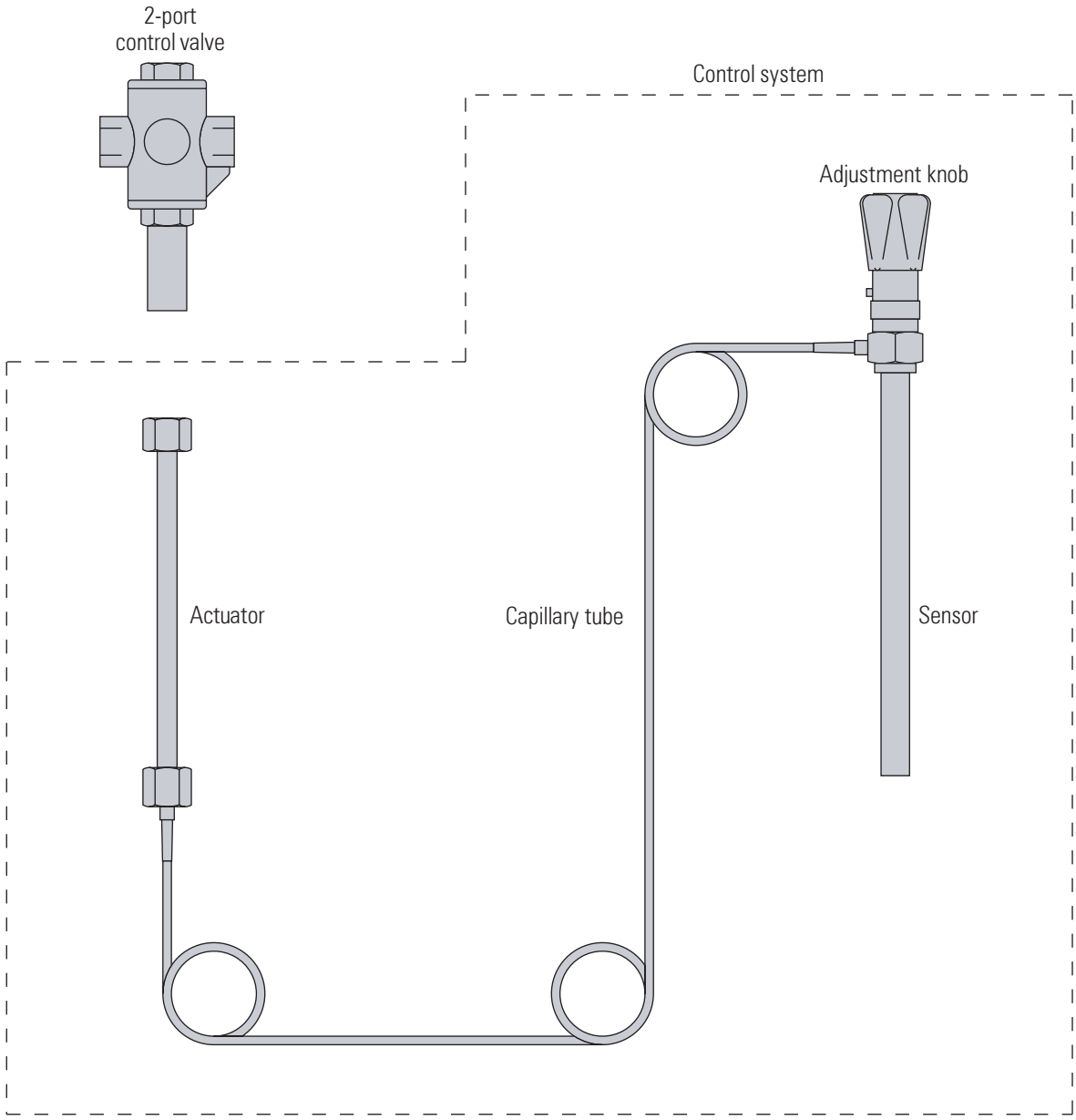


Fig. 7.1.1 Components of a typical self-acting temperature control system

The self-acting principle

If a temperature sensitive fluid is heated, it will expand. If it is cooled, it will contract. In the case of a self-acting temperature control, the temperature sensitive fluid fill in the sensor and capillary will expand with a rise in temperature (see Figure 7.1.2).

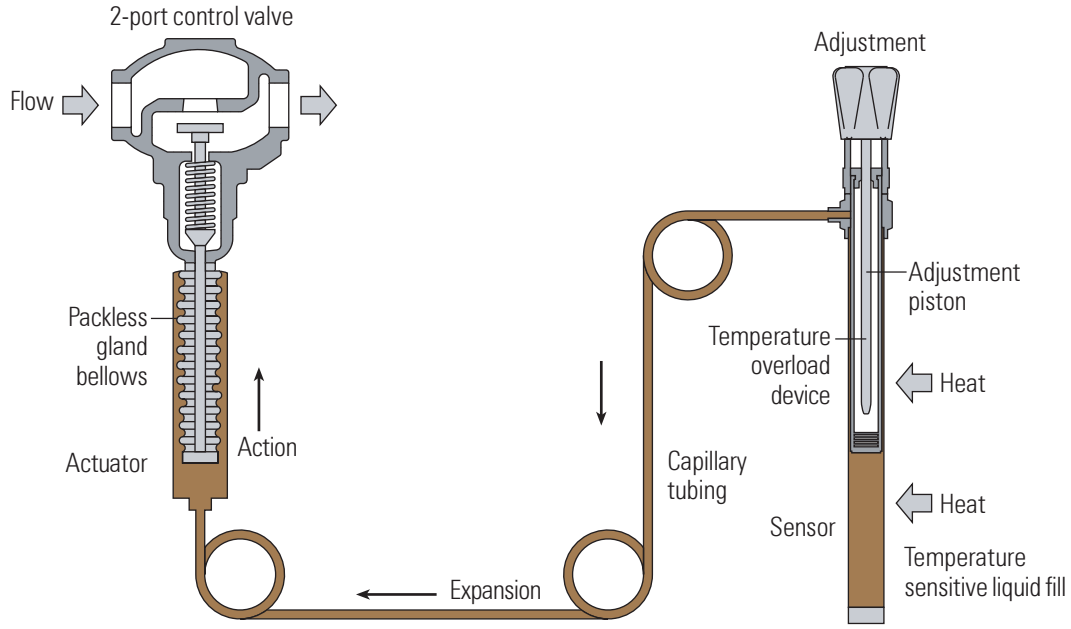


Fig. 7.1.2 Schematic drawing showing the expansive action of the liquid fill when heat is applied to the sensor

The force created by this expansion (or contraction in the case of less heat being applied to the sensor) is transferred via the capillary to the actuator, thereby opening or closing the control valve, and in turn controlling the flow of fluid through the control valve. The hydraulic fluid remains as a liquid.

There is a linear relationship between the temperature change at the sensor and the amount of movement at the actuator. Thus, the same amount of movement can be obtained for each equal unit rise or fall in temperature. This means that a self-acting temperature control system gives 'proportional control'.

To lower the set temperature

The adjustment knob is turned clockwise to insert the piston further into the sensor. This effectively reduces the amount of space for the liquid fill, which means that the valve is closed at a lower temperature. The set temperature will therefore be lower. On control systems with dial-type adjustments, the same effect will be achieved (typically) by using a screwdriver to turn the adjustment screw clockwise.

To raise the set temperature

The adjustment knob is turned anticlockwise to decrease the length of the piston inserted in the sensor. This increases the amount of space for the liquid fill, which means that a higher temperature will be needed to cause the fill to expand sufficiently to close the control valve. The set temperature will therefore be higher.

Again, typically for a dial-type adjustment, a screwdriver is used to turn the adjustment screw anticlockwise.

Protection against high temperatures

In the event of a temperature overrun above the set temperature (possible causes of which might be a leaking control valve, incorrect adjustment, or a separate additional heat source); a series of disc springs housed inside the piston will absorb the excess expansion of the fill. This will prevent the control system from rupturing. When the temperature overrun has ceased, the disc springs will return to their original position and the control system will function as normal. Overrun is typically 30°C to 50°C above the set temperature, according to the control type.

Vapour tension systems

A vapour tension control system has a sensing system filled with a mixture of liquid and vapour. An increase in the sensor temperature boils off a greater proportion of the vapour from the liquid held within it, increasing the vapour pressure in the sensor and capillary system. This increase in pressure is transmitted through the capillary to a bellows or diaphragm assembly at the opposite end (see Figure 7.1.3).

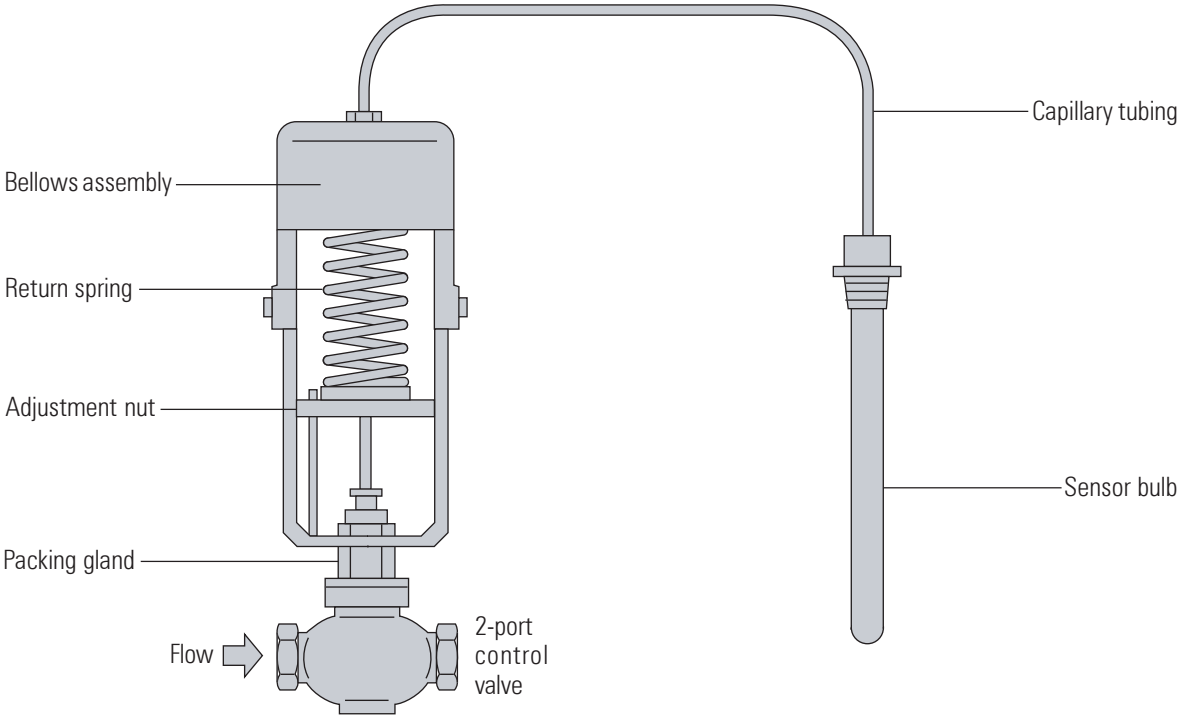


Fig. 7.1.3 Diagram showing a typical vapour tension temperature control system

A vapour tension system follows a unique pressure/temperature saturation curve for the fluid contained by the system. All fluids have a relationship between pressure and their boiling temperature. The result can be plotted by a saturation curve. The saturation curve for water can be seen in Figure 7.1.4.

Figure 7.1.4 illustrates how a 5°C temperature change at 150°C will cause a 0.65 bar change in system pressure. At the bottom of the scale, a 5°C temperature change only results in a 0.18 bar change in system pressure. Thus for the same temperature change, the valve will move a greater amount at the top end of the temperature range than at the bottom end.

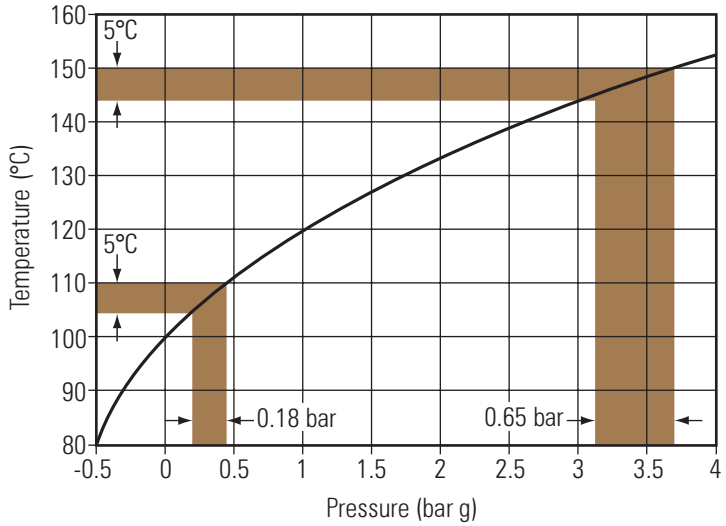


Fig. 7.1.4 Vapour pressure curve for water

Therefore to move a valve from fully open to fully closed requires a greater temperature change at the bottom end of the range than at the top. Manufacturers of these types of vapour tension control systems often suggest that the control be used only at the top end of its range, but this means that to cover a reasonable temperature span, different fills are used (including water, methyl alcohol and benzene).

Alternatively, a liquid filled system will give a true linear relationship between temperature change and valve movement, largely due to liquid being incompressible. The set temperature can be calibrated in degrees and not simply by a series of numbers. There is no confusion over adjusting the set temperature; which reduces commissioning time. Also, adjustment, which is carried out by altering the amount of space available for the liquid fill, can be carried out anywhere between the control valve and the sensor. This is not so with vapour tension systems, which can usually only be adjusted at the control valve.

- Vapour tension control valves sometimes leak through the stem. To avoid the extra cost of having a second bellows sealing mechanism, most manufacturers of vapour tension controls use a mechanical seal on the valve stem. These tend to be either too loose, causing leaks; or too tight, causing too much spindle friction and the valve to stick.
- In liquid systems, because the valve movement is truly proportional to temperature change and the valve seal is frictionless, the temperature control has a very high rangeability and can control at very light loads.

Liquid self-acting temperature control valves

The valves for use with self-acting temperature control systems can be divided into three groups:

- Normally open two-port valves.
- Normally closed two-port valves.
- Three-port mixing or diverting valves.

Normally open two-port control valves

These valves are for heating applications, which is the most common type of application. They are held in the open position by a spring. Once the system is in operation, any increase in temperature, detected by the sensor, will cause the fill to expand and begin to close the valve, restricting the flow of the heating medium.

Normally closed two-port control valves

These valves are for cooling applications. They are held in the closed position by a spring. When the system is in operation, any increase in temperature will cause the fill to expand and begin to open the valve, allowing the cooling medium to flow.

Force required to close a self-acting control valve

The required closing force on the valve plug is the product of the valve orifice area and differential pressure as shown in Equation 7.1.1. **Note** that for two-port steam valves, differential pressure should be taken as the upstream absolute steam pressure; whereas for two-port water valves it will be the maximum pump gauge pressure minus the pressure loss along the pipe between the pump and the valve inlet.

$$\text{Force on valve stem (newton)} = \frac{\pi d^2}{40} \times \Delta P \qquad \textbf{Equation 7.1.1}$$

Where:
d = Diameter of valve orifice (mm)
ΔP = Differential pressure (bar)

Example 7.1.1

Calculate the force required to shut the valve if a steam valve orifice is 20 mm diameter and the steam pressure is 9 bar g. (The maximum differential pressure is 9 + 1 = 10 bar absolute).

$$\text{Force on valve stem} = \frac{\pi\, d^2}{40} \times \Delta P$$

$$\text{Force on valve stem} = \frac{\pi\, (20)^2}{40} \times 10$$

$$\textbf{Force on valve stem} = \textbf{314 N}$$

This means that the actuator must provide at least 314 newton to close the control valve against the upstream steam pressure of 9 bar g.

It can be seen from Example 7.1.1 that the force required to shut the valve increases with the square of the diameter. There is a limited amount of force available from the actuator, which is why the maximum pressure against which a valve is able to shut decreases with an increase in valve size.

This would effectively limit self-acting temperature controls to low pressures in sizes over DN25, if it were not for a balancing facility. Balancing can be achieved by means of a bellows or a double seat arrangement.

Bellows balanced valves

In a bellows balanced valve, a balancing bellows with the same effective area as the seat orifice is used to counteract the forces acting on the valve plug. A small hole down the centre of the valve stem forms a balance tube, allowing pressure from upstream of the valve plug to be fed to the bellows housing (see Figure 7.1.5). Similarly, the forces on the valve plug pressurise the inside of the bellows. The differential pressure across the bellows is therefore the same as the differential pressure across the valve plug, but since the forces act in opposite directions they cancel each other out.

The balancing bellows may typically be manufactured from either:

- Phosphor bronze.
- Stainless steel, which permits higher pressures and temperatures.

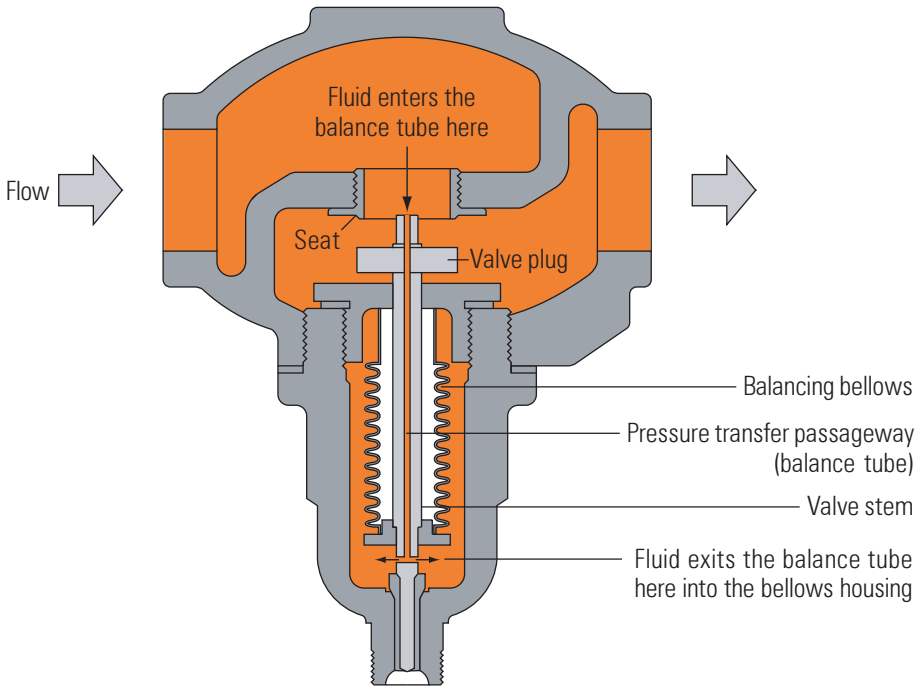


Fig. 7.1.5 Two-port, normally open, bellows balanced valve

Double-seated control valves

Double-seated control valves are useful when high capacity flow is required and tight shut-off is not needed. They can close against higher differential pressures than single seated valves of the same size. This is because the control valve comprises two valve plugs on a common spindle with two corresponding seats, as shown in Figure 7.1.6. The forces acting on the two valve plugs are almost balanced. Although the differential pressure is trying to keep one plug off its seat, it is pushing the other plug onto its seat.

However, the tolerances necessary to manufacture the component parts of the control valve make it difficult to achieve a tight shut-off. This is not helped by the lower valve plug and seat being smaller than its upper counterpart, which enables removal of the whole assembly for servicing.

Also, although the body and the valve shuttle are the same material, small variations in the chemistry of the individual parts can result in subtle variations in the coefficients of expansion, which adversely affects shut-off. A double-seated control valve should not be used as a safety device with a high limit safeguard.

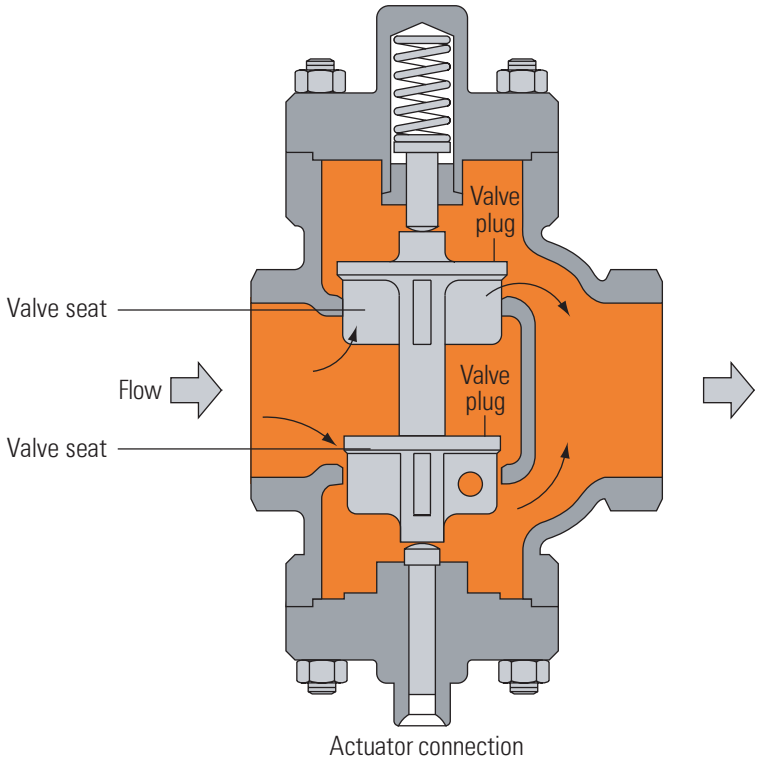


Fig. 7.1.6 Schematic of a double seated (normally closed) self-acting control valve

Control valves with internal fixed bleed holes

A normally closed valve will usually require a fixed bleed (Figure 7.1.7) to allow a small amount of flow through the control valve when it is fully shut. Normally closed self-acting control valves are sometimes referred to as being reverse acting (RA).

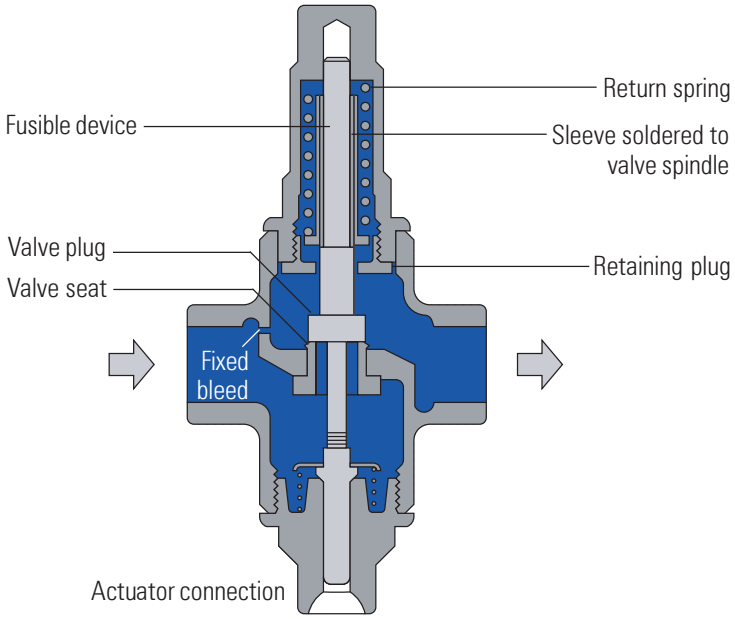


Fig. 7.1.7 Normally closed control valve with fixed bleed

A typical application for this type of valve is to control the flow of cooling water (coolant) for an industrial engine such as an air compressor (Figure 7.1.8). The control valve, controlling the flow of coolant through the engine, is upstream of the engine and the temperature sensor registers its temperature as it leaves the engine.

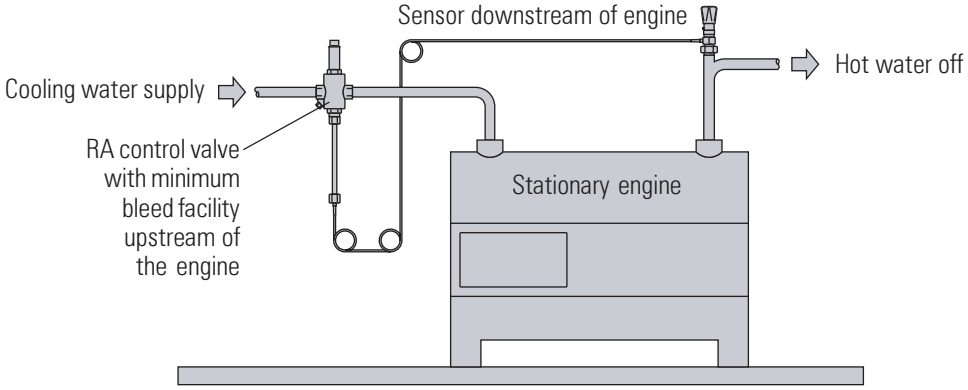


Fig. 7.1.8 Engine or compressor cooling system

If the coolant leaving the engine is hotter than the set point, the control valve opens to allow more coolant through the valve. However, once the water leaving the engine reaches the required set temperature the valve will shut again. Without a bleedhole, the coolant would no longer flow and would continue to pick up heat from the engine. Without the downstream sensor detecting any temperature rise, the engine is likely to overheat.

If the control valve has a fixed diameter bleed hole, enough cooling water can flow through the valve to allow the downstream sensor to register a representative temperature when the valve is shut. This feature is essential when the sensor is remote from the application heat source.

A normally closed valve might also have an optional fusible device (see Figure 7.1.7). The device melts in the event of excess heat, removing the spring tension on the valve plug and opening the valve to allow the cooling water to enter the system. It is usual with this kind of safety device, that once the fusible device has melted, it cannot be repaired and must be replaced.

Three-port control valves

Most of the control valves used with self-acting control systems are two-port. However, Figure 7.1.9 illustrates a self-acting piston type three-port control valve. The advantage of this type of valve design allows the same valve to be used for either mixing or diverting water applications; this is not normally the case with valves requiring electric or pneumatic actuators.

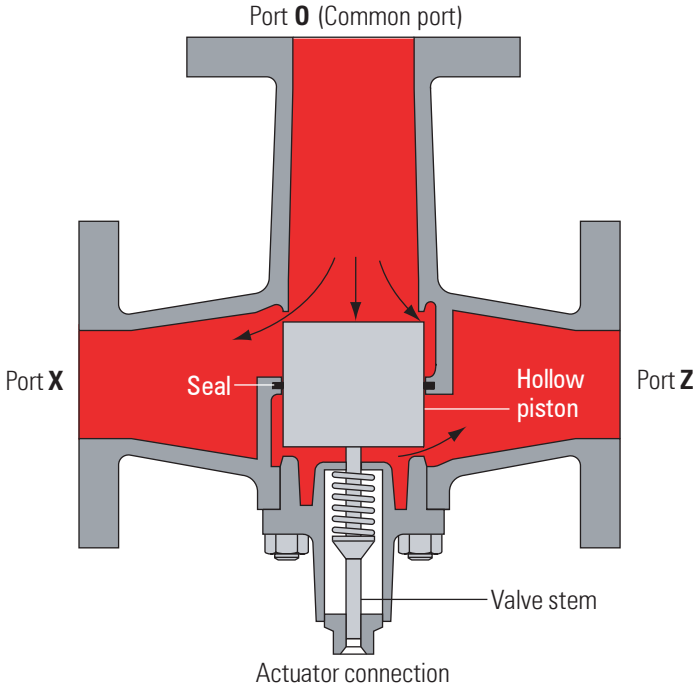


Fig. 7.1.9 Three-port control valve

The most common applications are for water heating, but three-port control valves may also be used on cooling applications such as air chillers, and on pumped circuits in heating, ventilating and air conditioning applications.

When a three-port control valve is used as a mixing valve (see Figure 7.1.10), the constant volume port 'O' is used as the common outlet.

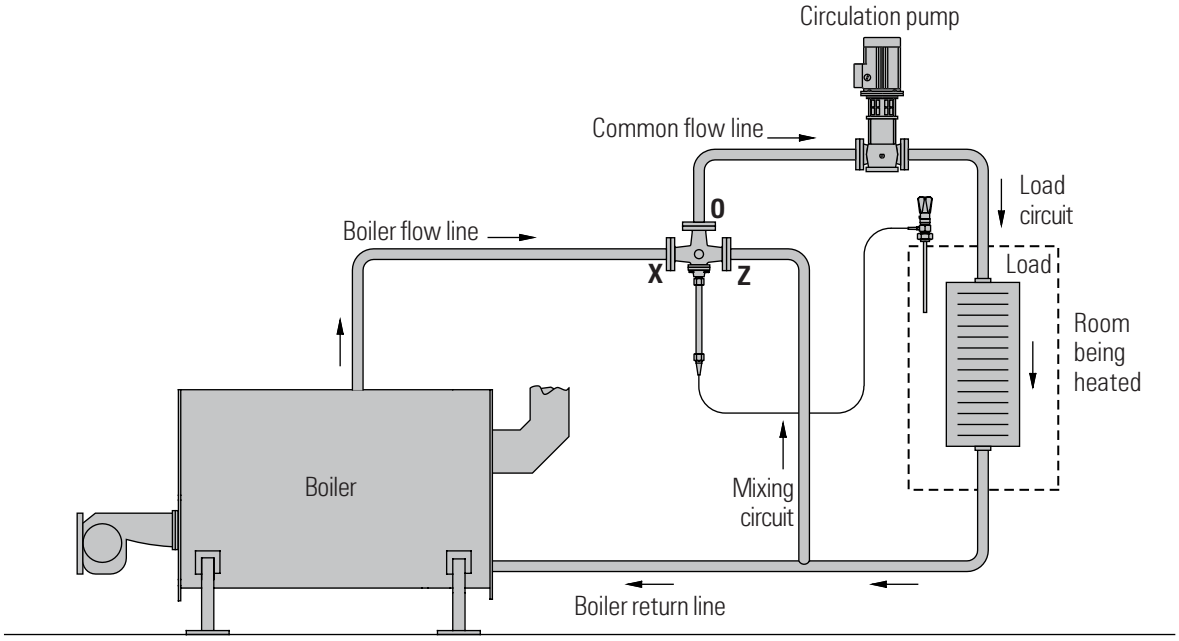


Fig. 7.1.10 Typical three-port control valve used in a mixing application

When a three-port control valve is used as a diverting valve (see Figure 7.1.11), the constant volume port is used as the common inlet

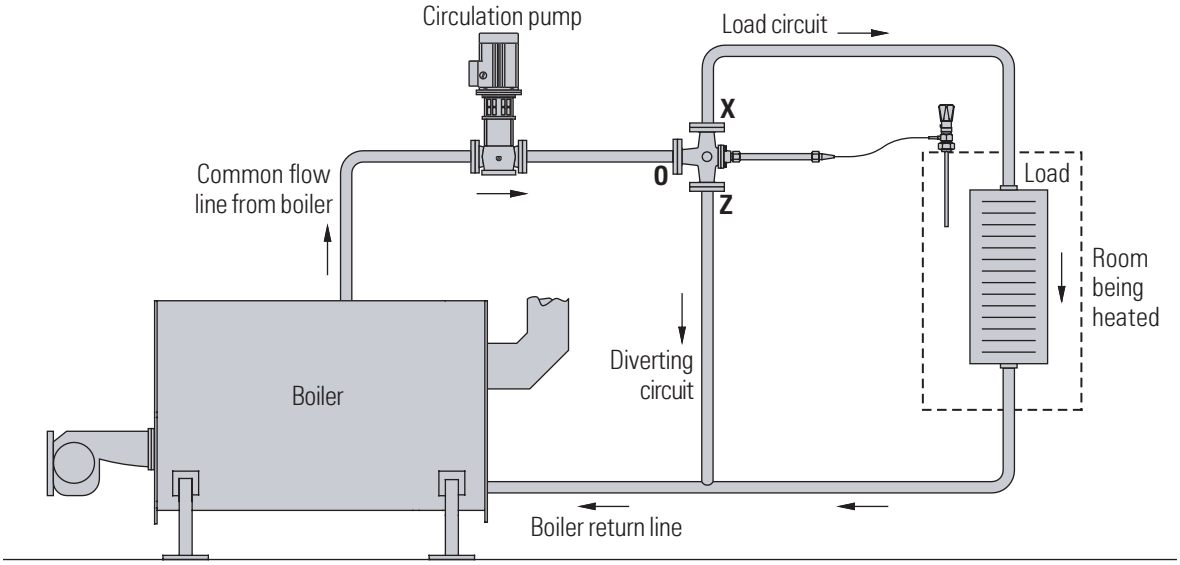


Fig. 7.1.11 Typical three-port control valve used in a diverting application

Self-contained three port control valves

Another type of three-port self-acting control valve contains an integral temperature sensing device and thus requires no external temperature controller to operate.

It can be used to protect Low Temperature Hot Water (LTHW) boilers from fire tube corrosion during start-up sequences when the temperature of the secondary return water is low (see Figure 7.1.12). At start-up, the valve allows cold secondary water to bypass the external system and flow through the boiler circuit. This allows water in the boiler to heat up quickly, minimising the condensation of water vapour in the flue gases. As the boiler water heats up, it is slowly blended with water from the main system, thus maintaining protection while the complete system is brought slowly up to temperature.

This type of control valve may also be used on cooling systems such as those found on air compressors (Figure 7.1.13).

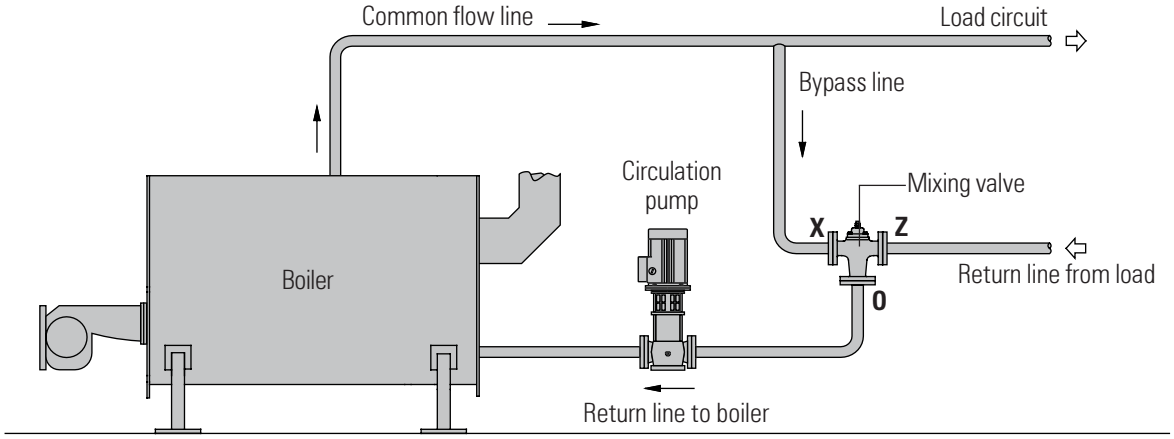


Fig. 7.1.12 Self contained three-port control valve reducing fire tube corrosion

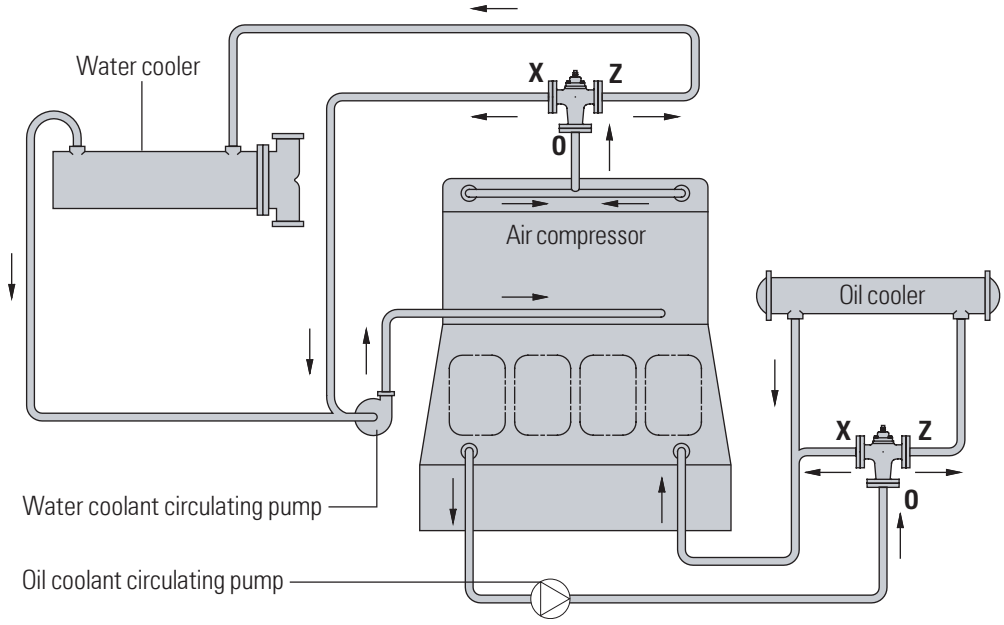


Fig. 7.1.13 Self-contained three-port valves used to control water and oil cooling systems on an air compressor

Questions

1. Name the components of a self-acting temperature control system.

a | Control valve and actuator ☐

b | Control valve, actuator and sensor ☐

c | Control valve, actuator, capillary tube and sensor ☐

d | Control valve, actuator and capillary tube ☐

2. What is the purpose of overtemperature protection within the self-acting control system?

a | To protect the valve from high temperature steam ☐

b | To protect the liquid fill in the capillary from boiling ☐

c | To protect the control system from irreversible damage ☐

d | To protect the application from overtemperature ☐

3. If the liquid expands with temperature, how can cooling control be achieved?

a | By fitting two control valves in parallel fashion ☐

b | It cannot because expanding liquid can only shut a control valve ☐

c | By using a bellows balanced control valve ☐

d | By using a normally closed control valve that opens with rising temperature ☐

4. Why do larger control valves tend only to close against lower pressures?

a | The control valve orifice is larger and needs a higher force to close ☐

b | The PN rating of larger control valves is less than smaller control valves ☐

c | The actuators are not designed to operate with high pressures ☐

d | The higher forces involved can rupture the capillary tubing ☐

5. Name two solutions which allow larger control valves to operate at high pressures.

a | Large actuators and large sensors ☐

b | Bellows balanced control valves or double-seated control valves ☐

c | It is not possible to allow larger control valves to operate at higher pressures ☐

d | Larger springs or a higher density capillary fluid ☐

6. Why are three-port self-acting control valves used?

a | To mix or divert liquids especially water ☐

b | To dump steam to waste under fault conditions ☐

c | Where cooling applications are required ☐

d | When large valves are required to meet large capacities ☐

Answers

1: c, 2: c, 3: d, 4: a, 5: b, 6: a

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Module 7.2

Typical Self-acting Temperature Control Valves and Systems

Typical Self-acting Temperature Control Valves and Systems

Typical self-acting temperature control systems

The required temperature for the system in Figure 7.2.1 is adjusted at the sensor. It is the most common type of self-acting temperature control configuration, and most other self-acting control designs are derived from it.

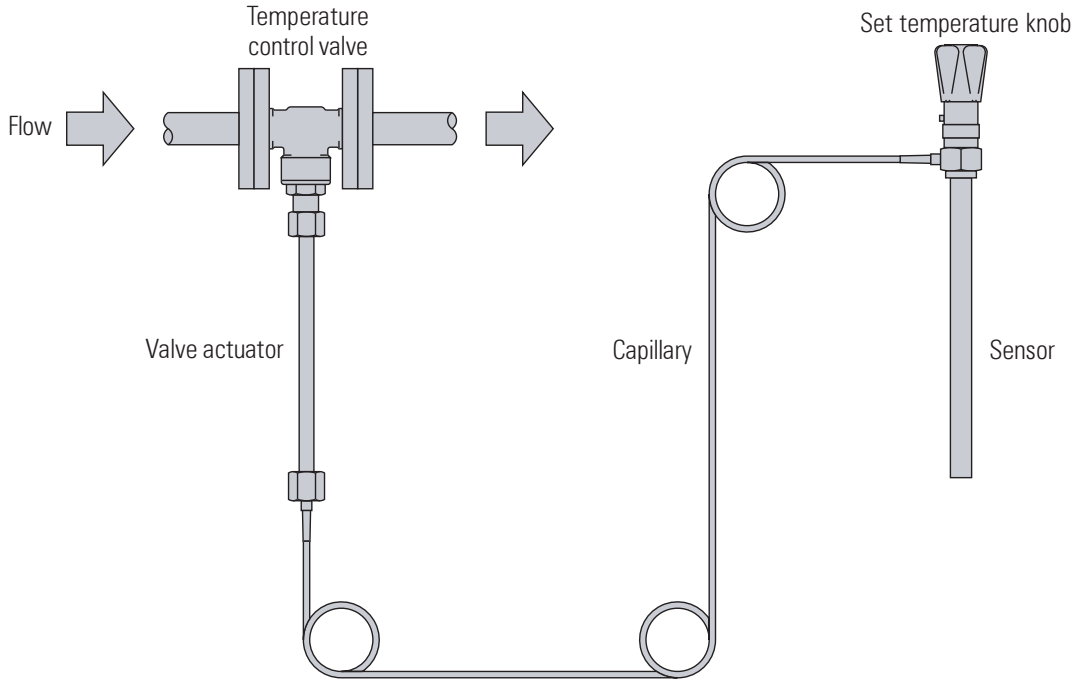


Fig. 7.2.1 Adjustment at sensor

Figure 7.2.2 illustrates a design which is adjusted at the actuator end of the system. It is worth noting that this system is limited to 1" (DN25) temperature control valves. This configuration is useful where the control valve position is more accessible than the sensor position.

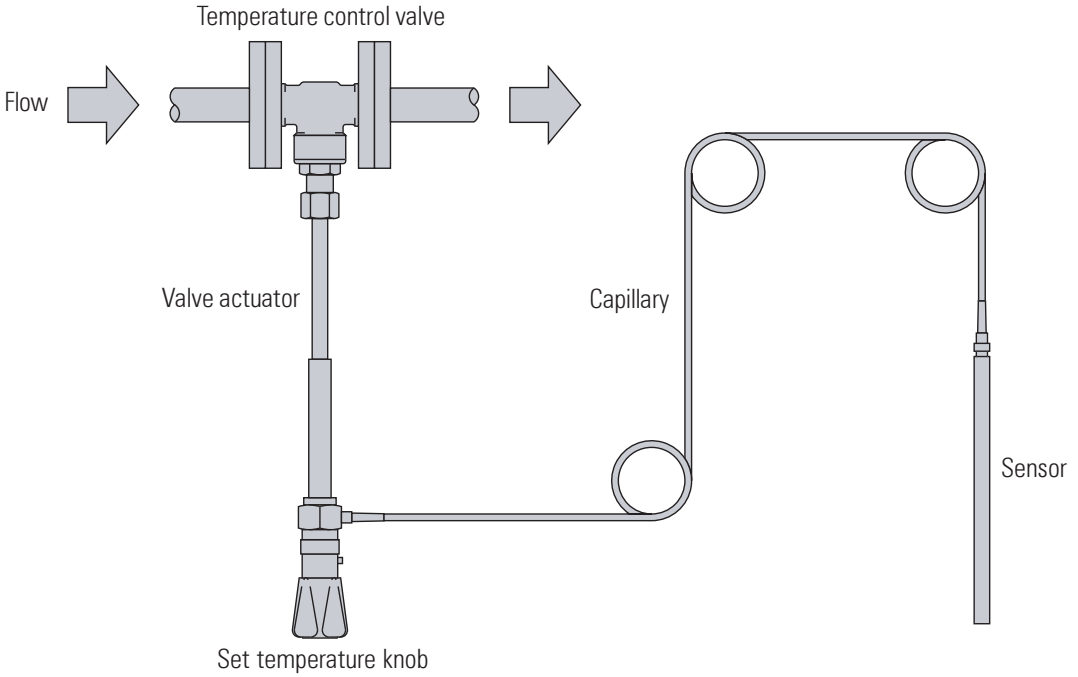


Fig. 7.2.2 Adjustment at actuator

Figure 7.2.3 depicts a third configuration which is similar to the one in Figure 7.2.1 but where the adjustment is located between the sensor and the temperature control valve actuation. This type of system is referred to as remote adjustment, and is helpful when either the control valve or the sensor, or both, are likely to be inaccessible once the control valve has been installed.

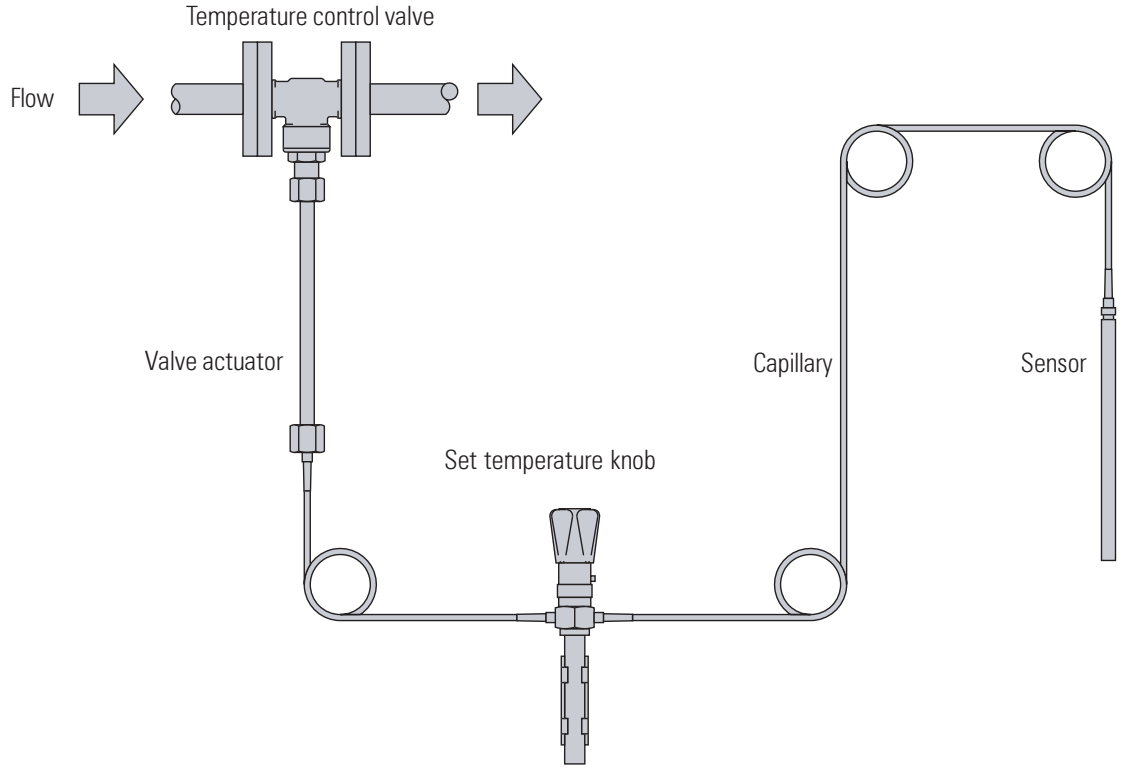


Fig. 7.2.3 Remote adjustment

Capillaries

It should be noted that capillaries of 10 metres or more in length may slightly affect the accuracy of the control. This is because a larger amount of capillary fluid is subjected to ambient temperature. When the ambient temperature changes a lot, it can affect the temperature setting. If long lengths of capillary are run outside, it is recommended they are lagged to minimise this effect.

Pockets

Pockets (sometimes called thermowells) can be fitted into pipework or vessels. These enable the sensor to be removed easily from the controlled medium without the need to drain the system. Pockets will tend to slow the response of the system and, where the heat load can change quickly, should be filled with an appropriate conducting medium to increase the heat transfer to the sensor.

Pockets fitted to systems which have relatively steady or slow changing load conditions do not usually need a conducting medium. Pockets are available in mild steel, copper, brass or stainless steel. Long pockets of up to 1 metre in length are available for special applications and in glass for corrosive applications. However, these longer pockets are only suitable for use where the adjustment head is not fitted at the sensor end.

Enhancements for self-acting temperature control systems

Overheat protection by a high limit cut-out device

A separate overheat protection system, as shown in Figure 7.2.4, is available to comply with local health and safety regulations or to prevent product spoilage. The purpose of the high limit cut-out device is to shut off the flow of the heating medium in the pipe, thereby preventing overheating of the process. It was originally developed to prevent overheating in domestic hot water services (DHWS) which supply general purpose hot water users, such as hospitals, prisons and schools. However, it is also used for industrial process applications.

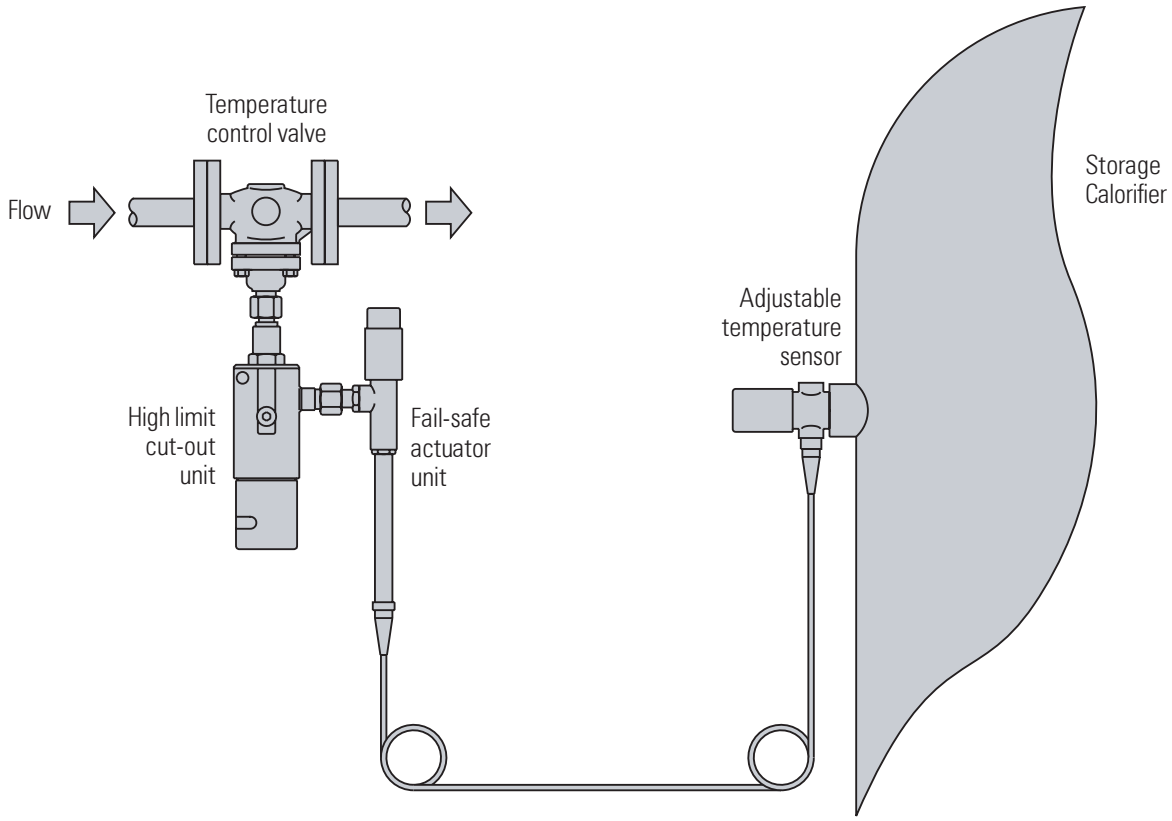


Fig. 7.2.4 High limit cut-out unit with fail-safe control system

The system is driven by a self-acting control system, which releases a compressed spring in the high limit cut-out unit and snaps the isolating valve shut if the pre-set high limit temperature is exceeded.

The fail-safe actuator unit does not drive the control valve directly, but a shuttle mechanism in the high limit cut-out unit instead. When the temperature is below the set point, the mechanism lies dormant. A certain amount of shuttle travel is allowed for in either direction, to avoid spurious activation of the system.

However, when the system temperature rises above the adjustable high limit temperature, the actuator drives the shuttle, displacing the trigger, which then releases the spring in the high limit cut-out unit. This causes the control valve to snap shut. Once the fault has been rectified, and after the system has cooled below the set temperature, the high limit cut-out can be manually reset, using a small lever. The system can also be connected to an alarm system via an optional microswitch.

The high limit system also has a fail-safe facility. If the capillary is damaged and loses fluid, a spring beyond the shuttle is released, pushing it the other way. This will also activate the cut-out and shut the control valve.

The trigger temperature can be adjusted between 0°C and 100°C.

The fail-safe actuator unit shown in Figure 7.2.5 is only suitable for use with a high limit cut-out unit. The systems shown in Figures 7.2.1, 7.2.2 and 7.2.3 can also be used with the cut-out unit but they will not fail-safe. Figure 7.2.5 shows the high limit cut-out unit attached to a separate valve to the temperature control valve. This is preferable because the high limit valve remains fully open during normal operation and is less likely to harbour dirt under the valve seat. The high limit valve should be line size to reduce pressure drop in normal use, and should be fitted upstream of the self-acting (or other) control valve and as close to it as possible.

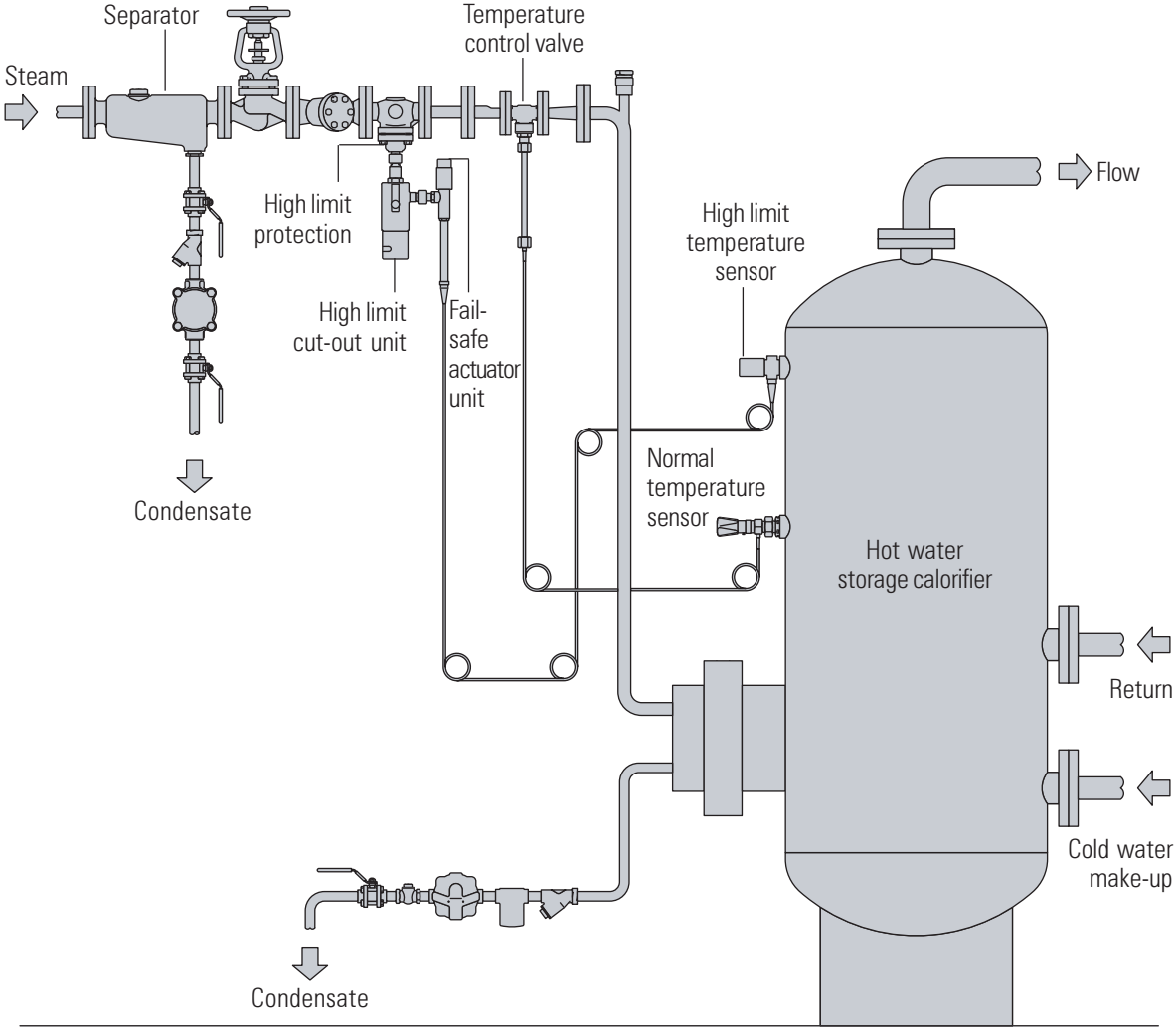


Fig. 7.2.5 Typical arrangement showing a high limit cut-out on DHWS heat exchanger

For heating applications, the high limit valve must be fitted in series with the temperature control valve, as shown in Figure 7.2.5. However, in cooling applications, the temperature control valve and high limit valve will both be of the normally-open type and must be fitted in parallel with each other, not in series.

The following valves can be used with the high limit system:

- Two-port valves, normally open for heating systems.
- Two-port valves, normally closed for cooling systems.
- Three-port valves.

Valves having a ball shaped plug cannot be used with the cut-out unit. This is because the closing operation could drive the ball into the seat and damage the valve.

Also, a double seated valve should not be used with this system because it does not have tight shut-off.

Typical self-acting 2-port temperature control valves

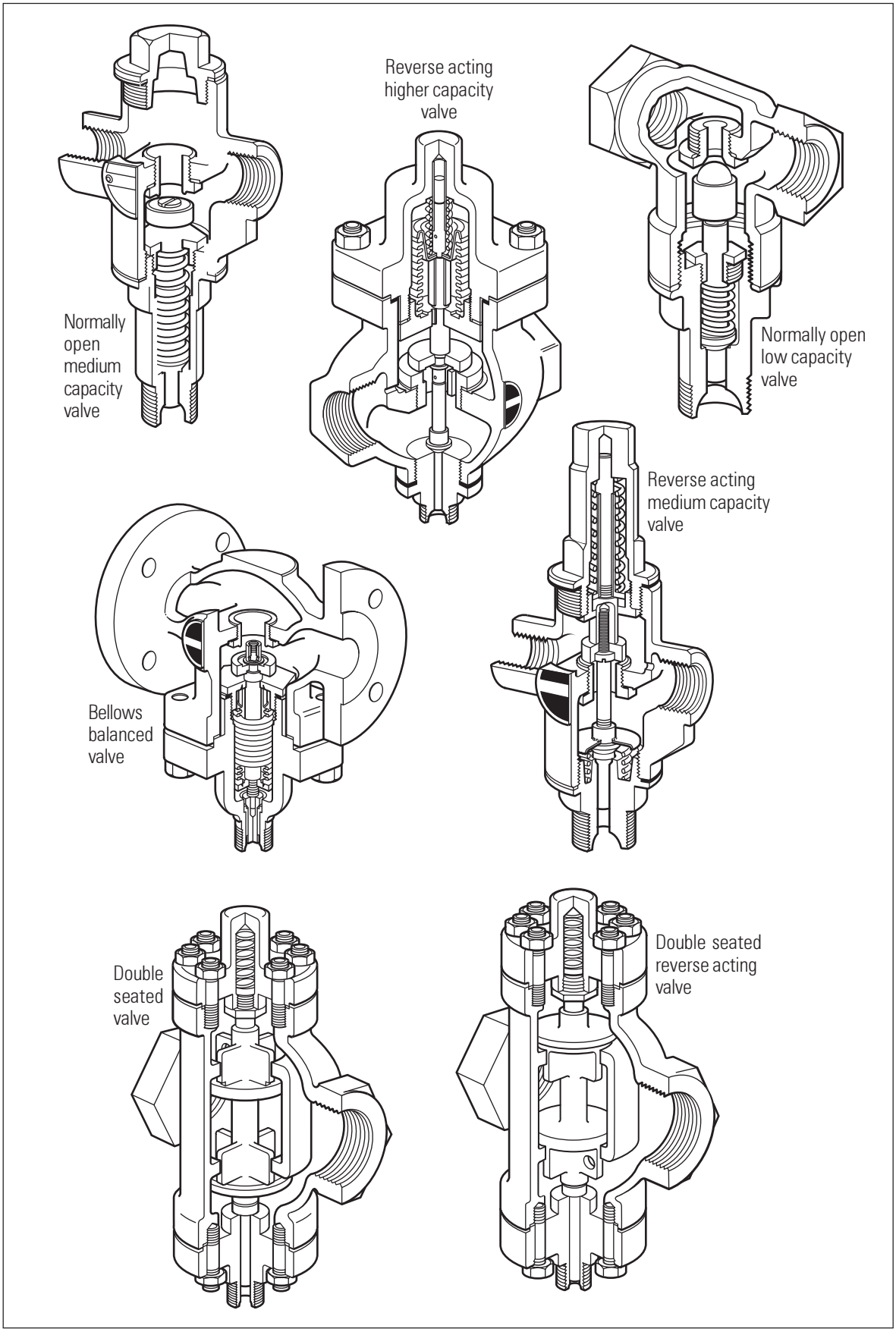


Fig. 7.2.6 Typical self-acting 2-port temperature control valves

Self-acting temperature control ancillaries

Twin sensor adaptor

A twin sensor adaptor, Figure 7.2.7, allows one valve to be operated by a control system with the option of having a manual isolation facility.

The adaptor can be used with both 2-port and 3-port control valves. The advantage offered by the adaptor is that the cost of a separate valve is saved. However, it is not recommended that temperature control and safeguard high limit protection be provided with a common valve, as there is no protection against failure of the valve itself.

Manual actuator

A manual adaptor as shown in Figure 7.2.8, is designed to be used with 2-port and 3-port control valves. It can also be used in conjunction with a twin sensor adaptor and a self-acting temperature control system, allowing manual shutdown without interfering with the control settings, as shown in Figure 7.2.7

Spacer

A spacer (Figure 7.2.9) enables the system to operate at higher temperatures. Each control valve and temperature control system has its own limiting conditions. A spacer, when fitted between the control system and any 2-port or 3-port control valve (except DN80 and DN100 3-port valves), enables the system to operate at a maximum of 350°C, providing that the control valve itself is able to tolerate such high temperatures.

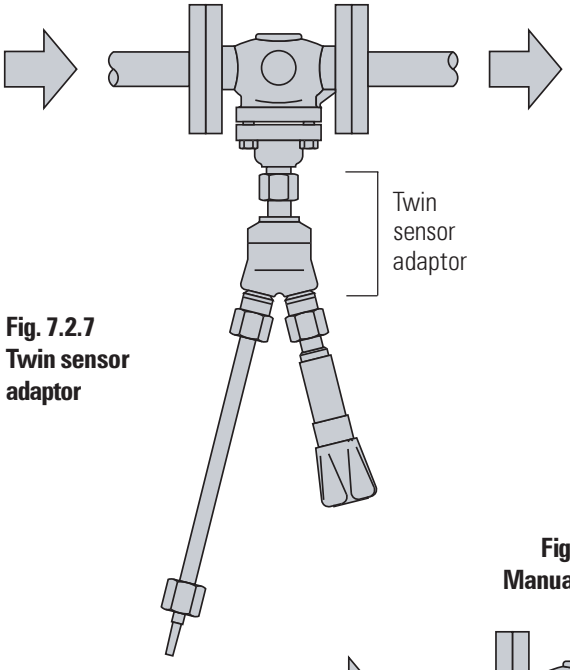


Fig. 7.2.7
Twin sensor adaptor

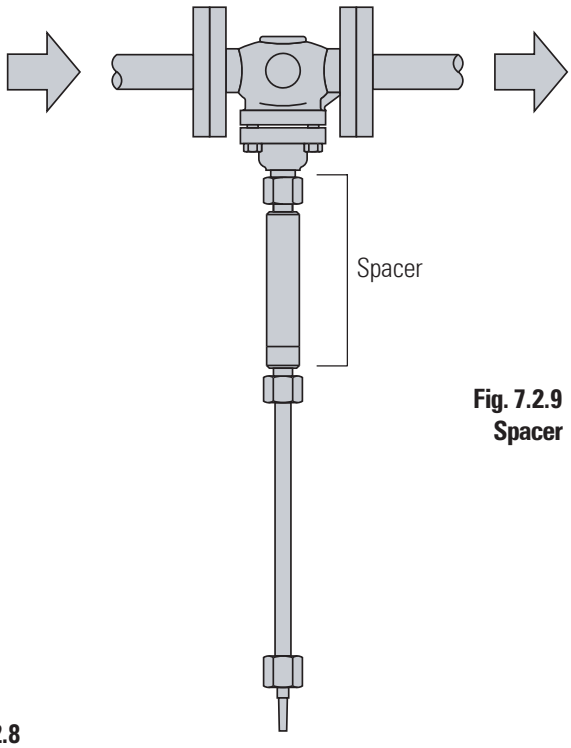
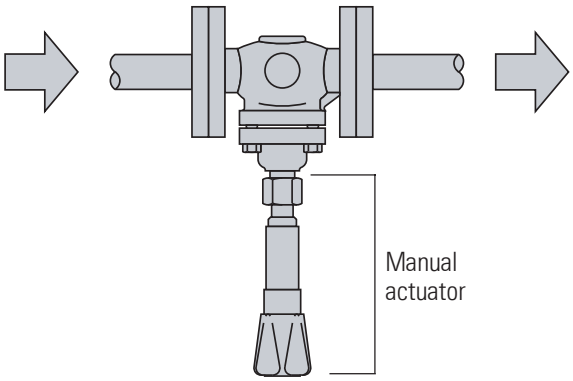


Fig. 7.2.9
Spacer

Fig. 7.2.8
Manual actuator



Typical environments and applications

Environments suitable for self-acting temperature controls:

- Any environment where the sophistication of electrical and pneumatic controls is not required. Especially suited to dirty and hazardous areas.
- Areas remote from any power source.
- For the accurate control of storage or constant load applications, or for variable load applications where high accuracy is not required.

Industries using self-acting temperature controls:

Foods

- Milling, heater battery temperature control (non-hazardous).
- Abattoirs - washing down etc.
- Manufacture of oils and fats - storage tank heating.

Industrial

- Metal plating - tank heating.
- Tank farms - heating.
- Refineries.
- Industrial washing.
- Steam and condensate systems.
- Laundries.

Heating, ventilation and air conditioning (HVAC)

- Domestic hot water and heating services in nursing homes, hospitals, leisure centres and schools, prisons and in horticulture for frost protection.

The most commonly encountered applications for self-acting temperature controls:

Boiler houses

- Boiler feedwater conditioning or direct steam injection heating to boiler feedtank.
- Stand-by generator cooling systems.

Non-storage calorifiers

- 2-port temperature control and overheat protection, (steam or water).
- 3-port temperature control and overheat protection (water only).
- 2-port time / temperature control (steam only).

Storage calorifiers

- 2-port temperature or time/temperature control and overheat protection (steam or water).
- 3-port control and overheat protection (water only).

Injection (or bleed-in) systems

- 2-port or 3-port injection system.

Heating systems

- Basic mixing valve and compensating control.
- Zoned compensating controls.
- Basic compensator plus internal zone controls.
- Control of overhead radiant strip or radiant panels.

Warm air systems

- Heater battery control via room sensor, air-off sensor or return air sensor.
- Compensating control on air-input unit.
- Low limit and high limit control.
- Frost protection to a heater battery.

Fuel oil control

- Bulk tank heating coil control.
- Control of line heaters.
- Control of steam tracer lines.

Process control

- Acid pickling tank.
- Plating vat.
- Process liquor boiling tank.
- Brewing plant detergent tank.
- Drying equipment, for example, laundry cabinet or wool hank dryer, chemical plant drying stove for powder and cake, tannery plant drying oven.
- Continuous or batch process reaction pan.
- Food industry jacketed pan.

Cooling applications

- Diesel engine cooling.
- Rotary vane compressor oil cooler control.
- Hydraulic and lubricating oil coolers.
- Cooling control on cold water to single-stage compressor.
- Closed circuit compressor cooling control.
- Air aftercooler control.
- Air cooler battery control.
- Jacketed vessel water cooling control.
- Degreaser cooling water control.

Special applications

- Control for reducing fireside corrosion and thermal stress in LTHW boilers.
- Hot water cylinder control.
- Temperature limiting.

Applications for the high limit safeguard system

- Preventing temperature overrun on hot water services, or heating calorifiers, in accordance with many Health and Safety Regulations. Good examples include prisons, hospitals and schools. An optional BMS/EMS interface to flag high temperature trip is available.

Questions

1. Where is a self-acting temperature control system adjusted?

a | Locally to the control valve ☐
b | Locally to the sensor ☐
c | Remotely, at a point between the control valve and sensor ☐
d | Any of the above ☐

2. Why are sensor pockets sometimes used?

a | To protect the sensor from overheating ☐
b | To allow the sensor to be removed without draining the system ☐
c | To contain any leakage of liquid fill from the sensor ☐
d | To enable small sensors to fit into large diameter pipes ☐

3. How can fail-safe temperature protection be achieved?

a | By fitting two control valves in series ☐
b | By fitting a proprietary spring-loaded actuator and control valve ☐
c | By setting the control system at a lower temperature ☐
d | By fitting a cooling valve in parallel with the heating valve ☐

4. What does a proprietary fail-safe protection device do?

a | It protects the control valve from high operating temperatures ☐
b | It protects the steam system from overpressure ☐
c | It protects the water system from overtemperature ☐
d | It allows one valve to act as a control and high limit valve ☐

5. For what application is a self-acting temperature control system not suitable?

a | An application with slow changes in heat load ☐
b | An application in a hazardous area ☐
c | An application with fast and frequent changes in heat load ☐
d | A warm air system such as a heater battery control ☐

6. What is the purpose of a twin sensor adaptor?

a | To close the control valve under fault conditions ☐
b | To allow two control valves to be operated by one controller ☐
c | To allow one control valve to be operated by two controllers ☐
d | To allow both heating and cooling with one valve ☐

Answers

1: d, 2: b, 3: b, 4: c, 5: c, 6: c

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Module 7.3

Self-acting Pressure Controls and Applications

Self-acting Pressure Controls and Applications

Why reduce steam pressure?

The main reason for reducing steam pressure is rather fundamental. Every item of steam using equipment has a maximum allowable working pressure (MAWP). If this is lower than the steam supply pressure, a pressure reducing valve must be employed to limit the supply pressure to the MAWP. In the event that the pressure reducing valve should fail, a safety valve must also be incorporated into the system.

This is not, however, the only occasion when a pressure reducing valve can be used to advantage.

Most steam boilers are designed to work at relatively high pressures and should not be run at lower pressures, since wet steam is likely to be produced. For this reason, it is usually more economic in the long term to produce and distribute steam at a higher pressure, and reduce pressure upstream of any items of plant designed to operate at a lower pressure.

This type of arrangement has the added advantage that relatively smaller distribution mains can be used due to the relatively small volume occupied by steam at high pressure.

Since the temperature of saturated steam is closely related to its pressure, control of pressure can be a simple but effective method of providing accurate temperature control. This fact is used to good effect on applications such as sterilisers and contact dryers where the control of surface temperature is difficult to achieve using temperature sensors.

Plant operating at low steam pressure:

- Can tend to reduce the amount of steam produced by the boiler due to the higher enthalpy of evaporation in lower pressure steam.
- Will reduce the loss of flash steam produced from open vents on condensate collecting tanks.

Most pressure reducing valves currently available can be divided into the following two main groups:

- Direct acting valves.
- Pilot-operated valves.

Direct acting valves

Smaller capacity direct acting pressure reducing valves (Figure 7.3.1)

Method of operation

On start-up and with the adjustment spring relaxed, upstream pressure, aided by a return spring, holds the valve head against the seat in the closed position. Rotating the handwheel in a clockwise direction causes a downward movement, which compresses the control spring and extends the bellows to set the downstream pressure.

This downward movement is transmitted via a pushrod, which causes the main valve to open. Steam then passes through the open valve into the downstream pipework and surrounds the bellows.

As downstream pressure increases, it acts through the bellows to counteract the adjustment spring force, and closes the main valve when the set pressure is reached. The valve plug modulates in an attempt to achieve constant pressure.

In order to close the valve, there must be a build-up of pressure around the bellows. This requires an increase in downstream pressure above the set pressure in proportion to the steam flow.

The downstream pressure will increase as the load falls and will be highest when the valve is closed. This change in pressure relative to a change in load means that the downstream pressure will only equal the set pressure at one load. The actual downstream pressure compared to the set point is the proportional offset; it will increase relative to the load, and this is sometimes referred to as 'droop'.

The total pressure available to close the valve consists of the downstream pressure acting on the underside of the bellows plus the inlet pressure acting on the underside of the main valve itself and the small force produced by the return spring. The control spring force must therefore be larger than the reduced pressure and inlet pressure and return spring for the downstream pressure to be set.

Any variation in the inlet pressure will alter the force it produces on the main valve and so affect the downstream pressure.

This type of pressure reducing valve has two main drawbacks in that:

1. It suffers from proportional offset as the steam flow changes

2. It has relatively low capacity.

It is nevertheless perfectly adequate for a substantial range of simple applications where accurate control is not essential and where steam flow is fairly small and reasonably constant.

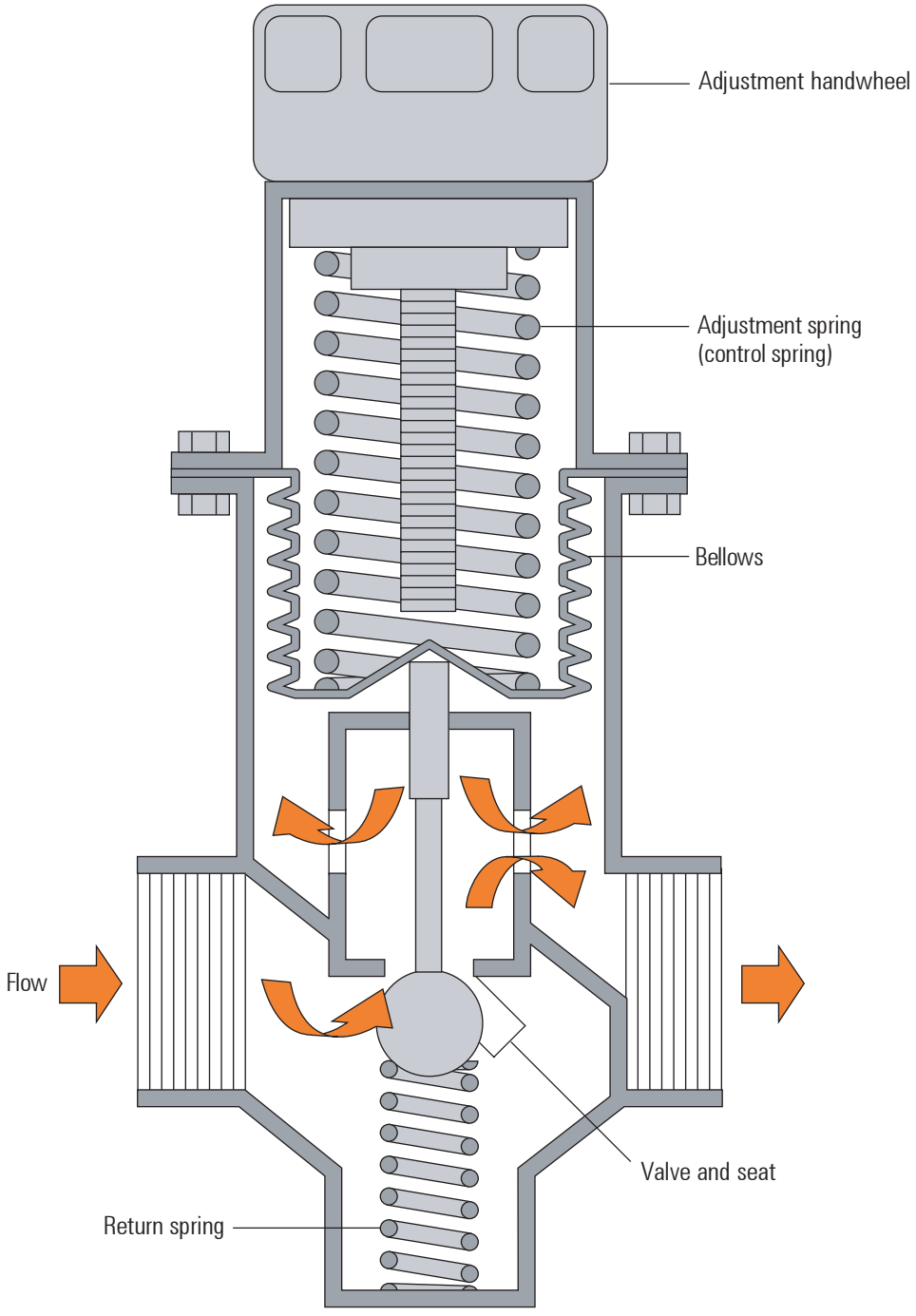


Fig. 7.3.1 Small capacity direct acting pressure reducing valve

Larger capacity direct acting pressure reducing valves (Figure 7.3.2)

Larger capacity direct acting pressure reducing valves are also available for use on larger capacity plant, or on steam distribution mains. They differ slightly to the smaller capacity valves in that the actuator force is provided by pressure acting against a flexible diaphragm inside the actuator rather than a bellows.

As these are not pilot-operated, they will incur a change in downstream pressure as the steam flow changes, and this should be taken into careful consideration when selecting and sizing the valve.

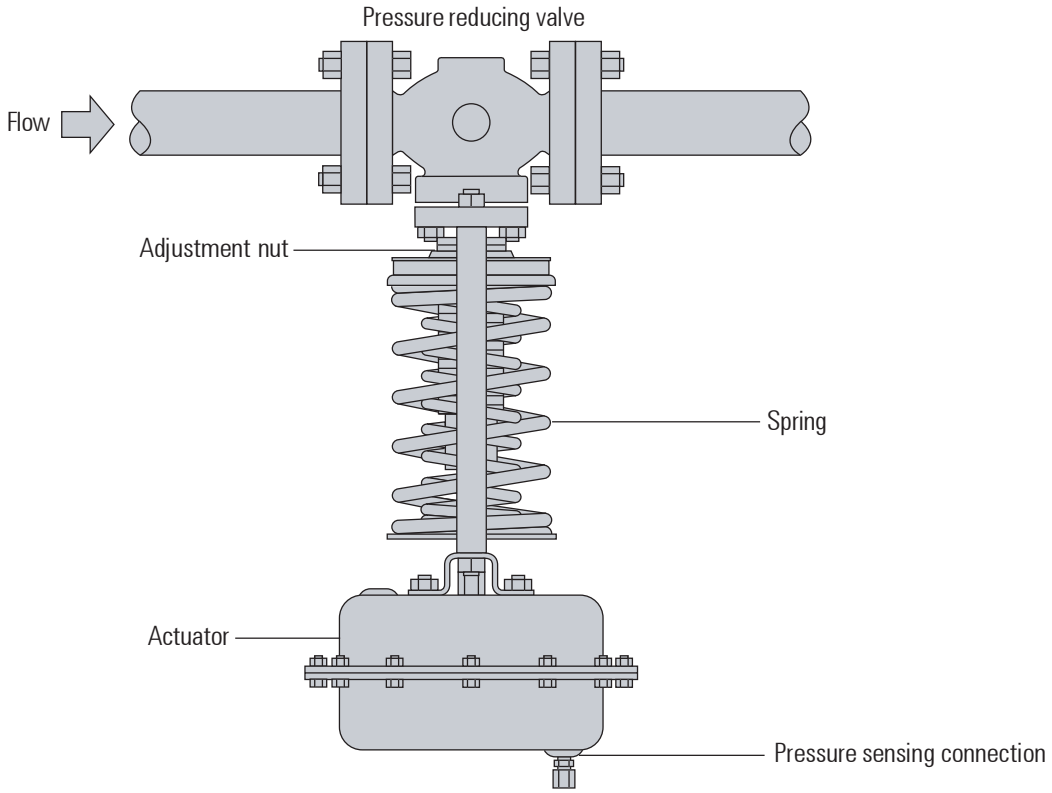


Fig. 7.3.2 Large capacity direct acting pressure reducing valve

This type of valve is installed with the actuator below the pipe when used with steam, and has a water seal pot to stop high steam temperatures from reaching and damaging the actuator's flexible diaphragm, which is commonly made out of neoprene. A typical installation for the reduction of steam mains pressure is shown in Figure 7.3.3.

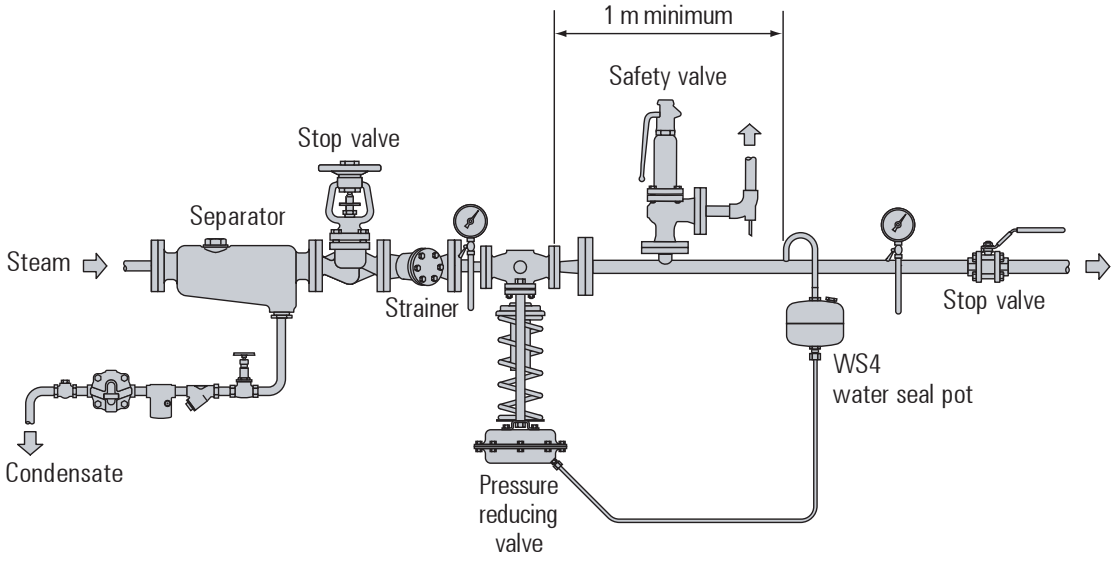


Fig. 7.3.3 Typical steam pressure reducing station for a large capacity direct acting pressure reducing valve

Pilot-operated valves

Where accurate control of pressure or a large flow capacity is required, a pilot-operated pressure reducing valve can be used. Such a valve is shown schematically in Figure 7.3.4. A pilot-operated pressure reducing valve will usually be smaller than a direct acting valve of the same capacity.

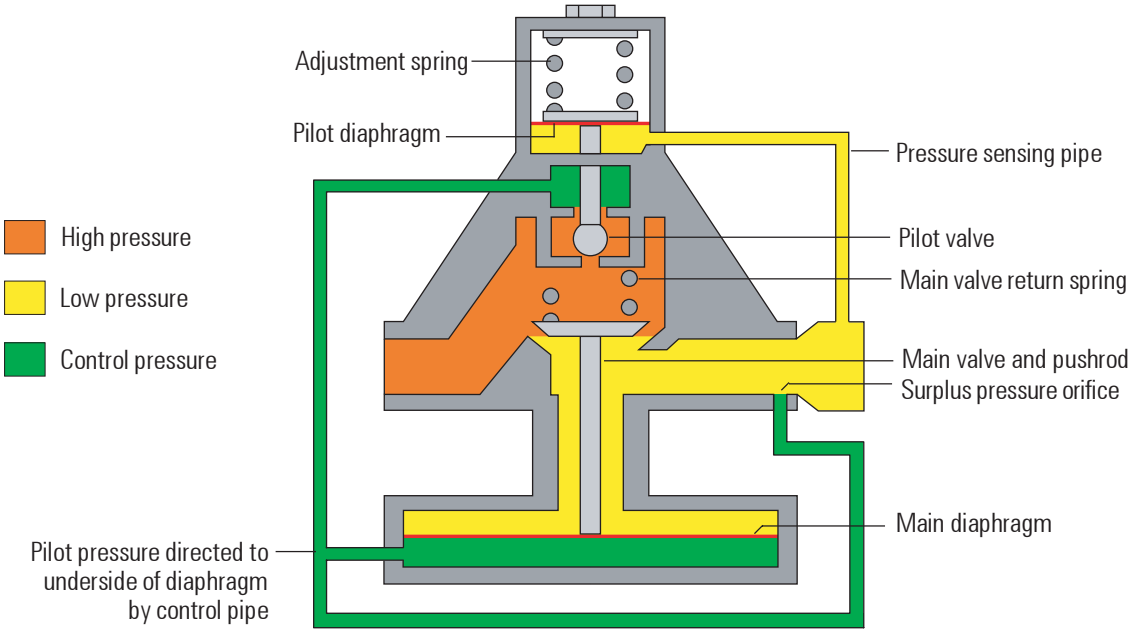


Fig. 7.3.4 Pilot-operated pressure reducing valve

A pilot-operated pressure reducing valve works by balancing the downstream pressure via a pressure sensing pipe against a pressure adjustment control spring. This moves a pilot valve to modulate a control pressure. The control pressure transmitted via the pilot valve is proportional to the pilot valve opening, and is directed, via the control pipe to the underside of the main valve diaphragm. The diaphragm moves the pushrod and the main valve in proportion to the movement of the pilot valve. Although the downstream pressure and pilot valve position are proportional (as in the direct acting valve), the mechanical advantage given by the ratio of the areas of the main diaphragm to the pilot diaphragm offers accuracy with small proportional offset.

Under stable load conditions, the pressure under the pilot diaphragm balances the force set on the adjustment spring. This settles the pilot valve, allowing a constant pressure under the main diaphragm. This ensures that the main valve is also settled, giving a stable downstream pressure.

When downstream pressure rises, the pressure under the pilot diaphragm is greater than the force created by the adjustment spring and the pilot diaphragm moves up. This closes the pilot valve and interrupts the transmission of steam pressure to the underside of the main diaphragm. The top of the main diaphragm is subjected to downstream pressure at all times and, as there is now more pressure above the main diaphragm than below, the main diaphragm moves down pushing the steam underneath into the downstream pipework via the control pipe and surplus pressure orifice. The pressure either side of the main diaphragm is balanced, and a small excess force created by the main valve return spring closes the main valve.

Any variations in load or pressure will immediately be sensed on the pilot diaphragm, which will act to adjust the position of the main valve accordingly, ensuring a constant downstream pressure.

The pilot-operated design offers a number of advantages over the direct acting valve. Only a very small amount of steam has to flow through the pilot valve to pressurise the main diaphragm chamber and fully open the main valve. Thus only very small changes in control pressure are necessary to produce large changes in flow. The fall in downstream pressure relative to changes in steam flow is therefore small, typically less than three hundredths of a bar (3 kPa; 0.5 psi) from fully open to fully closed.

Although any rise in upstream pressure will apply an increased closing force on the main valve, the same rise in pressure will act on the underside of the main diaphragm and will balance the effect. The result is a valve which gives close control of downstream pressure regardless of variations on the upstream side.

In some types of pilot-operated valve, a piston replaces the main diaphragm. This can be advantageous in bigger valves, which would require very large size main diaphragms. However, problems with the piston sticking in its cylinder are common, particularly in smaller valves.

It is important for a strainer and separator to be installed immediately prior to any pilot-operated control valve, as clean dry steam will prolong its service life.

Selection and installation of pressure reducing valves

The first essential is to select the best type of valve for a given application.

Small loads where accurate control is not vital should be met by using simple direct acting valves. In all other cases, the pilot-operated valve is the best choice, particularly if there are periods of no demand when the downstream pressure must not be allowed to rise.

Oversizing should be avoided with all types of control valve and this is equally true of reducing valves. A valve plug working close to its seat when passing wet steam can suffer wiredrawing and premature erosion. In addition, any small movement of the oversized valve plug will produce a relatively large change in the flow through the valve, making it more difficult for the valve to control accurately.

A smaller, correctly sized reducing valve will be less prone to wear and will provide more accurate control. Where it is necessary to make big reductions in pressure or to cope with wide fluctuations in load, it may be preferable to use two or more valves in series or in parallel.

Although reliability and accuracy depend on correct selection and sizing, pressure reducing valves also depend on correct installation. Figure 7.3.5 illustrates an ideal arrangement for the installation of a pilot-operated pressure reducing valve.

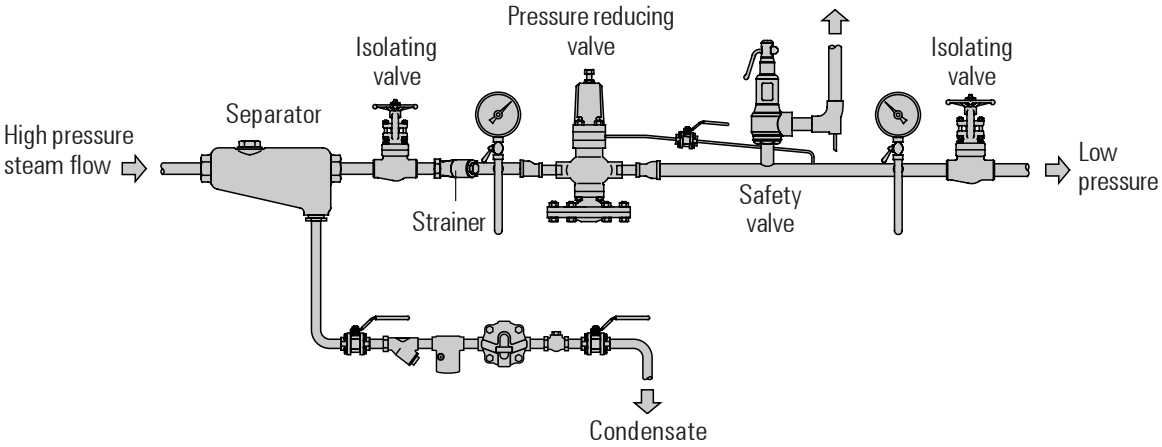


Fig. 7.3.5 Typical steam pressure reducing valve station

Many reducing valve problems are caused by the presence of moisture or dirt. A steam separator and strainer with fine mesh screen, if fitted before the valve, will help to prevent such problems. The strainer is fitted on its side to prevent the body filling with water and to ensure that the full area of the screen is effective. Large isolation valves will also benefit from being installed on their side for the same reason.

All upstream and downstream pipework and fittings must be adequately sized to ensure that the only appreciable pressure drop occurs across the reducing valve itself. If the isolating valves are the same size as the reducing valve connections, they will incur a larger pressure drop than if they are sized to match the correctly sized, larger diameters of the upstream and downstream pipework.

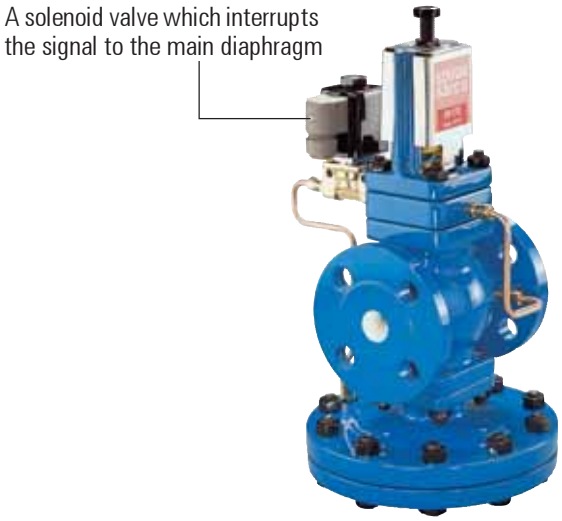
If the downstream pipework or any connected plant is incapable of withstanding the maximum possible upstream pressure, then a safety valve or relief valve must be fitted on the downstream side. This valve should be set at, or below, the maximum allowable working pressure of the equipment, but with a sufficient margin above its normal operating pressure. It must be capable of handling the full volume of steam that could pass through the fully open reducing valve, at the maximum possible upstream pressure.

Pilot operation also allows the reducing valve to be relatively compact compared to other valves of similar capacity and accuracy, and allows a variety of control options, such as on-off operation, dual pressure control, pressure and temperature control, pressure reducing and surplussing control, and remote manual adjustment. These variations can be seen in Figure 7.3.6.

Direct acting and pilot-operated control valves can be used to control either upstream or downstream pressures. Pressure maintaining valves (and surplussing valves) sense upstream pressure, while pressure reducing valves sense downstream pressure.



Basic pilot-operated pressure reducing valve



With on-off control



With temperature control



With dual pressure control

Fig. 7.3.6 Four complementary versions of pilot-operated pressure reducing valve

Summary of pressure reducing valves

A valve that senses and controls the downstream pressure is often referred to as a 'let-down' valve or 'pressure reducing valve' (PRV). Such valves can be used to maintain constant steam pressure onto a control valve, a steam flowmeter, or directly onto a process.

Pressure reducing valves are selected on capacity and type of application.

Table 7.3.1 Typical characteristics for different types of pressure reducing valve

Direct acting		Pilot-operated
Bellows operated	Diaphragm operated	
Small capacity	Very large capacity	Large capacity
Compact	Relatively large	Compact for capacity
Low cost	Robust	Extremely accurate
Steady load	Steady load	Varying loads
Coarse control	Coarse control	Fine control

Pressure maintaining valves

Some applications require that upstream pressure is sensed and controlled and this type of valve is often referred to as a 'Pressure Maintaining Valve' or 'PMV'. Pressure maintaining valves are also known as surplussing valves or spill valves in certain applications.

An example of a PMV application would be where steam generation plant is undersized, and yet steam flow is critical to the process. If steam demand is greater than the boiler output, or suddenly rises when the boiler burner is off, the boiler pressure will drop; progressively wetter steam will be supplied to the plant and the boiler operation may be jeopardised. If the boiler can operate at its design pressure, optimum steam quality will be maintained.

This can be achieved by fitting PMVs on each non-critical application (perhaps heating plant or domestic hot water plant), thereby introducing a controlled diversity to the plant. These will then progressively shut down if upstream pressure falls, giving priority to essential services. Should all supplies be considered essential, a variety of options are available, each of which has a different cost implication.

The cheapest solution might be to fit a PMV in the boiler steam outlet, (see PMV 1 in Figure 7.3.7). This will maintain a minimum steam pressure in the boiler, regulate maximum flow from the boiler and, in so doing, retain good quality steam to the plant.

If it is possible to shut off non-essential equipment during times of peak loading, PMVs can be installed in distribution lines or branch lines supplying these areas of the plant. When the steam boiler becomes overloaded, the non-essential supplies are gradually shut down by PMV 2 allowing the boiler to maintain steam flow to the 'essential' plant at the proper pressure.

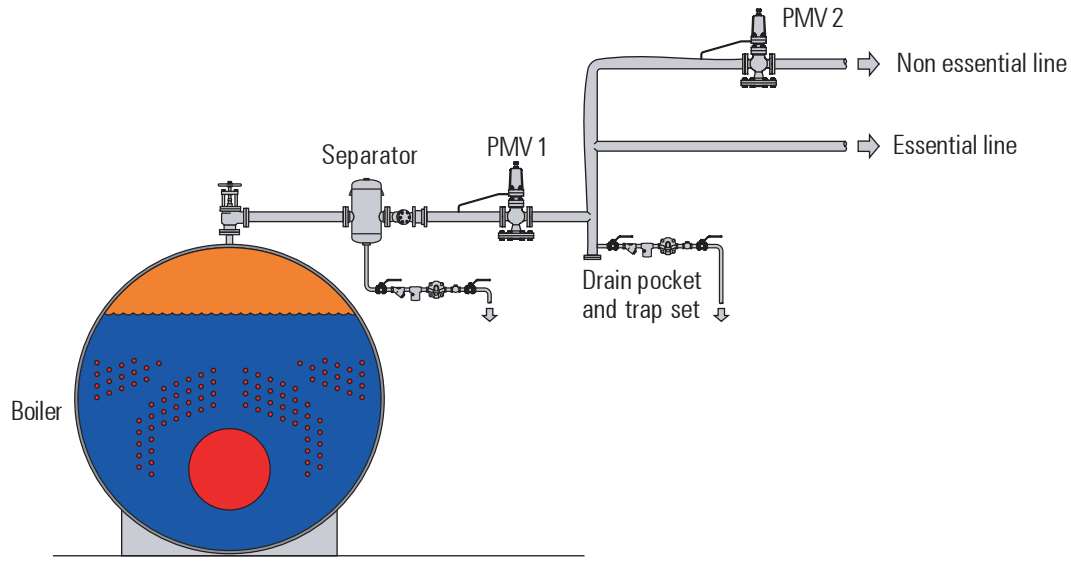


Fig. 7.3.7 Alternative positions for PMVs

It should be recognised that a PMV will not always cure the problems caused by insufficient boiler capacity. Sometimes, when there is little plant diversity, only one real alternative is available, which is to increase the generating capacity by adding another boiler.

However, there are occasions when the cheaper alternative of a steam accumulator is possible. This allows excess boiler energy to be stored during periods of low load. When the boiler is overloaded, the accumulator augments the boiler output by allowing a controlled release of steam to the plant (see Figure 7.3.8).

In Figure 7.3.8, the boiler is designed to generate steam at 10 bar g, which is distributed at both 10 bar g and 5 bar g to the rest of the plant.

PRV 1 is a pressure reducing valve, and is sized to pass the boiler capacity minus the high pressure steam load.

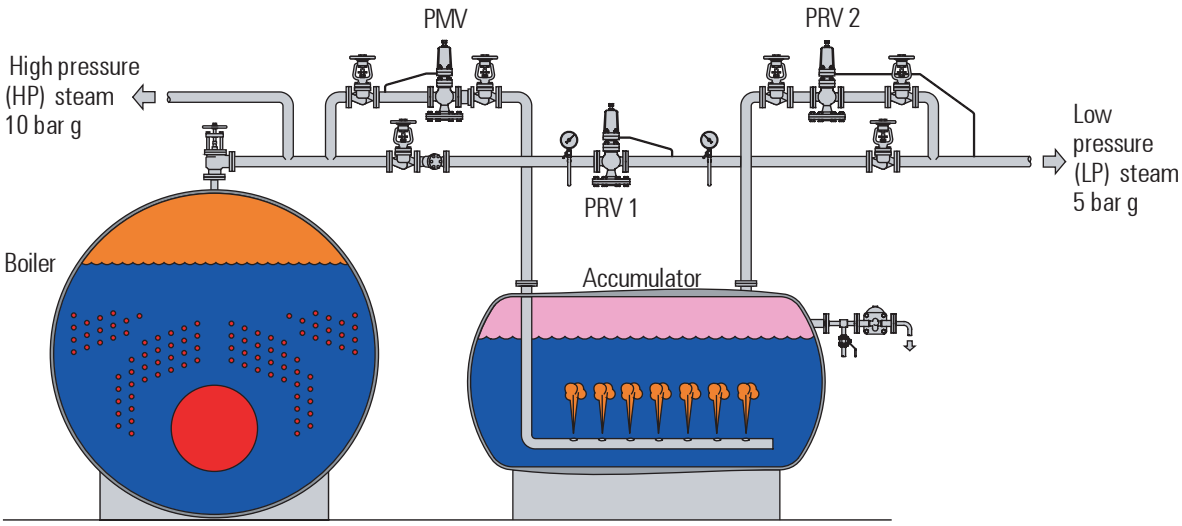


Fig. 7.3.8 Typical boiler and accumulator arrangement

For sizing purposes, the capacity of the pressure reducing valve PRV 2 should equal the maximum discharge rate and time for which the accumulator has been designed to operate, whilst the differential pressure for design purposes should be the difference between the minimum operating accumulator pressure and the LP (Low pressure) distribution pressure. In this example, PRV 2 would probably be set to open at about 4.8 bar g.

PMV is a pressure maintaining valve whose size is determined by the recharging time required by the accumulator and the available surplus boiler capacity during recharging. When recharging, the pressure drop across the PMV is likely to be relatively small, so the PMV is likely to be quite large, typically the same size as the line in which it is installed. The PMV is usually set to operate just below the boiler maximum pressure setting.

When the total plant load is within the boiler capacity, PRV 2 is shut and the boiler supplies the LP steam load through PRV 1 which is set to control slightly higher than PRV2. Any excess steam available in the boiler will cause the boiler pressure to rise above the PMV set point, and the PMV will open to recharge the accumulator. Recharging will continue until the accumulator pressure equals the boiler pressure, or until the plant load is such that the boiler pressure again drops below the PMV set point.

Should the LP steam load continue to increase, causing the LP pressure to drop below PRV 2 set point, PRV 2 will open to provide steam from the accumulator, in turn supplementing the steam flowing through PRV 1.

There is more than one way in which to design an accumulator installation; each will depend upon the circumstances involved, and will have a cost implication. The subject of accumulators is discussed in more detail in Module 3.22 ‘Steam accumulators’.

Pressure surplussing valves

The ability to sense upstream pressure may be used to release surplus pressure from a steam system in a controlled and safe manner. The surplussing valve is essentially the same as a PMV, opening when an increase in upstream pressure is sensed. The surplussing valve is sometimes referred to as a 'dump' valve when releasing steam to atmosphere.

A 'surplussing valve' is often used to control the maximum pressure in a flash recovery system. Should the demand for flash steam be less than the available supply, the flash pressure will rise and the surplussing valve will open to release any excess steam to atmosphere. The surplussing valve will be set to operate at a pressure below the safety valve setting.

Important: Whilst this allows the controlled release of steam to atmosphere, it does not replace the need for a safety valve, should the plant conditions require it.

In Figure 7.3.9 the PRV replenishes any shortfall of flash steam generated by the high pressure (HP) condensate, and the surplussing valve releases any excess flash steam to either a condenser or to atmosphere.

The safety valve is sized on the full capacity of the PRV plus the capacity of the steam traps and any other source feeding into the flash vessel.

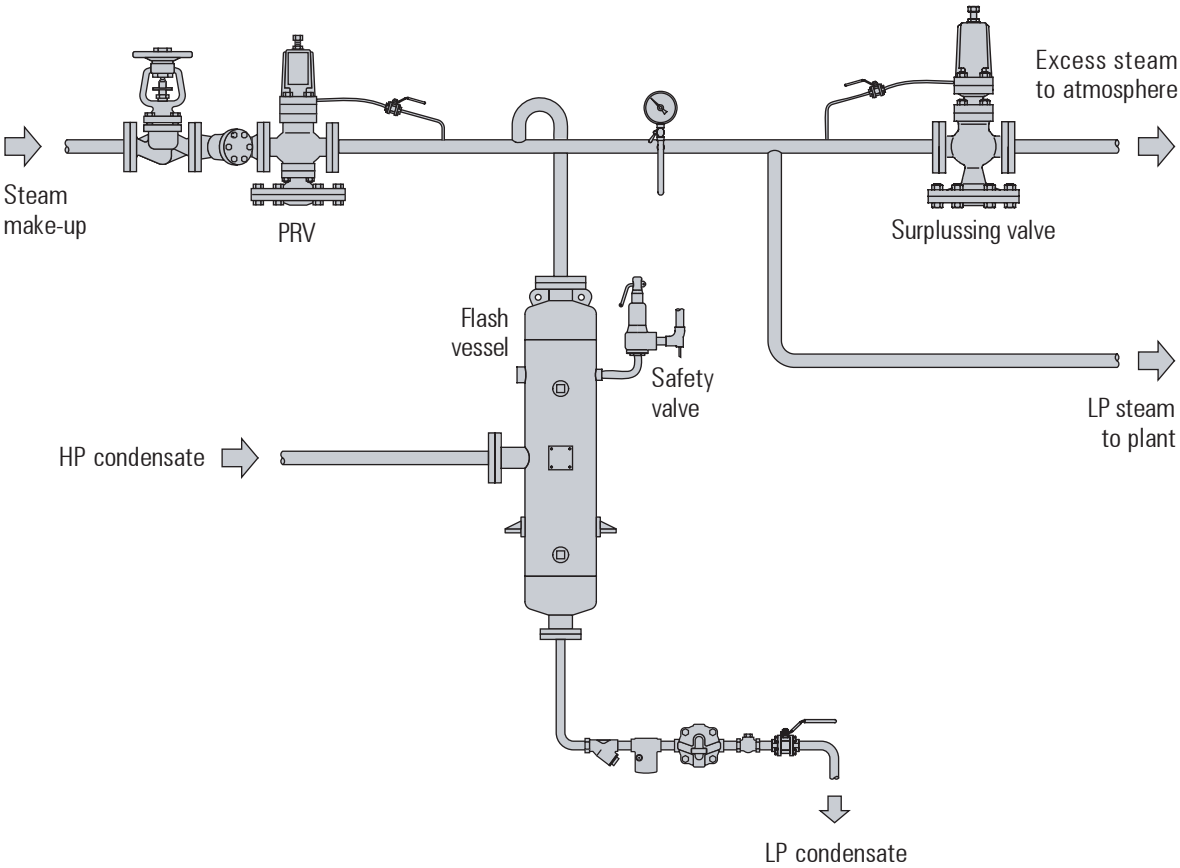


Fig. 7.3.9 Typical surplussing valve on a flash vessel application

Questions

1. In a self-acting pressure control system, which of the following is proportional to the control valve opening?

a| The deviation of the downstream pressure from the set point ☐
b| The difference between upstream and downstream pressure ☐
c| The difference between upstream pressure and the set point ☐
d| The spring force ☐

2. What is 'proportional offset'?

a| The rise in downstream pressure as flow increases through the control valve ☐
b| The fall in downstream pressure as flow decreases through the control valve ☐
c| The difference between the set point and actual downstream pressure ☐
d| The rise in upstream pressure when the control valve shuts ☐

3. Name an advantage that a pilot-operated pressure reducing valve has over a direct acting pressure reducing valve?

a| It is usually smaller for the same capacity ☐
b| It has a much lower proportional offset ☐
c| It is more accurate over large changes in load ☐
d| All of the above ☐

4. What is the basic difference between a PRV and a PMV?

a| A PRV reduces pressure and a PMV increases pressure ☐
b| As downstream pressure drops, a PRV will close and a PMV will open ☐
c| As the sensed pressure drops, a PRV will open and a PMV will close ☐
d| As upstream pressure drops, a PRV will close and a PMV will open ☐

5. What can a PMV be used for?

a| To reduce non-essential loads, maintaining steam distribution pressure ☐
b| To maintain boiler pressure under overload conditions ☐
c| To exhaust surplus steam from a flash steam system ☐
d| All of the above ☐

6. Which of the following can a PMV not be used as?

a| A safety valve ☐
b| A pressure maintaining valve ☐
c| A pressure surplussing valve ☐
d| A pressure dump valve ☐

Answers

1: a, 2: c, 3: d, 4: c, 5: d, 6: a

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Module 8.1

Pressure Control Applications

Pressure Control Applications

There are many reasons for reducing steam pressure:

- Steam boilers are usually designed to work at high pressures in order to reduce their physical size. Operating them at lower pressures can result in reduced output and 'carryover' of boiler water. It is, therefore, usual to generate steam at higher pressure.
- Steam at high pressure has a relatively higher density, which means that a pipe of a given size can carry a greater mass of steam at high pressure, than at low pressure. It is usually preferable to distribute steam at high pressure as this allows smaller pipes to be used throughout most of the distribution system.
- Lower condensing pressures at the point of use tend to save energy. Reduced pressure will lower the temperature of the downstream pipework and reduce standing losses, and also reduce the amount of flash steam generated when condensate from drain traps is discharging into vented condensate collecting tanks.

  It is worth noting that if condensate is continuously dumped to waste, perhaps because of the risk of contamination, less energy will be lost if the condensing pressure is lower.
- Because steam pressure and temperature are related, control of pressure can be used to control temperature in some processes. This fact is recognised in the control of sterilisers and autoclaves, and is also used to control surface temperatures on contact dryers, such as those found in papermaking and corrugator machines. Pressure control is also the basis of temperature control in heat exchangers.
- For the same heating duty, a heat exchanger designed to operate on low-pressure steam will be larger than one designed to be used on high-pressure steam. The low-pressure heat exchanger might be less expensive because of a lower design specification.
- The construction of plant means that each item has a maximum allowable working pressure (MAWP). If this is lower than the maximum possible steam supply pressure, the pressure must be reduced so that the safe working pressure of the downstream system is not exceeded.
- Many plants use steam at different pressures. A 'stage' system where high-pressure condensate from one process is flashed to steam for use in another part of the process is usually employed to save energy. It may be necessary to maintain continuity of supply in the low pressure system at times when not enough flash steam is being generated. A pressure reducing valve is ideally suited for this purpose.

Direct operating, self-acting pressure reducing valve – bellows type

Description

With this self-acting type of pressure controller, the downstream (control) pressure is balanced (via a bellows) against a spring force.

Advantages:

1. Inexpensive.
2. Small.
3. Easy to install.
4. Very robust, giving long life with minimum maintenance.
5. Tolerant of imperfect steam conditions.
6. Self-acting principle means that no external power is required.

Disadvantages:

1. Proportional only control.
2. Proportional band is 30% to 40% of the upstream pressure.
3. Wide proportional band means that maximum flow is only achieved when the downstream pressure has dropped considerably. This means that the reduced pressure will vary depending on flowrate.
4. Limited in size.
5. Limited flowrate.
6. Variation in upstream pressure will result in variation in downstream pressure.

Applications:

Non-critical, moderate load applications with constant running flowrates, for example:

1. Small jacketed pans.
2. Tracer lines.
3. Ironers.
4. Small tanks.
5. Acid baths.
6. Small storage calorifiers.
7. Unit heaters.
8. Small heater batteries.
9. OEM equipment.

Points to note:

1. Different versions for steam, compressed air, and water.
2. Soft seat versions may be available for use on gases.
3. A wide range of body materials means that particular standards, applications and preferences can be satisfied.
4. A wide proportional band means care is needed if the safety valve needs to be set close to the working pressure.

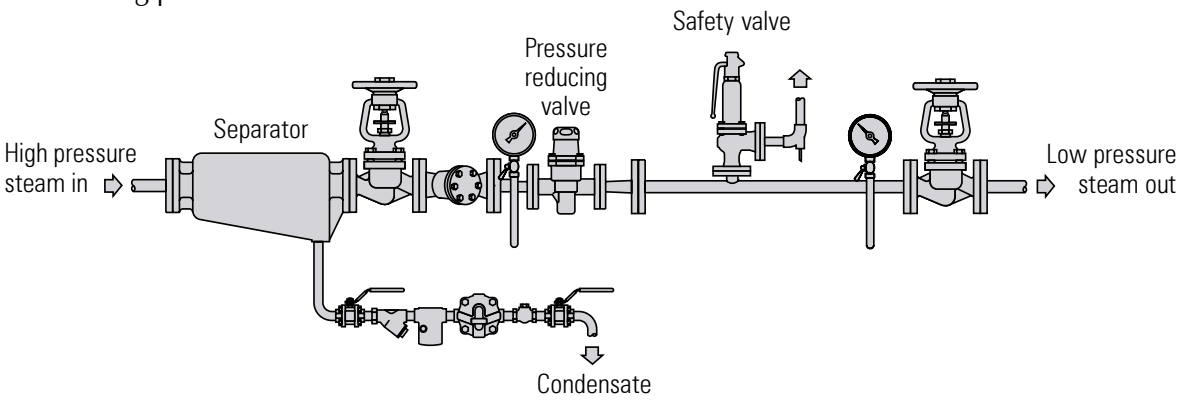


Fig. 8.1.1 General arrangement of a direct operating, self-acting pressure reducing station

Direct operating, self-acting pressure reducing valve – diaphragm type

Description:

With this self-acting type of pressure controller, the downstream (control) pressure is balanced (via a diaphragm) against a spring force.

Advantages:

1. Very robust.
2. Tolerant to wet and dirty steam.
3. Available in large sizes, so high flowrates are possible.
4. Easy to set and adjust.
5. Simple design means easy maintenance.
6. Self-acting principle means that no external power is required.
7. Able to handle pressure drops of 50:1 in small sizes, and 10:1 in large sizes.

Disadvantages:

1. Large proportional band means that close control of downstream pressure is improbable with large changes in load.
2. Relatively high purchase cost, but lifetime cost is low.
3. Bulky.

Applications:

1. Distribution mains.
2. Boiler houses.

Points to note:

1. Because the diaphragm is subject to fairly low temperature limitations, a water seal is required on steam applications. This adds to the cost slightly.
2. Because of the large proportional band, this type of valve is better suited to reducing steam pressure to plant areas rather than individual plant items.
3. A bellows sealed stem ensures zero maintenance and zero emissions.
4. Although wide proportional band provides stability, care is needed if a safety valve needs to be set close to the apparatus working pressure.
5. Suitable for liquid applications.
6. More expensive than a pilot operated valve, but less expensive than a pneumatic control system.

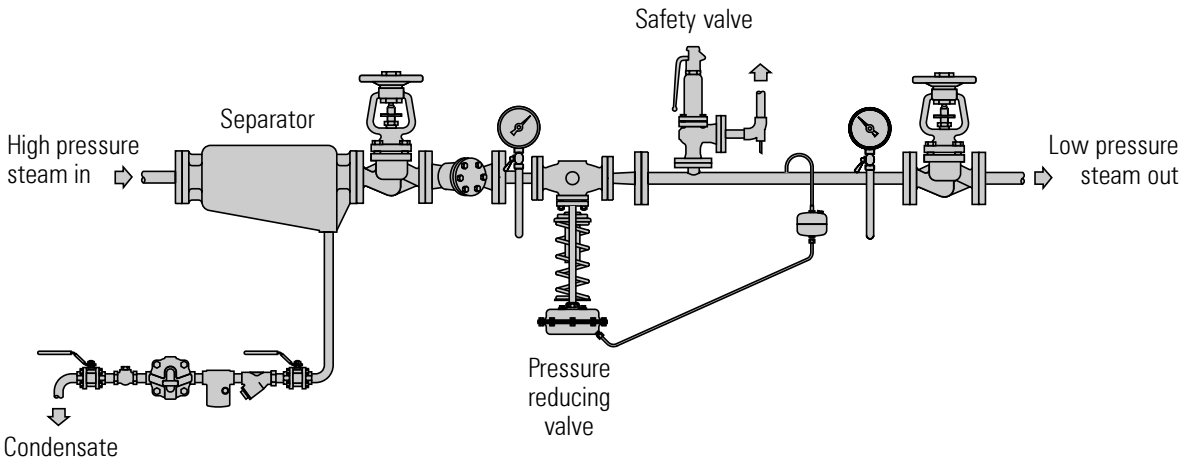


Fig. 8.1.2 General arrangement of a direct operating, self-acting pressure reducing station

Pilot operated, self-acting pressure reducing valve

Description

These have a more complex self-acting design, and operate by sensing the downstream pressure via a pilot valve, which in turn operates the main valve.

The effect is a very narrow proportional band, typically less than 200 kPa.

This, together with low hysterisis, results in very tight and repeatable control of pressure, even with widely varying flowrates.

Advantages:

1. Accurate and consistent pressure control, even at high and variable flowrates.
2. A variety of pilot valves may be used on one main valve. Pilot valve options include electrical override, multi-pilot for a choice of control pressures, a surplussing option and remote control, as well as different temperature/pressure control combinations.
3. Self-acting principle means that no external power is required.
4. Tolerant of varying upstream pressure.

Disadvantages:

1. More expensive than bellows operated direct acting controls.
2. Small clearances mean that steam must be clean and dry to ensure longevity, but this can be achieved by fitting a strainer and separator before the pressure reducing valve.

Applications:

1. A system which requires accurate and consistent pressure control, and installations which have variable and medium flowrates. For example: autoclaves, highly rated plant such as heat exchangers and calorifiers.
2. A system where installation space is limited.

Points to note:

1. Installation must include a strainer and separator.
2. Size for size, pilot operated valves are more expensive than bellows type self-acting controls, but cheaper than diaphragm type self-acting controls.
3. Size for size, they have higher capacity than bellows type self-acting controls, but less than diaphragm type self-acting controls.
4. Can be installed before temperature control valves to maintain a constant upstream pressure, and hence stabilise control.
5. Not suitable for liquid applications.
6. Do not use if the plant is subject to vibration, or other equipment is causing pulses in flow.

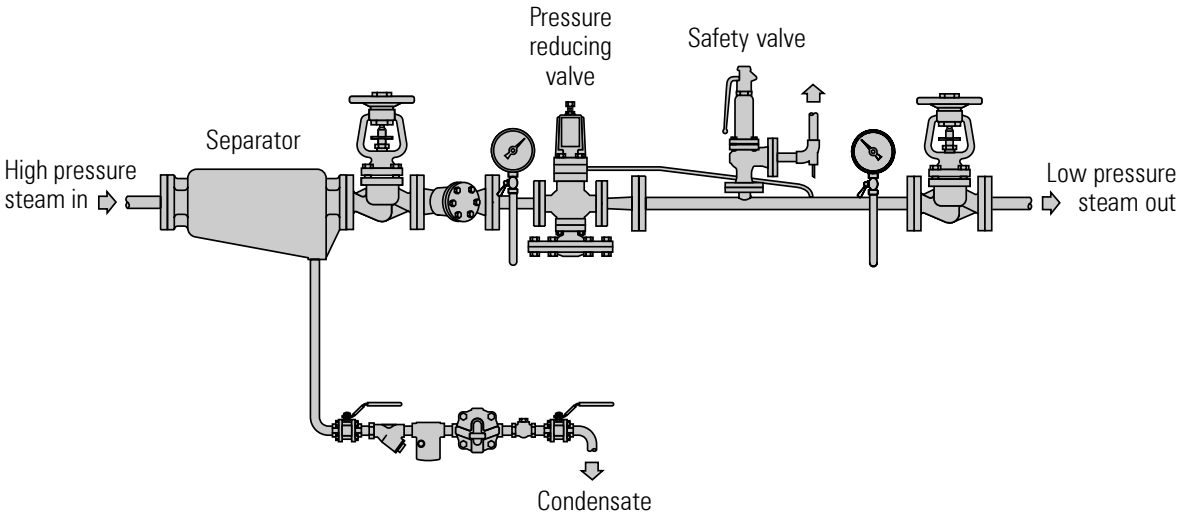


Fig. 8.1.3 General arrangement of a pilot operated, self-acting pressure reducing station

Pressure reduction – pneumatic

Description:

These control systems may include:

- P + I + D functions to improve accuracy under varying load conditions.
- Set point(s), which may be remotely adjusted.

Advantages:

1. Very accurate and flexible.
2. No limit on valve size within the limits of the valve range.
3. Acceptable 50:1 flow rangeability (typically for a globe control valve).
4. Suitable for hazardous environments.
5. No electrical supply required.
6. Fast operation means they respond well to rapid changes in demand.
7. Very powerful actuation being able to cope with high differential pressures across the valve.

Disadvantages:

1. More expensive than self-acting controls.
2. More complex than self-acting controls.
3. Not directly programmable.

Applications:

A system which requires accurate and consistent pressure control, and installations which have variable and high flowrates and/or variable or high upstream pressure. For example: autoclaves, highly rated plant such as large heat exchangers and calorifiers.

Points to note:

1. A clean, dry air supply is required.
2. A skilled workforce is required to install the equipment, and instrument personnel are required for calibration and commissioning.
3. The control is 'stand-alone', and cannot communicate with PLCs (Programmable Logic Controllers).
4. The failure mode can be important. For example, a spring-to-close on air failure is normal on steam systems.

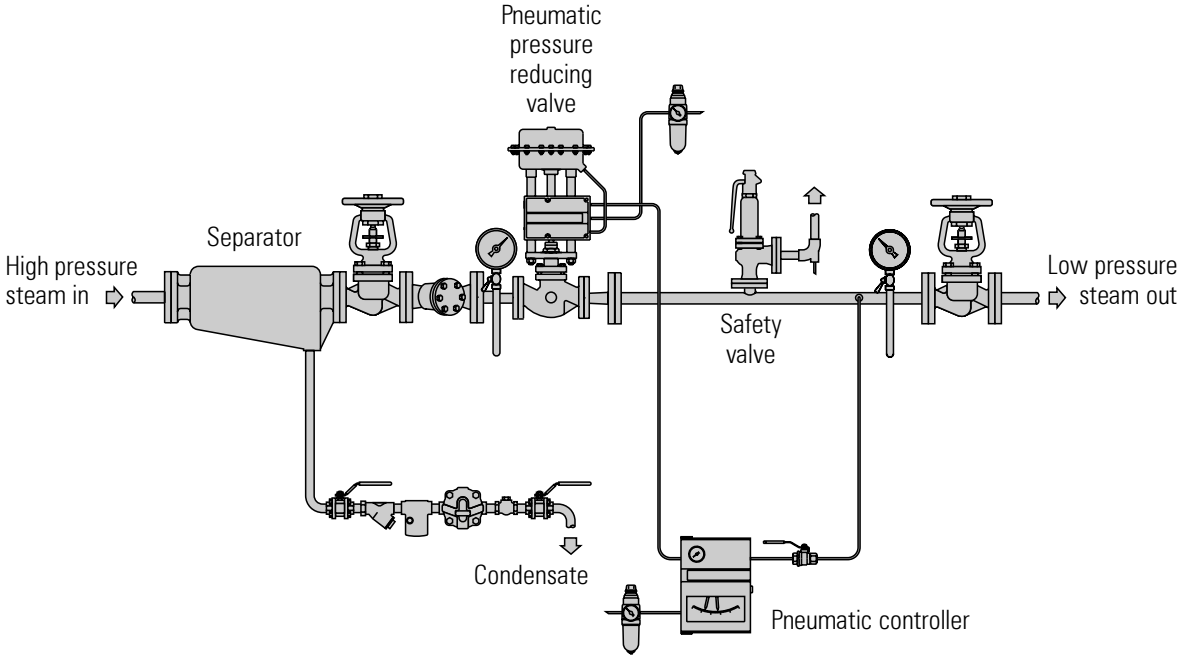


Fig. 8.1.4 General arrangement of a pneumatic pressure reducing station

Pressure reduction – electropneumatic

Description

These control systems may include:

- P + I + D functions to improve accuracy under varying load conditions.
- Set point(s) which may be remotely adjusted, with the possibility of ramps between set points.

Advantages:

1. Very accurate and flexible.
2. Remote adjustment and read-out.
3. No limit on valve size within the limits of the valve range.
4. Acceptable 50:1 flow rangeability (typically for a globe control valve).
5. Fast operation – rapid response to changes in demand.
6. Very powerful actuation being able to cope with high differential pressures across the valve.

Disadvantages:

1. More expensive than self-acting or pneumatic controls.
2. More complex than self-acting or pneumatic controls.
3. Electrical control signal required. Costly for hazardous areas.

Applications:

A system which requires accurate and consistent pressure control, and installations which have variable and high flowrates and/or variable or high upstream pressure, including autoclaves, highly rated plant such as large heat exchangers and calorifiers, and main plant pressure reducing stations.

Points to note:

1. A clean, dry air supply is required.
2. A skilled workforce is required to install the equipment, and instrument personnel are required for calibration and commissioning.
3. Can be part of a sophisticated control system involving PLCs, chart recorders and SCADA systems.
4. Always consider the failure mode, for example, spring-to-close on air failure is normal on steam systems.

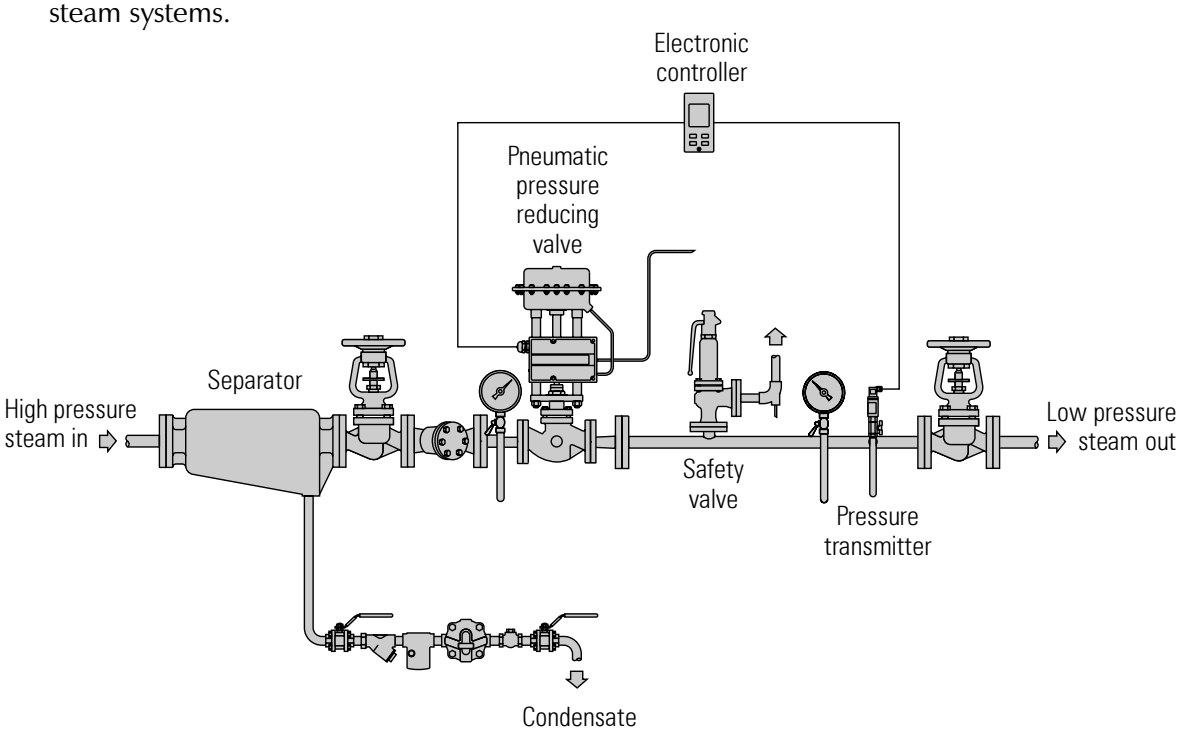


Fig. 8.1.5 General arrangement of an electropneumatic pressure reducing station

Pressure reduction – electric

Description:

These control systems may include:

- P + I + D functions to improve accuracy under varying load conditions.
- Set point(s), which may be remotely adjusted.

Advantages:

1. Both controller and valve actuator can communicate with a PLC.
2. No compressed air supply is required.

Disadvantages:

1. If a spring return actuator is required, the available shut-off pressure may be limited.
2. Relatively slow actuator speed, so only suitable for applications where the load changes slowly.

Applications:

1. Slow opening / warm-up systems with a ramp and dwell controller.
2. Pressure control of large autoclaves.
3. Pressure reduction supplying large steam distribution systems.

Points to note:

1. Safety: If electrical power is lost the valve position cannot change unless a spring return actuator is used.
2. Spring return actuators are expensive and bulky, with limited shut-off capability.

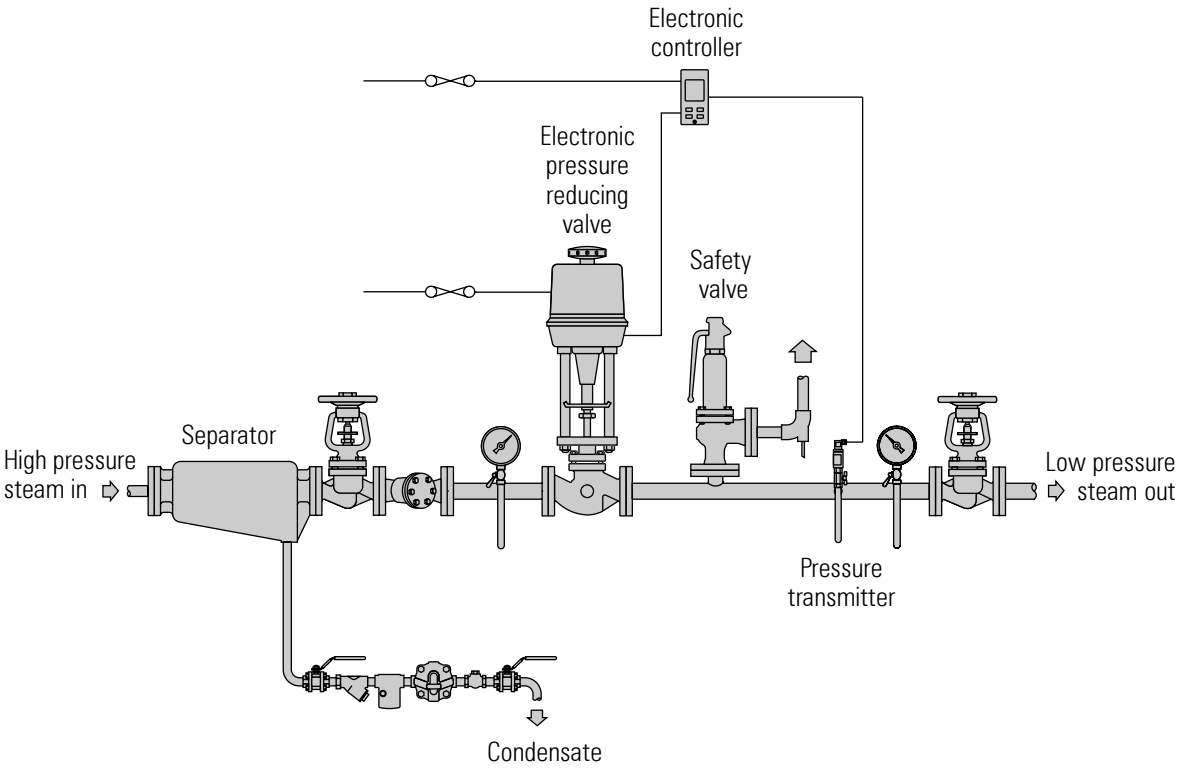


Fig. 8.1.6 General arrangement of an electric pressure reducing station

Pressure reduction (other possibilities) – Parallel pressure reducing stations

Description:

Pressure reducing stations may be configured as shown below for one of two reasons:

1. The valves are serving a critical application for which downtime is unacceptable
The equipment is operated on a 'one in operation, one on stand-by' basis to cover for breakdown and maintenance situations

2. The turndown ratio between the maximum and minimum flowrates is very high
The equipment is operated on a pressure sequence principle with one valve set at the ideal downstream pressure, and the other at a slightly lower pressure.

When demand is at a maximum, both valves operate; when flow is reduced, the valve set at the lower pressure shuts off first, leaving the second valve to control.

Point to note:

The valves selected for this type of application will require narrow proportional bands (such as pilot operated pressure reducing valves or electro-pneumatic control systems) to avoid the downstream pressure dropping too much at high flow rates.

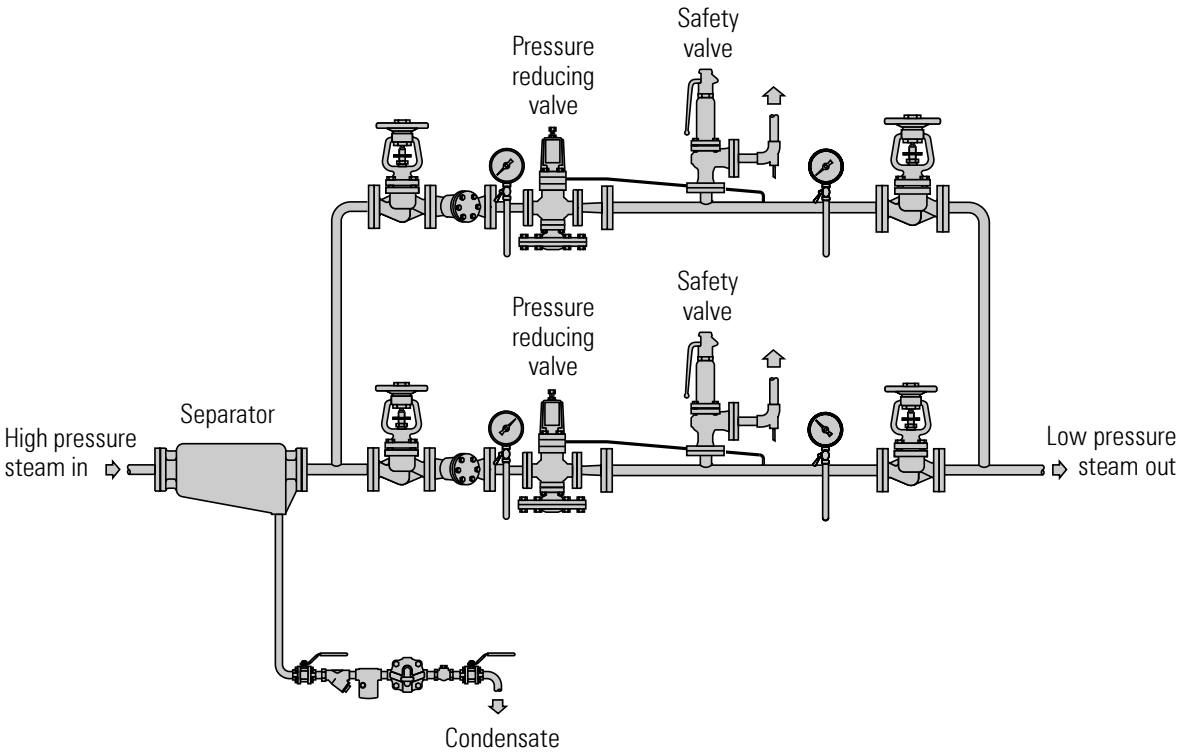


Fig. 8.1.7 Parallel pressure reducing station

Pressure reduction (other possibilities) – Series pressure reducing stations

A pressure reducing station may be configured in this manner if the ratio between the upstream and downstream pressure is very high, and the control systems selected have a low turndown ability. 10:1 is recommended as a practical maximum pressure ratio for this type of reducing valve.

Consider the need to drop pressure from 25 bar g to 1 bar g. The primary reducing valve might reduce pressure from 25 bar g to 5 bar g, which constitutes a pressure ratio of 5:1. The secondary reducing valve would drop pressure from 5 bar g to 1 bar g, also 5:1. Both valves in series provide a pressure ratio of 25:1.

It is important to check the allowable pressure turndown ratio on the selected reducing valve, this may be 10:1 on a self-acting valve, but can be much higher on electrically or pneumatically operated valves. Be aware that high pressure drops might have a tendency to create high noise levels. Refer to Module 6.4 for further details.

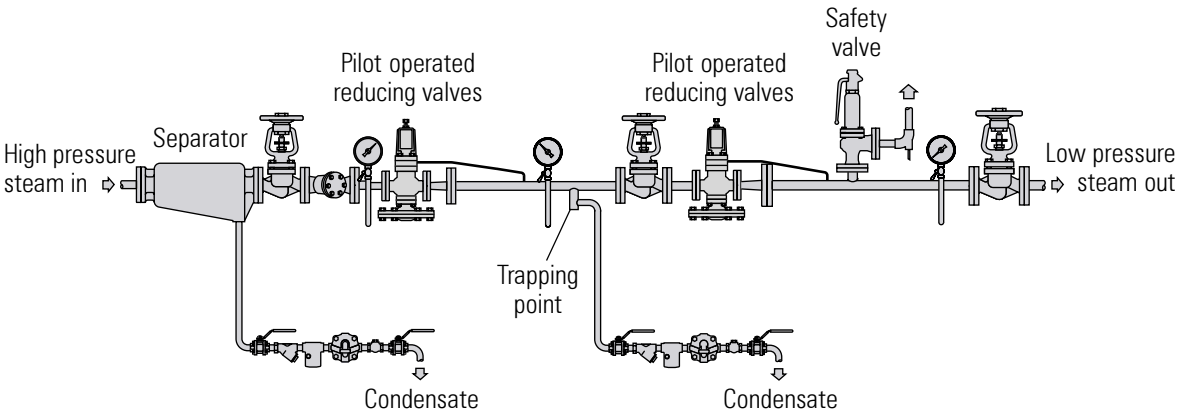


Fig. 8.1.8 Typical series pressure reducing station

The trapping point between the two reducing valves (Figure 8.1.8) is to stop a build up of condensate under no-load conditions. If this were not fitted, radiation losses would cause condensate to fill the connecting pipe, which would cause waterhammer the next time the load increased.

Desuperheaters

Desuperheating is the process by which superheated steam is either restored to its saturated state, or its superheated temperature is reduced. Further coverage of desuperheaters is given in Block 15.

The system in Figure 8.1.9 illustrates an arrangement of a pressure reducing station with a direct contact type pipeline desuperheater.

In its basic form, good quality water (typically condensate) is directed into the superheated steam flow, removing heat from the steam, causing a drop in the steam temperature.

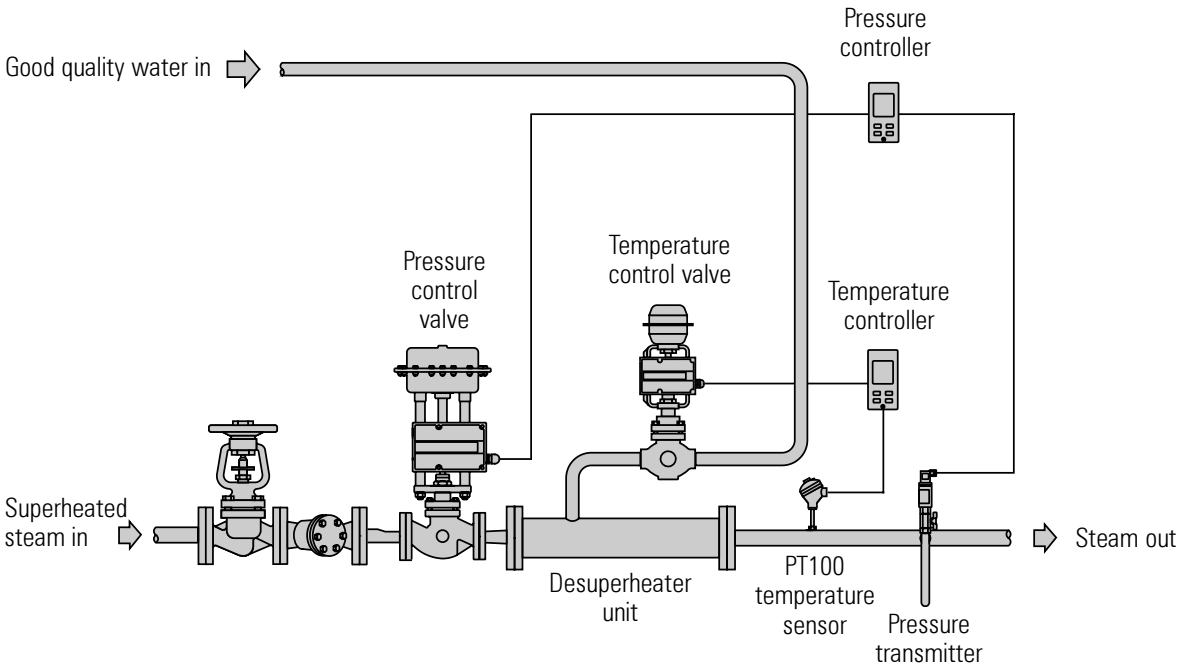


Fig. 8.1.9 Simple steam atomising desuperheater station

It is impractical to reduce the steam temperature to its saturated value, as the control system is unable to differentiate between saturated steam and wet steam at the same temperature.

Because of this, the temperature is always controlled at a value higher than the relevant saturation temperature, usually at 5°C to 10°C above saturation.

For most applications, the basic system as shown in Figure 8.1.9 will work well. As the downstream pressure is maintained at a constant value by the pressure control loop, the set value on the temperature controller does not need to vary; it simply needs to be set at a temperature slightly above the corresponding saturation temperature.

However, sometimes a more complex control system is required, and is shown in Figure 8.1.10. Should there be a transient change in the superheated steam supply pressure, or a change in the water supply temperature, the required water/steam flow ratio will also need to change.

A change in the water/steam flow ratio will also be required if the downstream pressure changes, as is sometimes the case with certain industrial processes.

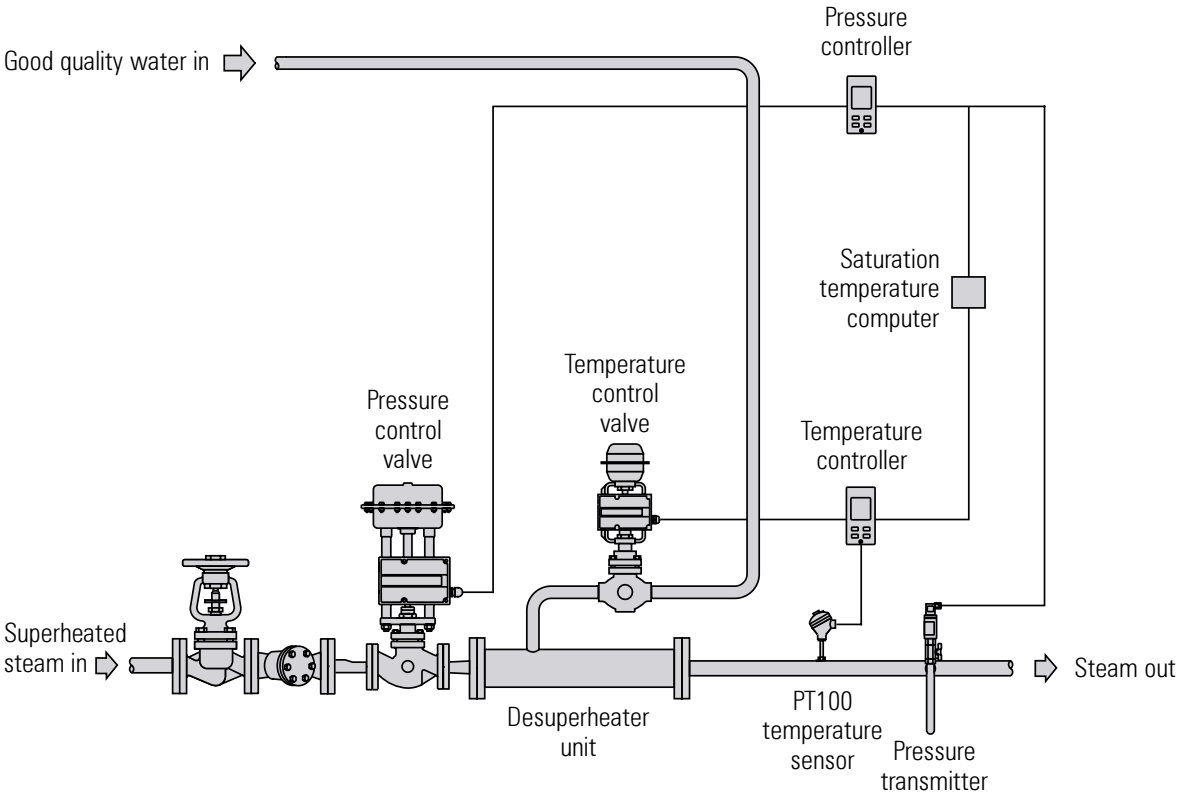


Fig. 8.1.10 Steam atomising desuperheater station with downstream pressure/temperature compensation

The system shown in Figure 8.1.10 works by having the pressure controller set at the required downstream pressure and operating the steam pressure control valve accordingly.

The 4-20 mA signal from the pressure transmitter is relayed to the pressure controller and the saturation temperature computer, from which the computer continuously calculates the saturation temperature for the downstream pressure, and transmits a 4-20 mA output signal to the temperature controller in relation to this temperature.

The temperature controller is configured to accept the 4-20 mA signal from the computer to determine its set point at 5°C to 10°C above saturation. In this way, if the downstream pressure varies due to any of the reasons mentioned above, the temperature set point will also automatically vary. This will maintain the correct water/steam ratio under all load or downstream pressure conditions.

Controlling pressure to control temperature

Description

These are applications which utilise the predictable relationship between saturated steam pressure and its temperature.

Advantages:

1. The pressure sensor may be located in the steam space, or close to the control valve rather than in the process medium itself. This is an advantage where it is difficult to measure the process temperature.
2. This arrangement can be used to control a number of different elements from a single point.

Disadvantage:

1. Control is 'open loop', in that the sensor is not measuring the actual product temperature.

Applications:

1. Autoclaves and sterilisers
2. Presses and calenders
3. Constant pressure plant, for example, jacketed pans, unit heaters, and steam-jacketed pipes.

Point to note:

Good air venting is essential (refer to Module 11.12 for further details)

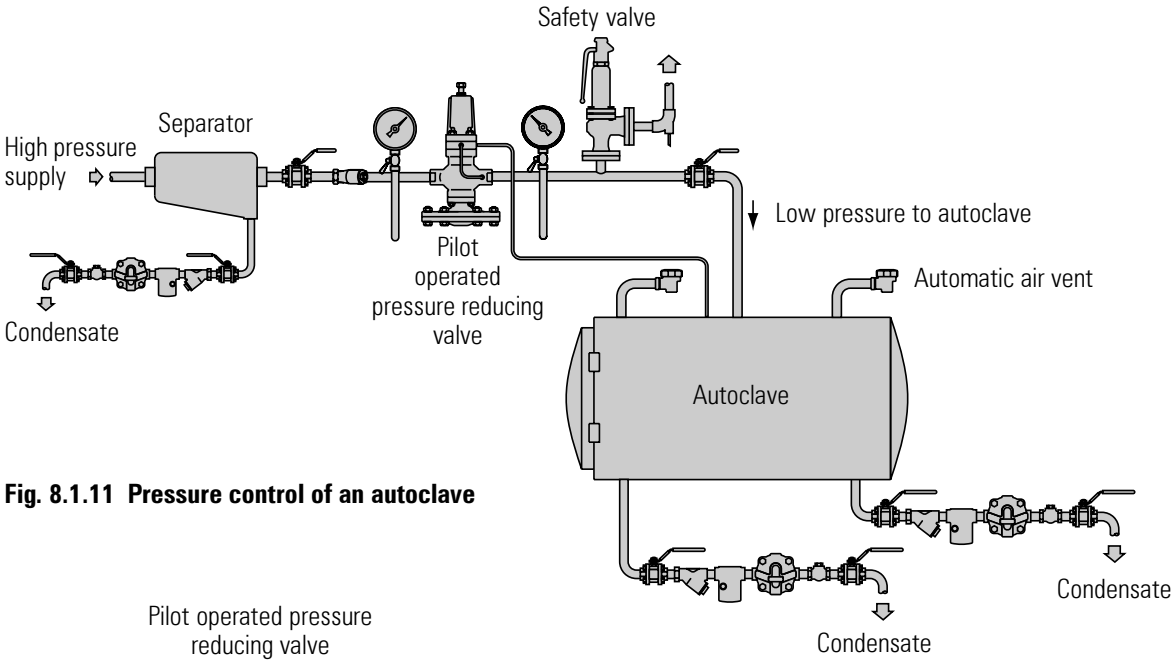


Fig. 8.1.11 Pressure control of an autoclave

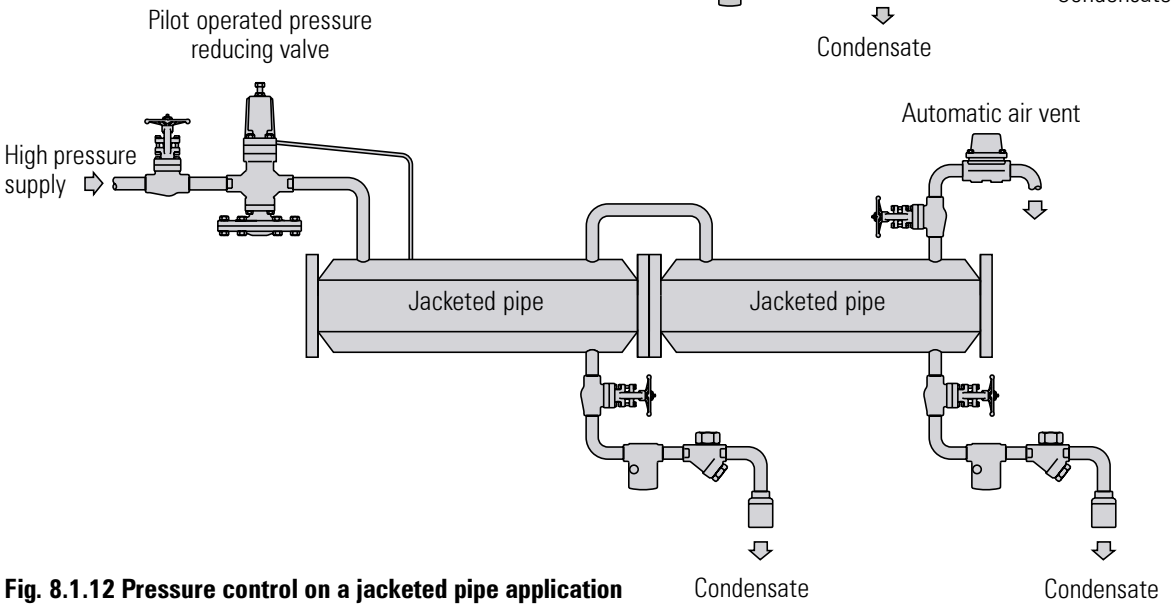


Fig. 8.1.12 Pressure control on a jacketed pipe application

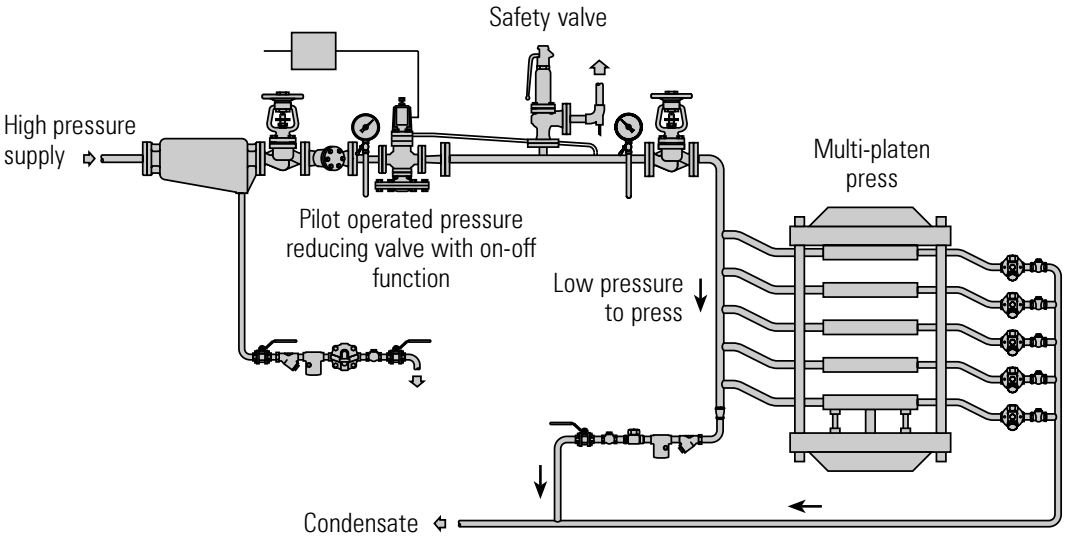


Fig. 8.1.13 Pressure control on a multi platen press

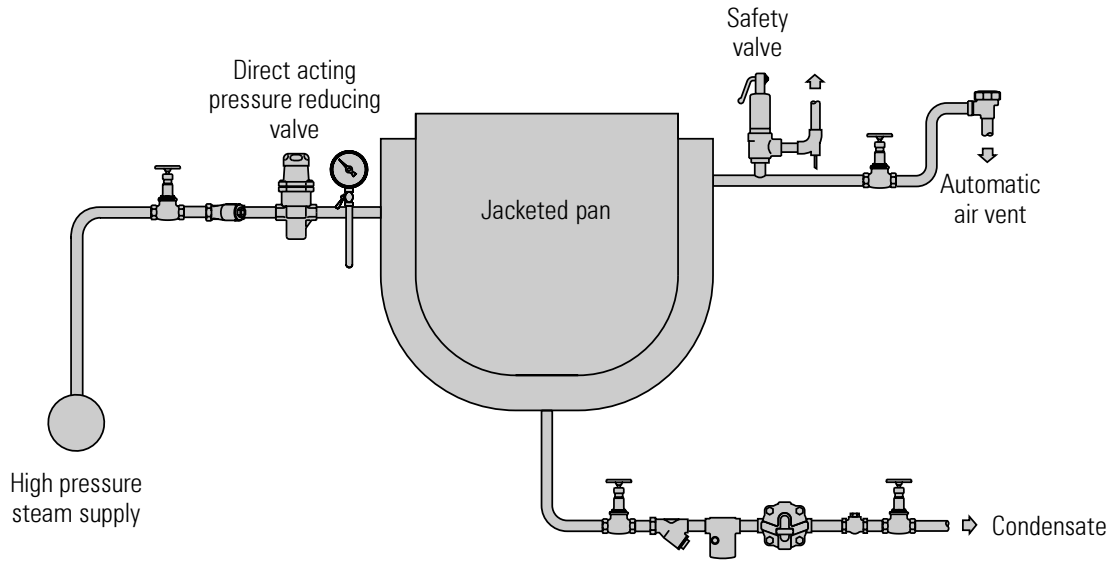


Fig. 8.1.14 Pressure / temperature control on a jacketed pan

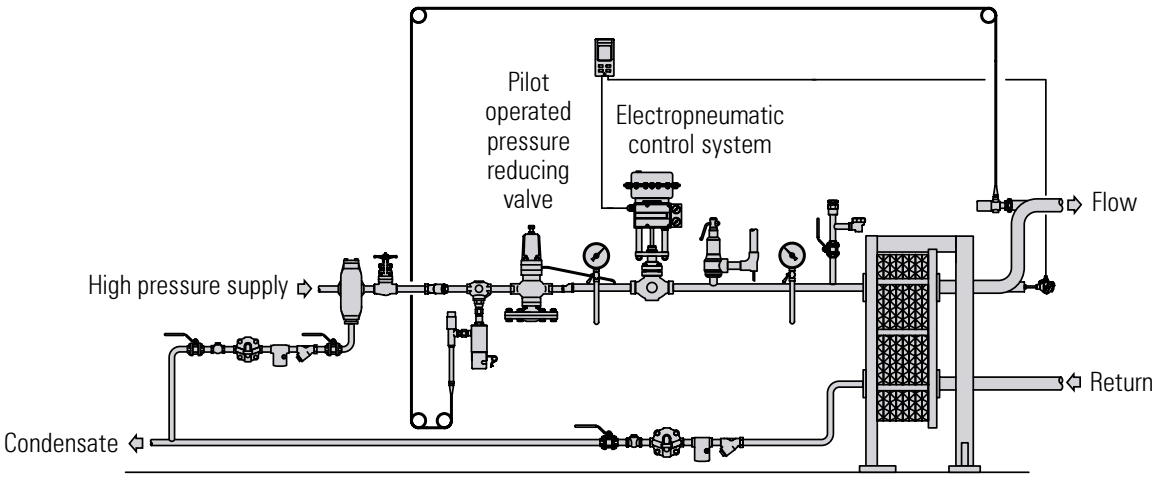


Fig. 8.1.15 Constant pressure steam supply to a control valve supplying a plate heat exchanger

Differential pressure control

Description

In these applications the control valve will open and close to maintain a set differential pressure between two points.

Advantages:

1. A constant differential steam pressure is maintained in the system.
2. The differential pressure ensures that condensate is actively purged from the heat exchange system. This is particularly important where accumulated condensate could act as a heat barrier, and create a temperature gradient across the heat transfer surface.

   This temperature gradient could, in turn, result in a distorted or poorly heated product.
3. Different operating temperatures can be achieved.

Disadvantage:

A complex system is required if efficiency is to be maintained. This might involve flash vessels and/or thermo-compressors, as well as downstream applications which use the lower pressure pass-out steam.

Application:

Blow-through drying rolls in a paper mill.

Point to note:

A special controller or differential pressure transmitter is required to accept two inputs; one from the primary steam supply and the other from the flash vessel. In this way, the pressure differential between the flash vessel and the primary steam supply is maintained under all load conditions.

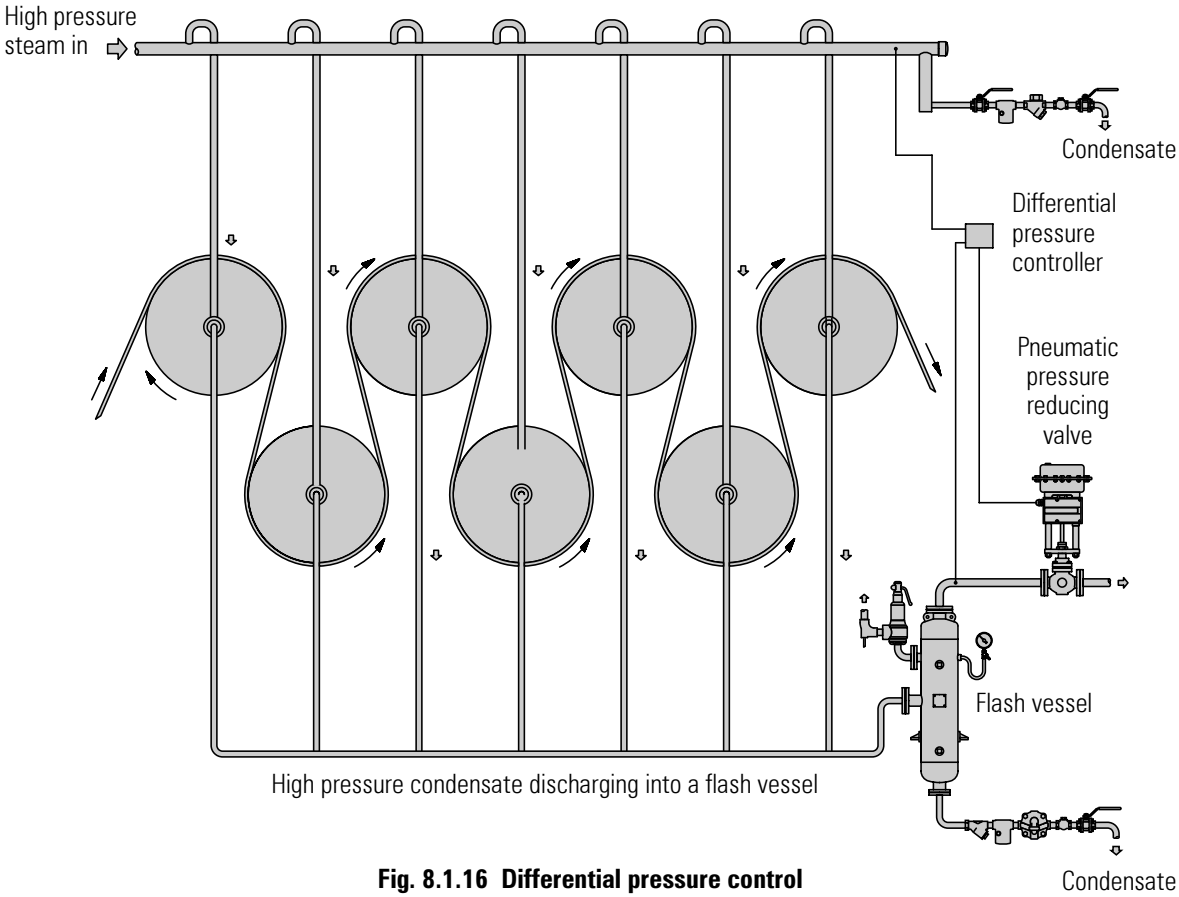


Fig. 8.1.16 Differential pressure control

Surplussing control

Description

The objective is to maintain the pressure upstream of the control valve. Surplussing valves are discussed in further detail in Module 7.3, 'Self-acting pressure controls and applications'.

Applications:

1. Boilers on plants where the load can change by a large proportion over a very short period. The sudden reduction in boiler pressure may result in increased turbulence and rapid flashing of the boiler water, and large quantities of water being carried over into the pipework system.
2. Accumulators where surplus boiler output is used to heat a mass of water under pressure. This stored energy is then released when the boiler has insufficient capacity.

Points to note:

1. Minimum pressure drop is usually required over the fully open control valve; this may mean a 'line size' valve is needed.
2. Not all self-acting controls are suitable for this application and it is important to consult the manufacturer before use.

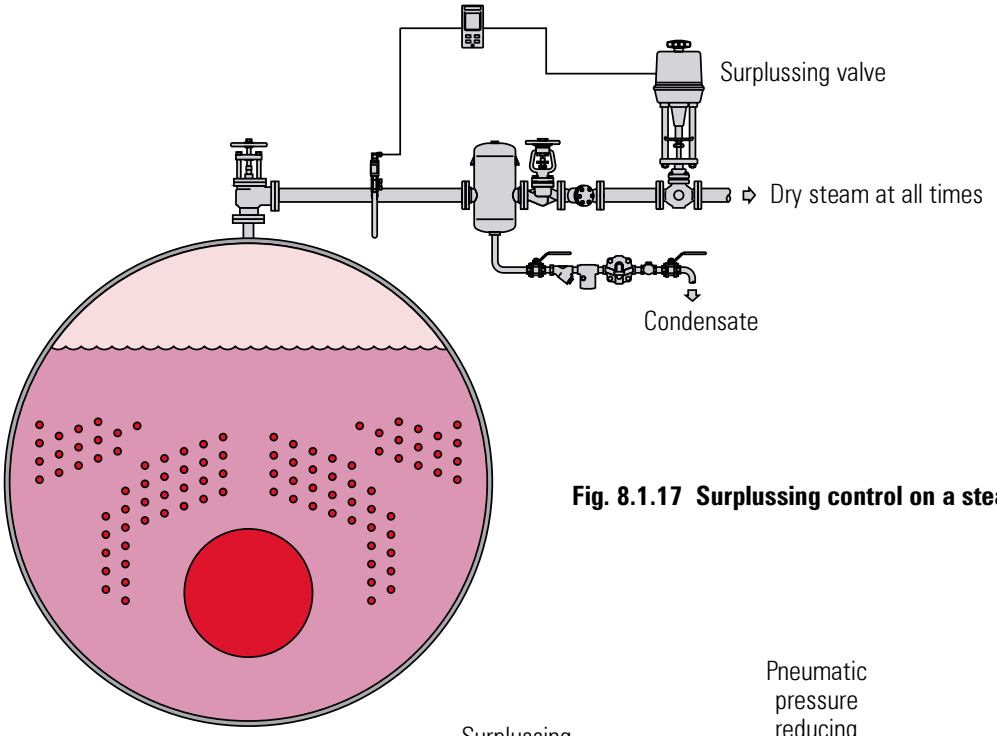


Fig. 8.1.17 Surplussing control on a steam boiler

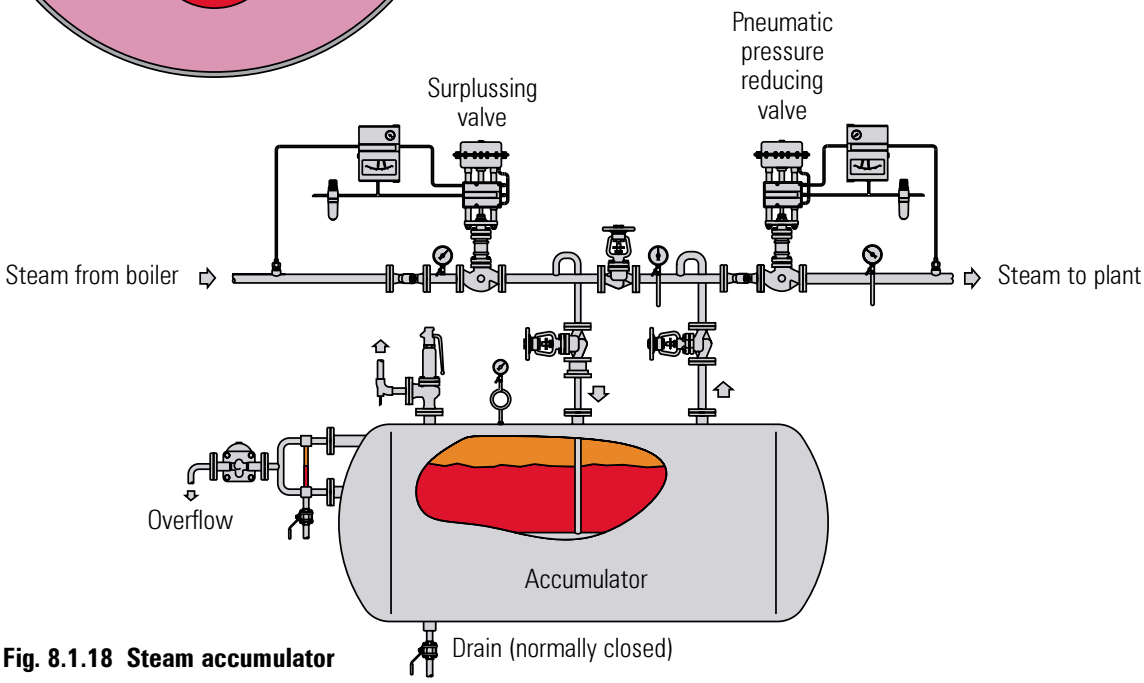


Fig. 8.1.18 Steam accumulator

Cascade control – Limiting pressure and temperature with one valve

Description

Where it is necessary to control two variables with one valve it is necessary to employ two separate controllers and sensors. It is always the case that the control valve accepts its control signal from the slave controller.

The slave controller is configured to accept two input signals, and its set point will change (within defined limits) depending on the electrical output signal from the master controller.

This form of control is very important where the pressure to the apparatus must be limited, despite the heat demand.

Application:

The steam heated plate heat exchanger shown in Figure 8.1.19 is heating water circulating in a secondary system. The heat exchanger has a maximum working pressure, consequently this is limited to that value in the slave controller.

In order to control the secondary water temperature, a master controller and temperature transmitter monitors the heat exchanger outflow temperature and sends a 4-20 mA signal to the slave controller, which is used to vary the slave set point, between pre-determined limits.

Points to note:

1. An adequate pressure margin must exist between the set pressure of the safety valve and the pressure limitation imposed by the controller.
2. The safety valve must not be used as a device to limit pressure in the heat exchanger; it must only be used as a safety device.

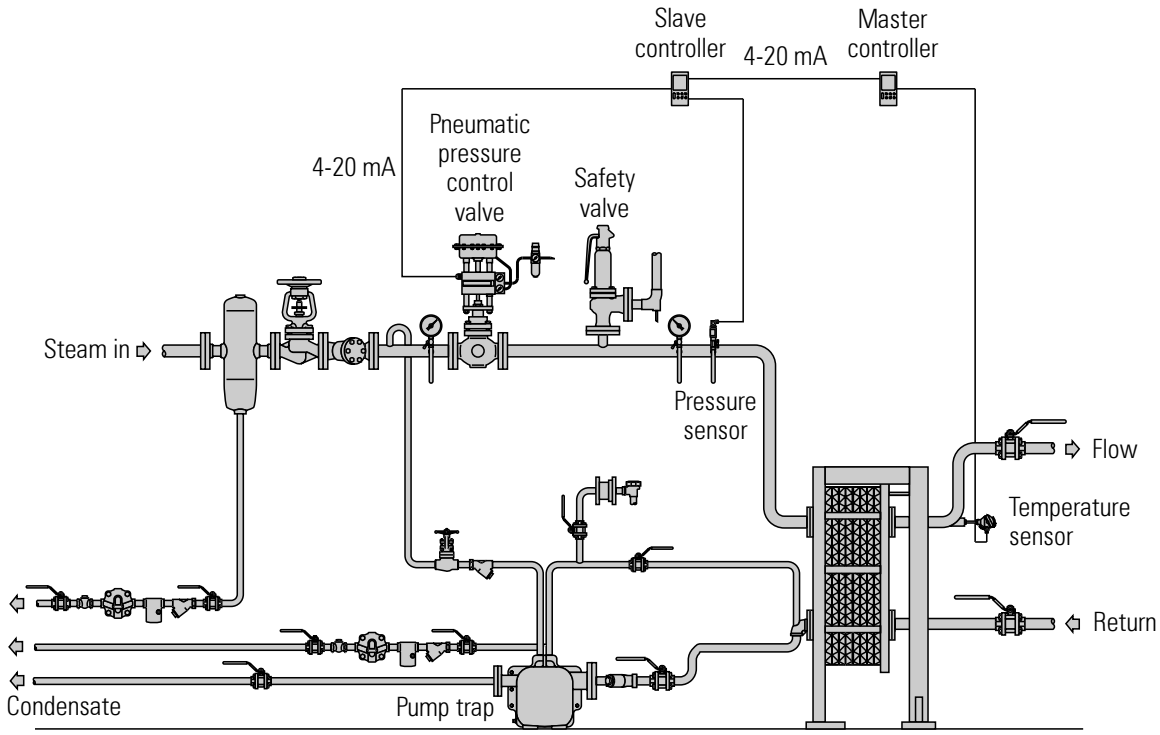


Fig. 8.1.19 Cascaded controllers on the steam supply to a heat exchanger

Cascade control – Combined pressure reduction and surplussing with one valve

Description

The objective is to reduce steam pressure but not at the expense of overloading the available supply capacity.

Application:

The upstream pipework is a high-pressure distribution pipe possibly from a distribution manifold or steam boiler supplying plant of a non-essential nature (Figure 8.1.20). Should the demand be higher than the supply capacity, the valve closes and throttles the steam flow, maintaining the pressure in the upstream pipework.

The master controller is set at the normal expected supply pressure. If the master detects a drop in upstream pressure below its set value (due to an increase in demand) it reduces the set point in the slave controller, in proportion to pre-determined limits.

The slave closes the valve until the steam demand falls to allow the upstream pressure to re-establish to the required value. When this is achieved, the set point of the slave controller is set at its original value.

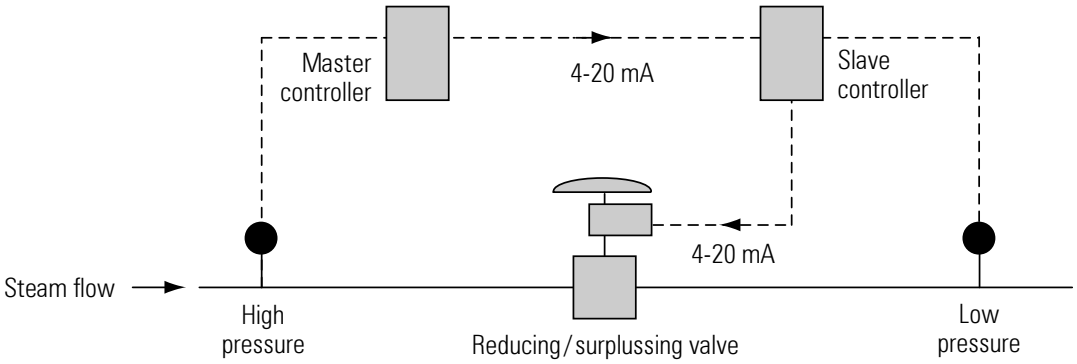


Fig. 8.1.20 General schematic arrangement of a reducing/surplussing valve

Typical settings

The output from the master controller is direct acting, that is, when the upstream pressure is at or above its proportional band, the master's output signal is maximum at 20 mA; when at the bottom of, or below the proportional band, the control signal is minimum at 4 mA.

When the control signal is 20 mA, the slave set point is the required downstream pressure; when the signal is 4 mA, the slave set point is at a pre-determined minimum.

Consider the 'normal' upstream pressure to be 10 bar g, and the maximum allowable downstream pressure to be 5 bar g. The minimum allowable upstream pressure is 8.5 bar g, which means that if this pressure is reached the valve is fully shut. The minimum reduced pressure is set at 4.6 bar g.

These conditions are recorded in Table 8.1.1

Table 8.1.1

P1 bar g	P1 and Master output signal	Master output signal mA	Master output signal and slave set point	Slave set point bar g
10.0	Output signal	20	Output signal	5.0
9.5		20		5.0
9.0		12		4.8
8.5		4		4.6
8.0	Upstream pressure	4	Slave set point	4.6

Cascade control – Limiting and controlling temperature with one valve

Description

The main objective is to limit and regulate the temperature to a particular process, where steam is the available heat source but it cannot be used directly to heat the final product for operational reasons.

Application:

A typical application is a dairy cream pasteuriser requiring a pasteurisation temperature of 50°C. Because of the low control temperature, if steam were applied directly to the pasteurisation heat exchanger, it is possible that the relatively large amount of heat in the steam would make control difficult, causing the system temperatures to oscillate, overheating and spoiling the cream.

To overcome this problem, the system in Figure 8.1.21 shows two heat exchangers. The pasteuriser is heated by hot water supplied from the primary steam heated heat exchanger.

However, even with this arrangement, if only the master controller operated the valve, a time lag would be introduced into the system, and poor control might again be the result.

Two controllers are therefore used, working in cascade, each receiving a 4-20 mA signal from their respective temperature transmitters.

The slave controller is used to control the final temperature of the product within clearly defined limits (perhaps between 49°C and 51°C). These values are altered by the master controller relative to the product temperature such that, if the product temperature increases, the slave set point reduces in proportion.

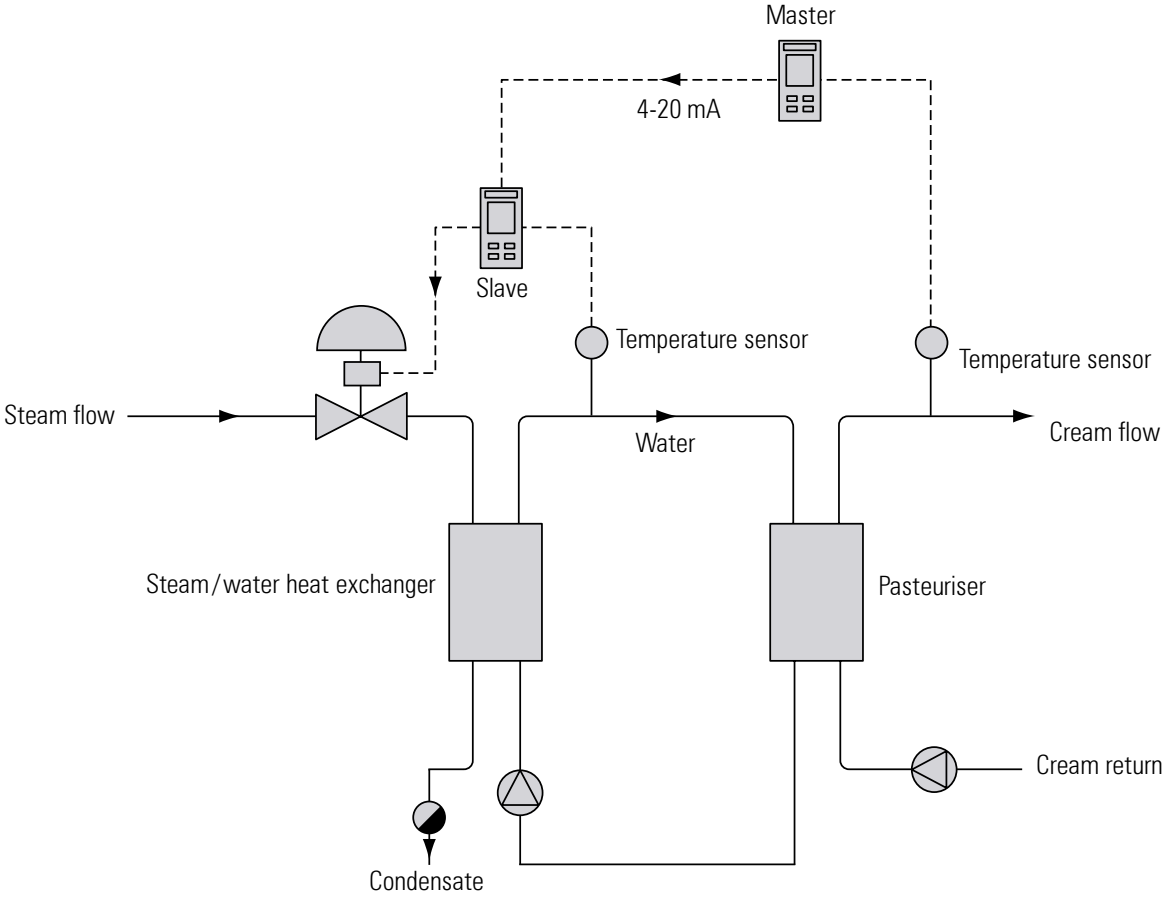


Fig. 8.1.21 Schematic diagram showing a pasteuriser control using the cascade principle

Questions

1. What is MAWP?

a | Maximum attenuated working pressure ☐

b | Minimum allowable working pressure ☐

c | Maximum allowable with pressure ☐

d | Maximum allowable working pressure ☐

2. One large and one small steam-heated heat exchanger have exactly the same heating duty. Which will operate at the lower pressure?

a | The smaller one ☐

b | The larger one ☐

c | They will both operate at the same pressure ☐

d | There is not enough information to answer the question ☐

3. Name one disadvantage of a direct acting pressure reducing valve

a | It only has proportional control ☐

b | It has proportional and integral control but no derivative control ☐

c | It operates in an on/off fashion ☐

d | An external power source is required for it to operate ☐

4. What type of pressure reducing station is required when the pressure ratio is greater than 10:1

a | A parallel station ☐

b | A pilot operated station ☐

c | A series station ☐

d | A surplussing station ☐

5. Why is cascade control used?

a | To control the flow of water over a weir ☐

b | When more than one input is necessary to secure good control ☐

c | When more than one valve is required to secure control ☐

d | When two pressures are being sampled ☐

6. Why is it sometimes necessary to reduce pressure?

a | To increase the pipe size ☐

b | Because the apparatus pressure is lower than the supply pressure ☐

c | Because the boiler pressure is too high ☐

d | To increase the steam flowrate ☐

Answers

1: d, 2: b, 3: a, 4: c, 5: b, 6: b

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Module 8.2

Temperature Control for Steam Applications

Temperature control for steam applications

There are a number of reasons for using automatic temperature controls for steam applications:

1. For some processes, it is necessary to control the product temperature to within fairly close limits to avoid the product or material being processed being spoilt.
2. Steam flashing from boiling tanks is a nuisance that not only produces unpleasant environmental conditions, but can also damage the fabric of the building. Automatic temperature controls can keep hot tanks just below boiling temperature.
3. Economy.
4. Quality and consistency of production.
5. Saving in manpower.
6. Comfort control, for space heating.
7. Safety.
8. To optimise rates of production in industrial processes.

The temperature control system employed should be matched to the system, and capable of responding to the changes in heat load. For example:

- On a low thermal mass system experiencing fast load changes, the control system needs to be able to react quickly.
- On massive systems, such as oil storage tanks, which experience slow changes in temperature, the control may only have to respond slowly.
- The temperature control system selected may need to be capable of coping with the start-up load without being too big, to provide accurate control under running conditions.

Direct operating, self-acting temperature control

Description

The direct operating, self-acting type of temperature control uses the expansion of liquid in a sensor and capillary to change the valve position.

Advantages:

1. Inexpensive.
2. Small.
3. Easy to install and commission.
4. One trade installation.
5. Very robust and extremely reliable.
6. Tolerant of imperfect steam conditions and of being oversized.
7. Self-acting principle means that no external power is required.
8. Simple to size and select.
9. Many options are available, such as different capillary lengths and temperature ranges.

Disadvantages:

1. The control is 'stand-alone', and cannot communicate with a remote controller or PLC (Programmable Logic Controller), although a high temperature cut-out may signal closure via a switch.
2. Limited sizes.
3. Limited pressure ratings.
4. Limited turndown.
5. Sensors tend to be much larger than the pneumatic and electronic equivalents and also much slower acting.

Applications:

Applications would include those with low and constant running flowrates:

1. Small jacketed pans.
2. Tracer lines.
3. Ironers.
4. Small tanks.
5. Acid baths.
6. Small storage calorifiers.
7. Small heater batteries.
8. Unit heaters.

Point to note:

The proportional band is influenced by the size of the valve.

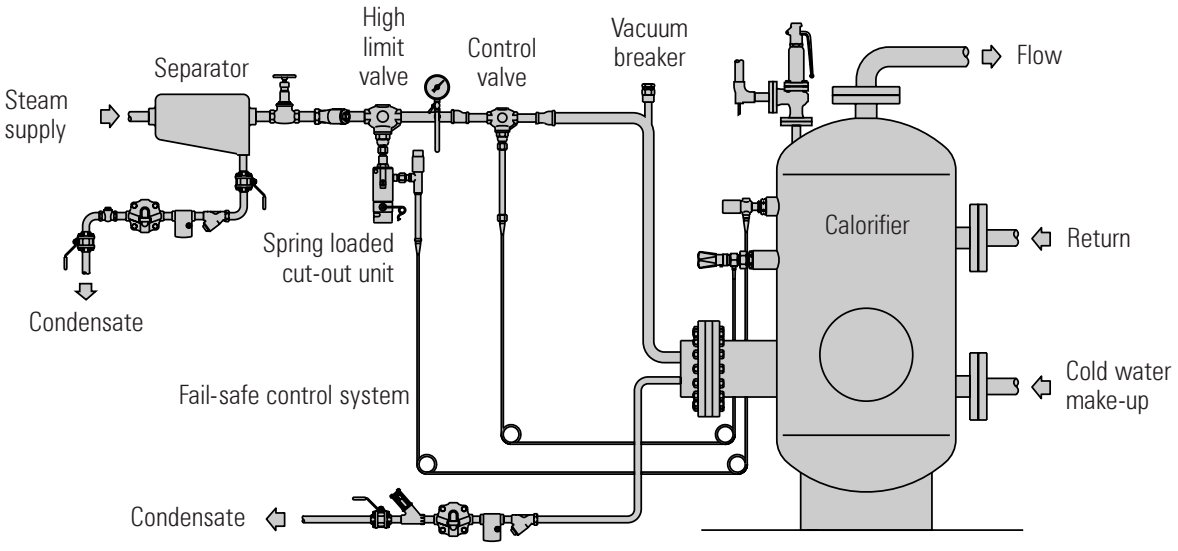


Fig. 8.2.1 General arrangement of a direct operating, self-acting temperature control system on a DHWS (Domestic Hot Water Services) storage calorifier

Pilot operated, self-acting temperature control

Description

The pilot operated self-acting type of temperature controller uses the expansion of liquid in a sensor and capillary to operate a pilot valve, which in turn changes the main valve position.

Advantages:

1. Easy to install and commission.
2. One trade installation.
3. Very robust.
4. Self-acting principle means that no external power is required.
5. Simple to size and select.
6. Remote adjustment (option).
7. Can be switched on and off (option).
8. Dual set point (option).

Disadvantages:

1. The control is 'stand-alone', and cannot communicate with a PLC.
2. Small clearances within the valve body mean that steam should be clean and dry to ensure longevity, but this can easily be achieved by fitting a separator and strainer before the valve.
3. Proportional only control, however, the proportional offset is much smaller than for direct operating, self-acting controls.

Applications:

1. Jacketed pans.
2. Tracer lines.
3. Tanks.
4. Acid baths.
5. Hot water storage calorifiers.
6. Heater batteries.
7. Unit heaters.

Points to note:

1. The temperature ranges of controllers tend to be narrower than direct operating, self-acting controls.
2. Installation must include a strainer and separator.

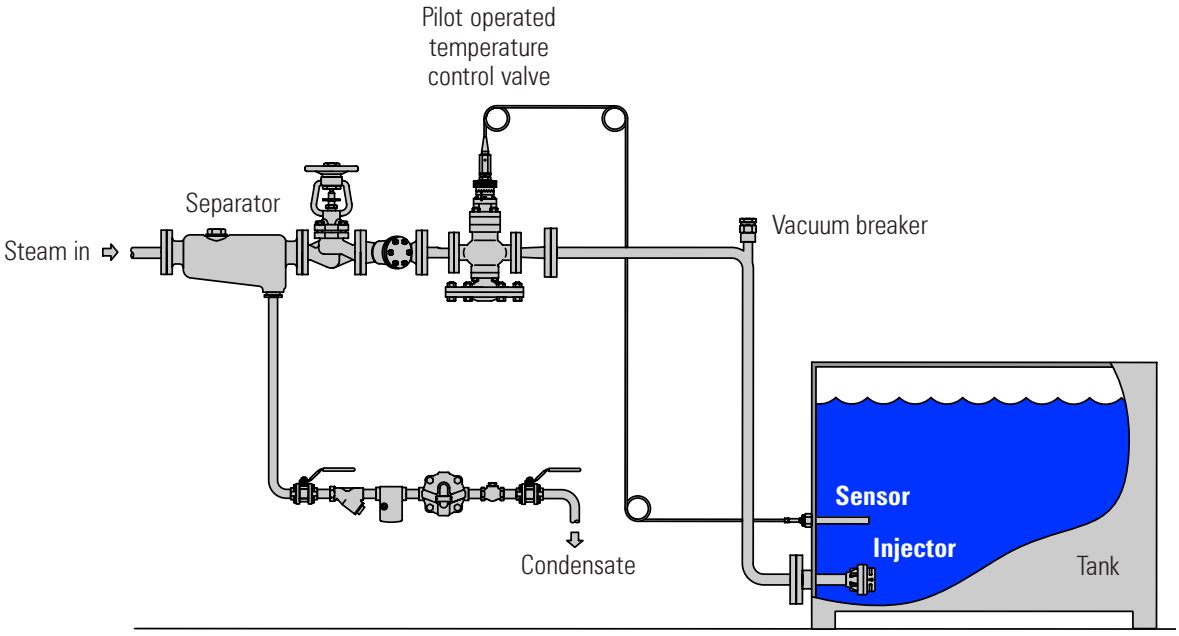


Fig. 8.2.2 General arrangement of a pilot operated, self-acting temperature control injecting steam into a tank

Pneumatic temperature control

Description

These control systems may include:

- P + I + D functions to improve accuracy under varying load conditions.
- Set point(s), which may be remotely adjusted.

Advantages:

1. Very accurate and flexible.
2. No limit on valve size within the limits of the valve range.
3. Excellent turndown ratio.
4. Suitable for hazardous environments.
5. No electrical supply required.
6. Fast operation means they respond well to rapid changes in demand.
7. Very powerful, and can cope with high differential pressures.

Disadvantages:

1. More expensive than direct operating controls.
2. More complex than direct operating controls.

Applications:

1. Which need accurate and consistent temperature control.
2. With variable and high flowrates, and/or variable upstream pressure.
3. Which require intrinsic safety.

Points to note:

1. A clean, dry air supply is required
2. A valve positioner is generally required except for the smallest and simplest of applications. Air is continually vented from the positioner and controller, and there is a need to ensure that this quiescent air flow is acceptable to the surroundings.
3. A skilled workforce is required to install the equipment, and instrument personnel for calibration and commissioning.
4. The control is 'stand-alone', and cannot directly communicate with a PLC.
5. The failure mode must always be considered. For example, 'spring-to-close' on air failure is normal on steam heating systems, 'spring-to-open' is normal on cooling systems.

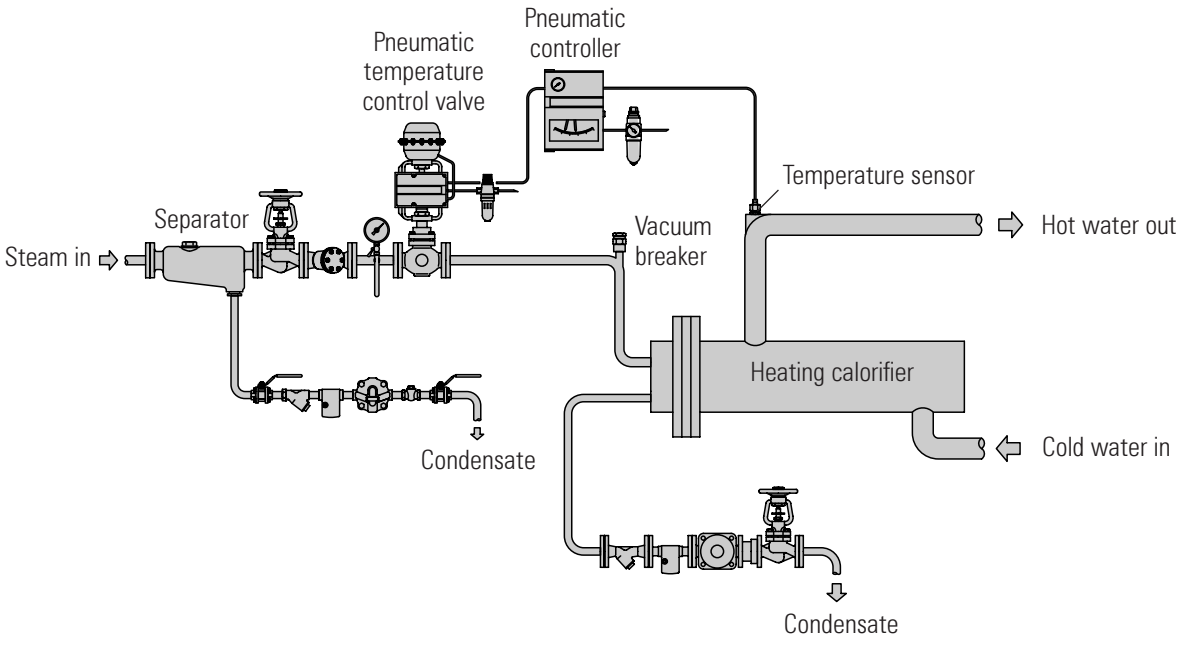


Fig. 8.2.3 General arrangement of a pneumatic temperature control system on a heating calorifier

Electropneumatic temperature control

Description

These control systems may include:

- P + I + D functions to improve accuracy under varying load conditions.
- Set point(s) may be remotely adjusted, with the possibility of ramps between set points.

Advantages:

1. Very accurate and flexible.
2. Remote adjustment and read-out.
3. No limit on valve size within the limits of the valve range.
4. Excellent turndown ratio.
5. Fast operation means they respond well to rapid changes in demand.
6. Very powerful, and can cope with high differential pressures.

Disadvantages:

1. More expensive than self-acting or pneumatic controls.
2. More complex than self-acting or pneumatic controls.
3. Electrical supply required.

Applications:

1. Which need accurate and consistent temperature control.
2. With variable and high flowrates, and/or variable upstream pressure.

Points to note:

1. A clean, dry air supply is required.
2. A skilled workforce is required to install the equipment, electrical personnel are required for power supplies, and instrument personnel to calibrate and commission.
3. Can be part of a sophisticated control system involving PLCs, chart recorders and SCADA systems.
4. The failure mode must always be considered. For example, 'spring-to-close' on air failure is normal on steam heating systems, 'spring-to-open' is normal on cooling systems.
5. Probably the most common control system - it has the sophistication of electronics with the pace / power of pneumatics.

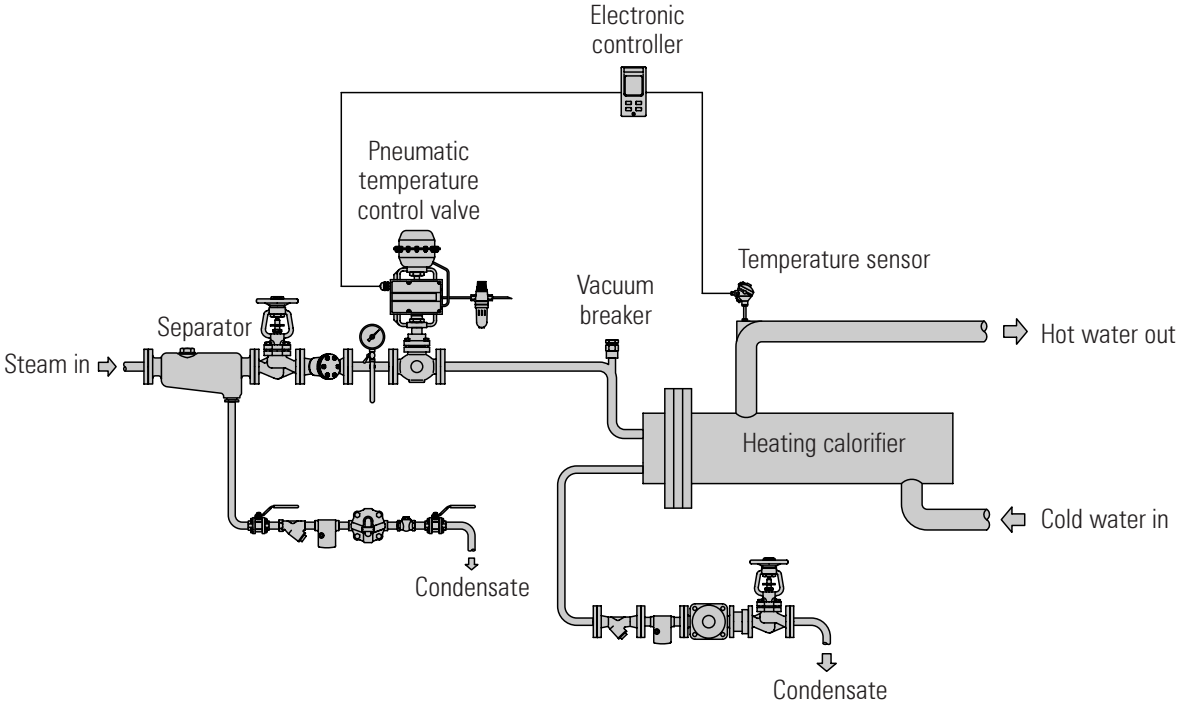


Fig. 8.2.4 General arrangement of an electropneumatic temperature control system on a heating calorifier

Electric temperature control

Description

These control systems may include:

- P + I + D functions to improve accuracy under varying load conditions.
- Set point(s), which may be remotely adjusted.

Advantages:

1. Both controller and valve actuator can communicate with a PLC.
2. No compressed air supply is required.

Disadvantage:

The relatively slow actuator speed means they are only suitable for applications where the load changes slowly.

Application:

Space heating of large volumes. For example; warehouses, workshops, aircraft hangars, etc.

Points to note:

1. Safety: If electrical power is lost the valve position will not change unless a spring return actuator is used.
2. Spring return actuators are expensive, bulky and can only shut off against a limited pressure.

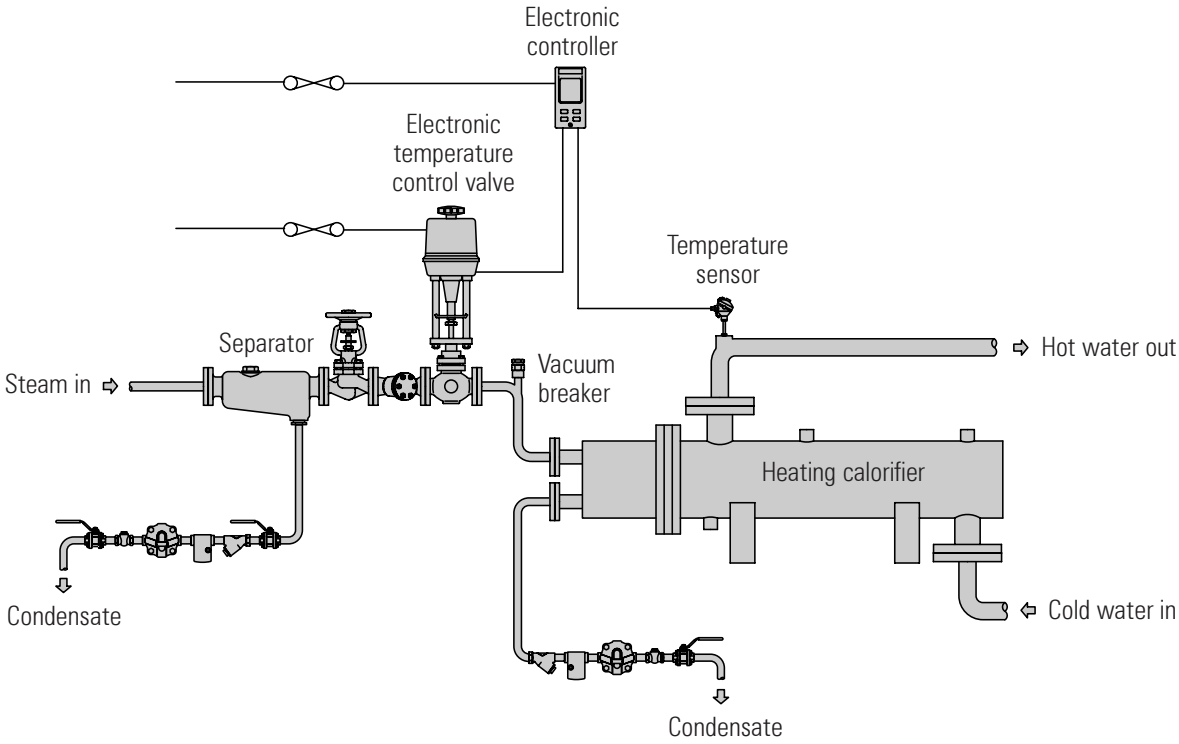


Fig. 8.2.5 General arrangement of an electric temperature control system on a heating calorifier

Temperature control (other possibilities) - Parallel temperature control station

Description

An arrangement, as shown in Figure 8.2.6, can be used where the ratio between maximum and minimum flowrates (the flowrate turndown) is greater than the maximum allowable for the individual temperature control valve.

For example, if a specific application has to be brought up to operating temperature very quickly, but the running load is small, and plant conditions dictate that self-acting controls must be used.

To satisfy the application:

1. A valve and controller, which could satisfy the running load, would be selected first, and set to the required temperature.
2. A second valve and controller, capable of supplying the additional load for warm-up would be selected, and set to a couple of degrees lower than the 'running load' valve. This valve is likely to be larger than the running load valve.

With this configuration:

1. When the process is cold, both control valves are open, allowing sufficient steam to pass to raise the product temperature within the required time period.
2. As the process approaches the required temperature, the 'warm-up' valve will modulate to closed, leaving the 'running load' valve to modulate and maintain the temperature.

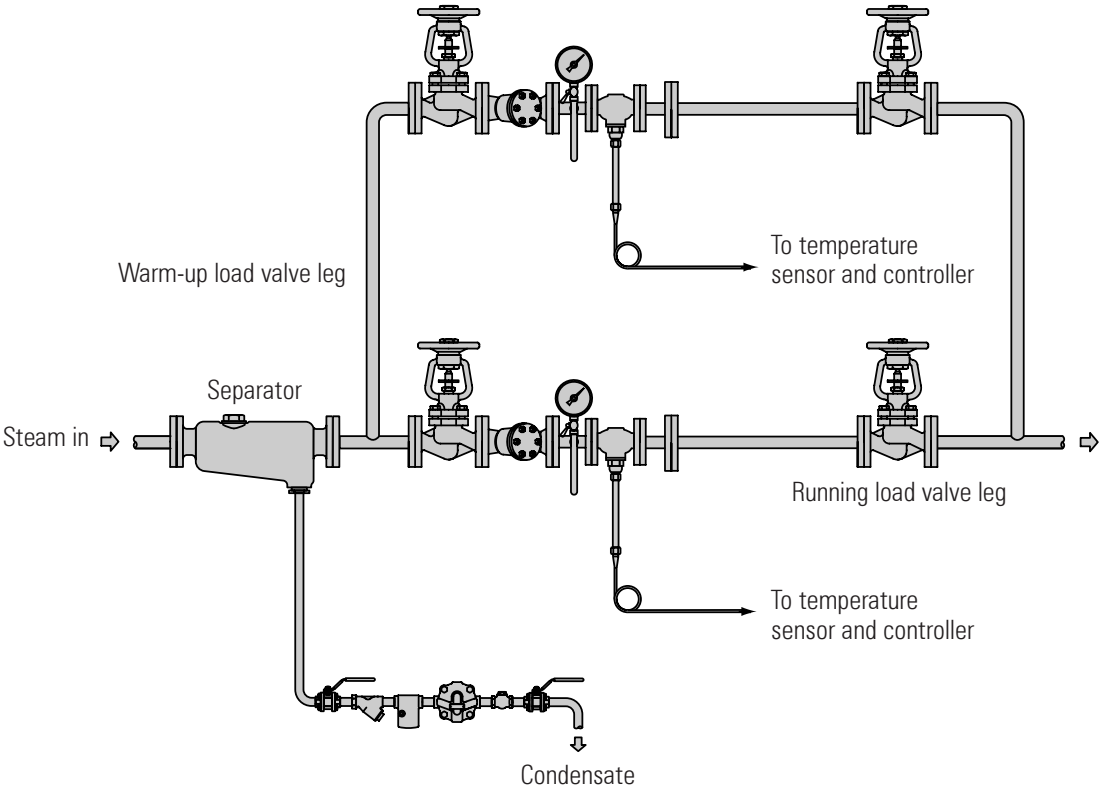


Fig. 8.2.6 General arrangement of a parallel temperature control station

High temperature fail safe control

Description

There are many applications where a totally independent high limit cut-out device is either desirable, or even a legal requirement.

Options:

1. A self-acting control, where the expansion of the fluid releases a compressed spring in a cut-out unit, and snaps the isolating valve shut if the preset high limit temperature is exceeded.

This particular type of self-acting control has additional advantages:

a. It can incorporate a microswitch for remote indication of operation.

b. It is best if it has to be reset manually, requiring personnel to visit the application and ascertain what caused the problem.

2. Spring-to-close electrical actuator where an overtemperature signal will interrupt the electrical supply and the valve will close. This may be accompanied by an alarm.

3. Spring-to-close pneumatic actuators where an overtemperature signal will cause the operating air to be released from the actuator. This may be accompanied by an alarm.

Application:

Domestic hot water services (DHWS) supplying general purpose hot water to users such as hospitals, prisons and schools.

Points to note:

1. There may be a legal requirement for the high temperature cut-out to be totally independent. This will mean that the high temperature cut-out device must operate on a separate valve.

2. Generally, the high temperature cut-out valve will be pipeline size, since a low pressure drop is required across the valve when it is open.

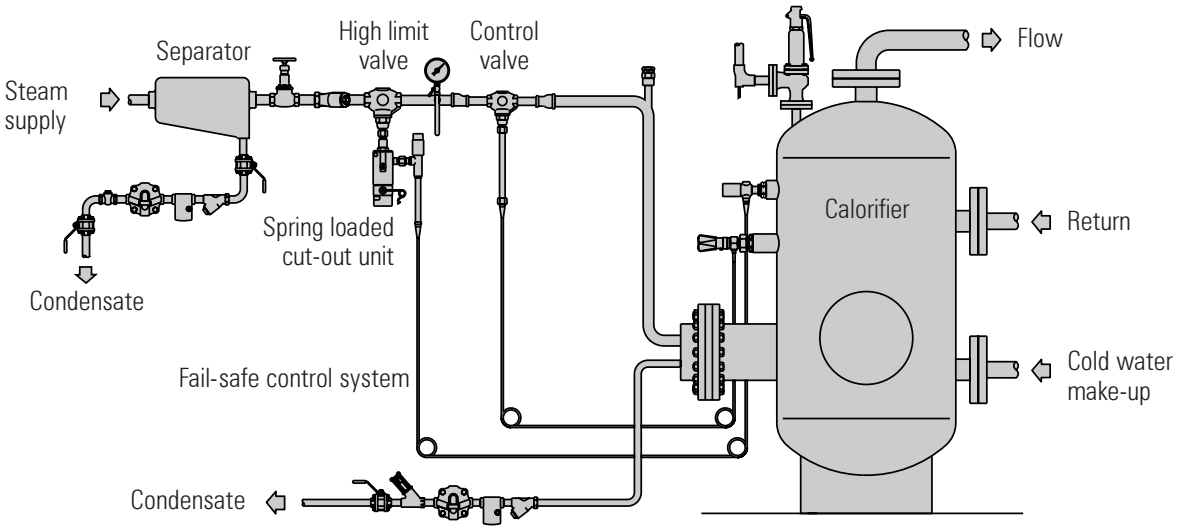


Fig. 8.2.7 General arrangement of a high temperature cut-out on a DHWS storage calorifier

Questions

1. Name one disadvantage of direct operating temperature control

a | It is relatively inexpensive ☐

b | The sensors tend to be large compared to EL (electronic) and PN (pneumatic) sensors ☐

c | Systems are difficult to size and select ☐

d | Systems are difficult to install and commission ☐

2. A temperature control application in a hazardous area, and which has low thermal mass, is subject to fast load changes and periods of inoperation. Which would be the best control solution from the following?

a | A direct operating temperature control system ☐

b | A pilot operated self-acting temperature control system ☐

c | A pneumatic temperature control system ☐

d | An electric temperature control system ☐

3. In Figure 8.2.6, the warm-up valve is shown in the upper leg of the parallel supply system. Is this logical?

a | Yes, otherwise condensate would tend to collect in the warm-up leg during low loads, when the warm-up valve would be shut ☐

b | Yes, it makes maintenance easier ☐

c | No, either leg is acceptable ☐

d | Yes, the warm-up valve needs more installation space ☐

4. Is the fail-safe self-acting high limit temperature cut-out only suitable for DHWS storage calorifiers?

a | Yes ☐

b | It is suitable for any application requiring high limit temperature control ☐

5. In Figure 8.2.5, a shell and tube heating calorifier uses electrical control. Is this really suitable for this type of application?

a | No, it was the only example drawing available ☐

b | No, the valve would not react quickly enough ☐

c | No, an electropneumatic system should always be chosen for this type of application, especially when steam is the energy provider ☐

d | Yes, because changes in load will occur slowly ☐

Answers

1: b, 2: c, 3: a, 4: b, 5: d

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Module 8.3

Level and Flow Control Applications

Level Control Applications

The control of liquid levels, for example in a process tank, is an important function. An example would be a hot water tank where water is removed, perhaps for washing down, and the level needs to be restored ready for the next wash cycle.

Control of water level and alarms for steam boilers is specifically excluded from this Module, and the reader is referred to Block 3 (The Boiler House), which deals with the subject in depth.

Many different types of level control systems are used in industry, covering a wide range of processes. Some processes will be concerned with media other than liquids, such as dry powders and chemical feedstock. The range of media is so wide that no single instrument is suitable for all applications.

Many systems are available to serve this wide range of applications. The following list is not exhaustive but, in most cases, the final control signal will be used to operate pumps or valves appropriate to the application:

- **Float operated types –** a float rises and falls according to the change in liquid level and operates switches at predetermined points in the range.
- **Solid probe types –** these measure conductivity or capacitance and are discussed in more detail in the following pages.
- **Steel rope capacitance types –** a flexible steel rope is suspended in the liquid, and the change in capacitance is measured relative to the change in water level.
- **Ultrasonic types –** a high frequency acoustic pulse is directed down from a transducer to the surface of the medium being measured and, by knowing the temperature and speed of sound in air, the time it takes for the pulse to rebound to the sensor is used to determine the level.
- **Microwave radar types –** similar in principle to the ultrasonic type but using high frequency electromagnetic energy instead of acoustic energy.
- **Hydrostatic types –** a pressure transmitter is used to measure the pressure difference between the confined hydrostatic pressure of the liquid head above the sensor and the outside atmospheric pressure. Changes in pressure are converted into a 4-20 mA output signal relative to the head difference.
- **Differential pressure types –** similar to hydrostatic but used where the application being measured is subjected to dynamic pressure in addition to static pressure. They are capable of measuring small changes in pressure in relation to the output signal range. Typical applications might be to measure the level of water in a boiler steam drum, or the level of condensate in a reboiler condensate pocket.
- **Magnetic types –** a float or cone is able to rise and fall along a stainless steel probe held in the tank fluid being measured. The float can interact magnetically with switches on the outside of the tank which send back information to the controller.
- **Torsion types –** a moving float spindle produces a change in torsion, measured by a torsion transducer.

It is important that the level control system is correct for the application, and that expert advice is sought from the manufacturer before selection.

It is not within the scope of this Module to discuss the pros and cons and potential applications of all the above control types, as the types of level control systems usually employed in the steam and condensate loop and its associated applications are float and solid probe types. The operation of float types is fairly self-explanatory, but conductivity and capacitance probes may require some explanation. Because of this, this section will mainly focus on conductivity and capacitance probe-type level controls.

Methods of achieving level control

There are three main methods of achieving level control:

- Non-adjustable on/off level control.
- Adjustable on/off level control.
- Modulating level control.

Non-adjustable on/off level control (Figure 8.3.1)

The final control element may be a pump which is switched on/off or a valve which is opened/closed.

Two main types of on/off level control systems are usually encountered; float operated types and types using conductivity probes. Float type level controls either rely upon the direct movement of a control valve, or upon electrical switches being operated by a float moving on the surface of the liquid. Conductivity probes (see Figure 8.3.1) may have several probe tips; the control points being located where the separate tips have been cut to different lengths.

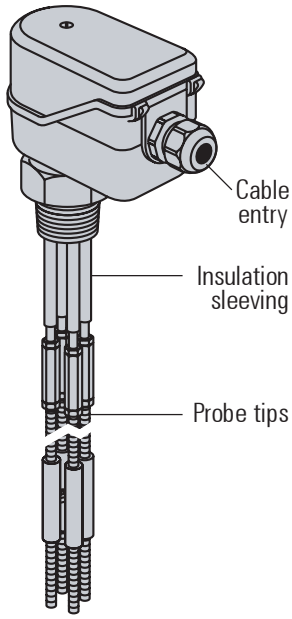


Fig. 8.3.1 A four tip level probe

Adjustable on/off level control (Figure 8.3.2)

Again, the final control element may be a pump which is switched on/off or a valve which is opened/closed.

One method used to adjust the control points is that of a capacitance probe (see Figure 8.3.2). The probe will monitor the level, with control points adjusted by the controller. Capacitance probes are not cut to length to achieve the required level and, of course, the whole probe length must be sufficient for the complete control range.

Modulating level control (Figure 8.3.2)

The final control element may be a valve that is adjusted to a point between fully open and fully closed, as a function of the level being monitored. Modulating level control cannot be achieved using a conductivity probe. Capacitance probes are ideal for this purpose (see Figure 8.3.2).

In systems of this type, the pump can run continuously, and the valve will permit appropriate quantities of liquid to pass. Alternatively, the final control element may be a variable speed drive on a pump. The speed of the drive may be adjusted over a selected range.

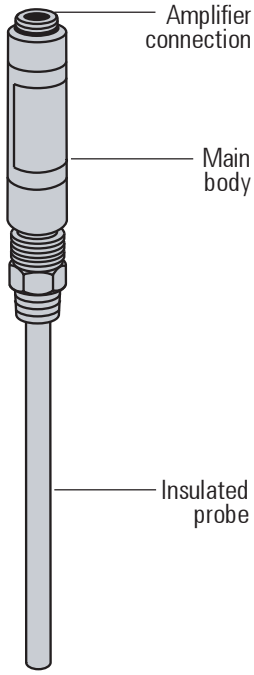


Fig. 8.3.2 A capacitance level probe

Alarms – are often required to warn of either:

- A high alarm where there is a danger of the tank overflowing and hot liquid being spilled, with the attendant danger to personnel.
- A low alarm where there is a danger of the tank water level becoming too low, with the potential to damage a pump drawing from the tank, or running out of liquid for the process.

Installation of floats and probes in turbulent conditions

In some tanks and vessels, turbulent conditions may exist, which can result in erratic and unrepresentative signals. If such conditions are likely to (or already) exist, it is recommended that floats or probes be installed within protection tubes. These have a dampening effect on the water level being sensed. The rest of this Module concerns itself with probes rather than floats for level control applications.

Non-adjustable on/off level control

Description

Non-adjustable on/off level control uses a conductivity probe connected to an electronic controller. The probe typically has three or four tips, each of which is cut to length during installation to achieve the required switching or alarm level (see Figure 8.3.3).

- When the tip of the probe is immersed in liquid it uses the relatively high conductivity of the water to complete an electrical circuit via the tank metalwork and the controller.
- When the water level drops below the tip, the circuit resistance increases considerably, indicating to the controller that the tip is not immersed in the liquid.
- In the case of a simple 'pumping in' system with on/off level control:
  - The valve is opened when the tank water level falls below the end of a tip.
  - The valve is closed when the water level rises to contact another tip.
  - Other tips may be used to activate low or high alarms.

Advantage:

A simple but accurate and relatively inexpensive method of level control.

Applications:

The system can be used for liquids with conductivities of 1 µS/cm or more, and is suitable for condensate tanks, feedwater tanks and process vats or vessels. Where the conductivity falls below this level it is recommended that capacitance based level controls are used.

Point to note:

If the tank is constructed from a non-conductive material, the electrical circuit may be achieved via another probe tip.

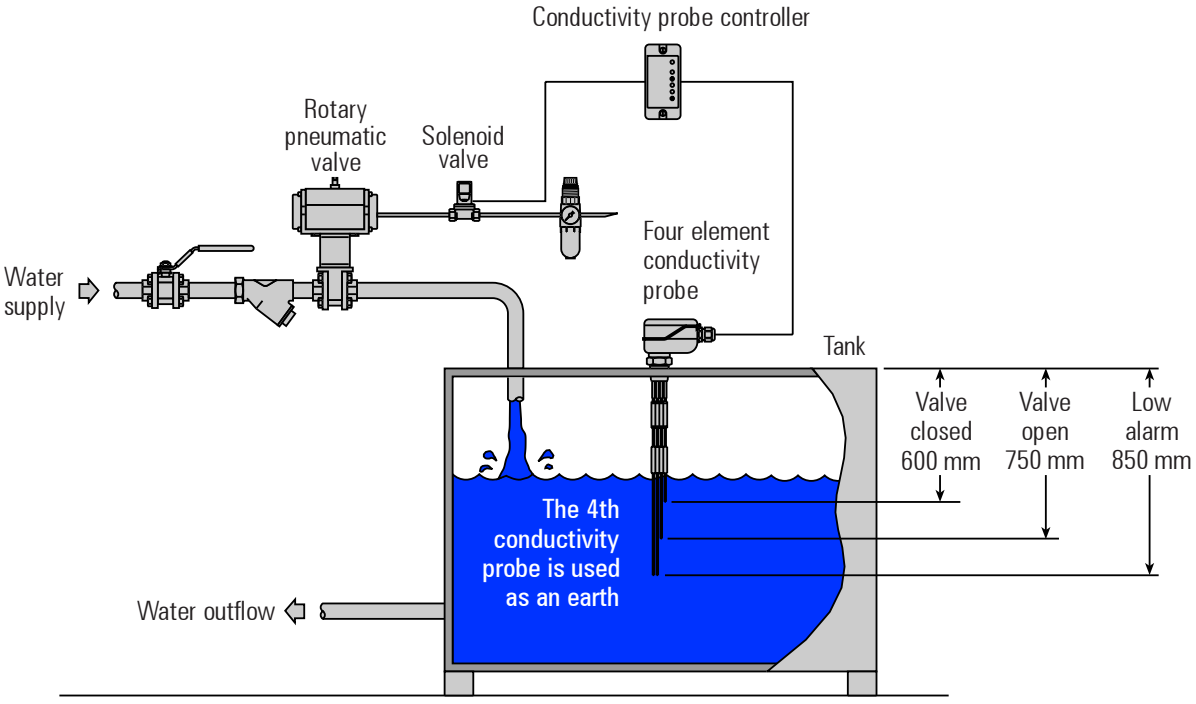


Fig. 8.3.3 General arrangement of a non-adjustable on/off level control system for a tank

Adjustable on/off level control

Description:

An adjustable on/off level control system consists of a controller and a capacitance probe (see Figure 8.3.4), and provides:

- Valve open/closed control plus one alarm point.
- Alternatively two alarms - high and low.

The levels at which the valve operates can be adjusted through the controller functions.

Advantage:

Adjustable on/off level control allows the level settings to be altered without shutting down the process.

Disadvantage:

More expensive than non-adjustable on/off control.

Application:

Can be used for most liquids, including those with low conductivities.

Point to note:

Can be used in situations where the liquid surface is turbulent, and the in-built electronics can be adjusted to prevent rapid on/off cycling of the pump (or valve).

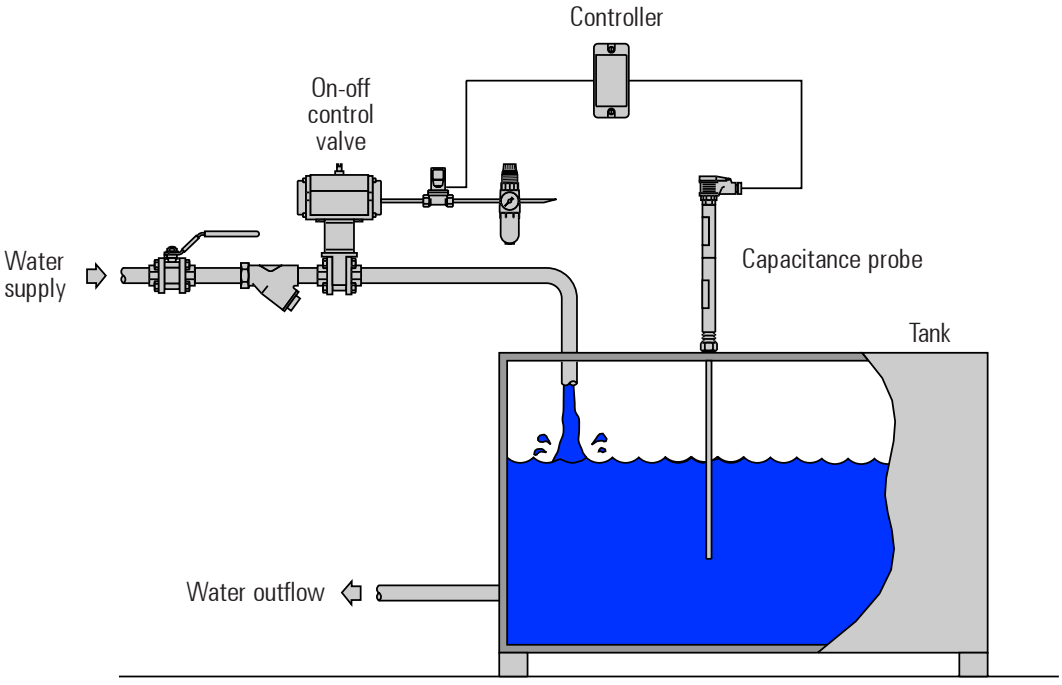


Fig. 8.3.4 General arrangement of an adjustable on/off level control system for a tank

Modulating level control

Description

A modulating level control system consists of a capacitance probe and appropriate controller, which provides a modulating output signal, typically 4-20 mA. Refer to Figure 8.3.5. This output signal may be used to affect a variety of devices including:

- Modulating a control valve.
- Operating a variable speed pump drive.

Advantages:

1. Because the probe and controller only provide a signal to which other devices respond, rather than providing the power to operate a device, there is no limit on the size of the application.
2. Steady control of level within the tank.

Disadvantages:

1. More expensive than a conductivity probe system.
2. More complex than a conductivity probe system.
3. Supply system must be permanently charged.
4. Less suitable for 'stand-by' operation.
5. Possibly greater electricity consumption.

Point to note:

To protect the supply pump from overheating when pumping against a closed modulating valve, a re-circulation or spill back line is provided to ensure a minimum flowrate through the pump (neither shown in Figure 8.3.5).

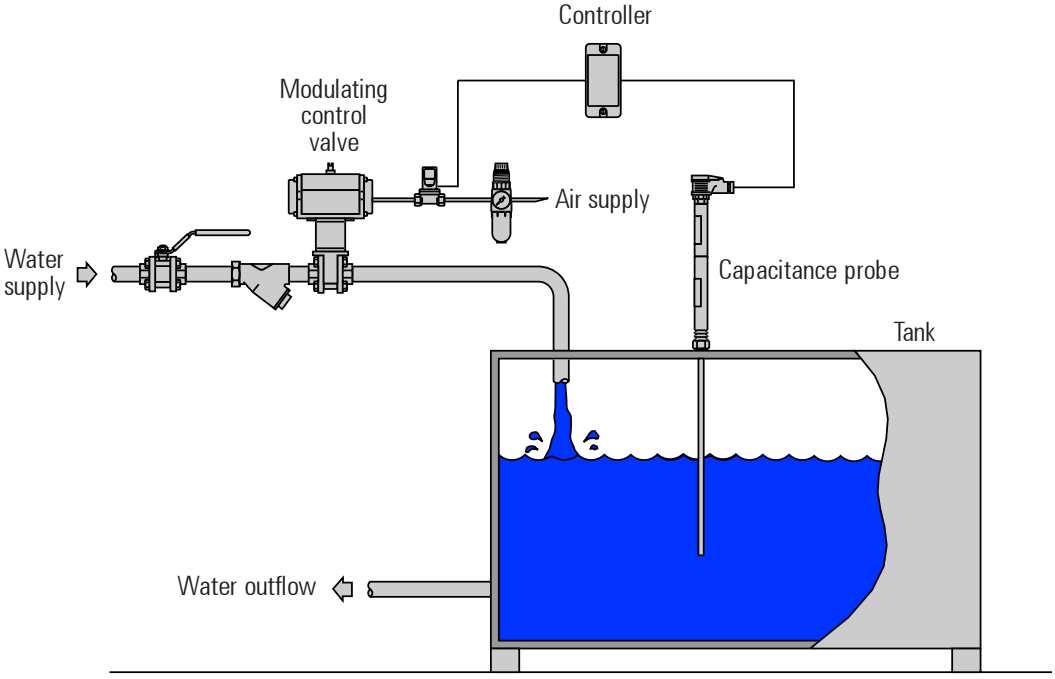


Fig. 8.3.5 General arrangement of a modulating control system maintaining the level in a tank

Steam flow control applications

The control of steam flow is less common than pressure and temperature control, but it is used in applications where the control of pressure or temperature is not possible or not appropriate to achieving the process objectives. The following sections give more information on measuring and controlling the flow of steam.

Flow control system

Typical applications:

1. Feed-forward systems on boiler plant, where the rate of steam flow from the boiler will influence other control points, for example: feedwater make-up rate, and burner firing rate.
2. Rehydration processes, where a measured quantity of steam (water) is injected into a product, which has been dried for transportation or storage. Examples of this can be found in the tobacco, coffee and animal feedstuff industries.
3. Batch processes, where it is known from experience that a measured quantity of steam will produce the desired result on the product.

The selection and application of components used to control flowrate require careful thought.

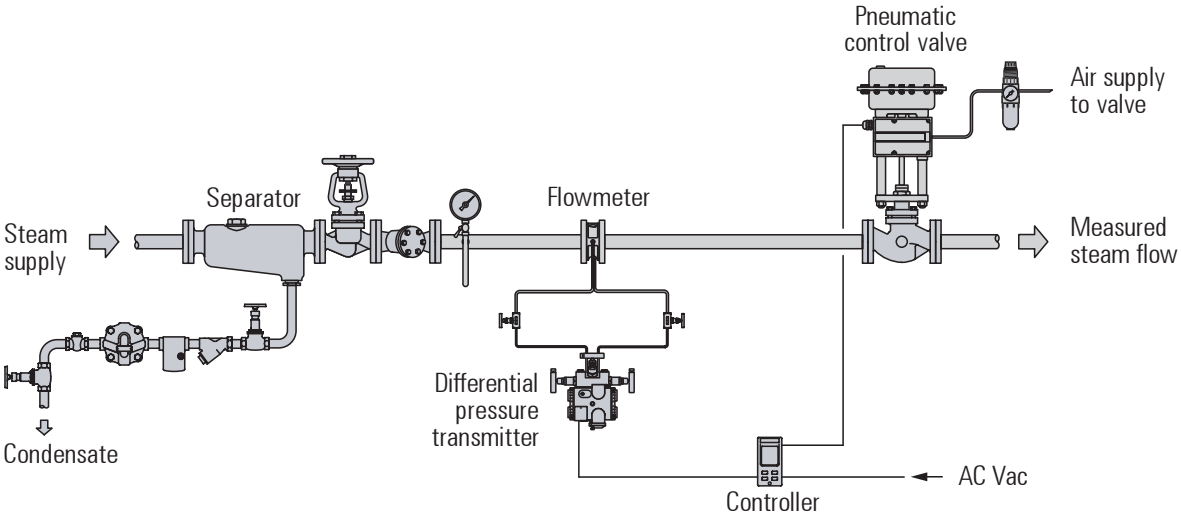


Fig. 8.3.6 General arrangement of a flow control system

The flowmeter (pipeline transducer)

The flowmeter is a pipeline transducer, which converts flow into a measurable signal. The most commonly used pipeline transducer is likely to relate flow to differential pressure. This pressure signal is received by another transducer (typically a standard DP (differential pressure) transmitter) converting differential pressure into an electrical signal. Some pipeline transducers are capable of converting flowrate directly to an electrical signal without the need for a DP transmitter.

Figure 8.3.6 shows a variable area flowmeter and standard DP transmitter relating differential pressure measured across the flowmeter into a 4 - 20 mA electrical signal. The standard DP transmitter is calibrated to operate at a certain upstream pressure; if this pressure changes, the output signal will not represent the flow accurately. One way to overcome this problem is to provide a pressure (or temperature) signal if the medium is saturated steam, or a pressure and temperature signal if the fluid is superheated steam, as explained in the next Section. Another way is to use a mass flow DP transmitter, which automatically compensates for pressure changes.

The possible need for a computer

If steam is the fluid in the pipeline, then other temperature and/or pressure sensors may be necessary to provide signals to compensate for variations in the supply pressure, as shown in Figure 8.3.7.

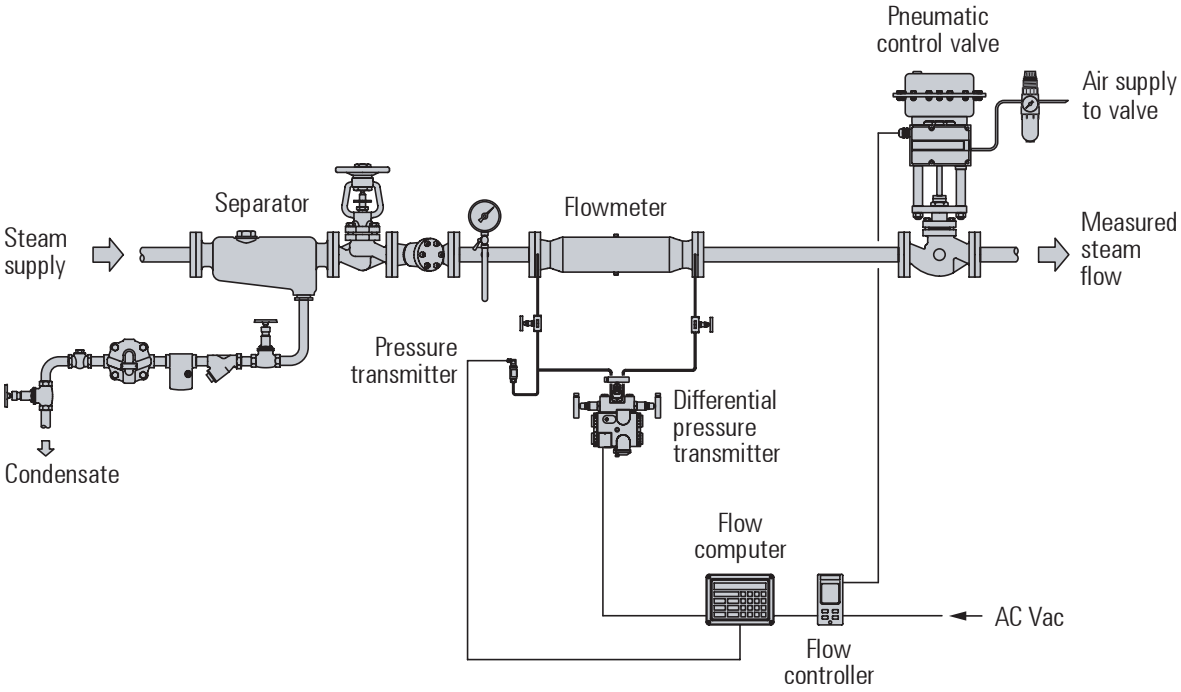


Fig. 8.3.7 General arrangement of a flow control system

Multiple inputs will mean that an additional flow computer (or PLC) containing a set of electronic steam tables must process the signals from each of these flow, pressure and temperature sensors to allow accurate measurement of saturated or superheated steam.

If a flow computer is not readily available to compensate for changes in upstream pressure, it may be possible to provide a constant pressure; perhaps by using an upstream control valve, to give stable and accurate pressure control (not shown in Figure 8.3.7).

The purpose of this pressure control valve is to provide a stable (rather than reduced) pressure, but it will inherently introduce a pressure drop to the supply pipe.

A separator placed before any steam flowmetering station to protect the flowmeter from wet steam will also protect the pressure control valve from wiredrawing.

Using a mass flow DP transmitter

By using a mass flow DP transmitter instead of a standard DP transmitter, the need for a computer to provide accurate measurement is not required, as shown in Figure 8.3.8.

This is because the mass flow transmitter carries its own set of steam tables and can compensate for any changes in saturated steam supply pressure.

However, a computer can still be used, if other important flowmetering information is required, such as, the times of maximum or minimum load, or is there is a need to integrate flow over a certain time period.

A controller is still required if flowrate is to be controlled, whichever system is used.

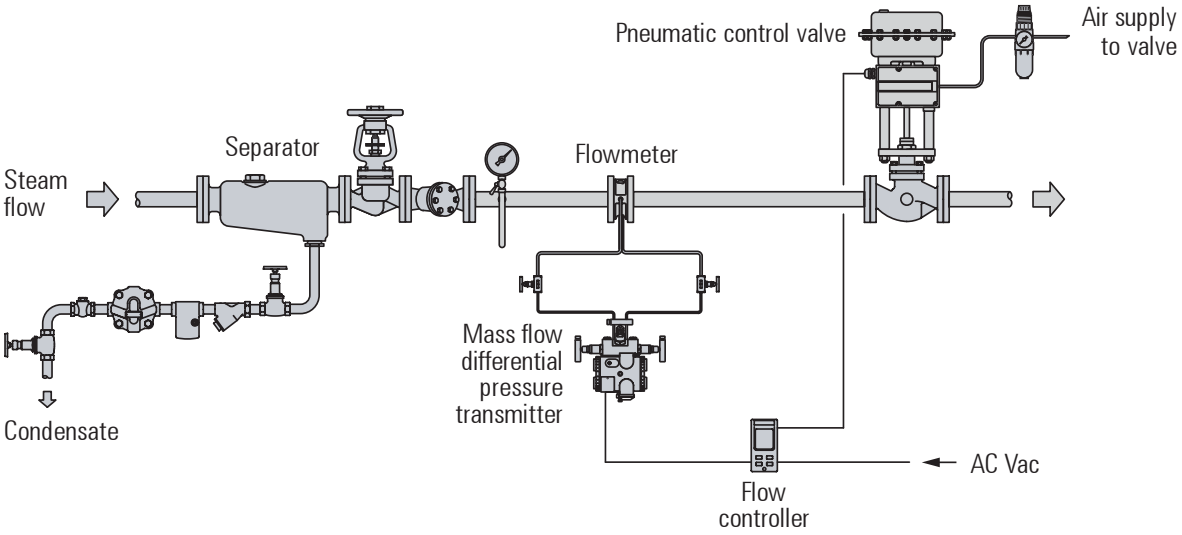


Fig. 8.3.8 General arrangement of a flow control system

The controller

Even if the output signal from the DP transmitter or computer is of a type that the control valve actuator can accept, a controller will still be required (as for any other type of control system) for the following reasons:

1. The output signal from certain flowmeters/computers has a long time repeat interval (approximately 3 seconds), which will give enough information for a chart recorder to operate successfully, but may not offer enough response for a control valve. This means that if the controller or PLC to which the transmitter signal is being supplied operates at higher speeds, then the process can become unstable.
2. PID functions are not available without a controller.
3. Selecting a set point would not be possible without a controller.
4. The signal needs calibrating to the valve travel - the effects of using either a greatly oversized or undersized valve without calibration, can easily cause problems.

Summary

It is usually better to install the flowmetering device upstream of the flow control valve. The higher pressure will minimise its size and allow it to be more cost effective. It is also likely that the flowmeter will be subjected to a more constant steam pressure (and density) and will be less affected by turbulence from the downstream flow control valve.

In some cases, the application may be required to control at a constant flowrate. This means that features, such as high turndown ratios, are not important, and orifice plate flowmeters are appropriate.

If the flowrate is to be varied by large amounts, however, then 'turndown' becomes an issue that must be considered.

The subject of Flowmetering is discussed in greater depth in Block 4.

Questions

1. Condensate has a conductivity of 0.1 µs/cm. Name the best choice of solid probe to give on/off level control for this application.

a| A single tip conductivity probe ☐
b| Two single tip conductivity probes ☐
c| A four tip conductivity probe ☐
d| A capacitance probe ☐

2. Name an advantage of modulating control over on/off control.

a| It tends to control at a steady level ☐
b| It allows the level settings to be altered without removing the probe ☐
c| It allows the alarm settings to be altered without removing the probe ☐
d| All of the above ☐

3. Why is a separator recommended before a flow control station?

a| It protects the pipeline transducer from the effects of a wet steam ☐
b| It protects the pressure control valve from wiredrawing ☐
c| It ensures that only dry steam is being measured ☐
d| All of the above ☐

4. Why is a flow computer recommended when controlling steam flow?

a| The system won't work without it ☐
b| It compensates for changes in supply pressure to give accuracy ☐
c| It contains a set of electronic steam tables ☐
d| All of the above ☐

5. What does a pipeline transducer actually do?

a| It always converts flow into a measurable signal ☐
b| It always converts flow into an electrical signal ☐
c| It always converts flow into a pressure signal ☐
d| It converts differential pressure into a flow signal ☐

6. What does a DP transmitter actually do?

a| It converts differential pressure into an electrical signal ☐
b| It converts an electrical signal into differential pressure ☐
c| It converts upstream pressure into an electrical signal ☐
d| It converts differential pressure into a flow signal ☐

Answers

1: d, 2: d, 3: d, 4: b, 5: a, 6: a

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Module 8.4

Control Installations

Control Installations

The service life and accuracy of a control system is influenced not just by the component parts, but also by the installation.

Temperature sensors

Sensor location

The position of the sensor is important, and it must be located where it can sense a representative pressure, temperature or level.

The length of the sensor must also be considered. If the sensor to be used is large or long, provision has to be made for this in the pipework into which it is installed.

Sensors for self-acting control systems can come in many different shapes and sizes. Generally, the sensors for electronic and pneumatic control systems are smaller than those for self-acting controls.

The next requirement is to position the sensor in a location where it is not susceptible to damage, and perhaps to fit it in a pocket if necessary.

The pocket must be long enough to enable the whole sensor to be immersed in the liquid. If, in Figure 8.4.1, the stub connector were longer, the sensor might not be properly immersed in the fluid.

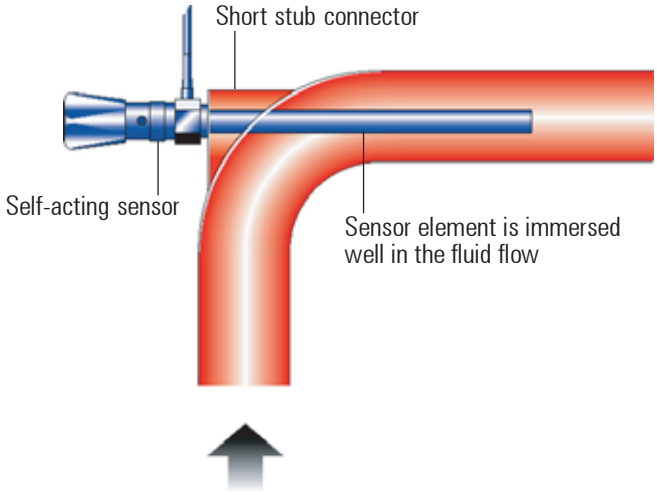


Fig. 8.4.1 A good installation with the sensor properly immersed in the fluid

Sensor protection

If the sensor is to be installed in a tank, it may be better to locate it close to one of the corners, where the greatest wall strength might be expected, with less chance of flexing.

With some fluids it is necessary to protect the sensor to prevent it from being corroded or dissolved. Pockets are usually available in various materials, including:

- Stainless steel.
- Mild steel.
- Copper and brass, which are suitable for the less severe applications.
- Heat resistant glass, which offers good general protection against corrosive products like acids and alkalis, but these can be fragile.

Self-acting control capillary tubes can usually be supplied covered with a PVC coating, which is useful in corrosive environments.

Where it is possible to fit the sensor through the side of the tank, the provision of a pocket also allows the sensor to be removed without draining the contents.

A pocket will tend to increase the time lag before the control can respond to changes in solution temperature, and it is important to make arrangements to keep this to a minimum. There will, for instance, be an air space between the sensor and the inside of the pocket, and air is an insulator. To overcome this, a heat conducting paste can be used to fill the space.

Controllers

The controller:

- Should be installed where it can be accessed and read by the authorised operator.
- Should be installed where it is safe from accidental damage inflicted by passing personnel or vehicles.
- Must be appropriate to the environment in terms of enclosure rating, hazardous gases and/or liquids.
- Must comply with standards relating to radio frequency interference.

Valves and actuators

The preferred actuator position will depend upon the type of control system used. For self-acting control valves, it is generally preferable if the actuator is fitted underneath the valve. Conversely, it is usually better to fit an electrical or pneumatic actuator above the valve, otherwise any leakage from the stem may result in process fluid, which may be hot or corrosive, spilling onto the actuator.

Horizontal fitting is not recommended as over a period of time:

- Uneven stem wear may occur.
- The valve plug may not present itself squarely to the valve seat.

The material construction of electric actuators must be appropriate to the environment in terms of the enclosure rating against excess moisture, and hazardous gases and liquids.

The valve and actuator will be heavier than an equivalent length of pipe, and will need adequate support.

It is important, before and after installation, to check that the valve is installed with its flow arrow in the correct direction.

Enough space must be left around the valve and actuator for maintenance, and to lift the actuator off the valve.

Radio frequency interference (RFI)

Radio frequency interference is electrical noise that can cause corruption of control signals and affect the operation of electronic controllers.

There are two forms of RFI:

- Continuous
- Impulse (transient).

Radio transmitters, computers, induction heaters, and other such equipment emit continuous high frequency radio interference.

Impulse interference is generated from electrical arcing, which can occur on the opening of switch contacts especially those responsible for switching inductive components, such as motors or transformers.

The control engineer is often most concerned about impulse interference. The pulses are of very high intensity and very short duration, and can disturb genuine electrical control signals.

Transmission of RFI

Radio interference can travel via two modes:

- Conduction.
- Radiation.

Conducted interference is communicated to the controller via mains supply cables. Having an interference suppressor in the supply as close to the controller as possible can reduce its effect.

Radiated interference is a greater problem because it is harder to counteract. This form of interference is like a broadcast transmission being picked up by 'aerials' naturally formed by the signal wiring, and then re-emitted within the controller box to more sensitive areas.

The electronic components within the controller can also receive transmissions directly, especially if the interference source is within 200 mm.

Effects of RFI

Controller types respond to different forms of interference in different ways.

Analogue controllers will usually respond to continuous rather than transient interference but will usually recover when the interference ceases. The symptoms of continuous interference are not easily recognisable because they usually influence the measurement accuracy. It is often difficult to distinguish between the effects of interference and the normal operation of the device.

Transient interference is more likely to affect relay outputs, as its occurrence is faster than that which the analogue circuits can respond.

Microprocessor based controllers are more subject to corruption from transient impulse interference but have a higher immunity to continuous interference.

The first indication that interference has occurred is often that the display has locked up, is scrambled or contains meaningless symbols in addition to the normal display.

More difficult symptoms to detect include measurement inaccuracies or incorrect actuator position, this may continue undetected until the system is clearly out of control.

Installation practice to limit RFI

The correct selection and installation of control signal wiring is vital to reduce susceptibility to RFI. Twisted pairs of wires are less susceptible to interference than parallel run cables (Figure 8.4.2). Earthed screened cables are even less susceptible to interference than twisted pairs of wires, but this cannot always be relied on, especially near high current cables.

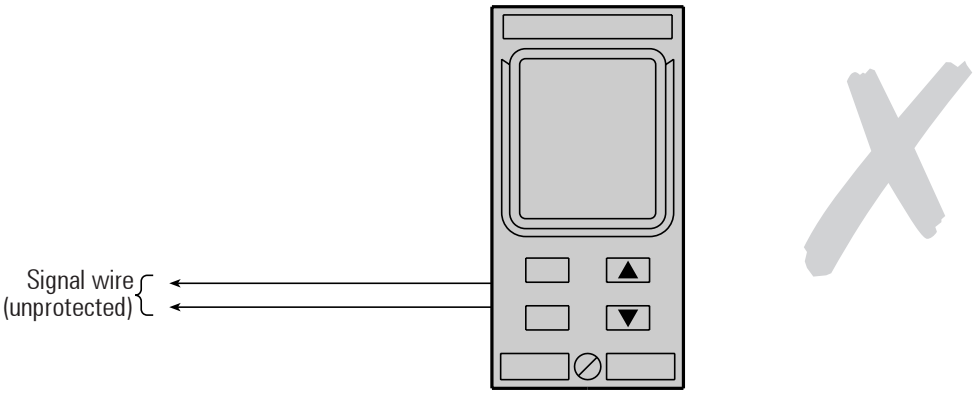


Fig. 8.4.2 Unprotected signal wire

Screened cable (Figures 8.4.3) should only be earthed at one end, see Figure 8.4.3 ('A' and 'B'); earthing at both ends will lead to a deterioration in this situation.

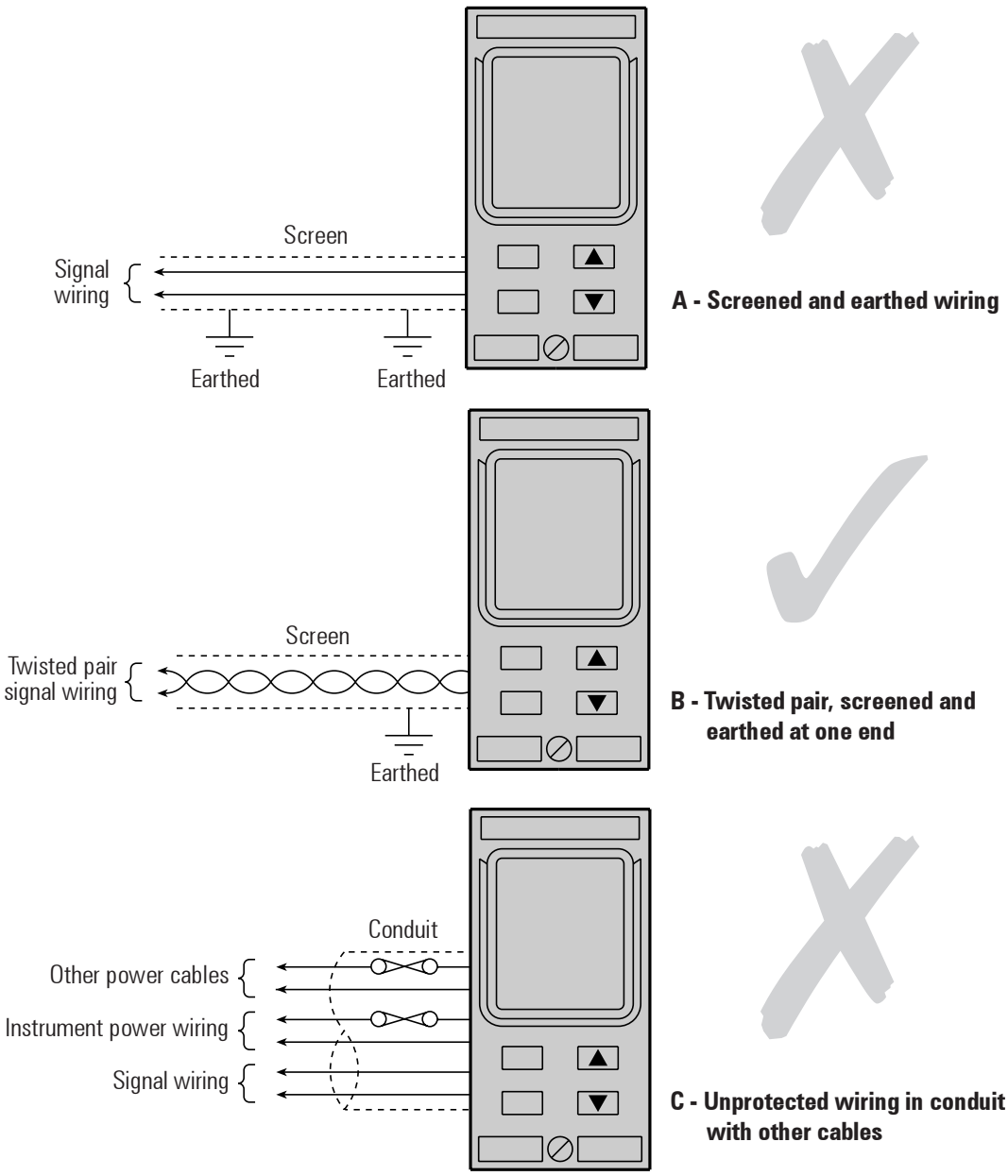


Fig. 8.4.3 Correct earthing of screened cable

Keeping wires separate from power wiring (Figure 8.4.4) can reduce pick-up via the signal wires. BS 6739: 1986 recommends that this separation should be at least 200 mm for instrument power wiring and 250 mm for other power cables.

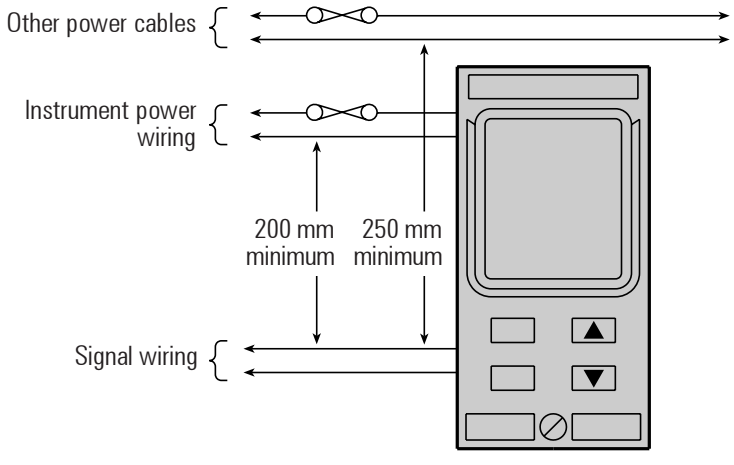


Fig. 8.4.4 Cable separation

It has been found in practice that signal wires can be run alongside/close to power wiring providing they are contained within their own earthed screen, see Figure 8.4.5.

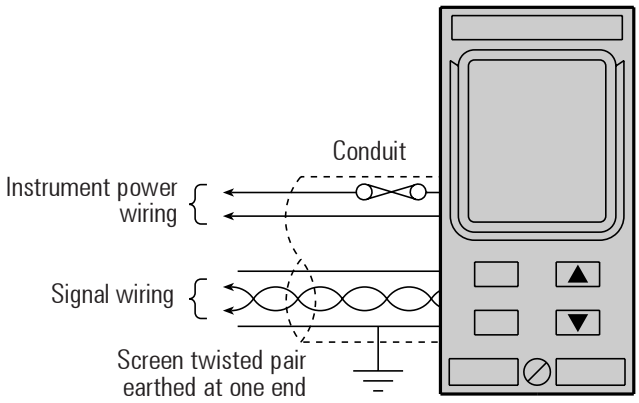


Fig. 8.4.5 Signal and power wiring in conduit

Impulse interference generated from electrical arcing can be reduced by means of an appropriate suppressor connected across switch contacts.

Pick-up via direct radiation can be reduced by installing the controllers at least 250 mm away from interference sources, such as contact breakers or mains switching relays.

Cable separation

The following information is reprinted from the British Standard Code of Practice for Instrumentation in Process Control systems: installation design and practice BS 6739: 1986:

Paragraph 10.7.4.2.2 - Separation from power cables

- Instrument cables should be routed above or below ground, separated from electrical power cables (i.e. ac, cables usually above 50 Vac with a 10 A rating).
- Parallel runs of cables should be avoided. However, where this is unavoidable, adequate physical separation should be provided.
- A spacing of 250 mm is recommended from ac power cables up to 10 A rating. For higher ratings, spacing should be increased progressively.
- Where it is unavoidable for signal and power cables to cross over each other, the cables should be arranged to cross at right angles with a positive means of separation of at least 250 mm.

Paragraph 10.7.4.2.3 - Separation between instrument cables

1. Categories 1 and 2 spaced 200 mm.

2. Categories 2 and 3 spaced 300 mm.

3. Categories 1 and 3 spaced 300 mm.

Cables are categorised as follows:

1. **Power cables ac -** Cables usually above 50 Vac with a 10 amp rating.
2. **Category 1. Instrument power and control wiring above 50 V -** This group includes ac and dc power supplies and control signals up to 10 A rating.
3. **Category 2. High-level signal wiring (5 V to 50 Vdc) -** This group includes digital signals, alarm signals, shutdown signals and high level analogue signals e.g. 4 - 20 mA.
4. **Category 3. Low-level signal wiring (below 5 Vdc) -** This group includes temperature signals and low-level analogue signals. Thermocouple wiring comes within this category.

Although it is not always practical, every effort should be made to achieve the recommended separations given.

Electrical protection standards

Electrical equipment such as electronic controllers must be suitable for the environment in which they are to be used. Hazardous environments may be found in oil refineries, offshore platforms, hospitals, chemical plants, mines, pharmaceutical plants and many others. The degree of protection will alter depending on the potential hazard, for example the risk of sparks or hot surfaces igniting flammable gases and vapours which may be present.

It is equally important to safeguard equipment against moisture, dust, water ingress, and severe changes in temperature.

Standards and procedures exist to reduce the chance of equipment inducing faults, which might otherwise start fires or initiate explosions in adjacent equipment.

Basic standards of protection have been devised to cater for specific environments.

IP ratings

The IP, or international protection rating stated for an enclosure, is a means of grading the protection level offered by the enclosure, by using two figures, as shown in Tables 8.4.1 and 8.4.2.

The first figure (see Table 8.4.1) refers to the protection offered against the intrusion of foreign objects such as levers, screwdrivers or even a person's hand. The range consists of seven digits commencing with 0, designating no protection offered from material objects or human intervention; up to 6, offering meticulous protection against the entry of dust or extremely fine particles.

Table 8.4.1 Degrees of protection offered by the 1st characteristic numeral

First characteristic numeral	Degree of protection	
	Short description	Definition
0	Non-protected	No special protection.
1	Protected against solid objects larger than 50 mm diameter.	A large surface of the human body, like a hand, but no protection against attempted deliberate access.
2	Protected against solid objects larger than 12 mm diameter.	Fingers, or similar objects, not exceeding 80 mm in length.
3	Protected against solid objects larger than 2.5 mm diameter.	Tools, wires etc of diameter greater than 2.5 mm.
4	Protected against solid objects larger than 1.0 mm diameter.	Tools, wires etc of diameter greater than 1.0 mm.
5	Dust protected.	Ingress of dust not prevented, but does not enter in sufficient quantity to interfere with satisfactory operation of the equipment.
6	Dust-tight.	No ingress of dust.

The second figure (see Table 8.4.2) indicates the degree of protection against water intrusion. The range commences with 0 meaning no protection against water. The highest is 8, giving optimum protection for equipment being continuously immersed in water.

Table 8.4.2 Degrees of protection offered by the 2nd characteristic numeral

First characteristic numeral	Degree of protection	
	Short description	Definition
0	Non-protected.	No special protection.
1	Protected against dripping water.	Dripping water shall have no harmful effect.
2	Protected against dripping water when tilted up to 15°.	Dripping water shall have no harmful effect when tilted at any angle up to 15° from its normal position.
3	Protected against spraying water.	Water falling as a spray at an angle up to 60° from the vertical shall have no harmful effect.
4	Protected against splashing water.	Water splashed against the enclosure from any direction shall have no harmful effect.
5	Protected against water jets.	Water projected by a nozzle against the enclosure shall have no harmful effect.
6	Protected against heavy seas.	Water from heavy seas or water projected in powerful jets shall not enter the enclosure in harmful quantities.
7	Protected against the effects of immersion.	Ingress of water in a harmful quantity shall not be possible when the enclosure is immersed in water under defined conditions of pressure and time.
8	Protected against submersion.	The equipment is suitable for continuous submersion in water under conditions which shall be specified by the manufacturer.

Example 8.4.1

An electrical enclosure having the following IP34 rating can be defined as follows:

Code letters	**IP**	An enclosure which has been given an International Protection rating.
1st characteristic numeral	**3**	Protects equipment inside the enclosure against ingress of solid foreign objects having a diameter of 2.5 mm and greater.
2nd characteristic numeral	**4**	Protects equipment inside the enclosure against harmful effects due to water splashed onto the enclosure from any direction.

It is not the intention of this Module to enter into detail regarding the subject of enclosure protection. The subject is discussed in much further depth in International Standards, BS EN 60529:1992 being one of them. The reader is advised to refer to such standards if information is required for specific purposes.

Explosion protected electrical equipment

It has been shown briefly how IP ratings cover two important areas of protection. There are, however, numerous other types of hazard to contend with. These may include corrosion, vibration, fire and explosion. The latter are likely to occur when electrical equipment produce sparks, operate at high temperatures, or arc; thus igniting chemicals, oils or gases.

In practice, it is difficult to determine whether or not an explosive atmosphere will be present at a specific place within a potentially hazardous area or plant. This problem has been resolved by assigning an area within the plant where flammable gases may be present to one of the following three hazardous zones:

- **Zone 1 -** An area where the explosive gas is continuously present or is present for long periods of time.
- **Zone 2 -** An area where the explosive gas is likely to occur during normal operation.
- **Zone 3 -** An area where the explosive gas is not likely to occur during normal operation and if it does, will exist only for a short period of time.

There have been many attempts to formulate internationally accepted standards of protection. The IEC (International Electrotechnical Commission) were the first to produce international standards in this area, however, CENELEC (European, Electrical Standards Co-ordination Committee) currently unites all the major European manufacturing countries under one set of standards.

Measurement and control equipment is covered by an intrinsic safety protection method, which is based upon the reduction of explosive risk by restricting the amount of electrical energy entering a hazardous area, and therefore does not, in principle, require special enclosures.

There are two categories of intrinsically-safe apparatus defined by the CENELEC and IEC, namely, EX ia and EX ib.

EX ia class

This classifies equipment as not being able to cause ignition under normal operational procedures, or as a result of a single fault or any two entirely independent faults occurring.

EX ib class

This classifies equipment as not being able to cause ignition under normal operational procedures, or as a result of a single fault occurring.

As with IP protection, this Module does not intend to discuss this subject in any great depth; it is a complex subject further complicated by the fact that groupings of equipment can be different in different countries.

It is suggested that, if the reader requires further information on this subject matter, he or she studies the appropriate relevant standard.

Questions

1. What is the main disadvantage of a self-acting sensor?

a| It is not available in various materials ☐
b| It cannot be fitted into a pocket ☐
c| It is generally larger than a EL (electrical) or PN (pneumatic) sensor ☐
d| It is not suitable for steam applications ☐

2. What can be done to improve the heat transfer efficiency between the process and the sensor when a sensor pocket is used?

a| Use a wider pocket ☐
b| Use a longer pocket ☐
c| Fill the sensor with distilled water ☐
d| Fill the sensor with a heat conducting paste or grease ☐

3. What is RFI and how is it transmitted?

a| Radio frequency interference; conduction and convection ☐
b| Radio frequency interference; conduction and radiation ☐
c| Radio frequency integration; conduction and radiation ☐
d| Radiographic friendly installation; conduction and radiation ☐

4. How can control signal wiring be installed to reduce RFI?

a| By earthing each end of the twisted signal cable ☐
b| By earthing the screen of a screened cable at both ends ☐
c| By earthing the screen of a screened cable at one of its ends ☐
d| By running it immediately alongside a mains power cable ☐

5. What is a category 1 cable as defined in BS 6739?

a| Instrument power and control wiring above 50 V ☐
b| High level signal wiring ☐
c| Low level signal wiring ☐
d| Cables above 50 V and a 10 A rating ☐

6. What minimum spacing is recommended between controllers and sources of RFI as defined in BS 6739?

a| 50 mm ☐
b| 100 mm ☐
c| 250 mm ☐
d| 1000 mm ☐

Answers

1: c, 2: d, 3: b, 4: c, 5: a, 6: c

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Module 9.1

Introduction to Safety Valves

Introduction

As soon as mankind was able to boil water to create steam, the necessity of the safety device became evident. As long as 2000 years ago, the Chinese were using cauldrons with hinged lids to allow (relatively) safer production of steam. At the beginning of the 14th century, chemists used conical plugs and later, compressed springs to act as safety devices on pressurised vessels.

Early in the 19th century, boiler explosions on ships and locomotives frequently resulted from faulty safety devices, which led to the development of the first safety relief valves.

In 1848, Charles Retchie invented the accumulation chamber, which increases the compression surface within the safety valve allowing it to open rapidly within a narrow overpressure margin. Today, most steam users are compelled by local health and safety regulations to ensure that their plant and processes incorporate safety devices and precautions, which ensure that dangerous conditions are prevented.

The primary function of a safety valve is therefore to protect life and property.

The principle type of device used to prevent overpressure in plant is the safety or safety relief valve. The safety valve operates by releasing a volume of fluid from within the plant when a predetermined maximum pressure is reached, thereby reducing the excess pressure in a safe manner. As the safety valve may be the only remaining device to prevent catastrophic failure under overpressure conditions, it is important that any such device is capable of operating at all times and under all possible conditions.

Safety valves should be installed wherever the maximum allowable working pressure (MAWP) of a system or pressure-containing vessel is likely to be exceeded. In steam systems, safety valves are typically used for boiler overpressure protection and other applications such as downstream of pressure reducing controls. Although their primary role is for safety, safety valves are also used in process operations to prevent product damage due to excess pressure. Pressure excess can be generated in a number of different situations, including:

- An imbalance of fluid flowrate caused by inadvertently closed or opened isolation valves on a process vessel.
- Failure of a cooling system, which allows vapour or fluid to expand.
- Compressed air or electrical power failure to control instrumentation.
- Transient pressure surges.
- Exposure to plant fires.
- Heat exchanger tube failure.
- Uncontrollable exothermic reactions in chemical plants.
- Ambient temperature changes.

The terms ‘safety valve’ and ‘safety relief valve’ are generic terms to describe many varieties of pressure relief devices that are designed to prevent excessive internal fluid pressure build-up. A wide range of different valves is available for many different applications and performance criteria. Furthermore, different designs are required to meet the numerous national standards that govern the use of safety valves.

A listing of the relevant national standards can be found at the end of this module.

In most national standards, specific definitions are given for the terms associated with safety and safety relief valves. There are several notable differences between the terminology used in the USA and Europe. One of the most important differences is that a valve referred to as a ‘safety valve’ in Europe is referred to as a ‘safety relief valve’ or ‘pressure relief valve’ in the USA. In addition, the term ‘safety valve’ in the USA generally refers specifically to the full-lift type of safety valve used in Europe.

The ASME / ANSI PTC25.3 standards applicable to the USA define the following generic terms:

- **Pressure relief valve -** A spring-loaded pressure relief valve which is designed to open to relieve excess pressure and to reclose and prevent the further flow of fluid after normal conditions have been restored. It is characterised by a rapid-opening 'pop' action or by opening in a manner generally proportional to the increase in pressure over the opening pressure. It may be used for either compressible or incompressible fluids, depending on design, adjustment, or application.

  This is a general term, which includes safety valves, relief valves and safety relief valves.

- **Safety valve -** A pressure relief valve actuated by inlet static pressure and characterised by rapid opening or pop action.

  Safety valves are primarily used with compressible gases and in particular for steam and air services. However, they can also be used for process type applications where they may be needed to protect the plant or to prevent spoilage of the product being processed.

- **Relief valve -** A pressure relief device actuated by inlet static pressure having a gradual lift generally proportional to the increase in pressure over opening pressure.

  Relief valves are commonly used in liquid systems, especially for lower capacities and thermal expansion duty. They can also be used on pumped systems as pressure overspill devices.

- **Safety relief valve -** A pressure relief valve characterised by rapid opening or pop action, or by opening in proportion to the increase in pressure over the opening pressure, depending on the application, and which may be used either for liquid or compressible fluid.

  In general, the safety relief valve will perform as a safety valve when used in a compressible gas system, but it will open in proportion to the overpressure when used in liquid systems, as would a relief valve.

The European standards (BS 6759 and DIN 3320) provide the following definition:

- **Safety valve -** A valve which automatically, without the assistance of any energy other than that of the fluid concerned, discharges a certified amount of the fluid so as to prevent a predetermined safe pressure being exceeded, and which is designed to re-close and prevent the further flow of fluid after normal pressure conditions of service have been restored.

Typical examples of safety valves used on steam systems are shown in Figure 9.1.1.

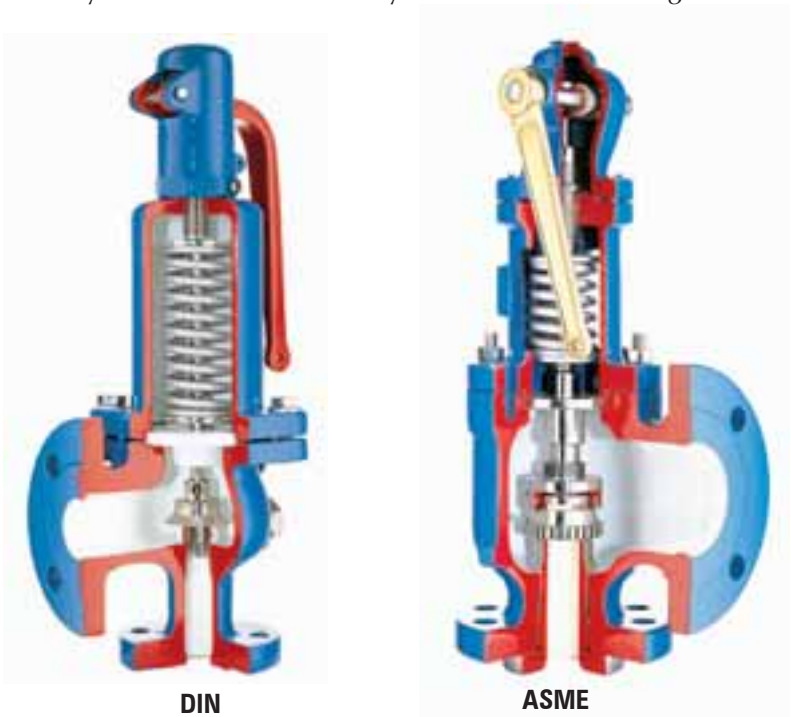


Fig. 9.1.1 Typical safety valves

Safety valve design

The basic spring loaded safety valve, referred to as 'standard' or 'conventional' is a simple, reliable self-acting device that provides overpressure protection.

The basic elements of the design consist of a right angle pattern valve body with the valve inlet connection, or nozzle, mounted on the pressure-containing system. The outlet connection may be screwed or flanged for connection to a piped discharge system. However, in some applications, such as compressed air systems, the safety valve will not have an outlet connection, and the fluid is vented directly to the atmosphere.

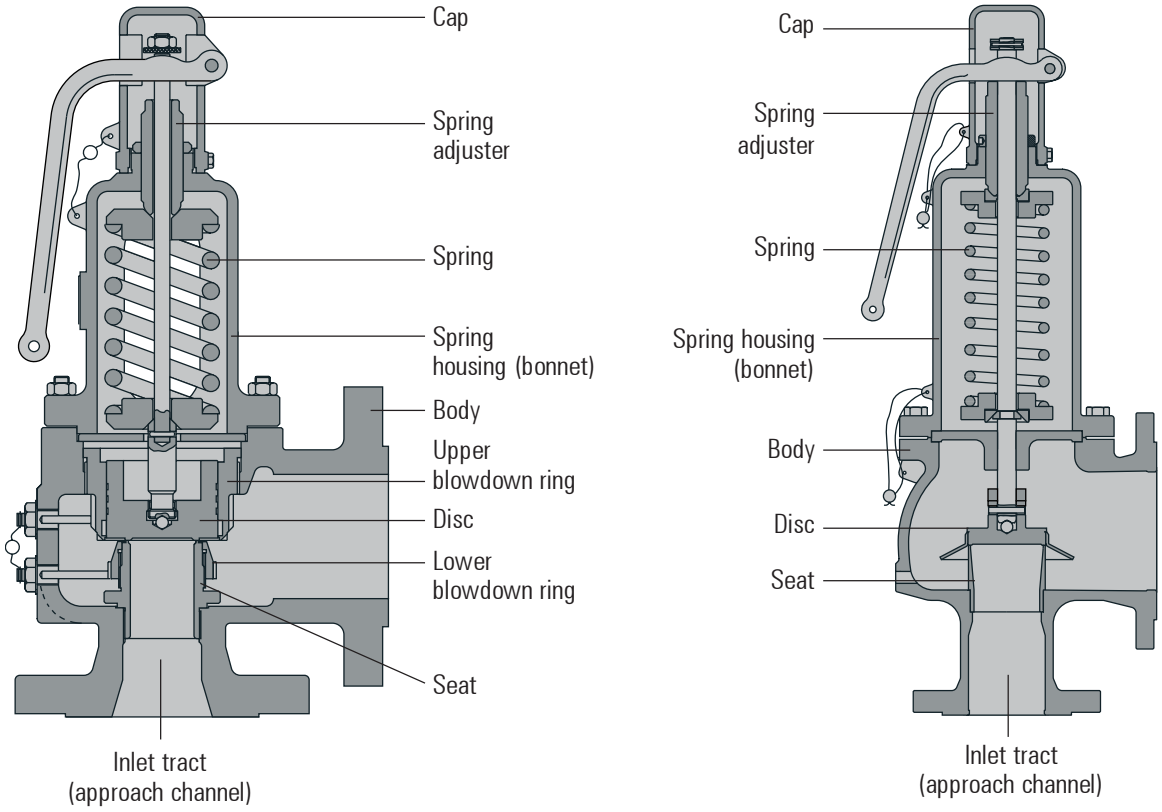


Fig. 9.1.2 Typical safety valve designs

The valve inlet (or approach channel) design can be either a full-nozzle or a semi-nozzle type. A full-nozzle design has the entire 'wetted' inlet tract formed from one piece. The approach channel is the only part of the safety valve that is exposed to the process fluid during normal operation, other than the disc, unless the valve is discharging.

Full-nozzles are usually incorporated in safety valves designed for process and high pressure applications, especially when the fluid is corrosive.

Conversely, the semi-nozzle design consists of a seating ring fitted into the body, the top of which forms the seat of the valve. The advantage of this arrangement is that the seat can easily be replaced, without replacing the whole inlet.

The disc is held against the nozzle seat (under normal operating conditions) by the spring, which is housed in an open or closed spring housing arrangement (or bonnet) mounted on top of the body. The discs used in rapid opening (pop type) safety valves are surrounded by a shroud, disc holder or huddling chamber which helps to produce the rapid opening characteristic.

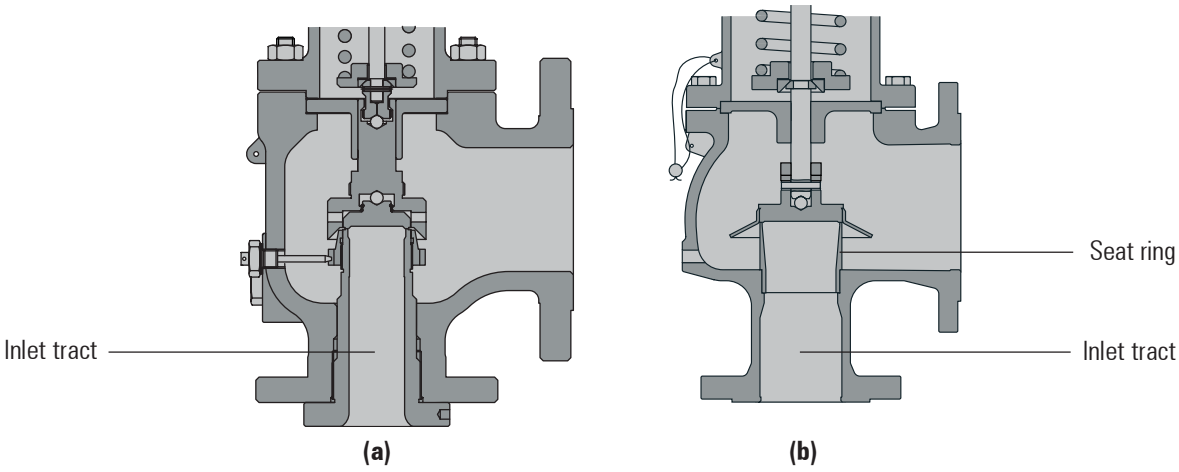


Fig. 9.1.3 A full-nozzle valve (a) and a semi-nozzle valve (b)

The closing force on the disc is provided by a spring, typically made from carbon steel. The amount of compression on the spring is usually adjustable, using the spring adjuster, to alter the pressure at which the disc is lifted off its seat.

Standards that govern the design and use of safety valves generally only define the three dimensions that relate to the discharge capacity of the safety valve, namely the flow (or bore) area, the curtain area and the discharge (or orifice) area (see Figure 9.1.4).

1. **Flow area -** The minimum cross-sectional area between the inlet and the seat, at its narrowest point. The diameter of the flow area is represented by dimension 'd' in Figure 9.1.4.

$$\text{Flow area} = \frac{\pi d^2}{4} \qquad \textbf{Equation 9.1.1}$$

2. **Curtain area -** The area of the cylindrical or conical discharge opening between the seating surfaces created by the lift of the disk above the seat. The diameter of the curtain area is represented by dimension 'd_1' in Figure 9.1.4.

$$\text{Curtain area} = \pi d_1 L \qquad \textbf{Equation 9.1.2}$$

3. **Discharge area -** This is the lesser of the curtain and flow areas, which determines the flow through the valve.

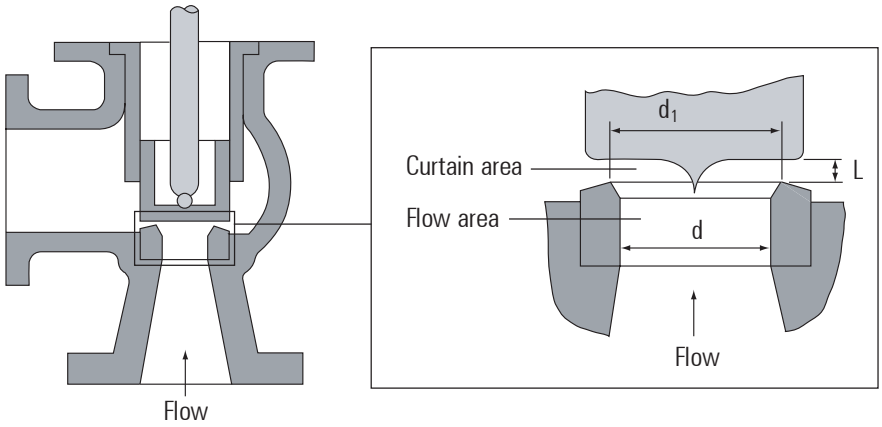


Fig. 9.1.4 Illustration of the standard defined areas

Valves in which the flow area and not the curtain area determines the capacity are known as full lift valves. These valves will have a greater capacity than low lift or high lift valves. This issue will be discussed in greater depth in Module 9.2.

Although the principal elements of a conventional safety valve are similar, the design details can vary considerably. In general, the DIN style valves (commonly used throughout Europe) tend to use a simpler construction with a fixed skirt (or hood) arrangement whereas the ASME style valves have a more complex design that includes one or two adjustable blowdown rings. The position of these rings can be used to fine-tune the overpressure and blowdown values of the valve.

For a given orifice area, there may be a number of different inlet and outlet connection sizes, as well as body dimensions such as centreline to face dimensions. Furthermore, many competing products, particularly of European origin have differing dimensions and capacities for the same nominal size.

An exception to this situation is found with steel ASME specification valves, which invariably follow the recommendations of the API Recommended Practice 526, where centreline to face dimensions, and orifice sizes are listed. The orifice area series are referred to by a letter. It is common for valves with the same orifice letter to have several different sizes of inlet and outlet connection. For example, 2″ x J x 3″ and 3″ x J x 4″ are both valves which have the same size ('J) orifice, but they have differing inlet and outlet sizes as shown before and after the orifice letter respectively. A 2″ x J x 3″ valve would have a 2″ inlet, a 'J' size orifice and a 3″ outlet.

This letter series is also referenced in other standards, for example, BS 6759 part 3, which deals with valves for process type applications and NFE- E 29-414.

Basic operation of a safety valve

Lifting

When the inlet static pressure rises above the set pressure of the safety valve, the disc will begin to lift off its seat. However, as soon as the spring starts to compress, the spring force will increase; this means that the pressure would have to continue to rise before any further lift can occur, and for there to be any significant flow through the valve.

The additional pressure rise required before the safety valve will discharge at its rated capacity is called the overpressure. The allowable overpressure depends on the standards being followed and the particular application. For compressible fluids, this is normally between 3% and 10%, and for liquids between 10% and 25%.

In order to achieve full opening from this small overpressure, the disc arrangement has to be specially designed to provide rapid opening. This is usually done by placing a shroud, skirt or hood around the disc. The volume contained within this shroud is known as the control or huddling chamber.

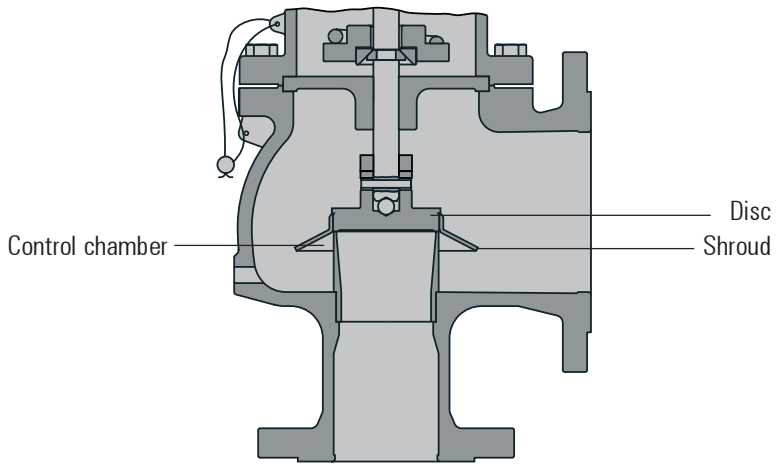


Fig. 9.1.5 Typical disc and shroud arrangement used on rapid opening safety valves

As lift begins (Figure 9.1.6b), and fluid enters the chamber, a larger area of the shroud is exposed to the fluid pressure. Since the magnitude of the lifting force (F) is proportional to the product of the pressure (P) and the area exposed to the fluid (A); (F = P x A), the opening force is increased. This incremental increase in opening force overcompensates for the increase in spring force, causing rapid opening. At the same time, the shroud reverses the direction of the flow, which provides a reaction force, further enhancing the lift.

These combined effects allow the valve to achieve its designed lift within a relatively small percentage overpressure. For compressible fluids, an additional contributory factor is the rapid expansion as the fluid volume increases from a higher to a lower pressure area. This plays a major role in ensuring that the valve opens fully within the small overpressure limit. For liquids, this effect is more proportional and subsequently, the overpressure is typically greater; 25% is common.

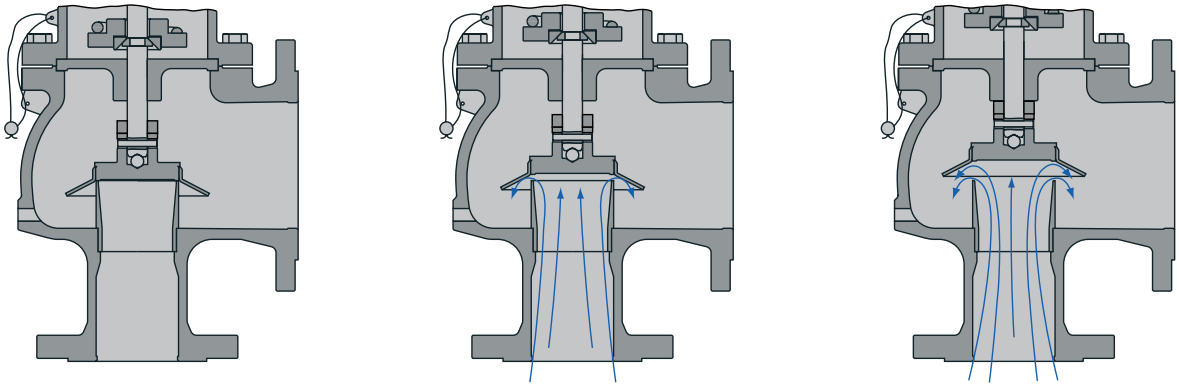


Fig. 9.1.6 Operation of a conventional safety valve

Reseating

Once normal operating conditions have been restored, the valve is required to close again, but since the larger area of the disc is still exposed to the fluid, the valve will not close until the pressure has dropped below the original set pressure. The difference between the set pressure and this reseating pressure is known as the 'blowdown', and it is usually specified as a percentage of the set pressure. For compressible fluids, the blowdown is usually less than 10%, and for liquids, it can be up to 20%.

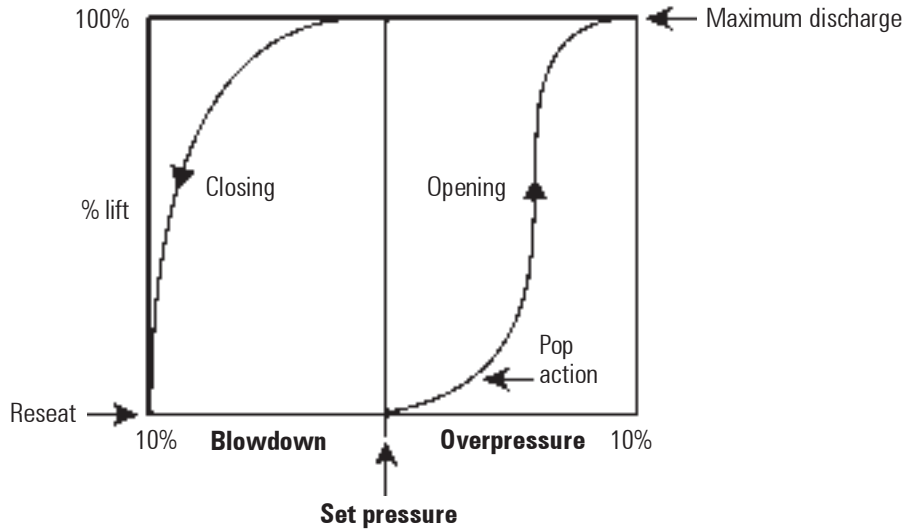


Fig. 9.1.7 Relationship between pressure and lift for a typical safety valve

The design of the shroud must be such that it offers both rapid opening and relatively small blowdown, so that as soon as a potentially hazardous situation is reached, any overpressure is relieved, but excessive quantities of the fluid are prevented from being discharged. At the same time, it is necessary to ensure that the system pressure is reduced sufficiently to prevent immediate reopening.

The blowdown rings found on most ASME type safety valves are used to make fine adjustments to the overpressure and blowdown values of the valves (see Figure 9.1.8). The lower blowdown (nozzle) ring is a common feature on many valves where the tighter overpressure and blowdown requirements require a more sophisticated designed solution. The upper blowdown ring is usually factory set and essentially takes out the manufacturing tolerances which affect the geometry of the huddling chamber.

The lower blowdown ring is also factory set to achieve the appropriate code performance requirements but under certain circumstances can be altered. When the lower blowdown ring is adjusted to its top position the huddling chamber volume is such that the valve will pop rapidly, minimising the overpressure value but correspondingly requiring a greater blowdown before the valve re-seats. When the lower blowdown ring is adjusted to its lower position there is minimal restriction in the huddling chamber and a greater overpressure will be required before the valve is fully open but the blowdown value will be reduced.

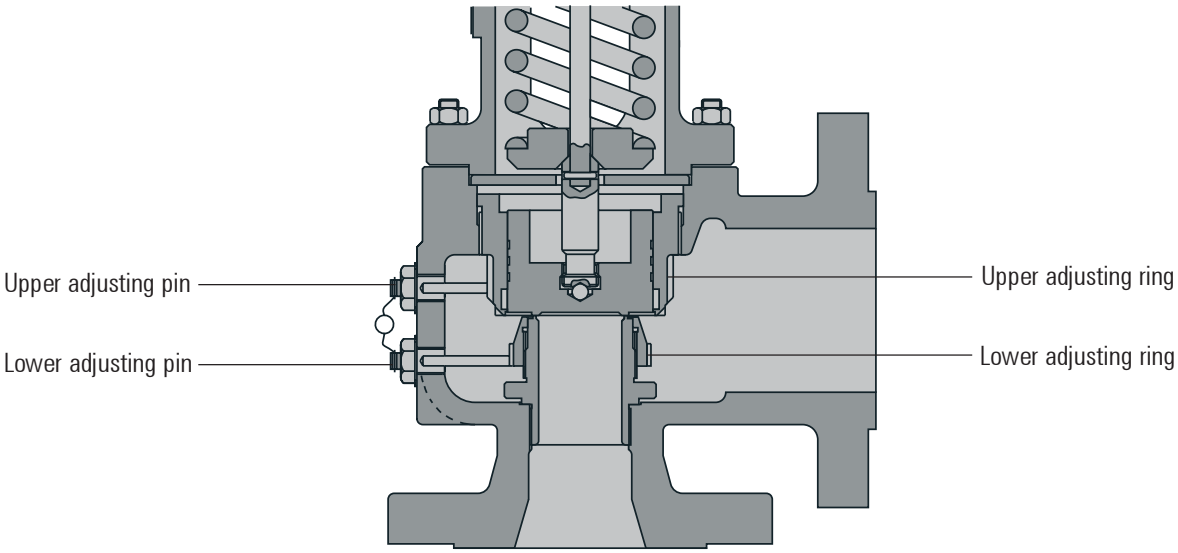


Fig. 9.1.8 The blowdown rings on an ASME type safety valve

Approval authorities

For most countries, there are independent bodies who will examine the design and performance of a product range to confirm conformity with the relevant code or standard. This system of third party approval is very common for any safety related products and is often a customer requirement before purchase, or a requirement of their insurance company.

The actual requirements for approval will vary depending on the particular code or standard. In some cases, revalidation is necessary every few years, in others approval is indefinite as long as no significant design changes are made, in which case the approval authority must be notified, and re-approval sought. In the USA, the National Board of Boiler and Pressure Vessel Inspectors represents the US and Canadian government agencies empowered to assure adherence to code construction and repair of boilers and pressure vessels.

Some of the more commonly encountered bodies are listed in Table 9.1.1.

Table 9.1.1 Approval authorities

Country	Abbreviation	Approval body
Germany	TÜV	Association of Technical Supervision
	DSRK	Deutsche Schiffs-Revision und Klassifikation
UK	SAFed	Safety Assessment Federation Type Approval Service (STAS) formerly Associated Offices Technical Committee AOTC and British Engine
		Lloyds Register of Shipping
France		CODAP
		APAVE
Belgium		Bureau Veritas
Netherlands		Dienst voor het Stoomwezen
Norway	DNV	Det Norske Veritas
Italy	ISPESL RINA	Institution of Prevention and Security Italian Register of Shipping
Korea		Ministry of Power and Resources Korean Register of Shipping
Canada		Ministry of Labour Canada
United States	NB	National Board of Boiler and Pressure Vessel Inspectors

Codes and Standards

Standards relevant to safety valves vary quite considerably in format around the world, and many are sections within codes relevant to Boilers or Pressure Containing Vessels. Some will only outline performance requirements, tolerances and essential constructional detail, but give no guidance on dimensions, orifice sizes etc. Others will be related to installation and application. It is quite common within many markets to use several in conjunction with each other.

Table 9.1.2 Standards relating to safety valves

Country	Standard No.	Description
Germany	AD-Merkblatt A2	Pressure Vessel Equipment safety devices against excess pressure - safety valves
	TRD 421	Technical Equipment for Steam Boilers Safeguards against excessive pressure - safety valves for steam boilers of groups I, III & IV
	TRD 721	Technical Equipment for Steam Boilers Safeguards against excessive pressure - safety valves for steam boilers of group II
UK	BS 6759	Part 1 specification for safety valves for steam and hot water Part 2 specification for safety valves for compressed air or inert gas Part 3 specification for safety valves for process fluids
France	AFNOR NFE-E 29-411 to 416	Safety and relief valves
	NFE-E 29-421	Safety and relief valves
Korea	KS B 6216	Spring loaded safety valves for steam boilers and pressure vessels
Japan	JIS B 8210	Steam boilers and pressure vessels - spring loaded safety valves
Australia	SAA AS1271	Safety valves, other valves, liquid level gauges and other fittings for boilers and unfired pressure vessels
USA	ASME I	Boiler Applications
	ASME III	Nuclear Applications
	ASME VIII	Unfired Pressure Vessel Applications
	ANSI/ASME PTC 25.3	Safety and Relief Valves - performance test codes
	API RP 520	Sizing selection and installation of pressure-relieving devices in refineries
		Part 1 Design
		Part 2 Installation
	API RP 521	Guide for pressure relieving and depressurising systems
	API STD 526	Flanged steel pressure relief valves
	API STD 527	Seat tightness of pressure relief valves
Europe	prEN ISO 4126*	Safety devices for protection against excessive pressure
International	ISO 4126	Safety valves - general requirements

*Note: pr = pre-ratification. This harmonised European standard is not offically issued.

For steam boiler applications there are very specific requirements for safety valve performance, demanded by national standards and often, insurance companies. Approval by an independent authority is often necessary, such as British Engine, TÜV or Lloyd's Register.

Safety valves used in Europe are also subject to the standards associated with the Pressure Equipment Directive (PED). Being classified as 'Safety accessories', safety valves are considered as 'Category 4' equipment, which require the most demanding level of assessment within the PED regime. This can usually be met by the manufacturer having an ISO 9000 quality system and the safety valve design and performance certified by an officially recognised approval authority referred to as a 'Notified Body'.

Questions

1. What is the primary function of a safety valve?

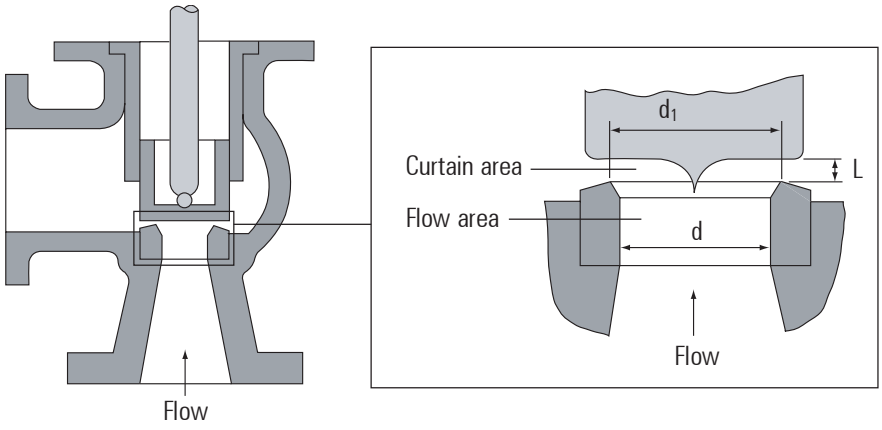
a| To maintain the pressure of a system within a specified range. ☐
b| To protect life and property ☐
c| To prevent product spoilage ☐
d| To allow the gradual release of overpressure ☐

2. What is the main operational difference between safety valves and relief valves?

a| Relief valves are characterised by a rapid opening or 'popping' type lift characteristic ☐
b| Safety valves are characterised by a gradual opening type lift characteristic ☐
c| Relief valves are characterised by a gradual opening type lift characteristic ☐
d| Safety valves will have a rapid opening lift characteristic when used on compressible fluid systems and a gradual opening characteristic when used on liquid systems ☐

3. Given the safety valve dimensions as indicated in the illustration below, what would the discharge area of the safety valve be?

Given:
d = 29 mm
d_1 = 35 mm
L = 5 mm



a| 550 mm²
b| 617 mm²
c| 661 mm²
d| 693 mm²

4. Which of the following factors combine to produce the rapid opening characteristic of most safety valves used in steam applications?

a| The rapid expansion of the steam as the fluid volume increases ☐
b| Exposure of a greater disc surface area to the steam ☐
c| The vectoring effect created by the shroud ☐
d| All of the above ☐

5. Blowdown rings are often found on ASME type pressure relief valves. What is the function of the lower or nozzle blowdown ring?

a| To adjust the blowdown value of the valve ☐

b| To adjust the set pressure of the valve ☐

c| To adjust the backpressure acting on the safety valve disc ☐

d| To adjust the overpressure and blowdown of the valve ☐

6. In which of the following applications should a full-nozzle valve be used?

a| On a process application where the fluid is corrosive ☐

b| On a steam system operating at 2 bar ☐

c| On a non-corrosive process fluid system where a significant amount of seat wear is predicted ☐

d| All of the above ☐

Answers

1:b, 2: c, 3: a, 4: d, 5: d, 6: a

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Module 9.2

Types of Safety Valves

Types of Safety Valves

There is a wide range of safety valves available to meet the many different applications and performance criteria demanded by different industries. Furthermore, national standards define many varying types of safety valve.

The ASME standard I and ASME standard VIII for boiler and pressure vessel applications and the ASME / ANSI PTC 25.3 standard for safety valves and relief valves provide the following definition. These standards set performance characteristics as well as defining the different types of safety valves that are used:

- **ASME I valve -** A safety relief valve conforming to the requirements of Section I of the ASME pressure vessel code for boiler applications which will open within 3% overpressure and close within 4%. It will usually feature two blowdown rings, and is identified by a National Board 'V' stamp.
- **ASME VIII valve -** A safety relief valve conforming to the requirements of Section VIII of the ASME pressure vessel code for pressure vessel applications which will open within 10% overpressure and close within 7%. Identified by a National Board 'UV' stamp.
- **Low lift safety valve -** The actual position of the disc determines the discharge area of the valve.
- **Full lift safety valve -** The discharge area is not determined by the position of the disc.
- **Full bore safety valve -** A safety valve having no protrusions in the bore, and wherein the valve lifts to an extent sufficient for the minimum area at any section, at or below the seat, to become the controlling orifice.
- **Conventional safety relief valve -** The spring housing is vented to the discharge side, hence operational characteristics are directly affected by changes in the backpressure to the valve.
- **Balanced safety relief valve -** A balanced valve incorporates a means of minimising the effect of backpressure on the operational characteristics of the valve.
- **Pilot operated pressure relief valve -** The major relieving device is combined with, and is controlled by, a self-actuated auxiliary pressure relief device.
- **Power-actuated safety relief valve -** A pressure relief valve in which the major pressure relieving device is combined with, and controlled by, a device requiring an external source of energy.

The following types of safety valve are defined in the DIN 3320 standard, which relates to safety valves sold in Germany and other parts of Europe:

- **Standard safety valve -** A valve which, following opening, reaches the degree of lift necessary for the mass flowrate to be discharged within a pressure rise of not more than 10%. (The valve is characterised by a pop type action and is sometimes known as high lift).
- **Full lift (Vollhub) safety valve -** A safety valve which, after commencement of lift, opens rapidly within a 5% pressure rise up to the full lift as limited by the design. The amount of lift up to the rapid opening (proportional range) shall not be more than 20%.
- **Direct loaded safety valve -** A safety valve in which the opening force underneath the valve disc is opposed by a closing force such as a spring or a weight.
- **Proportional safety valve -** A safety valve which opens more or less steadily in relation to the increase in pressure. Sudden opening within a 10% lift range will not occur without pressure increase. Following opening within a pressure of not more than 10%, these safety valves achieve the lift necessary for the mass flow to be discharged.

- **Diaphragm safety valve -** A direct loaded safety valve wherein linear moving and rotating elements and springs are protected against the effects of the fluid by a diaphragm.
- **Bellows safety valve -** A direct loaded safety valve wherein sliding and (partially or fully) rotating elements and springs are protected against the effects of the fluids by a bellows. The bellows may be of such a design that it compensates for influences of backpressure.
- **Controlled safety valve -** Consists of a main valve and a control device. It also includes direct acting safety valves with supplementary loading in which, until the set pressure is reached, an additional force increases the closing force.

The British Standard BS 6759 lists the following types of safety valve:

- **Direct loaded -** A safety valve in which the loading due to the fluid pressure underneath the valve disc is opposed only by direct mechanical loading such as a weight, a lever and weight, or a spring.
- **Conventional safety valve -** A safety valve of the direct loaded type, the set pressure of which will be affected by changes in the superimposed backpressure.
- **Assisted safety valve -** A direct loaded safety valve which, by means of a powered assistance mechanism, is lifted at a pressure below the unassisted set pressure and will, even in the event of failure of the assistance mechanism, comply with all the relevant requirements for safety valves.
- **Pilot operated (indirect loaded) safety valve -** The operation is initiated and controlled by the fluid discharged from a pilot valve, which is itself a direct loaded safety valve.
- **Balanced bellows safety valve -** A valve incorporating a bellows which has an effective area equal to that of the valve seat, to eliminate the effect of backpressure on the set pressure of the valve, and which effectively prevents the discharging fluid entering the bonnet space.
- **Balanced bellows safety valve with auxiliary piston -** A balanced bellows valve incorporating an auxiliary piston, having an effective area equal to the valve seat, which becomes effective in the event of bellows failure.
- **Balanced piston safety valve -** A valve incorporating a piston which has an area equal to that of the valve seat, to eliminate the effect of backpressure on the set pressure of the valve.
- **Bellows seal safety valve -** A valve incorporating a bellows, which prevents discharging fluid from entering the bonnet space.

In addition, the BS 759 standard pertaining to safety fittings for application to boilers, defines full lift, high lift and lift safety valves:

- **Lift safety valve (ordinary class) -** The valve member lifts automatically a distance of at least $\frac{1}{24}^{th}$ of the bore of the seating member, with an overpressure not exceeding 10% of the set pressure.
- **High lift safety valve -** Valve member lifts automatically a distance of at least $\frac{1}{12}^{th}$ of the bore of the seating member, with an overpressure not exceeding 10% of the set pressure.
- **Full lift safety valve -** Valve member lifts automatically to give a discharge area between 100% and 80% of the minimum area, at an overpressure not exceeding 5% of the set pressure.

The following table summarises the performance of different types of safety valve set out by the various standards.

Table 9.2.1 Safety valve performance summary

Standard		Fluid	Overpressure	Blowdown
A.D. Merkblatt A2		Steam	Standard 10% full lift 5%	10%
		Air or gas	Standard 10% full lift 5%	10%
		Liquid	10%	20%
ASME	I	Steam	3%	2-6%
	VIII	Steam	10%	7%
		Air or gas	10%	7%
		Liquid	10% **(see Note 3 below)**	
BS 6759	part 1	Steam	Standard 10% full lift 5%	10%
	part 2	Air or gas	10%	10%
	part 3	Liquid	10 – 25%	2.5 - 20%

Notes:
1. ASME blowdown values shown are for valves with adjustable blowdown.
2. BS 6759 blowdown values shown are for valves with non-adjustable blowdown.
3. 25% is often used for non-certified sizing calculations and 20% can be used for fire protection of storage vessels.

Conventional safety valves

The common characteristic shared between the definitions of conventional safety valves in the different standards, is that their operational characteristics are affected by any backpressure in the discharge system. It is important to note that the total backpressure is generated from two components; superimposed backpressure and the built-up backpressure:

- **Superimposed backpressure -** The static pressure that exists on the outlet side of a closed valve.

- **Built-up backpressure -** The additional pressure generated on the outlet side when the valve is discharging.

Subsequently, in a conventional safety valve, only the superimposed backpressure will affect the opening characteristic and set value, but the combined backpressure will alter the blowdown characteristic and re-seat value.

The ASME/ANSI standard makes the further classification that conventional valves have a spring housing that is vented to the discharge side of the valve. If the spring housing is vented to the atmosphere, any superimposed backpressure will still affect the operational characteristics. This can be seen from Figure 9.2.1, which shows schematic diagrams of valves whose spring housings are vented to the discharge side of the valve and to the atmosphere.

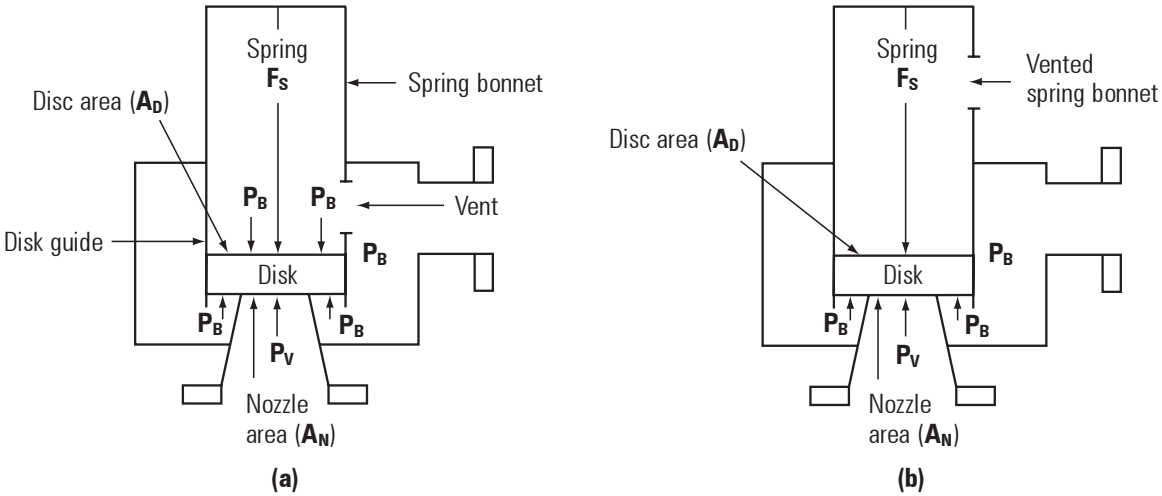


Fig. 9.2.1 Schematic diagram of safety valves with bonnets vented to (a) the valve discharge and (b) the atmosphere

By considering the forces acting on the disc (with area A_D), it can be seen that the required opening force (equivalent to the product of inlet pressure (P_V) and the nozzle area (A_N)) is the sum of the spring force (F_S) and the force due to the backpressure (P_B) acting on the top and bottom of the disc. In the case of a spring housing vented to the discharge side of the valve (an ASME conventional safety relief valve, see Figure 9.2.1 (a)), the required opening force is:

$$P_V A_N = F_S + P_B A_D - P_B (A_D - A_N) \text{ which simplies to Equation 9.2.1}$$

$$P_V A_N = F_S + P_B A_N \qquad \textbf{Equation 9.2.1}$$

Where:
P_V = Fluid inlet pressure
A_N = Nozzle area
F_S = Spring force
P_B = Backpressure

Therefore, any superimposed backpressure will tend to increase the closing force and the inlet pressure required to lift the disc is greater.

In the case of a valve whose spring housing is vented to the atmosphere (Figure 9.2.1b), the required opening force is:

$$P_V A_N = F_S - P_B (A_D - A_N) \qquad \textbf{Equation 9.2.2}$$

Where:
P_V = Fluid inlet pressure
A_N = Nozzle area
F_S = Spring force
P_B = Backpressure
A_D = Disc area

Thus, the superimposed backpressure acts with the vessel pressure to overcome the spring force, and the opening pressure will be less than expected.

In both cases, if a significant superimposed backpressure exists, its effects on the set pressure need to be considered when designing a safety valve system.

Once the valve starts to open, the effects of built-up backpressure also have to be taken into account. For a conventional safety valve with the spring housing vented to the discharge side of the valve, see Figure 9.2.1 (a), the effect of built-up backpressure can be determined by considering Equation 9.2.1 and by noting that once the valve starts to open, the inlet pressure is the sum of the set pressure, P_S, and the overpressure, P_O.

$$(P_S + P_O) A_N = F_S + P_B A_N \text{ which simplies to Equation 9.2.3}$$

$$P_S A_N = F_S + A_N (P_B - P_O) \qquad \textbf{Equation 9.2.3}$$

Where:
P_S = Set pressure of safety valves
A_N = Nozzle area
F_S = Spring force
P_B = Backpressure
P_O = Overpressure

Therefore, if the backpressure is greater than the overpressure, the valve will tend to close, reducing the flow. This can lead to instability within the system and can result in flutter or chatter of the valve.

In general, if conventional safety valves are used in applications, where there is an excessive built-up backpressure, they will not perform as expected. According to the API 520 Recommended Practice Guidelines:

- A conventional pressure relief valve should typically not be used when the built-up backpressure is greater than 10% of the set pressure at 10% overpressure. A higher maximum allowable built-up backpressure may be used for overpressure greater than 10%.

The British Standard BS 6759, however, states that the built-up backpressure should be limited to 12% of the set pressure when the valve is discharging at the certified capacity.

For the majority of steam applications, the backpressure can be maintained within these limits by carefully sizing any discharge pipes. This will be discussed in Module 9.4. If, however, it is not feasible to reduce the backpressure, then it may be necessary to use a balanced safety valve.

Balanced safety valves

Balanced safety valves are those that incorporate a means of eliminating the effects of backpressure. There are two basic designs that can be used to achieve this:

- **Piston type balanced safety valve.**
  Although there are several variations of the piston valve, they generally consist of a piston type disc whose movement is constrained by a vented guide. The area of the top face of the piston, A_P, and the nozzle seat area, A_N, are designed to be equal. This means that the effective area of both the top and bottom surfaces of the disc exposed to the backpressure are equal, and therefore any additional forces are balanced. In addition, the spring bonnet is vented such that the top face of the piston is subjected to atmospheric pressure, as shown in Figure 9.2.2.

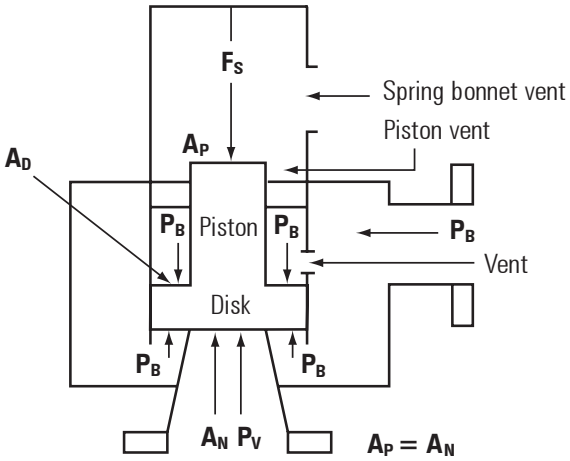


Fig. 9.2.2 Schematic diagram of a piston type balanced safety valve

By considering the forces acting on the piston, it is evident that this type of valve is no longer affected by any backpressure:

$$P_V A_N = F_S + P_B (A_D - A_P) - P_B (A_D - A_N)$$

Where:
P_V = Fluid inlet pressure
A_N = Nozzle area
F_S = Spring force
P_B = Backpressure
A_D = Disc area
A_P = Piston area

Since A_P equals A_N, the last two terms of the equation are equal in magnitude and cancel out of the equation. Therefore, this simplifies to Equation 9.2.4.

$$P_V A_N = F_S \qquad \textbf{Equation 9.2.4}$$

Where:
P_V = Fluid inlet pressure
A_N = Nozzle area
F_S = Spring force

- **Bellows type balanced safety valve.**
  A bellows with an effective area (A_B) equivalent to the nozzle seat area (A_N) is attached to the upper surface of the disc and to the spindle guide.

  The bellows arrangement prevents backpressure acting on the upper side of the disc within the area of the bellows. The disc area extending beyond the bellows and the opposing disc area are equal, and so the forces acting on the disc are balanced, and the backpressure has little effect on the valve opening pressure.

  The bellows vent allows air to flow freely in and out of the bellows as they expand or contract. Bellows failure is an important concern when using a bellows balanced safety valve, as this may affect the set pressure and capacity of the valve. It is important, therefore, that there is some mechanism for detecting any uncharacteristic fluid flow through the bellows vents. In addition, some bellows balanced safety valves include an auxiliary piston that is used to overcome the effects of backpressure in the case of bellows failure. This type of safety valve is usually only used on critical applications in the oil and petrochemical industries.

  In addition to reducing the effects of backpressure, the bellows also serve to isolate the spindle guide and the spring from the process fluid, this is important when the fluid is corrosive.

  Since balanced pressure relief valves are typically more expensive than their unbalanced counterparts, they are commonly only used where high pressure manifolds are unavoidable, or in critical applications where a very precise set pressure or blowdown is required.

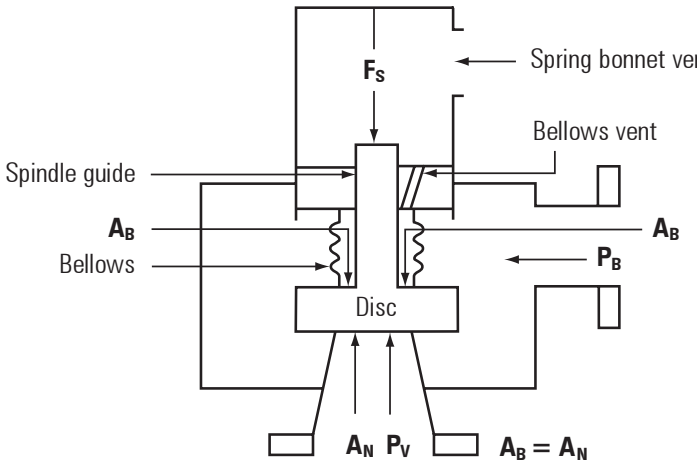


Fig. 9.2.3 Schematic diagram of the bellows balanced safety valve

Pilot operated safety valve

This type of safety valve uses the flowing medium itself, through a pilot valve, to apply the closing force on the safety valve disc. The pilot valve is itself a small safety valve.

There are two basic types of pilot operated safety valve, namely, the diaphragm and piston type.

The diaphragm type is typically only available for low pressure applications and it produces a proportional type action, characteristic of relief valves used in liquid systems. They are therefore of little use in steam systems, consequently, they will not be considered in this text.

The piston type valve consists of a main valve, which uses a piston shaped closing device (or obturator), and an external pilot valve. Figure 9.2.4 shows a diagram of a typical piston type, pilot operated safety valve.

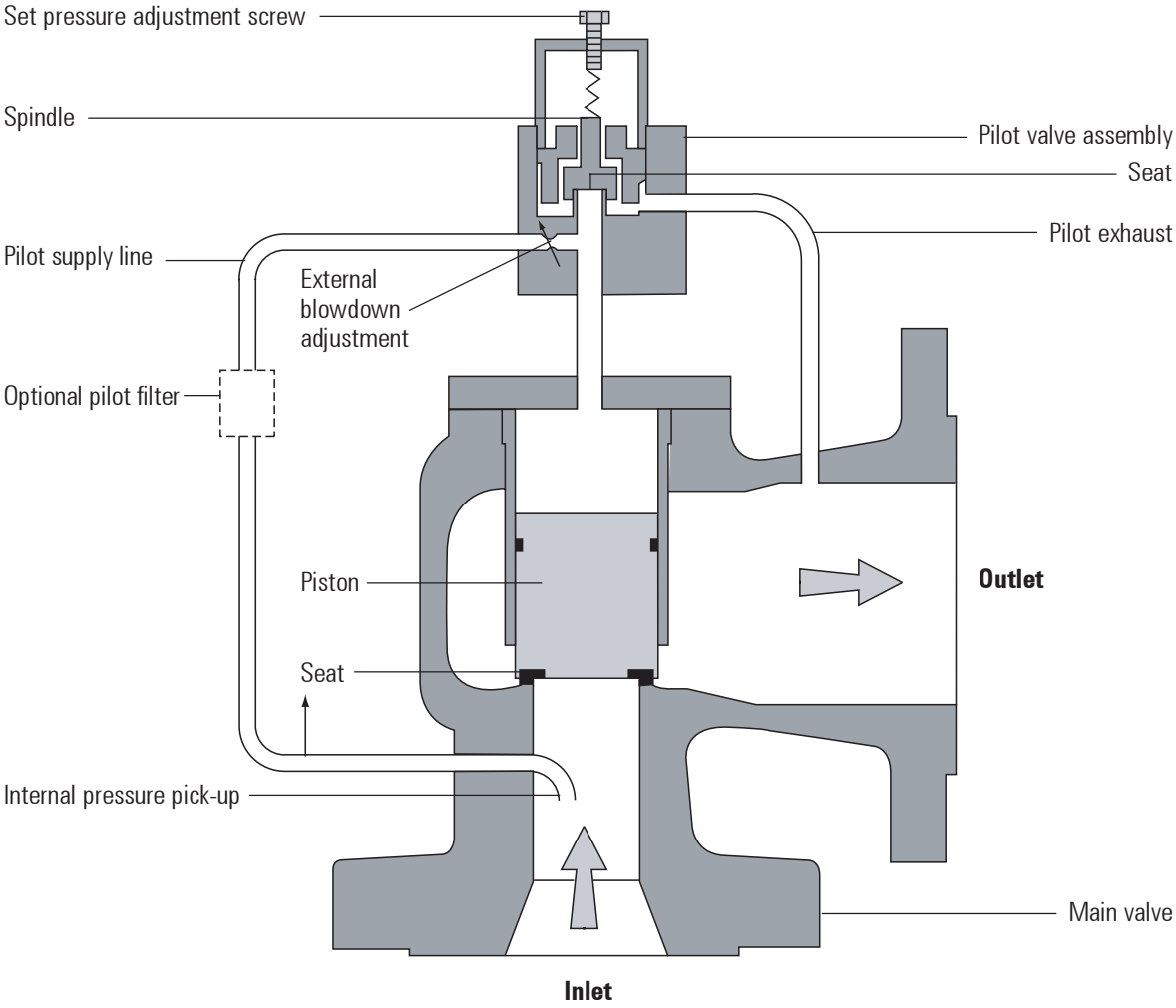


Fig. 9.2.4 A piston type, pilot operated safety valve

The piston and seating arrangement incorporated in the main valve is designed so that the bottom area of the piston, exposed to the inlet fluid, is less than the area of the top of the piston. As both ends of the piston are exposed to the fluid at the same pressure, this means that under normal system operating conditions, the closing force, resulting from the larger top area, is greater than the inlet force. The resultant downward force therefore holds the piston firmly on its seat.

If the inlet pressure were to rise, the net closing force on the piston also increases, ensuring that a tight shut-off is continually maintained. However, when the inlet pressure reaches the set pressure, the pilot valve will pop open to release the fluid pressure above the piston. With much less fluid pressure acting on the upper surface of the piston, the inlet pressure generates a net upwards force and the piston will leave its seat. This causes the main valve to pop open, allowing the process fluid to be discharged.

When the inlet pressure has been sufficiently reduced, the pilot valve will reclose, preventing the further release of fluid from the top of the piston, thereby re-establishing the net downward force, and causing the piston to reseat.

Pilot operated safety valves offer good overpressure and blowdown performance (a blowdown of 2% is attainable). For this reason, they are used where a narrow margin is required between the set pressure and the system operating pressure. Pilot operated valves are also available in much larger sizes, making them the preferred type of safety valve for larger capacities.

One of the main concerns with pilot operated safety valves is that the small bore, pilot connecting pipes are susceptible to blockage by foreign matter, or due to the collection of condensate in these pipes. This can lead to the failure of the valve, either in the open or closed position, depending on where the blockage occurs.

The British Standard BS 6759 states that all pilot operated safety valves should have at least two independent pilot devices, which are connected individually and arranged such that failure of either of the pilot will still enable the safety valve to continue to operate effectively.

Full lift, high lift and low lift safety valves

The terms full lift, high lift and low lift refer to the amount of travel the disc undergoes as it moves from its closed position to the position required to produce the certified discharge capacity, and how this affects the discharge capacity of the valve.

A full lift safety valve is one in which the disc lifts sufficiently, so that the curtain area no longer influences the discharge area. The discharge area, and therefore the capacity of the valve are subsequently determined by the bore area. This occurs when the disc lifts a distance of at least a quarter of the bore diameter. A full lift conventional safety valve is often the best choice for general steam applications.

The disc of a high lift safety valve lifts a distance of at least $\frac{1}{12}^{th}$ of the bore diameter. This means that the curtain area, and ultimately the position of the disc, determines the discharge area. The discharge capacities of high lift valves tend to be significantly lower than those of full lift valves, and for a given discharge capacity, it is usually possible to select a full lift valve that has a nominal size several times smaller than a corresponding high lift valve, which usually incurs cost advantages. Furthermore, high lift valves tend to be used on compressible fluids where their action is more proportional.

In low lift valves, the disc only lifts a distance of $\frac{1}{24}^{th}$ of the bore diameter. The discharge area is determined entirely by the position of the disc, and since the disc only lifts a small amount, the capacities tend to be much lower than those of full or high lift valves.

Materials of construction

Except when safety valves are discharging, the only parts that are wetted by the process fluid are the inlet tract (nozzle) and the disc. Since safety valves operate infrequently under normal conditions, all other components can be manufactured from standard materials for most applications. There are however several exceptions, in which case, special materials have to be used, these include:

- Cryogenic applications.
- Corrosive fluids.
- Where contamination of discharged fluid is not permitted.
- When the valve discharges into a manifold that contains corrosive media discharged by another valve.

The principal pressure-containing components of safety valves are normally constructed from one of the following materials:

- **Bronze -** Commonly used for small screwed valves for general duty on steam, air and hot water applications (up to 15 bar).
- **Cast iron -** Used extensively for ASME type valves. Its use is typically limited to 17 bar g.
- **SG iron -** Commonly used in European valves and to replace cast iron in higher pressure valves (up to 25 bar g).
- **Cast steel -** Commonly used on higher pressure valves (up to 40 bar g). Process type valves are usually made from a cast steel body with an austenitic full nozzle type construction.
- **Austenitic stainless steel -** Used in food, pharmaceutical or clean steam applications.

For extremely high pressure applications, pressure containing components may be forged or machined from solid.

For all safety valves, it is important that moving parts, particularly the spindle and guides are made from materials that will not easily degrade or corrode. As seats and discs are constantly in contact with the process fluid, they must be able to resist the effects of erosion and corrosion. For process applications, austenitic stainless steel is commonly used for seats and discs; sometimes they are 'stellite faced' for increased durability. For extremely corrosive fluids, nozzles, discs and seats are made from special alloys such as 'monel' or 'hastelloy'.

The spring is a critical element of the safety valve and must provide reliable performance within the required parameters. BS 6759 lists recommended materials, but most other standards just insist on sensible materials based on sound engineering practice. Standard safety valves will typically use carbon steel for moderate temperatures. Tungsten steel is used for higher temperature, non-corrosive applications, and stainless steel is used for corrosive or clean steam duty. For sour gas and high temperature applications, often special materials such as monel, hastelloy and 'inconel' are used.

Safety valve options and accessories

Due to the wide range of applications in which safety valves are used, there are a number of different options available:

Seating material

A key option is the type of seating material used. Metal-to-metal seats, commonly made from stainless steel, are normally used for high temperature applications such as steam. Alternatively, resilient discs can be fixed to either or both of the seating surfaces where tighter shut-off is required, typically for gas or liquid applications. These inserts can be made from a number of different materials, but Viton, nitrile or EPDM are the most common. Soft seal inserts are not recommended for steam use.

Table 9.2.2 Seating materials used in safety valves

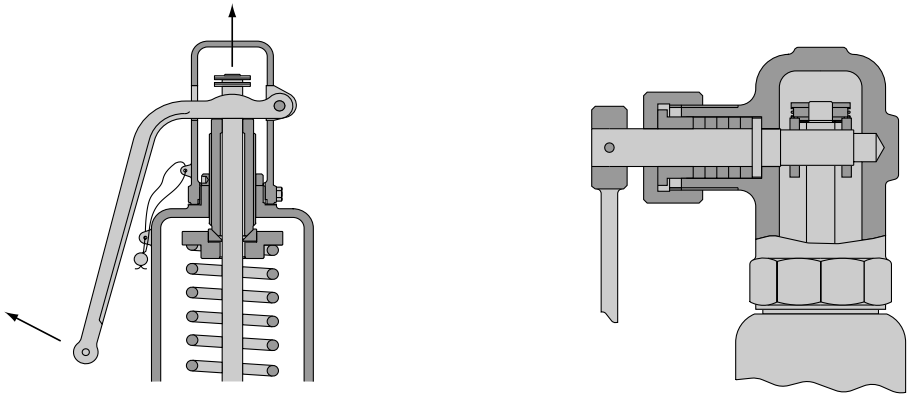
Seal material	Applications
EPDM	Water
Viton	High temperature gas applications
Nitrile	Air and oil applications
Stainless steel	Standard material, best for steam
Stellite	Wear resistant for tough applications

Levers

Standard safety valves are generally fitted with an easing lever, which enables the valve to be lifted manually in order to ensure that it is operational at pressures in excess of 75% of set pressure. This is usually done as part of routine safety checks, or during maintenance to prevent seizing. The fitting of a lever is usually a requirement of national standards and insurance companies for steam and hot water applications. For example, the ASME Boiler and Pressure Vessel Code states that pressure relief valves must be fitted with a lever if they are to be used on air, water over 60°C, and steam.

A standard or open lever is the simplest type of lever available. It is typically used on applications where a small amount of leakage of the fluid to the atmosphere is acceptable, such as on steam and air systems, (see Figure 9.2.5 (a)).

Where it is not acceptable for the media to escape, a packed lever must be used. This uses a packed gland seal to ensure that the fluid is contained within the cap, (see Figure 9.2.5 (b))



(a) Open **(b) Packed**

Fig. 9.2.5 Levers

For service where a lever is not required, a cap can be used to simply protect the adjustment screw. If used in conjunction with a gasket, it can be used to prevent emissions to the atmosphere, (see Figure 9.2.6).

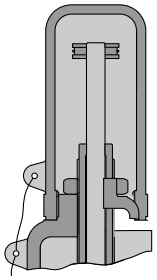


Fig. 9.2.6 A gas tight cap

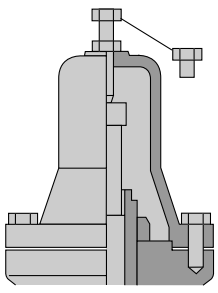


Fig. 9.2.7 A test gag

A test gag (Figure 9.2.7) may be used to prevent the valve from opening at the set pressure during hydraulic testing when commissioning a system. Once tested, the gag screw is removed and replaced with a short blanking plug before the valve is placed in service.

Open and closed bonnets

Unless bellows or diaphragm sealing is used, process fluid will enter the spring housing (or bonnet). The amount of fluid depends on the particular design of safety valve. If emission of this fluid into the atmosphere is acceptable, the spring housing may be vented to the atmosphere – an open bonnet. This is usually advantageous when the safety valve is used on high temperature fluids or for boiler applications as, otherwise, high temperatures can relax the spring, altering the set pressure of the valve. However, using an open bonnet exposes the valve spring and internals to environmental conditions, which can lead to damage and corrosion of the spring.

When the fluid must be completely contained by the safety valve (and the discharge system), it is necessary to use a closed bonnet, which is not vented to the atmosphere. This type of spring enclosure is almost universally used for small screwed valves and, it is becoming increasingly common on many valve ranges since, particularly on steam, discharge of the fluid could be hazardous to personnel.

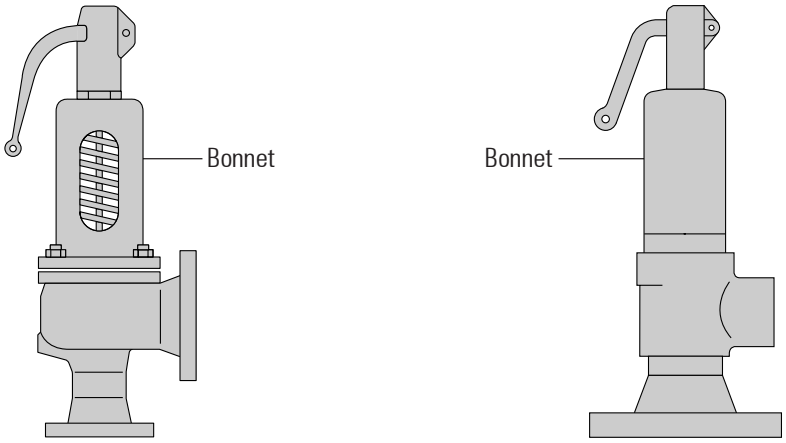


Fig. 9.2.8 Spring housings

Bellows and diaphragm sealing

Some safety valves, most commonly those used for water applications, incorporate a flexible diaphragm or bellows to isolate the safety valve spring and upper chamber from the process fluid, (see Figure 9.2.9).

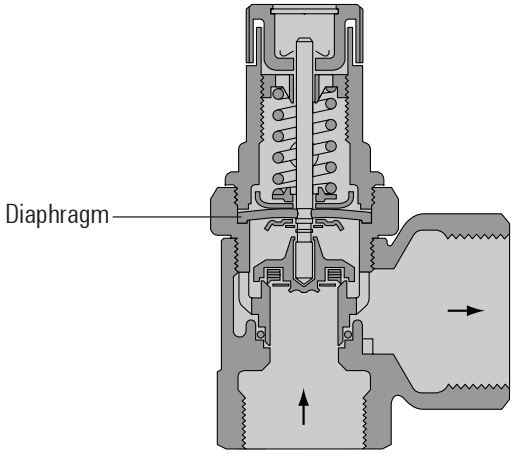


Fig. 9.2.9 A diaphragm sealed safety valve

An elastomer bellows or diaphragm is commonly used in hot water or heating applications, whereas a stainless steel one would be used on process applications employing hazardous fluids.

Questions

1. What is the typical maximum overpressure value for a standard safety valve used on steam applications, according to most national standards?

a| 5% ☐
b| 10% ☐
c| 15% ☐
d| 20% ☐

2. Superimposed backpressure affects which operational characteristic of a safety valve?

a| Blowdown ☐
b| Discharge capacity ☐
c| Set value ☐
d| All of the above ☐

3. Which type of conventional safety valve is most suitable for steam applications on the basis of its relationship between cost and discharge capacity?

a| Full lift ☐
b| High lift ☐
c| Low lift ☐
d| Full bore ☐

4. Which of the following statements about pilot operated safety valves are true?

i. Small margins of overpressure and blowdown are achievable

ii. The closing force increases as the inlet pressure increases, ensuring a tight shut-off

iii. Pilot operated valves can fail in the open or closed position due to the build up of condensate in the pilot connecting pipes

a| i only ☐
b| iii only ☐
c| i and ii ☐
d| i, ii and iii ☐

5. Which material would be most suitable for safety valves used on high pressure steam applications up to 25 bar?

a| Austenitic stainless steel ☐
b| SG iron ☐
c| Cast carbon steel ☐
d| Bronze ☐

6. Which of the following bonnet arrangements would be required on a system where it is important that none of the steam escapes?

a| Open bonnet and packed lever ☐
b| Closed bonnet and open lever ☐
c| Closed bonnet and packed lever ☐
d| Gas tight cap ☐

Answers

1:b, 2: c, 3: a, 4: d, 5: b, 6: c

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Module 9.3

Safety Valve Selection

Safety Valve Selection

As there is such a wide range of safety valves, there is no difficulty in selecting a safety valve that meets the specific requirements of a given application. Once a suitable type has been selected, it is imperative that the correct relieving pressure and discharge capacity are established, and a suitably sized valve and set pressure is specified.

The selection of a specific type of safety valve is governed by several factors:

- **Cost -** This is the most obvious consideration when selecting a safety valve for a non-critical application. When making cost comparisons, it is imperative to consider the capacity of the valve as well as the nominal size. As mentioned in the previous module, there can be large variations between models with the same inlet connection but with varying lift characteristics.
- **Type of disposal system -** Valves with an open bonnet can be used on steam, air or non-toxic gas, if discharge to the atmosphere, other than through the discharge system, is acceptable. A lifting lever is often specified in these applications.

  For gas or liquid applications, where escape to the atmosphere is not permitted, a closed bonnet must be specified. In such applications, it is also necessary to use either a closed/gas tight cap or packed lever.

  For applications with a significant superimposed backpressure (common in manifolds, typically seen in the process industry) a balancing bellows or piston construction is required.
- **Valve construction -** A semi-nozzle type construction should be used for non-toxic, non-corrosive type media at moderate pressures, whereas valves with the full nozzle type construction are typically used in the process industry for corrosive media or for extremely high pressures. For corrosive fluids or high temperatures, special materials of construction may also be required.
- **Operating characteristics -** Performance requirements vary according to application and the valve must be selected accordingly. For steam boilers, a small overpressure is required, usually 3% or 5%. For most other applications, 10% overpressure is required, but according to API 520, for special applications such as fire protection, larger valves with overpressures of 20% are allowed. For liquids, overpressures of 10% or 25% are common, and blowdown values tend to be up to 20%.
- **Approval -** For many valve applications, the end user will state the required code or standard for the construction and performance of the valve. This is usually accompanied by a requirement for approval by an independent authority, to guarantee conformance with the required standard.

Setting and sealing

In order to establish the set pressure correctly, the following terms require careful consideration:

- **Normal working pressure (NWP) -** The operating pressure of the system under full-load conditions.
- **Maximum allowable working pressure (MAWP) -** Sometimes called the safe working pressure (SWP) or design pressure of the system. This is the maximum pressure existing at normal operating conditions (relative to the maximum operating temperature) of the system.
- **Maximum allowable accumulation pressure (MAAP) -** The maximum pressure the system is allowed to reach in accordance with the specification of the design standards of the system. The MAAP is often expressed as a percentage of the MAWP.

  For steam using apparatus, the MAAP will often be 10% higher than the MAWP, but this is not always the case. If the MAWP is not readily available, the authority responsible for insuring the apparatus should be contacted. If the MAAP cannot be established, it must not be considered to be higher than the MAWP.
- **Set Pressure (P_S) -** The pressure at which the safety valve starts to lift.
- **Relieving pressure (P_R) -** This is the pressure at which the full capacity of the safety valve is achieved. It is the sum of the set pressure (P_s) and the overpressure (P_o).
- **Overpressure (P_O) -** The overpressure is the percentage of the set pressure at which the safety valve is designed to operate.

There are two fundamental constraints, which must be taken into account when establishing a safety valve set pressure:

1. The set pressure must be low enough to ensure that the relieving pressure never exceeds the maximum allowable accumulation pressure (MAAP) of the system.
2. The set pressure must be high enough to ensure that there is sufficient margin above the normal working pressure (NWP) to allow the safety valve to close. However, the set pressure must never be greater than the maximum allowable working pressure (MAWP).

In order to meet the first constraint, it is necessary to consider the relative magnitudes of the percentage overpressure and the percentage MAAP (expressed as a percentage of the MAWP). There are two possible cases:

- **The percentage overpressure of the safety valve is less than or equal to the percentage MAAP of the system -** This means that the set pressure can be made to equal the MAWP, as the relieving pressure will always be less than the actual MAAP.

  For example, if the safety valve overpressure was 5%, and the MAAP was 10% of the MAWP, the set pressure would be chosen to equal the MAWP. In this case, the relieving pressure (equal to the set pressure + 5% overpressure) would be less than the MAAP, which is acceptable.

  Note that if the percentage MAAP were higher than the percentage overpressure, the set pressure will still be made to equal the MAWP, as increasing it above the MAWP would violate the second constraint.
- **The percentage overpressure of the safety valve is greater than the percentage MAAP of the system -** In this case, making the set pressure equal to the MAWP will mean that the relieving pressure would be greater than the MAAP, so the set pressure must be lower than the MAWP.

  For example, if the safety valve overpressure was 25% and the percentage MAAP was only 10%, making the set pressure equal to the MAWP means that the relieving pressure would be 15% greater than the MAAP. In this instance, the correct set pressure should be 15% below the MAWP.

The following table summarises the determination of the set point based on the first constraint.

Table 9.3.1 Determination of the set pressure using safety valve overpressure and apparatus MAAP

Apparatus		Safety valve overpressure				
		5%	10%	15%	20%	25%
MAAP	20%	MAWP	MAWP	MAWP	MAWP	95% MAWP
	15%	MAWP	MAWP	MAWP	95% MAWP	90% MAWP
	10%	MAWP	MAWP	95% MAWP	90% MAWP	85% MAWP
	5%	MAWP	95% MAWP	90% MAWP	85% MAWP	80% MAWP

Unless operational considerations dictate otherwise, in order to meet the second constraint, the safety valve set pressure should always be somewhat above the normal working pressure with a margin allowed for the blowdown. A safety valve set just above the normal working pressure can lead to a poor shut-off after any discharge.

When the system operating pressure and safety valve set pressure have to be as close as possible to one another, a 0.1 bar minimum margin between reseat pressure and normal operating pressure is recommended to ensure a tight shut-off. This is called the 'shut-off margin'. In this case, it is important to take into account any variations in the system operating pressure before adding the 0.1 bar margin. Such variations can occur where a safety valve is installed after pressure reducing valves (PRVs) and other control valves, with relatively large proportional bands.

In practically all control systems, there is a certain amount of proportional offset associated with the proportional band (see Block 5, Control Theory, for more information regarding proportional offset). If a self-acting PRV is set under full-load conditions, the control pressure at no-load conditions can be significantly greater than its set pressure. Conversely, if the valve is set under no-load conditions, the full-load pressure will be less than its set pressure.

For example, consider a pilot operated PRV with a maximum proportional band of only 0.2 bar. With a control pressure of 5.0 bar set under full-load conditions, it would give 5.2 bar under no-load conditions. Alternatively, if the control pressure of 5.0 bar is set under no-load conditions, the same valve would exhibit a control pressure of 4.8 bar under full-load conditions.

When determining the set pressure of the safety valve, if the PRV control pressure is set under no-load conditions, then the proportional offset does not have to be taken into account. However, if the PRV control pressure is set under full-load conditions, it is necessary to consider the increase in downstream pressure as a result of the proportional offset of the PRV (see Example 9.3.1).

The amount of pressure control offset depends on the type of control valve and the pressure controller being used. It is therefore important to determine the proportional band of the upstream control valve as well as how this valve was commissioned.

Example 9.3.1
A safety valve, which is to be installed after a PRV, is required to be set as close as possible to the PRV working pressure. Given the parameters below, determine the most suitable safety valve set pressure:

PRV set pressure:	6.0 bar (set under full-load conditions)
PRV proportional band:	0.3 bar operating above the PRV working pressure
Safety valve blowdown:	10%

Answer:
Since it is necessary to ensure that the safety valve set pressure is as close to the PRV working pressure as possible, the safety valve is chosen so that its blowdown pressure is greater than the PRV working pressure (taking into account the proportional offset), and a 0.1 bar shut-off margin.

Firstly, the effect of the proportional offset needs to be considered; the normal maximum working pressure that will be encountered is:

$$6.0 \text{ bar} + 0.3 \text{ bar} = 6.3 \text{ bar (NWP)}$$

By adding the 0.1 bar shut-off margin, the blowdown pressure has to be 10% greater than 6.4 bar. For this example, this means that the safety valve's set pressure has to be:

$$6.4 \times \left(\frac{100}{100 - 10} \right) = 7.1 \text{ bar}$$

The set pressure would therefore be chosen as 7.04 bar, **provided that this does not exceed the MAWP of the protected system.**

Note that if the PRV were set at 6.0 bar under no-load conditions, and with a safety valve 10% blowdown, the safety valve set pressure would be:

$$(6.4 + 0.1) \times \left(\frac{100}{100 - 10} \right) = 6.8 \text{ bar}$$

Effects of backpressure on set pressure

For a conventional safety valve subject to a constant superimposed backpressure, the set pressure is effectively reduced by an amount equal to the backpressure. In order to compensate for this, the required set pressure must be increased by an amount equal to the backpressure. The cold differential set pressure (the pressure set on the test stand) will therefore be:

$$CDSP = RISP - CBP \qquad \textbf{Equation 9.3.1}$$

Where:
CDSP = Cold differential set pressure
RISP = Required installed set pressure
CBP = Constant backpressure

For variable superimposed backpressure, the effective set pressure could change as the backpressure varies, and a conventional valve could not be used if the variation were more than 10% to 15% of the set pressure. Instead, a balanced valve would have to be used.

The pressure level relationships for pressure relief valves as shown in the API Recommended Practice 520 is illustrated in Figure 9.3.1.

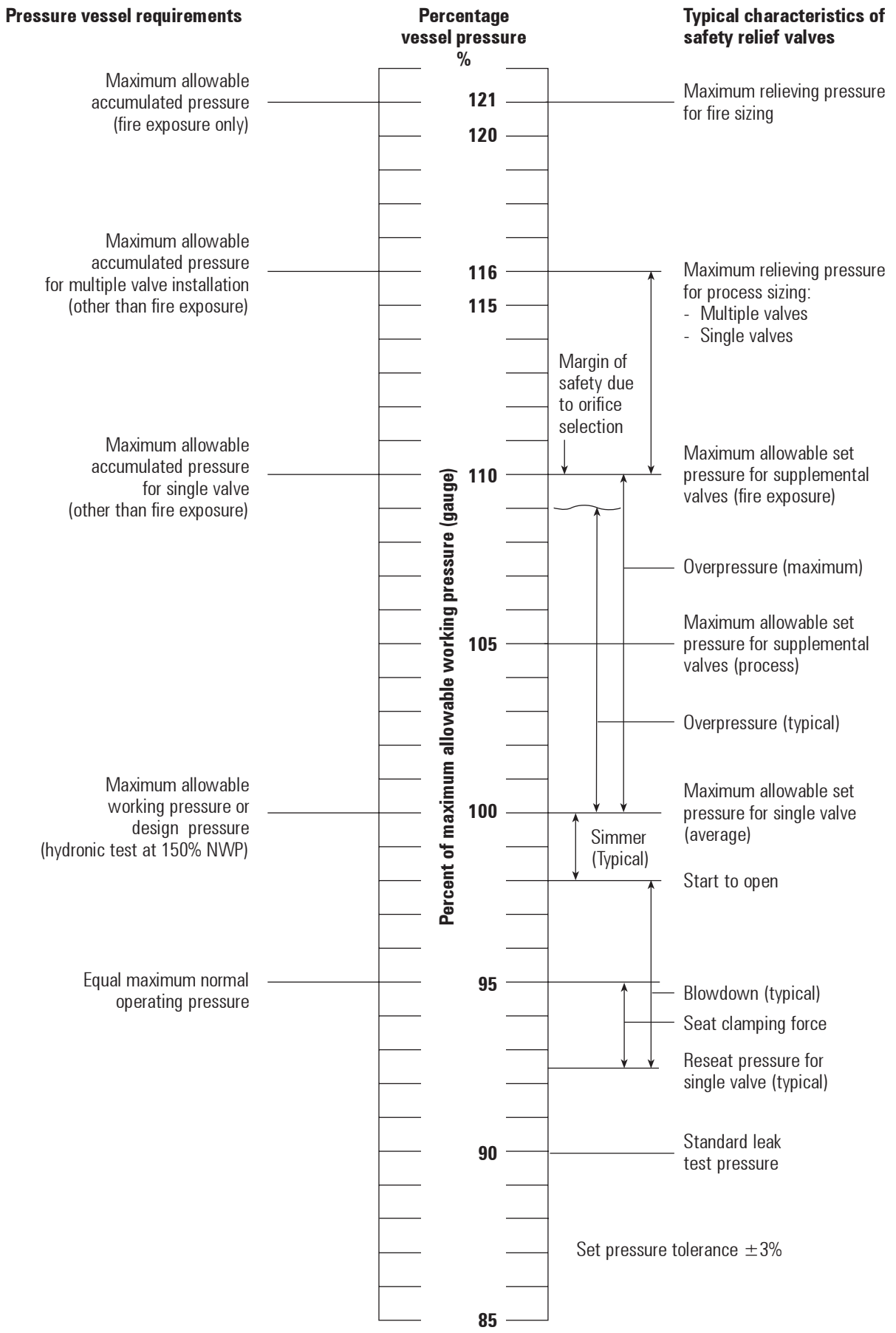


Fig. 9.3.1 Pressure level relationships for pressure relief valves (from API 520)

Setting a safety valve

For most types of safety valve, air or gas setting is permissible. A specially constructed test stand is usually employed, allowing easy and quick mounting of the safety valve, for adjustment, and subsequent locking and sealing of the valve at the required set pressure.

The most important requirement, in addition to the usual safety considerations is that instrument quality gauges are used and a regular calibration system is in place. All safety valve standards will specify a particular tolerance for the set pressure (which is typically around 3%) and this must be observed. It is also important that the environment is clean, dust free and relatively quiet.

The source of the setting fluid can vary from a compressed air cylinder to an intensifier and accumulator vessel running off an industrial compressed air main. In the latter case, the air must be clean, oil, and water free.

It is worth noting that there is no requirement for any sort of capacity test. The test stand simply enables the required set pressure to be ascertained. Usually this point is established by listening for an audible 'hiss' as the set point is reached. When making adjustments it is imperative for both metal seated and soft seated valves that the disc is not allowed to turn on the seat or nozzle, since this can easily cause damage and prevent a good shut-off being achieved. The stem should therefore be gripped whilst the adjuster is turned.

There is a fundamental difference in the allowable setting procedures for ASME I steam boiler valves. In order to maintain the National Board approval and to apply the 'V' stamp to the valve body, these valves must be set using steam on a rig capable not only of achieving the desired set pressure but also with sufficient capacity to demonstrate the popping point and reseat point. This must be done in accordance with an approved, and controlled, quality procedure. For ASME VIII valves (stamped on the body with 'UV'), if the setter has a steam setting facility, then these valves must also be set on steam. If not, then gas or air setting is permissible. For liquid applications with ASME VIII valves, the appropriate liquid, usually water, must be used for setting purposes.

In the case of valves equipped with blowdown rings, the set positions will need to be established and the locking pins sealed in accordance with the relevant manufacturer's recommendations.

Sealing

For valves not claiming any particular standard and with no reference to a standard on the name-plate or supporting literature there is no restriction on who can set the valve. Such valves are normally used to indicate that a certain pressure has been reached, and do not act as a safety device.

For valves that are independently approved by a notified body, to a specific standard, the setting and sealing of the valve is a part of the approval. In this case, the valve must be set by the manufacturer or an approved agent of the manufacturer working in accordance with agreed quality procedures and using equipment approved by the manufacturer or the notified body.

To prevent unauthorised alteration or tampering, most standards require provision to be made for sealing the valve after setting.

The most common method is to use sealing wire to secure the cap to the spring housing and the housing to the body. It may also be used to lock any blowdown adjuster ring pins into position.

The wire is subsequently sealed with a lead seal, which may bear the imprint of the setter's trademark.

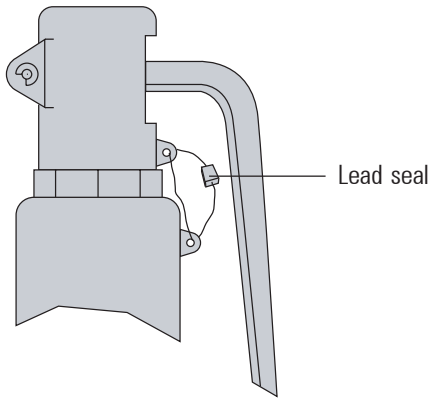


Fig. 9.3.2 Sealed cap showing a lead seal

Safety valve positioning

In order to ensure that the maximum allowable accumulation pressure of any system or apparatus protected by a safety valve is never exceeded, careful consideration of the safety valve's position in the system has to be made. As there is such a wide range of applications, there is no absolute rule as to where the valve should be positioned and therefore, every application needs to be treated separately.

A common steam application for a safety valve is to protect process equipment supplied from a pressure reducing station. Two possible arrangements are shown in Figure 9.3.3.

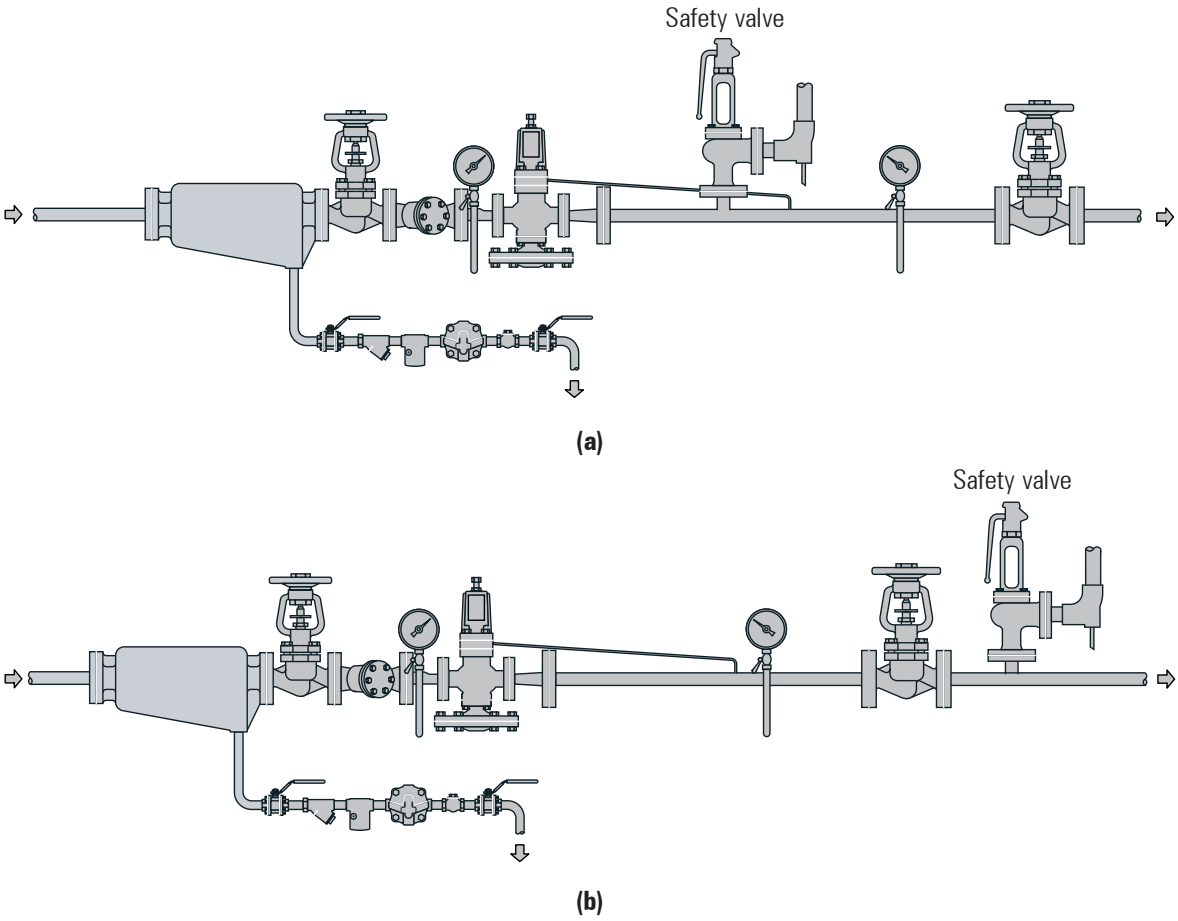


Fig. 9.3.3 Possible positioning of a safety valve in a pressure reducing station

The safety valve can be fitted within the pressure reducing station itself, that is, before the downstream stop valve, as in Figure 9.3.3 (a), or further downstream, nearer the apparatus as in Figure 9.3.3 (b). Fitting the safety valve before the downstream stop valve has the following advantages:

- The safety valve can be tested in-line by shutting down the downstream stop valve without the chance of downstream apparatus being over pressurised, should the safety valve fail under test.
- When the testing is carried out in-line, the safety valve does not have to be removed and bench tested, which is more costly and time consuming.
- When setting the PRV under no-load conditions, the operation of the safety valve can be observed, as this condition is most likely to cause 'simmer'. If this should occur, the PRV pressure can be adjusted to below the safety valve reseat pressure.
- Any additional take-offs downstream are inherently protected. Only apparatus with a lower MAWP requires additional protection. This can have significant cost benefits.

It is however sometimes practical to fit the safety valve closer to the steam inlet of any apparatus. Indeed, a separate safety valve may have to be fitted on the inlet to each downstream piece of apparatus, when the PRV supplies several such pieces of apparatus.

The following points can be used as a guide:

- If supplying one piece of apparatus, which has a MAWP pressure less than the PRV supply pressure, the apparatus must be fitted with a safety valve, preferably close-coupled to its steam inlet connection.
- If a PRV is supplying more than one apparatus and the MAWP of any item is less than the PRV supply pressure, either the PRV station must be fitted with a safety valve set at the lowest possible MAWP of the connected apparatus, or each item of affected apparatus must be fitted with a safety valve.
- The safety valve must be located so that the pressure cannot accumulate in the apparatus via another route, for example, from a separate steam line or a bypass line.

It could be argued that every installation deserves special consideration when it comes to safety, but the following applications and situations are a little unusual and worth considering:

- **Fire -** Any pressure vessel should be protected from overpressure in the event of fire. Although a safety valve mounted for operational protection may also offer protection under fire conditions, such cases require special consideration, which is beyond the scope of this text.
- **Exothermic applications -** These must be fitted with a safety valve close-coupled to the apparatus steam inlet or the body direct. No alternative applies.
- **Safety valves used as warning devices -** Sometimes, safety valves are fitted to systems as warning devices. They are not required to relieve fault loads but to warn of pressures increasing above normal working pressures for operational reasons only. In these instances, safety valves are set at the warning pressure and only need to be of minimum size. If there is any danger of systems fitted with such a safety valve exceeding their maximum allowable working pressure, they must be protected by additional safety valves in the usual way.

Example 9.3.2

In order to illustrate the importance of the positioning of a safety valve, consider an automatic pump trap (see Block 14) used to remove condensate from a heating vessel. The automatic pump trap (APT), incorporates a mechanical type pump, which uses the motive force of steam to pump the condensate through the return system. The position of the safety valve will depend on the MAWP of the APT and its required motive inlet pressure.

If the MAWP of the APT is more than or equal to that of the vessel, the arrangement shown in Figure 9.3.4 could be used.

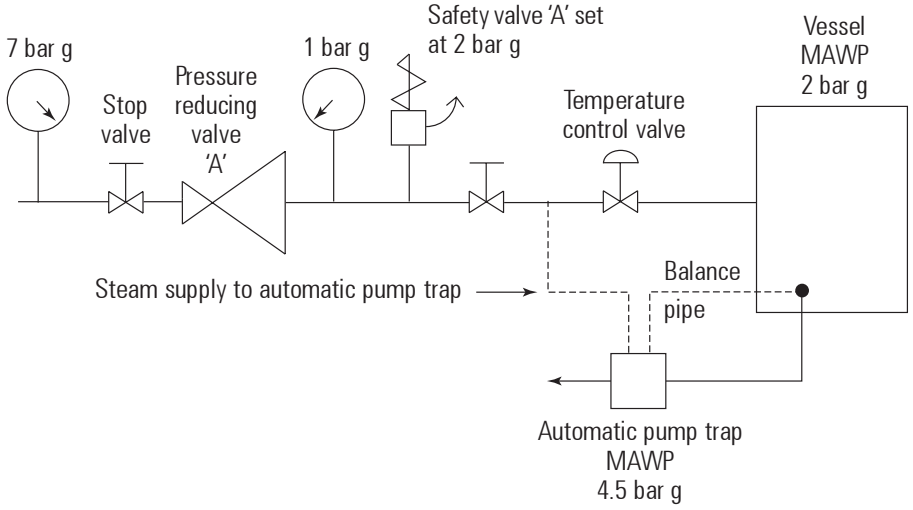


Fig. 9.3.4 Pressure reducing station arrangement for automatic pump trap and process vessel system

This arrangement is suitable if the pump-trap motive pressure is less than 1.6 bar g (safety valve set pressure of 2 bar g less 0.3 bar blowdown and a 0.1 bar shut-off margin). Since the MAWP of both the APT and the vessel are greater than the safety valve set pressure, a single safety valve would provide suitable protection for the system.

However, if the pump-trap motive pressure had to be greater than 1.6 bar g, the APT supply would have to be taken from the high pressure side of the PRV, and reduced to a more appropriate pressure, but still less than the 4.5 bar g MAWP of the APT. The arrangement shown in Figure 9.3.5 would be suitable in this situation.

Here, two separate PRV stations are used each with its own safety valve. If the APT internals failed and steam at 4 bar g passed through the APT and into the vessel, safety valve 'A' would relieve this pressure and protect the vessel. Safety valve 'B' would not lift as the pressure in the APT is still acceptable and below its set pressure.

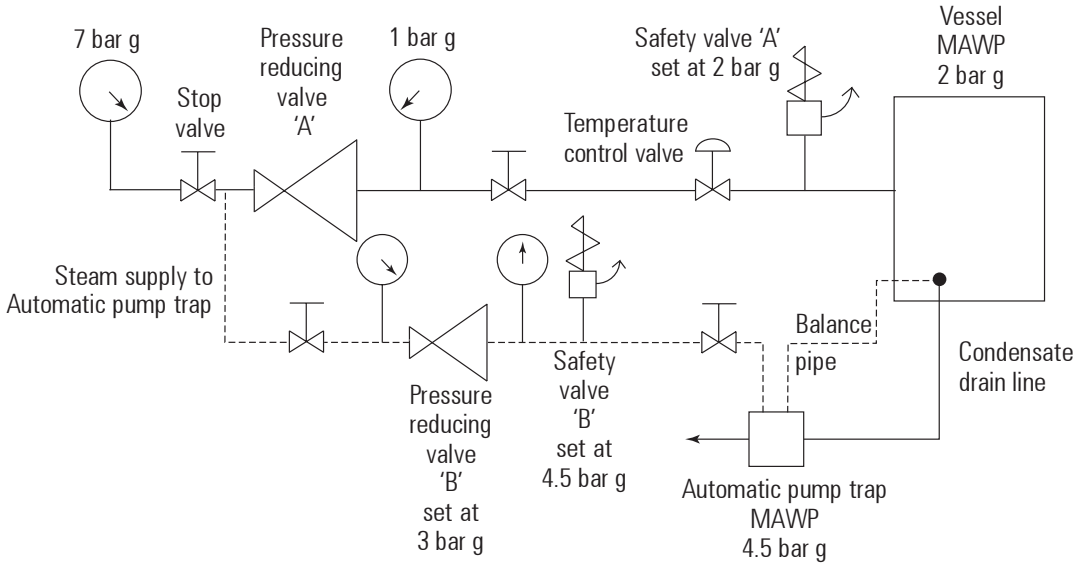


Fig. 9.3.5 The automatic pump trap and vessel system using two PRV stations

It should be noted that safety valve 'A' is positioned on the downstream side of the temperature control valve; this is done for both safety and operational reasons:

- **Safety -** If the internals of the APT failed, the safety valve would still relieve the pressure in the vessel even if the control valve were shut.
- **Operation -** There is less chance of safety valve 'A' simmering during operation in this position, as the pressure is typically lower after the control valve than before it.

Also, note that if the MAWP of the pump-trap were greater than the pressure upstream of PRV 'A', it would be permissible to omit safety valve 'B' from the system, but safety valve 'A' must be sized to take into account the total fault flow through PRV 'B' as well as through PRV 'A'.

Example 9.3.3
A pharmaceutical factory has twelve jacketed pans on the same production floor, all rated with the same MAWP. Where would the safety valve be positioned?

One solution would be to install a safety valve on the inlet to each pan (Figure 9.3.6). In this instance, each safety valve would have to be sized to pass the entire load, in case the PRV failed open whilst the other eleven pans were shut down.

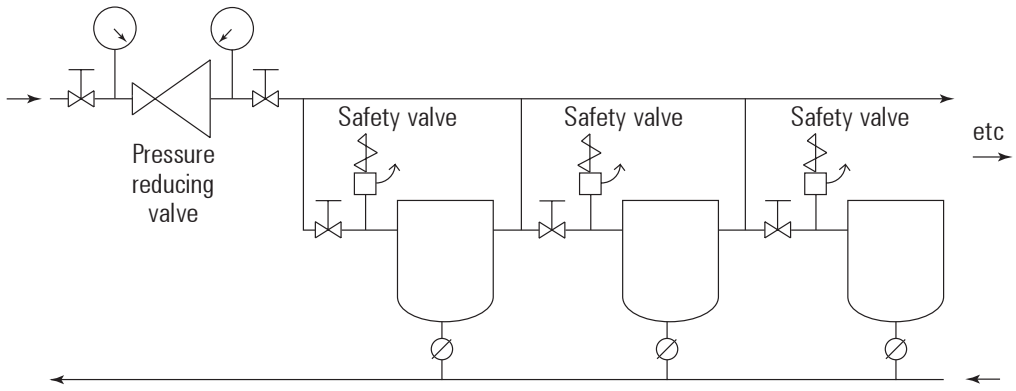


Fig. 9.3.6 Protection of the heating pans using individual safety valves

As all the pans are rated to the same MAWP, it is possible to install a single safety valve after the PRV.

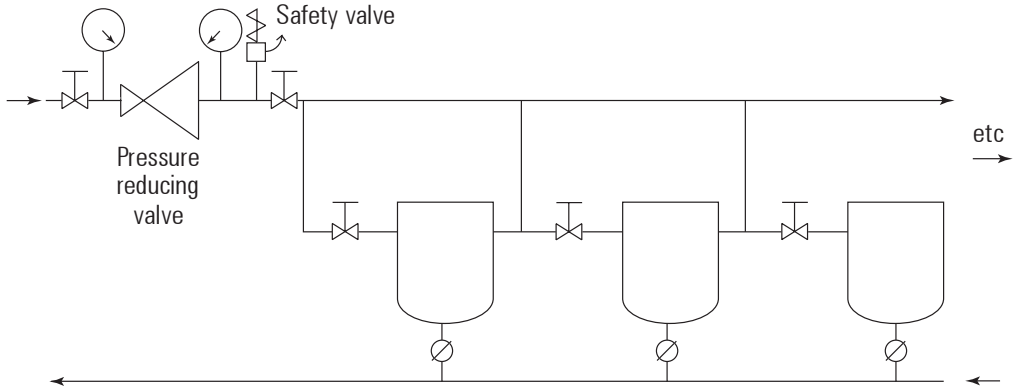


Fig. 9.3.7 Protection of heating pans using a single safety valve

If additional apparatus with a lower MAWP than the pans (for example, a shell and tube heat exchanger) were to be included in the system, it would be necessary to fit an additional safety valve. This safety valve would be set to an appropriate lower set pressure and sized to pass the fault flow through the temperature control valve (see Figure 9.3.8).

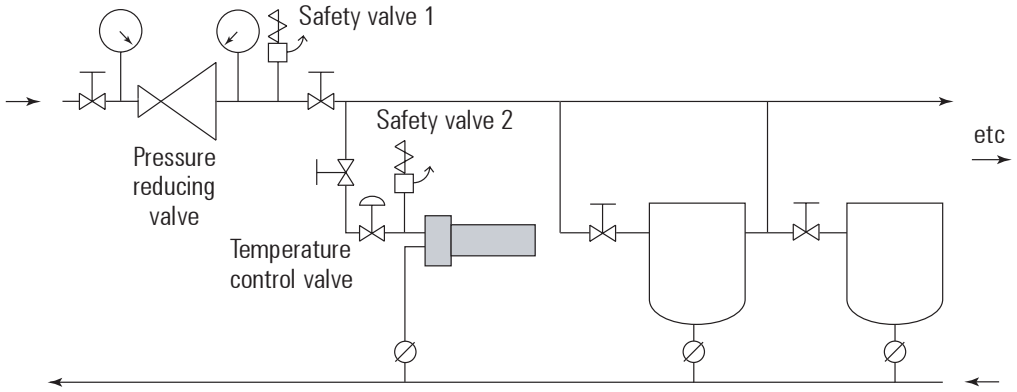


Fig. 9.3.8 Possible safety valve arrangement if additional apparatus was included in the system

Questions

1. Which of the following are the most important criteria in determining the set pressure of a safety valve?

i. The MAWP of the system must never be exceeded

ii. The MAAP of the system must never be exceeded

iii. The NWP of the system must never be exceeded

a| i only ☐
b| ii only ☐
c| i and ii ☐
d| i, ii and iii ☐

2. The manufacturer of a heating vessel states that the maximum allowable working pressure (MAWP) of the vessel is 6.0 bar g, and the maximum allowable accumulation pressure is 6.3 bar g (5% of the MAWP).

If a safety valve used to protect the vessel has an overpressure of 10%, which set pressure would be selected?

a| 5.7 bar ☐
b| 6.0 bar ☐
c| 6.3 bar ☐
d| 6.5 bar ☐

3. Determine the set pressure of a safety valve to be installed in a pressure reducing valve station, given the following conditions and ensuring that the set pressure is as close to the PRV working pressure as possible:

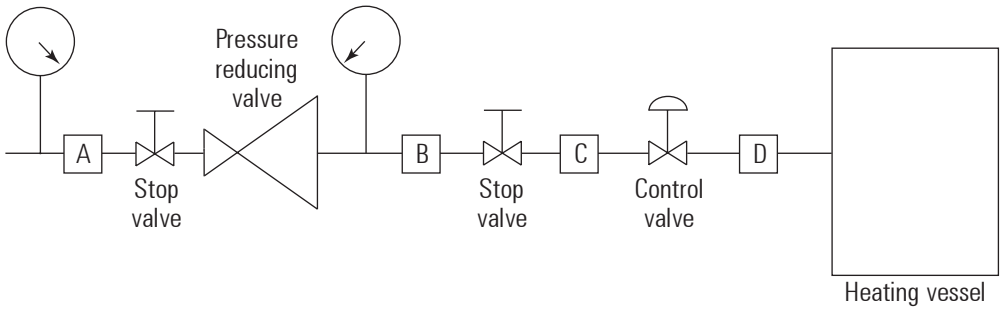
Normal working pressure	**7.4 bar g**
PRV proportional band	**0.2 bar**
Blowdown	**5%**
MAWP of downstream apparatus	**8.5 bar g**

a| 8.0 bar ☐
b| 8.1 bar ☐
c| 8.2 bar ☐
d| 8.5 bar ☐

4. The required set pressure of a conventional safety valve is 8.5 bar g, if however, the valve experiences a constant backpressure of 1.0 bar g, at which pressure should the valve be set on the test stand?

a| 7.5 bar ☐
b| 8.5 bar ☐
c| 9.5 bar ☐
d| 10.5 bar ☐

5. Which location would be the most appropriate position for a safety valve installed to protect a single temperature controlled heating vessel?



a| A ☐
b| B ☐
c| C ☐
d| D ☐

6. Who is permitted to adjust the settings of a safety valve approved by a notified body, to a specific standard?

a| Any suitable person provided with the necessary tools ☐
b| Only the certifying body ☐
c| Only the manufacturer ☐
d| The manufacturer or an agent approved by the manufacturer ☐

Answers

1:b, 2: a, 3: b, 4: a, 5: d, 6: d

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Module 9.4

Safety Valve Sizing

Safety Valve Sizing

A safety valve must always be sized and able to vent any source of steam so that the pressure within the protected apparatus cannot exceed the maximum allowable accumulated pressure (MAAP). This not only means that the valve has to be positioned correctly, but that it is also correctly set. The safety valve must then also be sized correctly, enabling it to pass the required amount of steam at the required pressure under all possible fault conditions.

Once the type of safety valve has been established, along with its set pressure and its position in the system, it is necessary to calculate the required discharge capacity of the valve. Once this is known, the required orifice area and nominal size can be determined using the manufacturer's specifications.

In order to establish the maximum capacity required, the potential flow through all the relevant branches, upstream of the valve, need to be considered.

In applications where there is more than one possible flow path, the sizing of the safety valve becomes more complicated, as there may be a number of alternative methods of determining its size. Where more than one potential flow path exists, the following alternatives should be considered:

- The safety valve can be sized on the maximum flow experienced in the flow path with the greatest amount of flow.
- The safety valve can be sized to discharge the flow from the combined flow paths.

This choice is determined by the risk of two or more devices failing simultaneously. If there is the slightest chance that this may occur, the valve must be sized to allow the combined flows of the failed devices to be discharged. However, where the risk is negligible, cost advantages may dictate that the valve should only be sized on the highest fault flow. The choice of method ultimately lies with the company responsible for insuring the plant.

For example, consider the pressure vessel and automatic pump-trap (APT) system as shown in Figure 9.4.1. The unlikely situation is that both the APT and pressure reducing valve (PRV 'A') could fail simultaneously. The discharge capacity of safety valve 'A' would either be the fault load of the largest PRV, or alternatively, the combined fault load of both the APT and PRV 'A'.

This document recommends that where multiple flow paths exist, any relevant safety valve should, at all times, be sized on the possibility that relevant upstream pressure control valves may fail simultaneously.

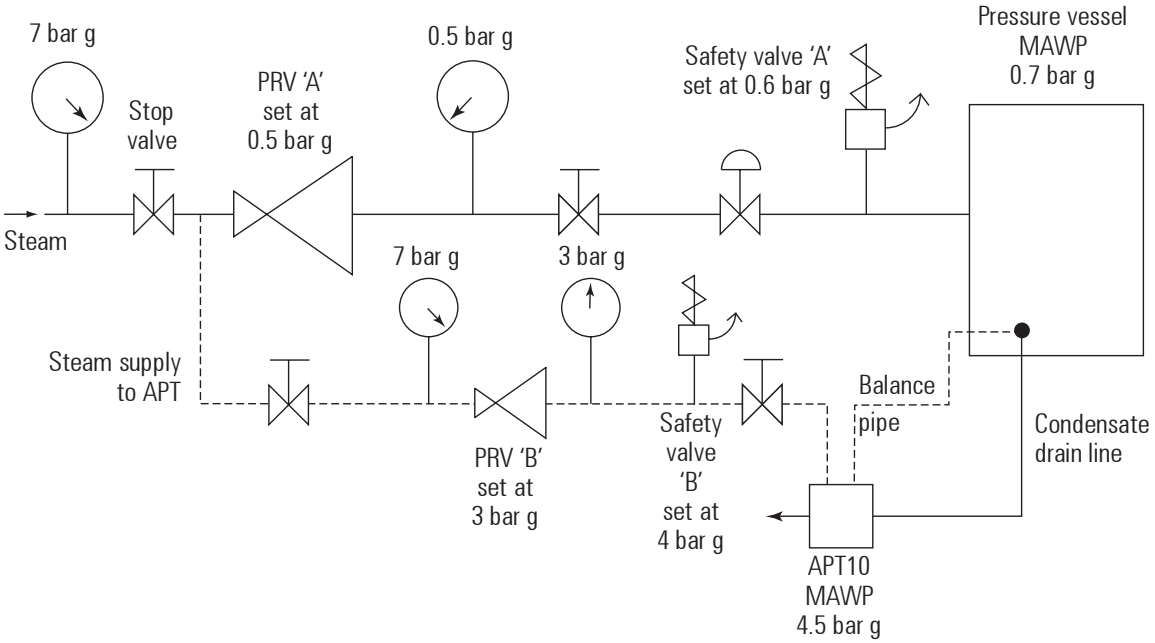


Fig. 9.4.1 An automatic pump-trap and pressure vessel system

Finding the fault flow

In order to determine the fault flow through a PRV or indeed any valve or orifice, the following need to be considered:

- The potential fault pressure - this should be taken as the set pressure of the appropriate upstream safety valve
- The relieving pressure of the safety valve being sized
- The full open capacity (K_{VS}) of the upstream control valve, see Equation 3.21.2

Example 9.4.1

Consider the PRV arrangement in Figure 9.4.2.

Where:

NWP = Normal working pressure
MAAP = Maximum allowable accumulated pressure
P_S = Safety valve set pressure
P_o = Safety valve overpressure
P_R = Safety valve relieving pressure

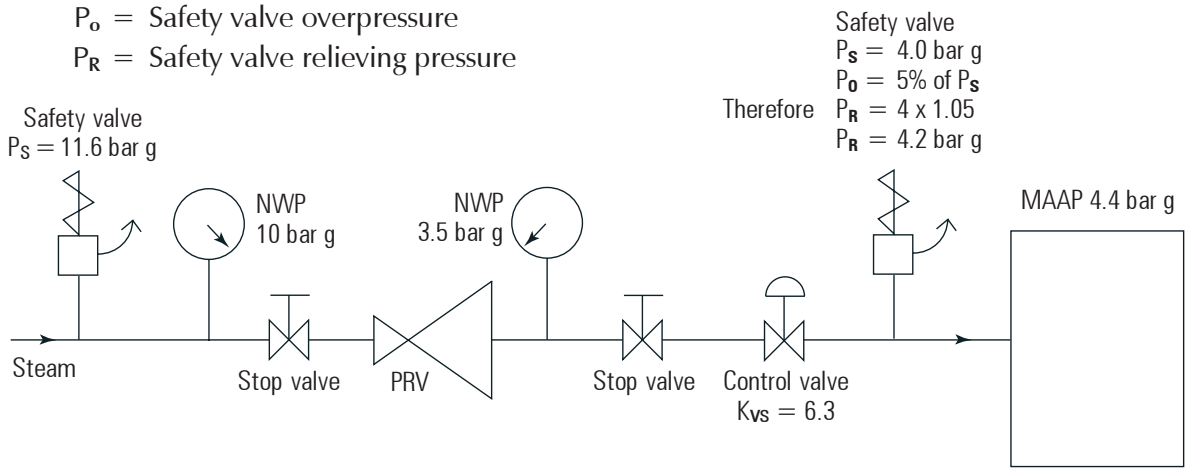


Fig. 9.4.2 Sizing a safety valve for a typical pressure reducing application

The supply pressure of this system (Figure 9.4.2) is limited by an upstream safety valve with a set pressure of 11.6 bar g. The fault flow through the PRV can be determined using the steam mass flow equation (Equation 3.21.2):

$$\dot{m}_s = 12\ K_V\ P_1 \sqrt{1 - 5.67\ (0.42 - \chi)^2} \qquad \textbf{Equation 3.21.2}$$

Where:

$\dot{m}_s$ = Fault load (kg/h)

K_V = PRV full open capacity index (K_{VS} = 6.3)

χ = Pressure drop = $\dfrac{P_1 - P_2}{P_1}$

P_1 = Fault pressure (taken as the set pressure of the upstream safety valve) (bar a)

P_2 = Relieving pressure of the apparatus safety valve (bar a)

Equation 3.21.2 is used when the pressure drop ratio is less than 0.42.

If the pressure drop ratio is 0.42 or greater, the mass flow is calculated using Equation 6.4.3

$$\dot{m}_s = 12\ K_V\ P_1 \qquad \textbf{Equation 6.4.3}$$

In this example:

$P_1 = 11.6$ bar g $= 12.6$ bar a

$P_2 = 4.2$ bar g $= 5.2$ bar a

Therefore: $\chi = \frac{12.6 - 5.2}{12.6} = 0.59$

Since χ is greater than 0.42, critical pressure drop occurs across the control valve, and the fault flow is calculated as follows using the formula in Equation 6.4.3:

$\dot{m}_s = 12\ K_V\ P_1$

$\dot{m}_s = 12 \times 6.3 \times 12.6$

Therefore: $\dot{m}_s$ **= 953 kg/h**

Consquently, the safety valve would be sized to pass at least 953 kg/h when set at 4 bar g.

Once the fault load has been determined, it is usually sufficient to size the safety valve using the manufacturer's capacity charts. A typical example of a capacity chart is shown in Figure 9.4.3. By knowing the required set pressure and discharge capacity, it is possible to select a suitable nominal size. In this example, the set pressure is 4 bar g and the fault flow is 953 kg/h. A DN32/50 safety valve is required with a capacity of 1 284 kg/h.

SV615 flow capacity for saturated steam in kilogrammes per hour (kg/h)

(calculated in accordance with BS 6759 at 5% overpressure)

Derated coefficient of discharge (K_{dr}) = 0.71

Valve size DN	15/20	20/32	25/40	32/50	40/65	50/80
Area (mm^2)	113	314	452	661	1 075	1 662
Set pressure (bar g)	Flow capacity for saturated steam kg/h					
0.5	65	180	259	379	616	953
1.0	87	241	348	508	827	1278
1.5	109	303	436	638	1037	1603
2.0	131	364	524	767	1247	1929
2.5	153	426	613	896	1458	2254
3.0	175	487	701	1026	1668	2579
3.5	197	549	790	1155	1879	2904
4.0	220	610	878	1284	2089	3230
4.5	242	672	967	1414	2299	3555
5.0	264	733	1055	1543	2510	3880
5.5	286	794	1144	1672	2720	4205
6.0	308	856	1232	1802	2930	4530
6.5	330	917	1321	1931	3141	4856
7.0	352	979	1409	2061	3351	5181
7.5	374	1040	1497	2190	3561	5506
8.0	396	1102	1586	2319	3772	5831

Fig. 9.4.3 A typical safety valve capacity chart

Where sizing charts are not available or do not cater for particular fluids or conditions, such as backpressure, high viscosity or two-phase flow, it may be necessary to calculate the minimum required orifice area. Methods for doing this are outlined in the appropriate governing standards, such as:

- AD-Merkblatt A2, DIN 3320, TRD 421
- ASME/API RP 520
- BS 6759 for steam, air/gases and liquids
- EN ISO 4126

The methods outlined in these standards are based on the coefficient of discharge, which is the ratio of the measured capacity to the theoretical capacity of a nozzle with an equivalent flow area.

$$K_d = \frac{\text{Actual flowing capacity}}{\text{Theoretical flowing capacity}} \qquad \textbf{Equation 9.4.1}$$

Where:

K_d = Coefficient of discharge

Coefficient of discharge

Coefficients of discharge are specific to any particular safety valve range and will be approved by the manufacturer. If the valve is independently approved, it is given a 'certified coefficient of discharge'.

This figure is often derated by further multiplying it by a safety factor 0.9, to give a derated coefficient of discharge. Derated coefficient of discharge is termed $K_{dr} = K_d \times 0.9$

When using standard methods of calculating the required orifice area, the following points may need to be considered:

- **Critical and sub-critical flow -** the flow of gas or vapour through an orifice, such as the flow area of a safety valve, increases as the downstream pressure is decreased. This holds true until the critical pressure is reached, and critical flow is achieved. At this point, any further decrease in the downstream pressure will not result in any further increase in flow. A relationship (called the critical pressure ratio) exists between the critical pressure and the actual relieving pressure, and, for gases flowing through safety valves, is shown by Equation 9.4.2.

$$\frac{P_B}{P_1} = \left(\frac{2}{k+1}\right)^{\left(\frac{k}{k-1}\right)} \qquad \textbf{Equation 9.4.2}$$

  Where:

  P_B = Critical backpressure (bar a)

  P_1 = Actual relieving pressure (bar a)

  k = Isentropic coefficient of the gas or vapour at the relieving conditions

  For gases, with similar properties to an ideal gas, 'k' is the ratio of specific heat of constant pressure (c_p) to constant volume (c_v), i.e. $c_p : c_v$. 'k' is always greater than unity, and typically between 1 and 1.4 (see Table 9.4.8).

  For steam, although 'k' is an isentropic coefficient, it is not actually the ratio of $c_p : c_v$. As an approximation for saturated steam, 'k' can be taken as 1.135, and superheated steam, as 1.3. As a guide, for saturated steam, critical pressure is taken as 58% of accumulated inlet pressure in absolute terms.

- **Overpressure -** Before sizing, the design overpressure of the valve must be established. It is not permitted to calculate the capacity of the valve at a lower overpressure than that at which the coefficient of discharge was established. It is however, permitted to use a higher overpressure (see Table 9.2.1, Module 9.2, for typical overpressure values). For DIN type full lift (Vollhub) valves, the design lift must be achieved at 5% overpressure, but for sizing purposes, an overpressure value of 10% may be used.

  For liquid applications, the overpressure is 10% according to AD-Merkblatt A2, DIN 3320, TRD 421 and ASME, but for non-certified ASME valves, it is quite common for a figure of 25% to be used.

- **Backpressure -** The sizing calculations in the AD-Merkblatt A2, DIN 3320 and TRD 421 standards account for backpressure in the outflow function,(Ψ), which includes a backpressure correction. The ASME/API RP 520 and BS 6759 standards, however, require an additional backpressure correction factor to be determined and then incorporated in the relevant equation.
- **Two-phase flow -** When sizing safety valves for boiling liquids (e.g. hot water) consideration must be given to vaporisation (flashing) during discharge. It is assumed that the medium is in liquid state when the safety valve is closed and that, when the safety valve opens, part of the liquid vaporises due to the drop in pressure through the safety valve. The resulting flow is referred to as two-phase flow.

  The required flow area has to be calculated for the liquid and vapour components of the discharged fluid. The sum of these two areas is then used to select the appropriate orifice size from the chosen valve range. (see Example 9.4.3)

  Many standards do not actually specify sizing formula for two-phase flow and recommend that the manufacturer be contacted directly for advice in these instances.

Sizing equations for safety valves designed to the following standards

The following methods are used to calculate the minimum required orifice area for a safety valve, as mentioned in the most commonly used national standards.

Standard - AD-Merkblatt A2, DIN 3320, TRD 421

Use Equation 9.4.3 to calculate the minimum required orifice area for a safety valve used on **steam applications:**

$$A_O = \frac{\chi \dot{m}}{\alpha_w P_R} \qquad \textbf{Equation 9.4.3}$$

Use Equation 9.4.4 to calculate the minimum required orifice area for a safety valve used on **air and gas applications:**

$$A_O = \frac{0.1791 \dot{m}}{\Psi \alpha_w P_R} \sqrt{\frac{T Z}{M}} \qquad \textbf{Equation 9.4.4}$$

Use Equation 9.4.5 to calculate the minimum required orifice area for a safety valve used on **liquid applications:**

$$A_O = \frac{0.6211 \dot{m}}{\alpha_w \sqrt{\rho \Delta P}} \qquad \textbf{Equation 9.4.5}$$

Where:

A_O = Minimum cross sectional flow area (mm^2)
$\dot{m}$ = Mass flow to be discharged (kg/h)
P_R = Absolute relieving pressure (bar a)
ΔP = P_R - P_B
P_B = Absolute backpressure (bar a)
T = Inlet temperature (K)
ρ = Density (kg/m^3) (see Appendix A at the back of this module)
M = Molar mass (kg/kmol) (see Appendix A at the back of this module)
Z = Compressibility factor (see Equation 9.4.6)
α_W = Outflow coefficient (specified by the manufacturer)
Ψ = Outflow function (see Figure 9.4.4)
χ = Pressure medium coefficient (see Figure 9.4.5)

The outflow function (Ψ) for air and gas applications

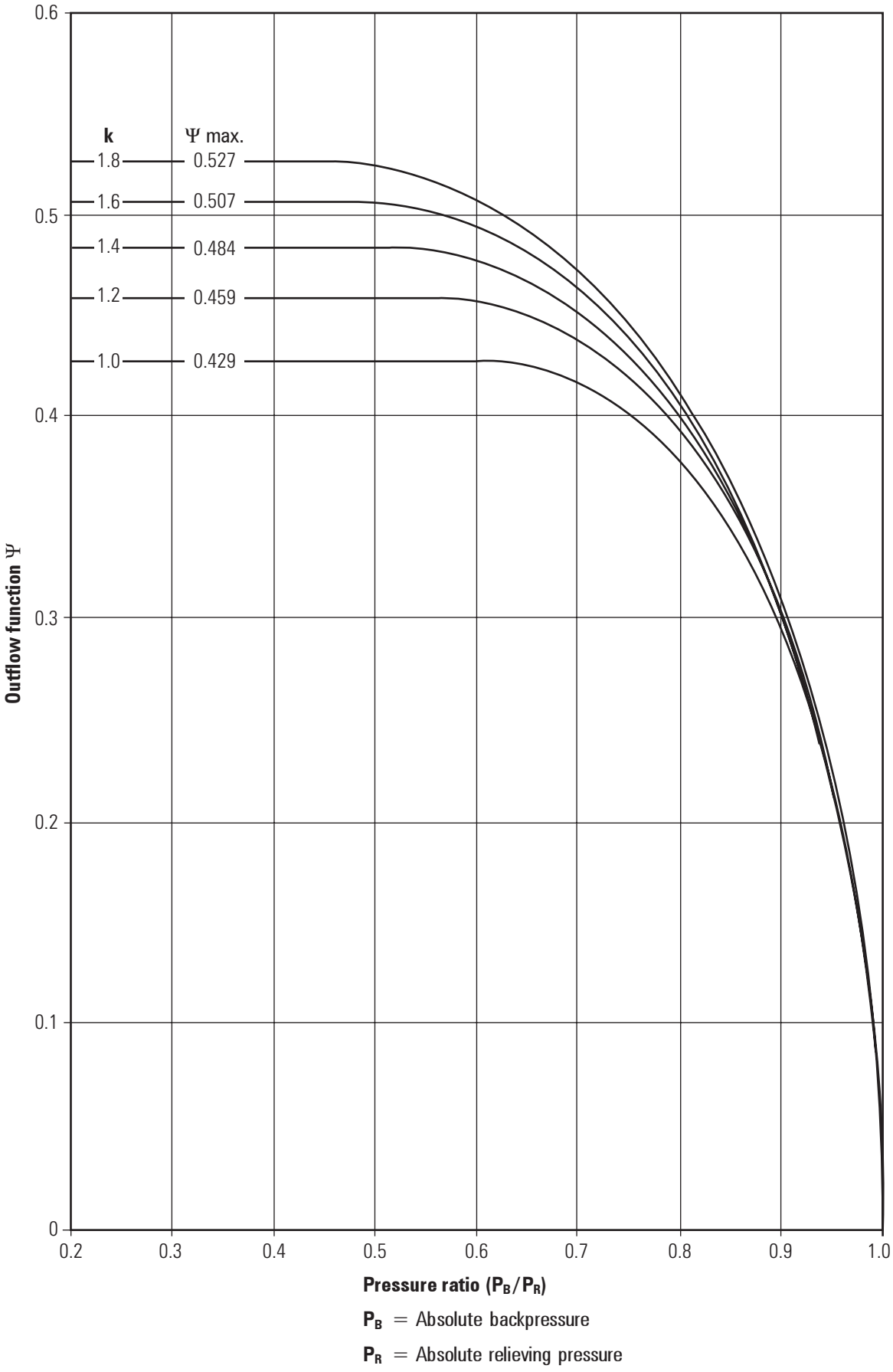


Fig. 9.4.4 The outflow function (Ψ) as used in AD-Merkblatt A2, DIN 3320 and TRD 421

Pressure medium coefficient (χ) for steam applications

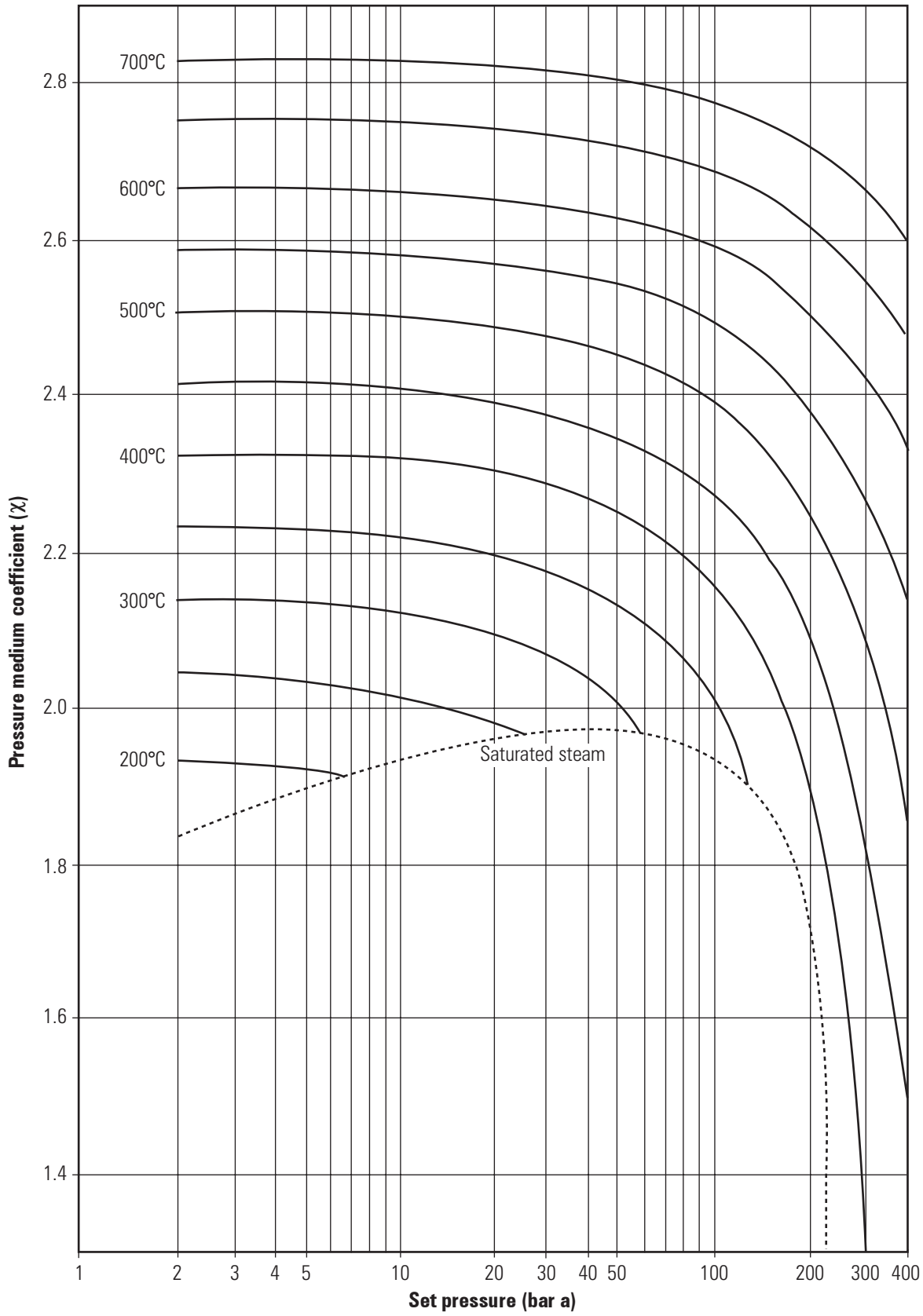


Fig. 9.4.5 Pressure medium coefficient (χ) for steam as used in AD-Merkblatt A2, DIN 3320, TRD 421

Compressibility factor (Z)

For gases, the compressibility factor, Z, also needs to be determined. This factor accounts for the deviation of the actual gas from the characteristics of an ideal gas. It is often recommended that Z = 1 is used where insufficient data is available. Z can be calculated by using the formula in Equation 9.4.6:

$$Z = \frac{10^5 \, P_R \, M \, \nu}{R_u \, T} \qquad \textbf{Equation 9.4.6}$$

Where:

Z = Compressibility factor

P_R = Safety valve relieving pressure (bar a)

ν = Specific volume of the gas at the actual relieving pressure and temperature (m^3/kg) (see Appendix A at the back of this module). Note: The specific volume of a gas will change with temperature and pressure, and therefore it must be determined for the operating conditions.

M = Molar mass (kg/kmol) (see Appendix A at the back of this module)

R_u = Universal gas constant (8 314 Nm/kmol K)

T = Actual relieving temperature (K)

Example 9.4.2

Determine the minimum required safety valve orifice area under the following conditions:

Medium:	Saturated steam
Discharge quantity ($\dot{m}$):	2 500 kg/h
Set pressure (P_s):	4 bar a
Backpressure:	Atmospheric pressure 1 bar a
Stated outflow coefficient (α_w):	0.7

It is first necessary to determine the pressure medium coefficient using Figure 9.4.5.

Pressure medium coefficient (χ): 1.88

Using Equation 9.4.3: $A_O = \dfrac{\chi \times \dot{m}}{\alpha_w \times P_s}$

Therefore: $\mathbf{A_O = \dfrac{1.88 \times 2500}{0.7 \times 4} = 1678\ mm}$

Consequently, the chosen safety valve would need an orifice area of at least 1 678 mm^2.

Two-phase flow

In order to determine the minimum orifice area for a two-phase flow system (e.g. hot water), it is first necessary to establish what proportion of the discharge will be vapour (n). This is done using the Equation 9.4.7:

$$n = \frac{h_{f1} - h_{f2}}{h_{fg2}} \qquad \textbf{Equation 9.4.7}$$

Where:

n = The proportion of discharge fluid which is vapour
h_{f1} = Enthalpy of liquid before the valve (kJ/kg)
h_{f2} = Enthalpy of liquid after the valve (kJ/kg)
h_{fg2} = Enthalpy of evaporation after the valve (kJ/kg)

For hot water, the enthalpy values can be obtained from steam tables.

In order to determine the proportion of flow, which is vapour, the discharge capacity is multiplied by n. The remainder of the flow will therefore be in the liquid state.

The area sizing calculation from Equations 9.4.3, 9.4.4 and 9.4.5 can then be used to calculate the required area to discharge the vapour portion and then the liquid portion. The sum of these areas is then used to establish the minimum required orifice area.

Example 9.4.3

Consider hot water under the following conditions:

Temperature:	160°C
Discharge quantity ($\dot{m}$):	3 900 kg/h
Set pressure (P_S):	10 bar g = 11 bar a
Backpressure (P_B):	Atmospheric
Density of water at 160°C (ρ):	908 kg/m³
$\Delta P = P_S - P_B$:	10 bar
Stated outflow coefficient (α_w):	0.7

Using steam tables, the proportion of vapour is first calculated:

h_{f1} = 675 kJ/kg (at 160°C)
h_{f2} = 417 kJ/kg (at 1 bar a, atmospheric pressure)
h_{fg2} = 2 258 kJ/kg (at 1 bar a, atmospheric pressure)

Using Equation 9.4.7: $n = \frac{h_{f1} - h_{f2}}{h_{fg2}}$

Therefore: $n = \frac{675 - 417}{2\,258} = 0.114\,3$

Capacity discharge as vapour (steam) = 0.114 3 x 3 900 kg/h = 446 kg/h
Capacity discharge as liquid (water) = 3 900 kg/h - 446 kg/h = 3 454 kg/h

Calculated area for vapour portion:

Using Equation 9.4.3: $A_O = \frac{\chi\,\dot{m}}{\alpha_w\,P_S}$ (where χ = Pressure medium coefficient at the set pressure)

Therefore: $A_{O\,Steam} = \frac{1.92 \times 446}{0.7 \times 11} = 111\ mm^2$

Calculated area for liquid portion:

Using Equation 9.4.5: $$A_O = \frac{0.6211 \times \dot{m}}{\alpha_w \sqrt{\rho \ \Delta P}}$$

$$\text{Therefore: } A_{O\,liquid} = \frac{0.6211 \times 3\,454}{0.7\sqrt{908 \times 10}} = 33\ mm^2$$

Total required discharge area = 111 + 33 = 144 mm²

Therefore, a valve must be selected with a discharge area greater than 144 mm².

Standard - ASME/API RP 520

The following formulae are used for calculating the minimum required orifice area for a safety valve according to ASME standards and the API RP 520 guidelines.

Use Equation 9.4.8 to calculate the minimum required orifice area for a safety valve used on **steam applications:**

$$A_O = \frac{\dot{m}}{51.5\ P_R\ K_d\ K_{SH}} \qquad \textbf{Equation 9.4.8}$$

Use Equation 9.4.9 to calculate the minimum required orifice area for a safety valve used on **air and gas applications:**

$$A_O = \frac{\dot{V}\sqrt{T\ Z\ G}}{1.175\ C_g\ K_d\ P_R\ K_B} \qquad \textbf{Equation 9.4.9}$$

Use Equation 9.4.10 to calculate the minimum required orifice area for a safety valve used on **liquid applications:**

$$A_O = \frac{\dot{V}^1}{38\ K_d\ K_\mu\ K_w}\ \sqrt{\frac{G}{P_R - P_B}} \qquad \textbf{Equation 9.4.10}$$

Where:

A_O = Required effective discharge area (in²)
$\dot{m}$ = Required mass flow through the valve (lb/h)
$\dot{V}$ = Required volume flow through the valve (ft³/min)
$\dot{V}^1$ = Required volume flow through the valve (U.S. gal/min)
P_R = Upstream relieving pressure (psi a)
P_B = Absolute backpressure (psi a)
C_g = Nozzle gas constant (see Table 9.4.1)
T = Relieving temperature (°R ≡ °F + 460)
G = Specific gravity (ratio of molar mass of the fluid to the molar mass of air (28.96 kg/kmol)) (see Appendix A at the back of this module)
Z = Compressibility factor (see Equation 9.4.6)
K_d = Effective coefficient of discharge (specified by the manufacturer)
K_{SH} = Superheat correction factor (see Table 9.4.2)
K_B = Backpressure correction factor for gas and vapour (see Figures 9.4.6 and 9.4.7)
K_W = Backpressure correction factor for liquids (bellows balanced valves only) (see Figure 9.4.8)
K_μ = Viscosity factor (see Figure 9.4.9)

Nozzle gas constant for ASME / API RP 520

Table 9.4.1 Nozzle gas constant (C_g) relative to isentropic constant (k) for air and gases

k	C_g	k	C_g	k	C_g	k	C_g
1.01	317	1.26	343	1.51	365	1.76	384
1.02	318	1.27	344	1.52	366	1.77	385
1.03	319	1.28	345	1.53	367	1.78	386
1.04	320	1.29	346	1.54	368	1.79	386
1.05	321	1.30	347	1.55	369	1.80	387
1.06	322	1.31	348	1.56	369	1.81	388
1.07	323	1.32	349	1.57	370	1.82	389
1.08	325	1.33	350	1.58	371	1.83	389
1.09	326	1.34	351	1.59	372	1.84	390
1.10	327	1.35	352	1.60	373	1.85	391
1.11	328	1.36	353	1.61	373	1.86	391
1.12	329	1.37	353	1.62	374	1.87	392
1.13	330	1.38	354	1.63	375	1.88	393
1.14	331	1.39	355	1.64	376	1.89	393
1.15	332	1.40	356	1.65	376	1.90	394
1.16	333	1.41	357	1.66	377	1.91	395
1.17	334	1.42	358	1.67	378	1.92	395
1.18	335	1.43	359	1.68	379	1.93	396
1.19	336	1.44	360	1.69	379	1.94	397
1.20	337	1.45	360	1.70	380	1.95	397
1.21	338	1.46	361	1.71	381	1.96	398
1.22	339	1.47	362	1.72	382	1.97	398
1.23	340	1.48	363	1.73	383	1.98	399
1.24	341	1.49	364	1.74	383	1.99	400
1.25	342	1.50	365	1.75	384	2.00	400

The nozzle gas constant C_g is calculated using Equation 9.4.11, for air and gas applications and applied to Equation 9.4.9.

$$C_g = 520 \sqrt{k \left(\frac{2}{k+1}\right)^{\left(\frac{k+1}{k-1}\right)}} \quad \text{for } k > 1$$

$$C_g = 315 \text{ for } k = 1$$

Equation 9.4.11

Superheat correction factors for ASME / API RP 520

Table 9.4.2 Superheat correction factors (K_{SH}) as used in ASME / API RP 520 (Imperial units)

Set pressure (psi g)	Temperature (°F)									
	300	400	500	600	700	800	900	1 000	1 100	1 200
15	1.00	0.98	0.93	0.88	0.84	0.80	0.77	0.74	0.72	0.70
20	1.00	0.98	0.93	0.88	0.84	0.80	0.77	0.74	0.72	0.70
40	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.74	0.72	0.70
60	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.75	0.72	0.70
80	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.75	0.72	0.70
100	1.00	0.99	0.94	0.89	0.84	0.81	0.77	0.75	0.72	0.70
120	1.00	0.99	0.94	0.89	0.84	0.81	0.78	0.75	0.72	0.70
140	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
160	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
180	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
200	1.00	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.70
220	1.00	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.70
240		1.00	0.95	0.90	0.85	0.81	0.78	0.75	0.72	0.70
260		1.00	0.95	0.90	0.85	0.81	0.78	0.75	0.72	0.70
280		1.00	0.96	0.90	0.85	0.81	0.78	0.75	0.72	0.70
300		1.00	0.96	0.90	0.85	0.81	0.78	0.75	0.72	0.70
350		1.00	0.96	0.90	0.86	0.82	0.78	0.75	0.72	0.70
400		1.00	0.96	0.91	0.86	0.82	0.78	0.75	0.72	0.70
500		1.00	0.96	0.92	0.86	0.82	0.78	0.75	0.73	0.70
600		1.00	0.97	0.92	0.87	0.82	0.79	0.75	0.73	0.70
800			1.00	0.95	0.88	0.83	0.79	0.76	0.73	0.70
1 000			1.00	0.96	0.89	0.84	0.78	0.76	0.73	0.71
1 250			1.00	0.97	0.91	0.85	0.80	0.77	0.74	0.71
1 500			1.00	1.00	0.93	0.86	0.81	0.77	0.74	0.71

Gas and vapour constant backpressure correction factor for ASME/API 520

The backpressure correction factor (K_B) is the ratio of the capacity with backpressure, C_1, to the capacity when discharging to atmosphere, C_2, see Equation 9.4.12.

$$K_B = \frac{C_1}{C_2}$$ **Equation 9.4.12**

The value of K_B can be established using the curves shown in Figure 9.4.6 to Figure 9.4.8. These are applicable to set pressures of 50 psi g and above. For a given set pressure, these values are limited to a backpressure less than the critical pressure, namely, critical flow conditions. For sub-critical flow and backpressures below 50 psi g, the manufacturer should be consulted for values of K_B.

▫ Balanced bellows valves

$$\% \text{ of gauge backpressure} = \frac{P_B}{P_S} \times 100$$ **Equation 9.4.13**

Where:
P_B = Backpressure (psi g)
P_S = Set pressure (psi g)

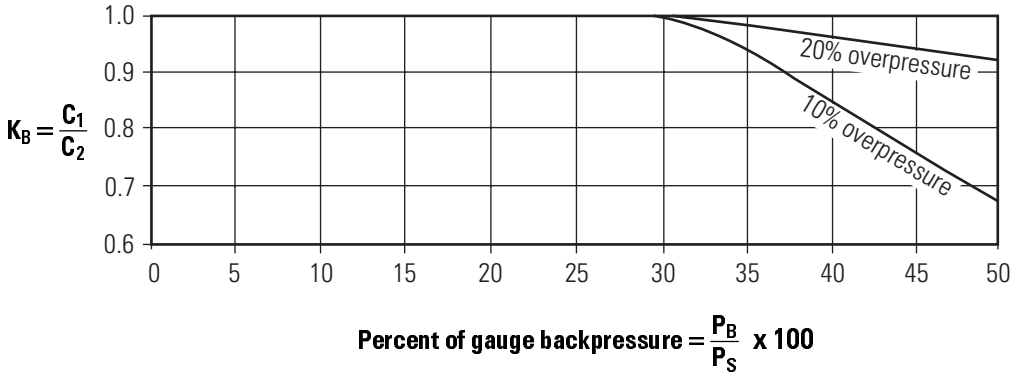


Fig. 9.4.6 Constant backpressure correction factor (K_B) for gas and vapour as used in ASME / API RP 520 for balanced bellows valves

▫ Conventional valves

$$\% \text{ of gauge backpressure} = \frac{P_B}{P_R} \times 100$$ **Equation 9.4.14**

Where:
P_B = Backpressure (psi g)
P_R = Relieving pressure (psi g)

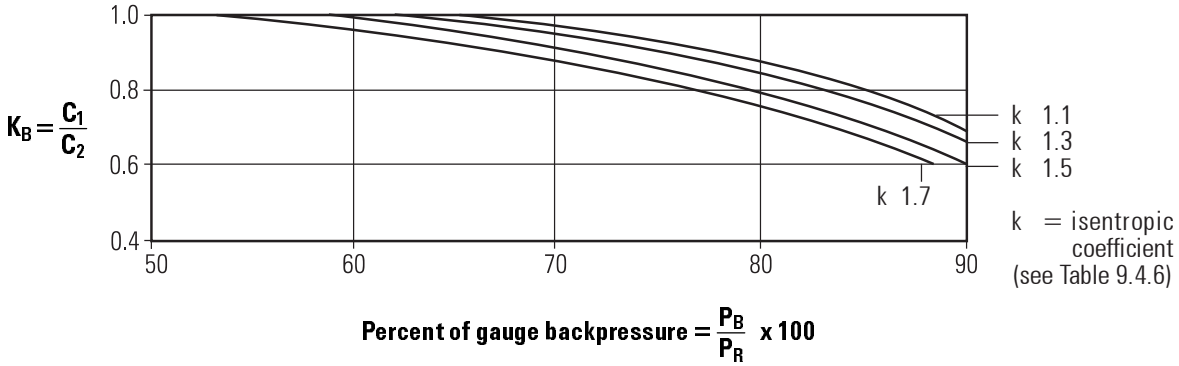
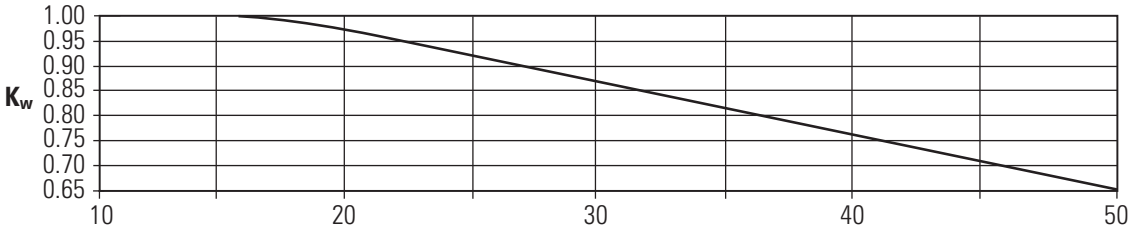


Fig. 9.4.7 Constant backpressure correction factor (K_B) for gas and vapour as used in ASME / API RP 520 for conventional valves

Liquid constant backpressure correction factor for ASME/API RP 520

- **Balanced bellows valves**



$$\text{Percent of gauge backpressure} = \frac{\text{Backpressure}}{\text{Set pressure}} = \frac{P_B}{P_S} \times 100$$

Fig. 9.4.8 Constant backpressure correction factor (K_w) for liquids as used in ASME/API RP 520 for balanced bellows valves

Viscosity correction factor for ASME/API RP 520 and BS 6759

This is used to make allowances for high viscosity fluids. In order to account for this, the valve size must first be established, assuming the fluid is non-viscous. Once the size has been selected, the Reynolds number for the valve is calculated and used to establish the correction factor from Figure 9.4.9.

The valve size should then be checked to ensure that the original size chosen would accommodate the flow after the viscous correction factor has been applied. If not this process should be repeated with the next largest valve size.

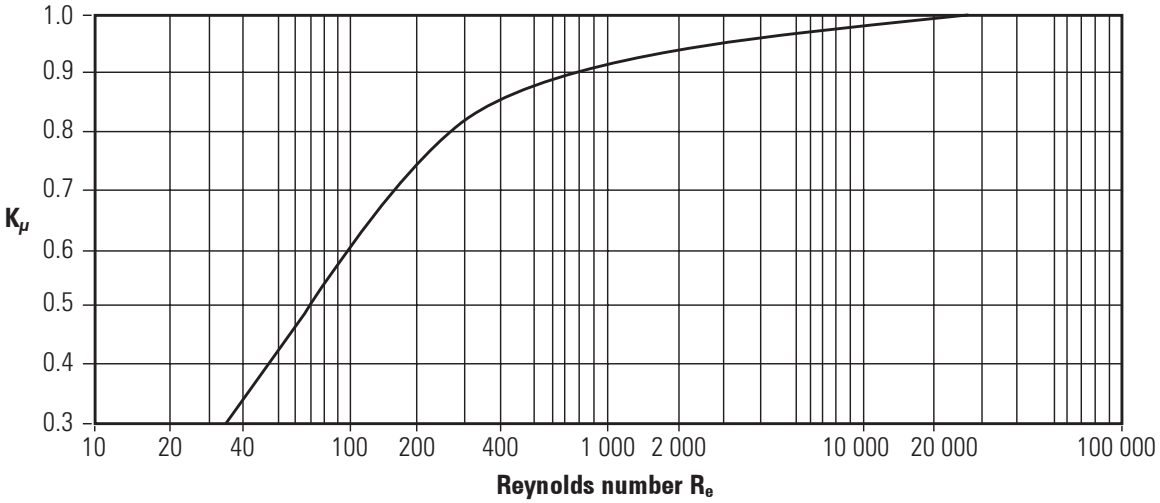


Fig. 9.4.9 Viscosity correction factor (Kμ) as used in ASME/API RP 520 and BS 6759

The Reynolds number can be calculated using Equations 9.4.15 and 9.4.16:

Metric units	$R_e = 0.3414 \frac{\dot{m}}{\mu\sqrt{A_O}}$	**Equation 9.4.15**

Imperial units	$R_e = \frac{2\,800\,G\,\dot{V}}{\mu\sqrt{A_O}}$	**Equation 9.4.16**

Where:

R_e = Reynolds number
$\dot{V}$ = Volume flow to be discharged (U.S. gal/min)
$\dot{m}$ = Mass flow to be discharged (kg/h)
μ = Dynamic viscosity (Imperial – cP, Metric – Pa s)
A_O = Discharge area (Imperial – in², Metric – mm²)

Standard - BS 6759

Use Equation 9.4.17 to calculate the minimum required orifice area for a safety valve used on **steam applications:**

$$A_O = \frac{\dot{m}}{0.525\ P_R\ K_{dr}\ K_{SH}} \qquad \textbf{Equation 9.4.17}$$

Use Equation 9.4.18 to calculate the minimum required orifice area for a safety valve used on **air applications:**

$$A_O = \frac{\dot{V}}{0.193\ P_R\ K_{dr}}\ \sqrt{\frac{T}{288}} \qquad \textbf{Equation 9.4.18}$$

Use Equation 9.4.19 to calculate the minimum required orifice area for a safety valve used on **gas applications:**

$$A_O = \frac{\dot{m}}{P_R\ C_g\ K_{dr}}\ \sqrt{\frac{Z\ T}{M}} \qquad \textbf{Equation 9.4.19}$$

Use Equation 9.4.20 to calculate the minimum required orifice area for a safety valve used on **liquid applications:**

$$A_O = \frac{\dot{m}}{1.61\ K_{dr}\ K_\mu\ \sqrt{\rho\ \Delta P}} \qquad \textbf{Equation 9.4.20}$$

Use Equation 9.4.21 to calculate the minimum required orifice area for a safety valve used on **hot water applications:**

$$A_O = \frac{\dot{Q}}{0.329\ P_R\ K_{dr}} \qquad \textbf{Equation 9.4.21}$$

Where:

A_O = Flow area (mm²)
$\dot{m}$ = Mass flow to be discharged (kg/h)
$\dot{V}$ = Volumetric flow to be discharged (l/s)
$\dot{Q}$ = Hot water heating capacity (kW)
C_g = Nozzle gas constant (see Table 9.4.3)
ΔP = P_R - P_B
P_R = Absolute relieving pressure (bar a)
P_B = Absolute backpressure (bar a)
T = Inlet temperature (K)
ρ = Density (kg/m³) (see Appendix A at the back of this module)
M = Molar mass (kg/kmol) (see Appendix A at the back of this module)
Z = Compressibility factor (see Equation 9.4.6)
K_{dr} = Derated coefficient of discharge (specified by the manufacturer)
K_{SH} = Superheat correction factor (see Table 9.4.4)
K_μ = Viscosity correction factor (see Figure 9.4.9)

Nozzle gas constant for BS 6759

Table 9.4.3 Nozzle gas constant (C_g) relative to isentropic coefficient (k) for gases

k	C_g	k	C_g	k	C_g
0.40	1.65	1.02	2.41	1.42	2.72
0.45	1.73	1.04	2.43	1.44	2.73
0.50	1.81	1.06	2.45	1.46	2.74
0.55	1.89	1.08	2.46	1.48	2.76
0.60	1.96	1.10	2.48	1.50	2.77
0.65	2.02	1.12	2.50	1.52	2.78
0.70	2.08	1.14	2.51	1.54	2.79
0.75	2.14	1.16	2.53	1.56	2.80
0.80	2.20	1.18	2.55	1.58	2.82
0.82	2.22	1.20	2.56	1.60	2.83
0.84	2.24	1.22	2.58	1.62	2.84
0.86	2.26	1.24	2.59	1.64	2.85
0.88	2.28	1.26	2.61	1.66	2.86
0.90	2.30	1.28	2.62	1.68	2.87
0.92	2.32	1.30	2.63	1.70	2.89
0.94	2.34	1.32	2.65	1.80	2.94
0.96	2.36	1.34	2.66	1.90	2.99
0.98	2.38	1.36	2.68	2.00	3.04
0.99	2.39	1.38	2.69	2.10	3.09
1.001	2.40	1.40	2.70	2.20	3.13

The nozzle gas constant C_g is calculated using Equation 9.4.22, for gases, and applied to Equation 9.4.19.

$$C_g = 3.948 \sqrt{k \left(\frac{2}{k+1}\right)^{\left(\frac{k+1}{k-1}\right)}} \qquad \textbf{Equation 9.4.22}$$

Superheat correction factor (K_{SH}) for BS 6759

Table 9.4.4 Superheat correction factors (K_{SH}) as used in BS 6759 (Metric units)

Set pressure (bar g)	Temperature (°C)									
	150	200	250	300	350	400	450	500	550	600
2	1.00	0.99	0.94	0.89	0.86	0.82	0.79	0.76	0.74	0.72
3	1.00	0.99	0.94	0.89	0.86	0.82	0.79	0.76	0.74	0.72
4	1.00	0.99	0.94	0.90	0.86	0.82	0.79	0.76	0.74	0.72
5	1.00	0.99	0.94	0.90	0.86	0.82	0.79	0.76	0.74	0.72
6		0.99	0.94	0.90	0.86	0.82	0.79	0.76	0.74	0.72
7		0.99	0.95	0.90	0.86	0.82	0.79	0.77	0.74	0.72
8		1.00	0.95	0.90	0.86	0.82	0.79	0.77	0.74	0.72
9		1.00	0.95	0.90	0.86	0.83	0.79	0.77	0.74	0.72
10		1.00	0.95	0.90	0.86	0.83	0.79	0.77	0.74	0.72
11		1.00	0.95	0.90	0.86	0.83	0.79	0.77	0.74	0.72
12		1.00	0.95	0.90	0.86	0.83	0.79	0.77	0.74	0.72
13		1.00	0.96	0.91	0.86	0.83	0.80	0.77	0.74	0.72
14		1.00	0.96	0.91	0.86	0.83	0.80	0.77	0.74	0.72
16		1.00	0.96	0.91	0.87	0.83	0.80	0.77	0.74	0.72
18			0.96	0.91	0.87	0.83	0.80	0.77	0.74	0.72
20			0.97	0.91	0.87	0.83	0.80	0.77	0.74	0.72
24			0.98	0.92	0.87	0.84	0.80	0.77	0.74	0.72
28			0.99	0.92	0.87	0.84	0.80	0.77	0.75	0.72
34			0.99	0.93	0.88	0.84	0.80	0.77	0.75	0.72
40			1.00	0.94	0.89	0.84	0.81	0.78	0.75	0.72
56				0.96	0.90	0.86	0.81	0.78	0.75	0.73
70				0.98	0.92	0.86	0.82	0.79	0.76	0.73
85				1.00	0.93	0.87	0.83	0.79	0.76	0.73
100				1.00	0.93	0.88	0.84	0.80	0.76	0.74

Standard - EN ISO 4126: 2004

Use Equation 9.4.23 to calculate the minimum required orifice area for a safety valve used on **dry saturated steam, superheated steam, and air and gas applications** at critical flow:

$$A = \frac{\dot{m}}{0.288\ 3\ C\ K_{dr}\ \sqrt{\frac{P_o}{\nu_g}}}$$ **Equation 9.4.23**

Use Equation 9.4.24 to calculate the minimum required orifice area for a safety valve used on **wet steam applications** at critical flow; Note: wet steam must have a dryness fraction greater than 0.9:

$$A = \frac{\dot{m}}{0.288\ 3\ C\ K_{dr}\ \sqrt{\frac{P_o}{\nu_g\ x}}}$$ **Equation 9.4.24**

Use Equation 9.4.25 to calculate the minimum required orifice area for a safety valve used on **air and gas applications** at sub-critical flow:

$$A = \frac{\dot{m}}{0.288\ 3\ C\ K_{dr}\ K_b\ \sqrt{\frac{P_o}{\nu_g}}}$$ **Equation 9.4.25**

Use Equation 9.4.26 to calculate the minimum required orifice area for a safety valve used on **liquid applications:**

$$A = \frac{\dot{m}}{1.61\ K_{dr}\ K_v\ \sqrt{\frac{P_o - P_b}{\nu_g}}}$$ **Equation 9.4.26**

Where:

A = Flow area (not curtain area) mm²
$\dot{m}$ = Mass flowrate (kg/h)
C = Function of the isentropic exponent (see Table 9.4.5)
K_{dr} = Certified derated coefficient of discharge (from manufacturer)
P_o = Relieving pressure (bar a)
P_b = Backpressure (bar a)
ν_g = Specific volume at relieving pressure and temperature (m³/kg)
x = Dryness fraction of wet steam
K_b = Theoretical correction factor for sub-critical flow (see Table 9.4.6)
K_v = Viscosity correction factor (see Figure 9.4.10)

Table 9.4.5 Value of C as a function of 'k' for steam, air and gas applications to the EN ISO 4126 standard. 'k' values are incorporated into the ISO 4126 standard: (Part 7). Alternatively, 'k' values can be obtained from the Spirax Sarco website steam tables.

k	C_g	k	C_g	k	C_g	k	C_g
0.40	1.647	0.90	2.301	1.40	2.703	1.90	2.991
0.41	1.665	0.91	2.311	1.41	2.710	1.91	2.996
0.42	1.682	0.92	2.320	1.42	2.717	1.92	3.001
0.43	1.700	0.93	2.330	1.43	2.723	1.93	3.006
0.44	1.717	0.94	2.339	1.44	2.730	1.94	3.010
0.45	1.733	0.95	2.349	1.45	2.736	1.95	3.015
0.46	1.750	0.96	2.358	1.46	2.743	1.96	3.020
0.47	1.766	0.97	2.367	1.47	2.749	1.97	3.025
0.48	1.782	0.98	2.376	1.48	2.755	1.98	3.030
0.49	1.798	0.99	2.386	1.49	2.762	1.99	3.034
0.50	1.813	1.00	2.401	1.50	2.768	2.00	3.039
0.51	1.829	1.01	2.404	1.51	2.774	2.01	3.044
0.52	1.844	1.02	2.412	1.52	2.780	2.02	3.049
0.53	1.858	1.03	2.421	1.53	2.786	2.03	3.053
0.54	1.873	1.04	2.430	1.54	2.793	2.04	3.058
0.55	1.888	1.05	2.439	1.55	2.799	2.05	3.063
0.56	1.902	1.06	2.447	1.56	2.805	2.06	3.067
0.57	1.916	1.07	2.456	1.57	2.811	2.07	3.072
0.58	1.930	1.08	2.464	1.58	2.817	2.08	3.076
0.59	1.944	1.09	2.472	1.59	2.823	2.09	3.081
0.60	1.957	1.10	2.481	1.60	2.829	2.10	3.085
0.61	1.971	1.11	2.489	1.61	2.843	2.11	3.090
0.62	1.984	1.12	2.497	1.62	2.840	2.12	3.094
0.63	1.997	1.13	2.505	1.63	2.846	2.13	3.099
0.64	2.010	1.14	2.513	1.64	2.852	2.14	3.103
0.65	2.023	1.15	2.521	1.65	2.858	2.15	3.107
0.66	2.035	1.16	2.529	1.66	2.863	2.16	3.112
0.67	2.048	1.17	2.537	1.67	2.869	2.17	3.116
0.68	2.060	1.18	2.545	1.68	2.874	2.18	3.121
0.69	2.072	1.19	2.553	1.69	2.880	2.19	3.125
0.70	2.084	1.20	2.560	1.70	2.886	2.20	3.129
0.71	2.096	1.21	2.568	1.71	2.891		
0.72	2.108	1.22	2.570	1.72	2.897		
0.73	2.120	1.23	2.583	1.73	2.902		
0.74	2.131	1.24	2.591	1.74	2.908		
0.75	2.143	1.25	2.598	1.75	2.913		
0.76	2.154	1.26	2.605	1.76	2.918		
0.77	2.165	1.27	2.613	1.77	2.924		
0.78	2.170	1.28	2.620	1.78	2.929		
0.79	2.187	1.29	2.627	1.79	2.934		
0.80	2.198	1.30	2.634	1.80	2.940		
0.81	2.209	1.31	2.641	1.81	2.945		
0.82	2.219	1.32	2.649	1.82	2.950		
0.83	2.230	1.33	2.656	1.83	2.955		
0.84	2.240	1.34	2.663	1.84	2.960		
0.85	2.251	1.35	2.669	1.85	2.965		
0.86	2.261	1.36	2.676	1.86	2.971		
0.87	2.271	1.37	2.683	1.87	2.976		
0.88	2.281	1.38	2.690	1.88	2.981		
0.89	2.291	1.39	2.697	1.89	2.986		

Table 9.4.6 Capacity correction factors for backpressure to the EN ISO 4126 standard for steam, air and gas applications

<k	Pb/Po																
	0.45	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.82	0.84	0.86	0.88	0.90	0.92	0.94	0.96	0.98
0.4	-	-	-	-	-	-	-	0.999	0.992	0.979	0.957	0.924	0.880	0.820	0.739	0.628	0.462
0.5	-	-	-	-	-	-	1.000	0.985	0.970	0.948	0.919	0.881	0.831	0.769	0.687	0.579	0.422
0.6	-	-	-	-	-	0.999	0.995	0.965	0.944	0.917	0.884	0.842	0.791	0.727	0.647	0.542	0.393
0.7	-	-	-	-	-	0.999	0.983	0.942	0.918	0.888	0.852	0.809	0.757	0.693	0.614	0.513	0.371
0.8	-	-	-	-	0.999	0.993	0.968	0.921	0.894	0.862	0.825	0.780	0.728	0.664	0.587	0.489	0.353
0.9	-	-	-	-	0.999	9.985	0.953	0.900	0.672	0.839	0.800	0.755	0.703	0.640	0.565	0.469	0.337
1.001	-	-	-	1.000	0.995	0.975	0.938	0.881	0.852	0.818	0.779	0.733	0.681	0.619	0.545	0.452	0.325
1.1	-	-	-	0.999	0.989	0.964	0.923	0.864	0.833	0.799	0.759	0.714	0.662	0.601	0.528	0.438	0.314
1.2	-	-	-	0.997	0.982	0.953	0.909	0.847	0.817	0.782	0.742	0.697	0.645	0.585	0.514	0.425	0.305
1.3	-	-	1.000	0.993	0.974	0.943	0.896	0.833	0.801	0.766	0.727	0.682	0.631	0.571	0.501	0.414	0.296
1.4	-	-	0.999	0.989	0.967	0.932	0.884	0.819	0.787	0.752	0.712	0.668	0.617	0.559	0.489	0.404	0.289
1.5	-	-	0.997	0.983	0.959	0.922	0.872	0.806	0.774	0.739	0.700	0.655	0.605	0.547	0.479	0.395	0.282
1.6	-	1.000	0.994	0.978	0.951	0.913	0.861	0.794	0.763	0.727	0.688	0.644	0.594	0.537	0.470	0.387	0.277
1.7	-	0.999	0.991	0.972	0.944	0.903	0.851	0.783	0.752	0.716	0.677	0.633	0.584	0.527	0.461	0.380	0.271
1.8	-	0.998	0.987	0.967	0.936	0.895	0.841	0.773	0.741	0.706	0.677	0.624	0.575	0.519	0.453	0.373	0.266
1.9	-	0.996	0.983	0.961	0.929	0.886	0.832	0.764	0.732	0.697	0.658	0.615	0.566	0.511	0.446	0.367	0.262
2.0	1.000	0.994	0.979	0.955	0.922	0.879	0.824	0.755	0.723	0.688	0.649	0.606	0.558	0.504	0.440	0.362	0.258
2.1	0.999	0.992	0.975	0.950	0.915	0.871	0.815	0.747	0.715	0.680	0.641	0.599	0.551	0.497	0.434	0.357	0.254
2.2	0.999	0.989	0.971	0.945	0.909	0.864	0.808	0.739	0.707	0.672	0.634	0.592	0.544	0.490	0.428	0.352	0.251

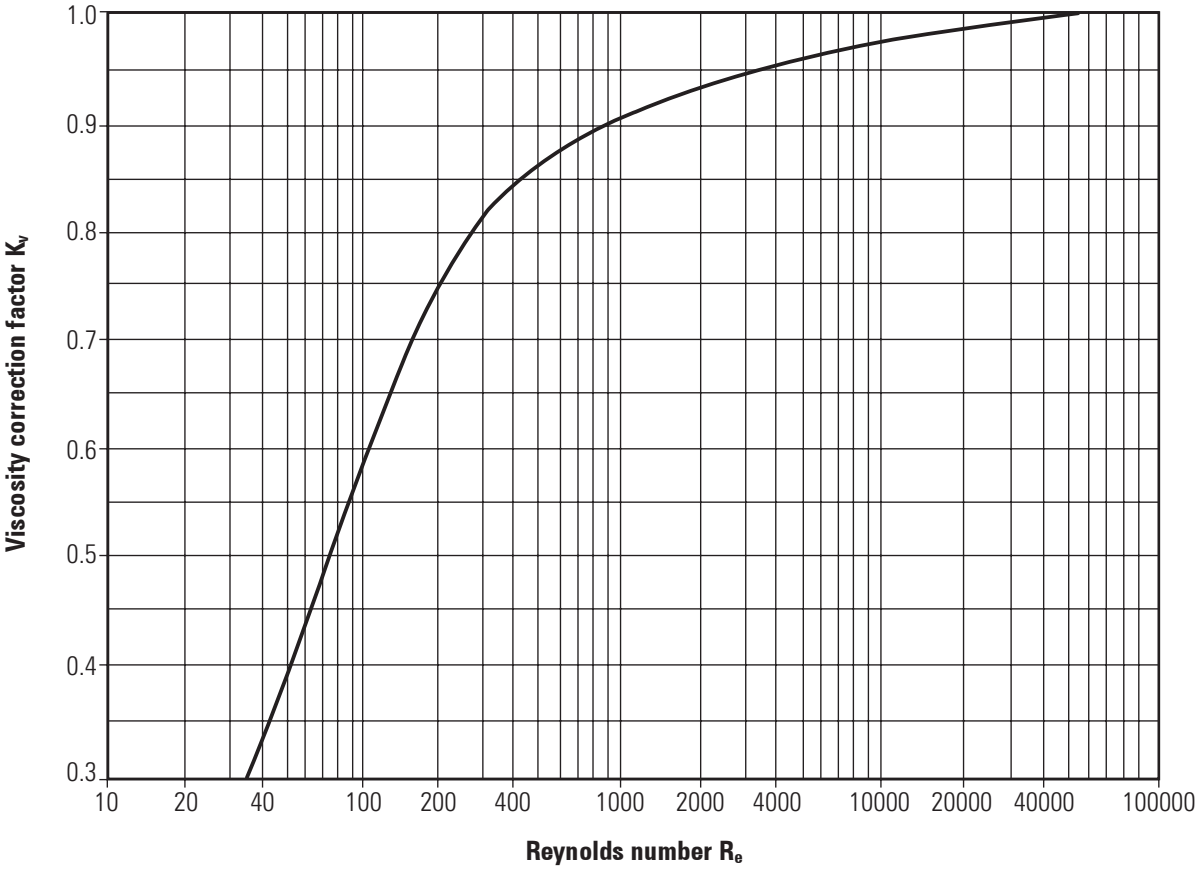


Fig. 9.4.10 Graph to determine K_V from the Reynold's No. for liquid applications to the EN ISO 4126 standard

Example 9.4.4

Size the minimum flow area required for a safety valve designed to EN ISO 4126 to relieve a superheated steam system of overpressure.

Steam system conditions

Relieving pressure : 20 bar g
ρ_o : 21 bar a
Steam temperature : 280°C
Flowrate to pass ($\dot{m}$) : 2 500 kg/h

It is necessary to obtain the following: C, K_{dr} and ν

From EN ISO 4126:7 C = 2.628
From the manufacturer K_{dr} = 0.71
From steam tables ν_g = 0.113 8 m³/kg

From Equation 9.4.23

$$A = \frac{\dot{m}}{0.288\,3\ C\ K_{dr}\ \sqrt{\frac{\rho_o}{\nu_g}}}$$

$$A = \frac{2500}{0.288\,3 \times 2.628 \times 0.71 \times \sqrt{\frac{21}{0.113\,8}}}$$

$$A = \frac{2\,500}{0.288\,3 \times 2.628 \times 0.71 \times 13.58}$$

A = 342 mm²

A = 342 mm²

Appendix A - Properties of industrial liquids

Table 9.4.7 Properties of some common industrial liquids

For specific gravity (G) used in ASME liquid sizing calculations, divide density by 998 (density of water).

Liquid	Chemical formula	Boiling point (°C) at 1.013 bar a	Density (kg/m³)
Acetone	$CH_2.CO.CH_3$	56.0	791
Ammonia	NH_3	- 33.4	609
Benzene	C_6H_6	80.0	879
Butalene	C_4H_8	- 6.3	600
Butane	C_4H_{10}	- 0.5	580
Carbon disulphide	CS_2	46.0	1 260
Carbon tetrachloride	CCl_4	76.7	1 594
20% caustic soda	NaOH		1 220
Crude oil			700 to 1 040
Diesel oil		175.0	880
Ethanol	C_2H_5OH	78.0	789
Freon 12	CF_2Cl_2	- 29.8	1 330
Glycol	$C_2H_4(OH)_2$	197.5	1 140
Light fuel oil		175.0	850
Heavy fuel oil		220.0 to 350.0	950
Kerosene		150.0 to 300.0	740
Methanol	C_3OH	65.0	792
Naphthalene	$C_{10}H_8$	218.0	1 145
Nitric acid	HNO_3	86.0	1 560
Propane	C_3H_8	- 42.0	500
Sulphurous acid	H_2SO_3	338.0	1 400
Toluene	$C_6H_5.CH_3$	111.0	867
Trichlorethylene	$CHCl.CCl_2$	87.0	1 464
Water	H_2O	100.0	998

Properties of industrial gases

Table 9.4.8 Properties of some common industrial gases

For specific gravity (G) used in ASME gas sizing calculations, divide molar mass by 28.96 (molar mass of air).

Gas	Chemical formula	Molar mass (M) kg/kmol	Isentropic coefficient (k) at 1.013 bar a and 0°C†	Specific volume (V) m^3/kg at 1.013 bar a and 0°C
Acetylene	C_2H_2	26.02	1.26	0.853
Air		28.96	1.40	0.773
Ammonia	NH_3	17.03	1.31	1.297
Argon	Ar	39.91	1.66	0.561
Benzene	C_6H_6	78.00	1.10	
Biphenyl oxide	$C_{12}H_{10}$	166.00	1.05*	0.0094*
Butane - n	C_4H_{10}	58.08	1.11	0.370
Butylene	C_4H_8	56.10	1.20	
Carbon disulphide		76.00	1.21	
Carbon dioxide	CO_2	44.00	1.30	0.506
Carbon monoxide	CO	28.00	1.40	0.800
Chlorine	Cl_2	70.91	1.35	0.311
Cyclohexane		84.00	1.08	
Ethane	C_2H_6	30.05	1.22	0.737
Ethylene	C_2H_4	28.03	1.25	0.794
Freon 12	Cf_2Cl_2	121.00	1.14	
Helium	He	4.00	1.66	
Hexane	C_6H_{14}	86.00	1.08	
Hydrogen	H_2	2.02	1.41	11.124
Hydrogen chloride	HCl	36.46	1.40	0.610
Hydrogen sulphide	H_2S	34.08	1.32	0.651
Isobutane	$CH(CH_3)_3$	58.05	1.11	0.375
Methane	CH_4	16.03	1.31	1.395
Methyl chloride	CH_3Cl	50.48	1.28	0.434
Natural gas		19.00	1.27	
Nitrogen	N_2	28.02	1.40	0.799
Nitrous oxide	N_2O	44.02	1.30	0.746
Oxygen	O_2	32.00	1.40	0.700
Pentane	C_5H_{12}	72.00	1.09	0.451
Propane	C_3H_8	44.06	1.13	0.498
Sulphur dioxide	SO_2	64.07	1.29	0.342
Dry saturated steam	H_2O	18.015	1.135 †	
Superheated steam	H_2O	18.015	1.30 †	

† These are typical values, not values at 1.013 bar and 0°C

* At 15°C

Questions

1. A process vessel is supplied with steam from a pressure reducing station through a temperature control valve. In order to protect the process vessel from overpressure, a safety valve is to be installed downstream of the control valve. Given the following conditions, determine the potential fault load.

Safety valve set pressure	**6.0 bar g**
Safety valve overpressure	**10%**
Control valve full open capacity (K_{VS})	**10.3**
Maximum possible upstream pressure	**12.5 bar g**
Vessel MAAP	**7.3 bar g**

a| 900 kg/h ☐
b| 1020 kg/h ☐
c| 1545 kg/h ☐
d| 1670 kg/h ☐

2. Using the sizing formulae from ASME/API RP 520, calculate the minimum required orifice diameter for a safety valve discharging superheated steam under the following conditions:

Relieving temperature	**700°F**
Discharge quantity	**88 500 lb/h**
Safety valve coefficient of discharge	**0.995**
Safety valve set pressure	**240 psi g**
Safety valve overpressure	**10%**
Safety valve relieving pressure	**278.7 psi a**

a| 6.7 in² ☐
b| 7.3 in² ☐
c| 7.9 in² ☐
d| 8.5 in² ☐

3. Using the sizing formulae from BS 6759, calculate the minimum required orifice diameter for a safety valve discharging air under the following conditions:

Relieving temperature	**50°C**
Discharge quantity	**28 800 m³/h**
Safety valve coefficient of discharge	**0.995**
Safety valve set pressure	**12 bar g**
Safety valve overpressure	**5%**

a| 18 140 mm² ☐
b| 11 680 mm² ☐
c| 49 770 mm² ☐
d| 52 250 mm² ☐

4. A safety valve is used to provide overpressure protection on an ammonia system. Using the AD-Merkblatt A2 standard calculations, determine the minimum required orifice area required for the following system parameters:

Discharge quantity	**4 000 kg/h**
Relieving pressure	**8.5 bar a**
Backpressure	**2 bar a**
Relieving temperature	**293 K**
Specific volume (8.5 bar a, 293 K)	**0.149 4 m^3/kg**
Outflow coefficient	**0.7**

a| 2 555 mm^2 ☐
b| 2 000 mm^2 ☐
c| 3 000 mm^2 ☐
d| 4 000 mm^2 ☐

5. A safety valve (with a relieving pressure, P_R, of 6 bar a and coefficient of discharge K_{dr}, of 0.76) is used to provide overpressure protection in a hot water system. The safety valve discharges the 160°C water against a backpressure of 2 bar a in a manifold system. Using the BS 6759 standard calculations and the concept of two-phase flow, determine the minimum orifice area required to discharge 5 000 kg/h of the hot water.

a| 60 mm^2 ☐
b| 90 mm^2 ☐
c| 160 mm^2 ☐
d| 220 mm^2 ☐

6. Determine the minimum required orifice area for a safety valve to be used on heavy fuel oil (density, ρ = 980 kg/m^3 and viscosity, μ = 1.05 Pa s), under the following conditions, using the BS 6759 standard method of calculation:

Discharge quantity	**10 000 kg/h**
Safety valve coefficient of discharge	**0.71**
Safety valve relieving pressure	**8 bar a**
Backpressure	**1 bar a (atmospheric)**

a| 90 mm^2 ☐
b| 110 mm^2 ☐
c| 130 mm^2 ☐
d| 150 mm^2 ☐

Answers

1:d, 2: b, 3: b, 4: a, 5: d, 6: c

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Module 9.5

Safety Valve Installation

Safety Valve Installation

Seat tightness

Seat tightness is an important consideration when selecting and installing a safety valve, as not only can it lead to a continuous loss of system fluid, but leakage can also cause deterioration of the sealing faces, which can lead to premature lifting of the valve.

The seat tightness is affected by three main factors; firstly by the characteristics of the safety valve, secondly by the installation of the safety valve and thirdly, by the operation of the safety valve.

Characteristics of the safety valve

For a metal-seated valve to provide an acceptable shut-off, the sealing surfaces need to have a high degree of flatness with a very good surface finish. The disc must articulate on the stem and the stem guide must not cause any undue frictional effects. Typical figures required for an acceptable shut-off for a metal seated valve are 0.5 µm for surface finish and two optical light bands for flatness. In addition, for a reasonable service life, the mating and sealing surfaces must have a high wear resistance.

Unlike ordinary isolation valves, the net closing force acting on the disc is relatively small, due to there being only a small difference between the system pressure acting on the disc and the spring force opposing it.

Resilient or elastomer seals incorporated into the valve discs are often used to improve shut-off, where system conditions permit. It should be noted, however, that a soft seal is often more susceptible to damage than a metal seat.

Safety valve installation

Seat damage can often occur when a valve is first lifted as part of the general plant commissioning procedure, because very often, dirt and debris are present in the system. To ensure that foreign matter does not pass through the valve, the system should be flushed out before the safety valve is installed and the valve must be mounted where dirt, scale and debris cannot collect.

It is also important on steam applications to reduce the propensity for leakage by installing the valve so that condensate cannot collect on the upstream side of the disc. This can be achieved by installing the safety valve above the steam pipe as shown in Figure 9.5.1.

When a safety valve is installed correctly, above the steam pipe, the safety valve inlet pipework is self-draining.

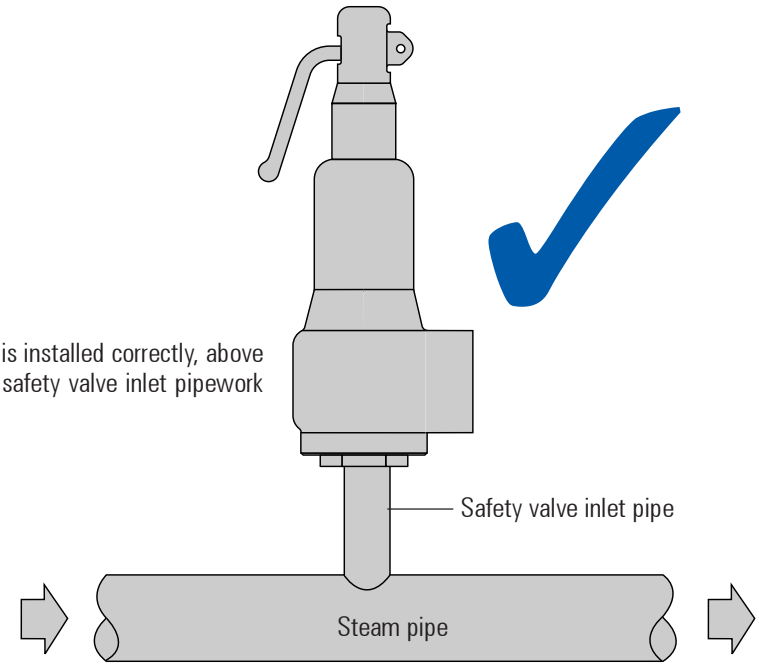


Fig. 9.5.1 Correct position of a safety valve on a steam system

Where safety valves are installed below the pipe, steam will condense, fill the pipe and wet the upstream side of the safety valve seat. This type of installation is not recommended but is shown in Figure 9.5.2 for reference purposes.

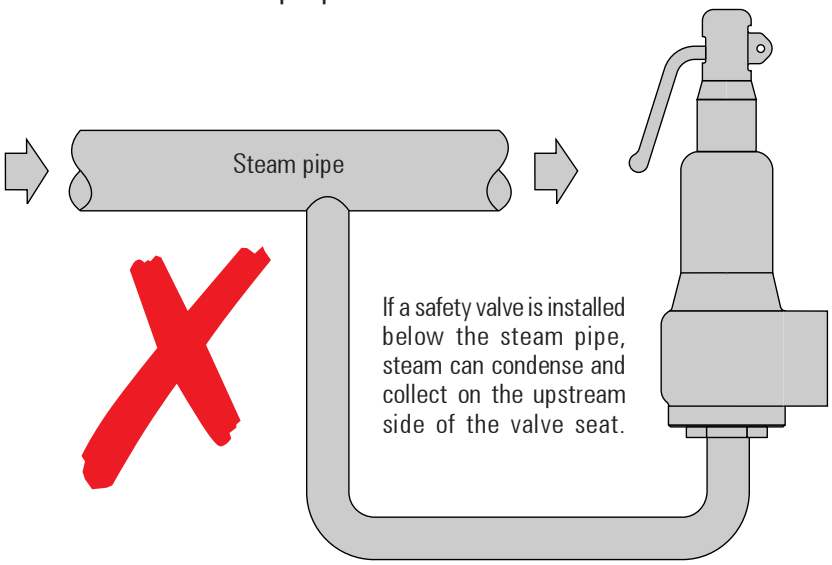


Fig. 9.5.2 Incorrect position of a safety valve on a steam system

Also, it is essential at all times to ensure that the downstream pipework is well drained so that downstream flooding (which can also encourage corrosion and leakage) cannot occur, as shown in Figure 9.5.3.

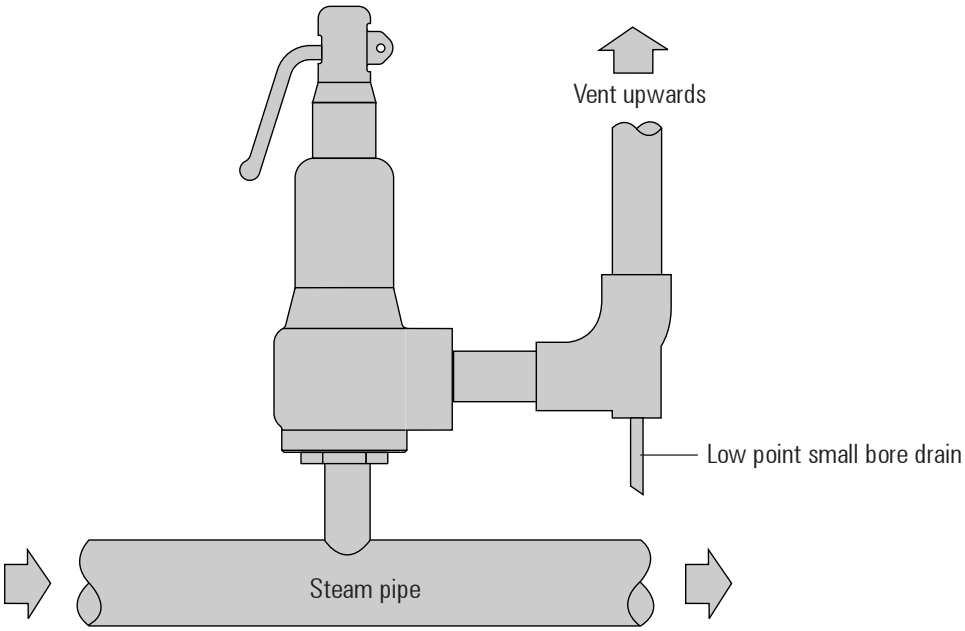


Fig. 9.5.3 Correct installation of a safety valve on a steam system

Operation of the safety valve

Leakage can also be experienced when there is dirt or scale sitting on the seating face. This usually occurs during the periodic lifting demanded by insurance companies and routine maintenance programs. Further lifting of the lever will generally clear any dirt that may be on the seating face.

The vast majority of safety valve seat leakage problems occur after initial manufacture and test. These problems typically result from damage during transit, and sometimes as a result of misuse and contamination, or because of poor installation.

Most safety valve standards do not include detailed shut-off parameters. For those that do, the requirements and recommended test procedures are usually based on the API 527 standard, which is commonly used throughout the safety valve industry.

The procedure for testing valves that have been set on air involves blocking all secondary leakage paths, whilst maintaining the valve at 90% of the set pressure on air (see Figure 9.5.4). The outlet of the safety valve is connected to a 6 mm internal diameter pipe, the end of which is held 12.7 mm below the surface of water contained in a suitable, transparent vessel. The number of bubbles discharged from this tube per minute is measured. For the majority of valves set below 70 bar g, the acceptance criteria is 20 bubbles per minute.

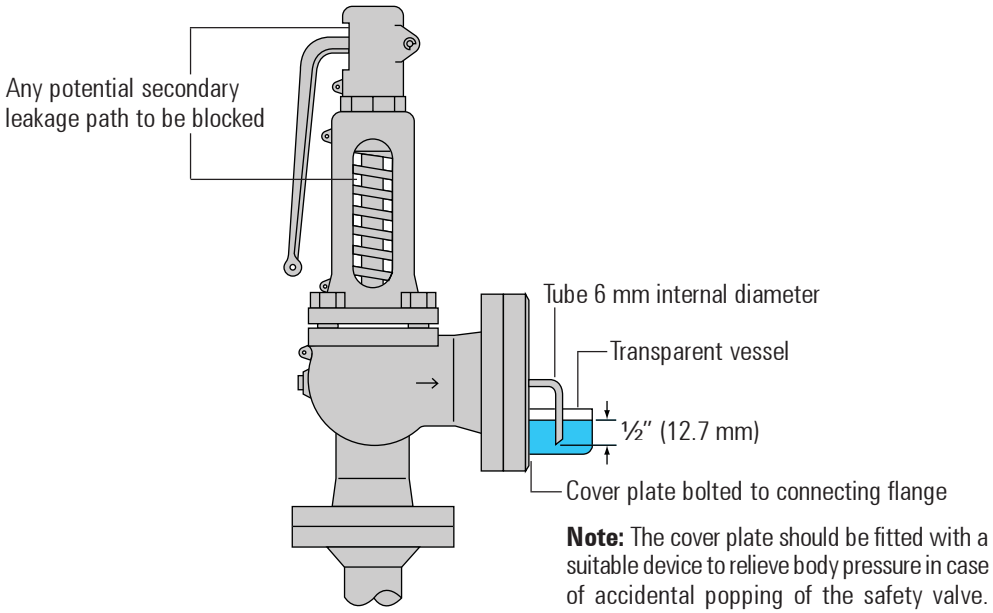


Fig. 9.5.4 Apparatus to test seat tightness with air

For valves set on steam or water, the leakage rate should be assessed using the corresponding setting media. For steam, there must be no visible leakage observed against a black background for one minute after a three-minute stabilisation period. In the case of water, there is a small leakage allowance, dependent on the orifice area, of 10 ml per hour per inch of the nominal inlet diameter.

The above procedure can be time consuming, so it is quite common for manufacturers to employ a test using alternative methods, for example, using accurate flow measuring equipment that is calibrated against the parameters set in API 527.

Under no circumstances should any additional load be applied to the easing lever nor should the valve be gagged in order to increase the seat tightness. This will affect the operating characteristics and can result in the safety valve failing to lift in overpressure conditions. If there is an unacceptable level of seat leakage, the valve can be refurbished or repaired, but only by authorised personnel, working with the approval of the manufacturer, and using information supplied by the manufacturer.

Commonly supplied spare parts typically include springs, discs and nozzles, resilient seals and gaskets. Many valves have seat rings which are not removable and these can sometimes be re-profiled and re-lapped in the body. However, it is important that the size of seat orifice is maintained exactly in line with the original drawings since this can alter the effective area and, subsequently affect the set pressure.

It is unacceptable for the disc to be lapped directly onto the seat in the body, since a groove will be created on the disc preventing a consistent shut-off after lifting.

In the case of resilient seal valves usually the seal (which is normally an 'O' ring or disc) can be changed in the disc assembly.

If Independent Authority Approval is to be maintained then it is mandatory that the repairer is acting as the manufacturer's approved agent. For ASME approved valves, the repairer must be independently approved by the National Board and is subsequently allowed to apply a 'VR' stamp, which indicates a valve has been repaired.

Marking

Safety valve standards are normally very specific about the information which must be carried on the valve. Marking is mandatory on both the shell, usually cast or stamped, and the name-plate, which must be securely attached to the valve. A general summary of the information required is listed below:

On the shell:

- Size designation.
- Material designation of the shell.
- Manufacturer's name or trademark.
- Direction of flow arrow.

On the identification plate:

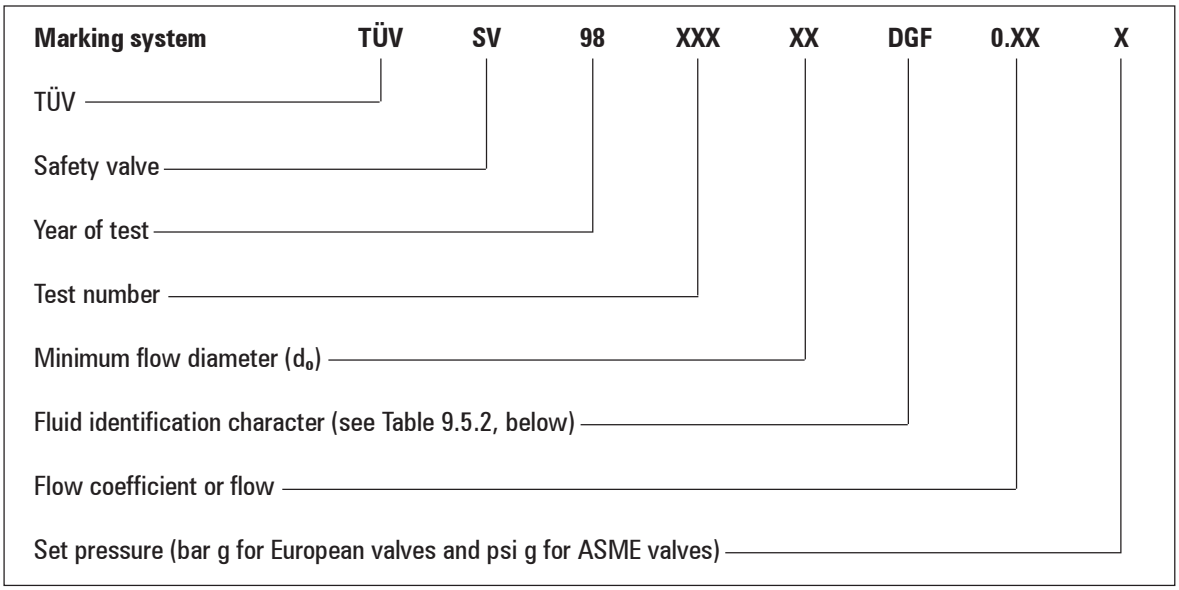
- Set pressure (in bar g for European valves and psi g for ASME valves).
- Number of the relevant standard (or relevant ASME stamp).
- Manufacturer's model type reference.
- Derated coefficient of discharge or certified capacity.
- Flow area.
- Lift and overpressure.
- Date of manufacture or reference number.

National Board approved ASME stamps are applied as follows:

V ASME I approved safety relief valves.
UV ASME VIII approved safety relief valves.
UD ASME VIII approved rupture disc devices.
NV ASME III approved pressure relief valves.
VR Authorised repairer of pressure relief valves.

Table 9.5.1 details the marking system required by TÜV and Table 9.5.2 details the fluid reference letters.

Table 9.5.1 Marking system used for valves approved by TÜV to AD-Merkblatt A2, DIN 3320 and TRD 421



The K_{dr} or α_w value can vary according to the relevant fluid and is either suffixed or prefixed by the identification letter shown in Table 9.5.2.

Table 9.5.2 Fluid types defined as steam, gas or liquid

For α_w
D (dampf) for steam
G (gase) for gas
F (flüssigkeiten) for liquids

For K_{dr}
S for steam
G for gas
L for liquids

Installation

Safety valves are precision items of safety equipment; they are set to close tolerances and have accurately machined internal parts. They are susceptible to misalignment and damage if mishandled or incorrectly installed.

Valves should be transported upright if possible and they should never be carried or lifted by the easing lever. In addition, the protective plugs and flange protectors should not be removed until actual installation. Care should also be taken during movement of the valve to avoid subjecting it to excessive shock as this can result in considerable internal damage or misalignment.

Inlet pipework

When designing the inlet pipework, one of the main considerations is to ensure that the pressure drop in this pipework is minimised. EN ISO 4126 recommends that the pressure drop be kept below 3% of the set pressure when discharging. Where safety valves are connected using short 'stub' connections, inlet pipework must be at least the same size as the safety valve inlet connection. For larger lines or any line incorporating bends or elbows, the branch connection should be at least two pipe sizes larger than the safety valve inlet connection, at which point it is reduced in size to the safety valve inlet size (see Figure 9.5.5a). Excessive pressure loss can lead to 'chatter', which may result in reduced capacity and damage to the seating faces and other parts of the valve. In order to reduce the pressure loss in the inlet, the following methods can be adopted:

- Increase the diameter of the pipe. (see Figure 9.5.5 (a)).
- Ensure that any corners are suitably rounded. The standard EN ISO 4126: Part 1 recommends that corners should have a radius of not less than one quarter of the bore (see Figure 9.5.5 (b)).
- Reduce the inlet pipe length.
- Install the valve at least 8 to 10 pipe diameters downstream from any converging or diverging 'Y' fitting, or any bend (see Figure 9.5.5 (c)).
- Never install the safety valve branch directly opposite a branch on the lower side of the steam line.
- Avoid take-off branches (such as for other processes) in the inlet piping, as this will increase the pressure drop.

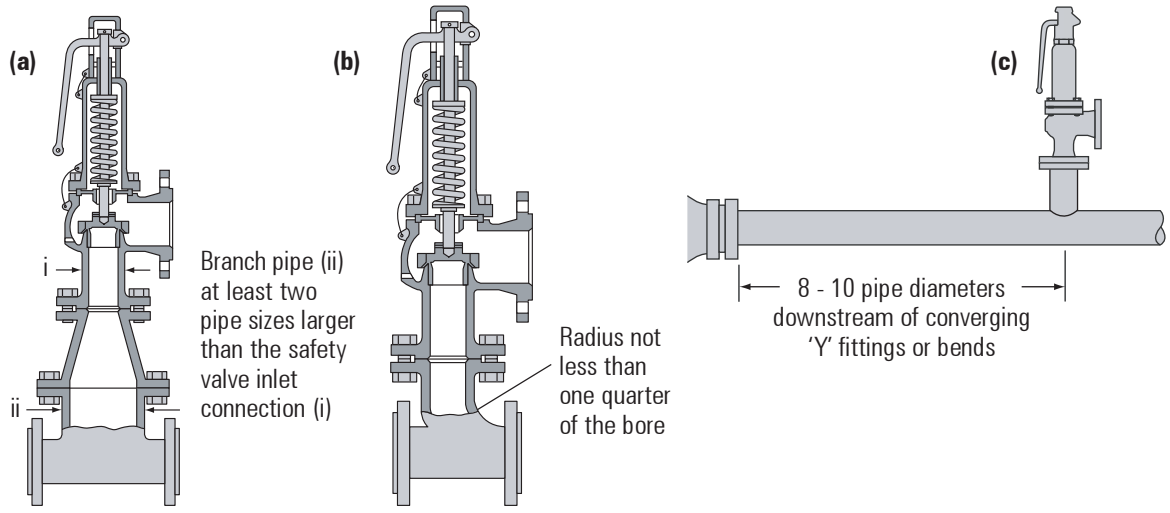


Fig. 9.5.5 Correct installations of safety valves

Safety valves should always be installed with the bonnet vertically upwards. Installing the valve in any other orientation can affect the performance characteristics.

The API Recommended Practice 520 guidelines also state that the safety valve should not be installed at the end of a long horizontal pipe that does not normally have flow through it. This can lead to the accumulation of foreign material or condensate in the pipe, which may cause unnecessary damage to the valve, or interfere with its operation.

Outlet pipework

There are two possible types of discharge system – open and closed. An open system discharges directly into the atmosphere whereas a closed system discharges into a manifold along with other safety valves.

It is recommended that discharge pipework should rise for steam and gas systems, whereas for liquids, it should fall. Horizontal pipework should have a downward gradient of at least 1 in 100 away from the valve ensuring that any discharge will be self-draining. It is important to drain any rising discharge pipework. Vertical rises will require separate drainage. Note: all points of system drainage are subject to the same precautions, notably that valve performance must not be affected, and any fluid must be discharged to a safe location.

It is essential to ensure that fluid cannot collect on the downstream side of a safety valve, as this will impair its performance and cause corrosion of the spring and internal parts. Many safety valves are provided with a body drain connection, if this is not used or not provided, then a small bore drain should be fitted in close proximity to the valve outlet (see Figure 9.5.3).

One of the main concerns in closed systems is the pressure drop or built-up backpressure in the discharge system. As mentioned in Module 9.2, this can drastically affect the performance of a safety valve. The EN ISO 4126: Part 1 standard states that the pressure drop should be maintained below 10% of the set pressure. In order to achieve this, the discharge pipe can be sized using Equation 9.5.1.

$$d = \sqrt[5]{\frac{L_e \dot{m}^2 \left(\frac{v_g + 2}{2}\right)}{0.08\ P}}$$ **Equation 9.5.1**

Where:
d = Pipe diameter (mm)
L_e = Equivalent length of pipe (m)
$\dot{m}$ = Discharge capacity (kg/h)
P = Safety valve set pressure (bar g) x 0.1
v_g = Specific volume of saturated steam at the pressure (P) (m^3/kg)

The pressure (P) should be taken as the maximum allowable pressure drop according to the relevant standard. In the case of EN ISO 4126: Part 1, this would be 10% of the set pressure and it is at this pressure v_g is taken.

Example 9.5.1
Calculate the nominal diameter of the discharge pipework for a safety valve required to discharge 1 000 kg/h of saturated steam; given that the steam is to be discharged into a vented tank via the pipework, which has an equivalent length of 25 m. The set pressure of the safety valve is 10 bar g and the acceptable backpressure is 10% of the set pressure. (Assume zero pressure drop along the tank vent).

Answer: If the maximum 10% backpressure is allowed, then the gauge pressure at the safety valve outlet will be:

$$\frac{10}{100} \times 10 \text{ bar g} = 1.0 \text{ bar g}$$

Using saturated steam tables, the specific volume at this pressure is, $v_g = 0.88\ m^3/kg$.

Applying Equation 9.5.1:

$$d = \sqrt[5]{\frac{L_e \dot{m}^2 \left(\frac{v_g + 2}{2}\right)}{0.08\ P}} = \sqrt[5]{\frac{25 \times 1000^2 \times \left(\frac{0.88 + 2}{2}\right)}{0.08 \times 1.0}} = \sqrt[5]{\frac{25 \times 1000^2 \times 1.44}{0.08}} = \mathbf{54\ mm}$$

Therefore, the pipework connected to the outlet of the safety valve should have an internal diameter of at least 54 mm. With schedule 40 pipe, this would require a DN65 pipe.

If it is not possible to reduce the backpressure to below 10% of the set pressure, a balanced safety valve should be used.

Balanced safety valves require that their bonnets be vented to atmosphere. In the case of the balanced bellows type, there will be no discharge of the process fluid, so they can be vented directly to the atmosphere. The main design consideration is to ensure that this vent will not become blocked, for example, by foreign material or ice. With the balanced piston type, consideration must be given to the fact that process fluid may be discharged through the bonnet vent. If discharging to a pressurised system, the vent has to be suitably sized, so that no backpressure exists above the piston.

Safety valves that are installed outside of a building for discharge directly into the atmosphere should be covered using a hood. The hood allows the discharge of the fluid, but prevents the build up of dirt and other debris in the discharge pipework, which could affect the backpressure. The hood should also be designed so that it too does not affect the backpressure.

Manifolds

Manifolds must be sized so that in the worst case (i.e. when all the manifold valves are discharging), the pipework is large enough to cope without generating unacceptable levels of backpressure. The volume of the manifold should ideally be increased as each valve outlet enters it, and these connections should enter the manifold at an angle of no greater than 45° to the direction of flow (see Figure 9.5.6). The manifold must also be properly secured and drained where necessary.

For steam applications, it is generally not recommended to use manifolds, but they can be utilised if proper consideration is given to all aspects of the design and installation.

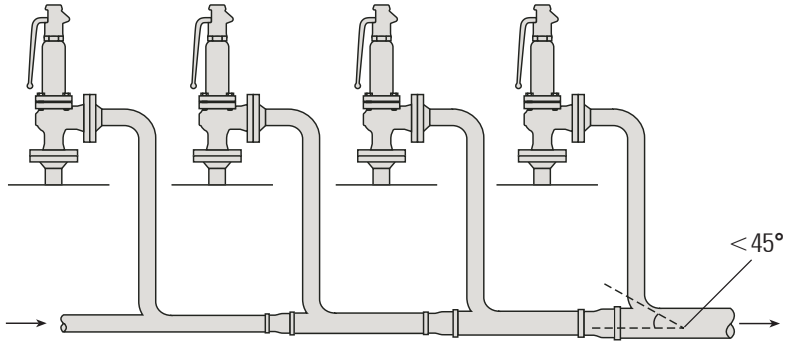


Fig. 9.5.6 A typical manifold discharge system

Reaction forces when discharging

In open systems, careful consideration must be given to the effects of the reaction forces generated in the discharge system when the valve lifts. In these systems, there will be significant resultant force acting in the opposite direction to that of discharge. It is important to prevent excessive loads being imposed on the valve or the inlet connection by these reaction forces, as they can cause damage to the inlet pipework. The magnitude of the reaction forces can be calculated using the formula in Equation 9.5.2:

$$F = 129\,\dot{m}\sqrt{\frac{k\,T}{(k+1)\,M}} + 0.1\,A\,P \qquad \textbf{Equation 9.5.2}$$

Where:
F = Reaction force at the point of discharge to atmosphere (newtons) (see Figure 9.5.4)
$\dot{m}$ = Discharge mass flowrate (kg/s)
k = Isentropic coefficient of the fluid
T = Fluid temperature (K)
M = Molar mass of the fluid (kg/kmol)
A = Area of the outlet at the point of discharge (mm²) (see Figure 9.5.7)
P = Static pressure at the outlet at the point of discharge (bar g)

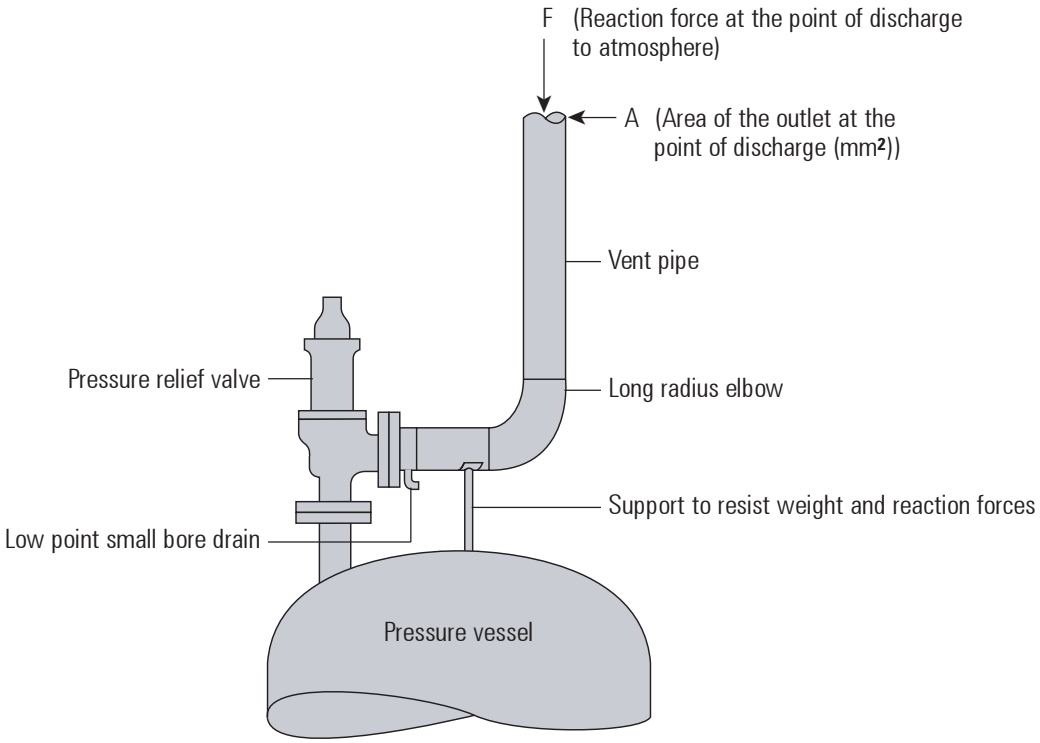


Fig. 9.5.7 Determination of the reaction forces generated in an open system

The reaction forces are typically small for safety valves with a nominal diameter of less than 75 mm, but safety valves larger than this usually have mounting flanges for a reaction bar on the body to allow the valve to be secured.

These reaction forces are typically negligible in closed systems, and they can therefore be ignored.

Regardless of the magnitude of the reaction forces, the safety valve itself should never be relied upon to support the discharge pipework itself and a support should be provided to resist the weight of the discharge pipework. This support should be located as close as possible to the centreline of the vent pipe (see Figure 9.5.7).

Figures 9.5.8 and 9.5.9 show typical safety valve installations for both open and closed systems.

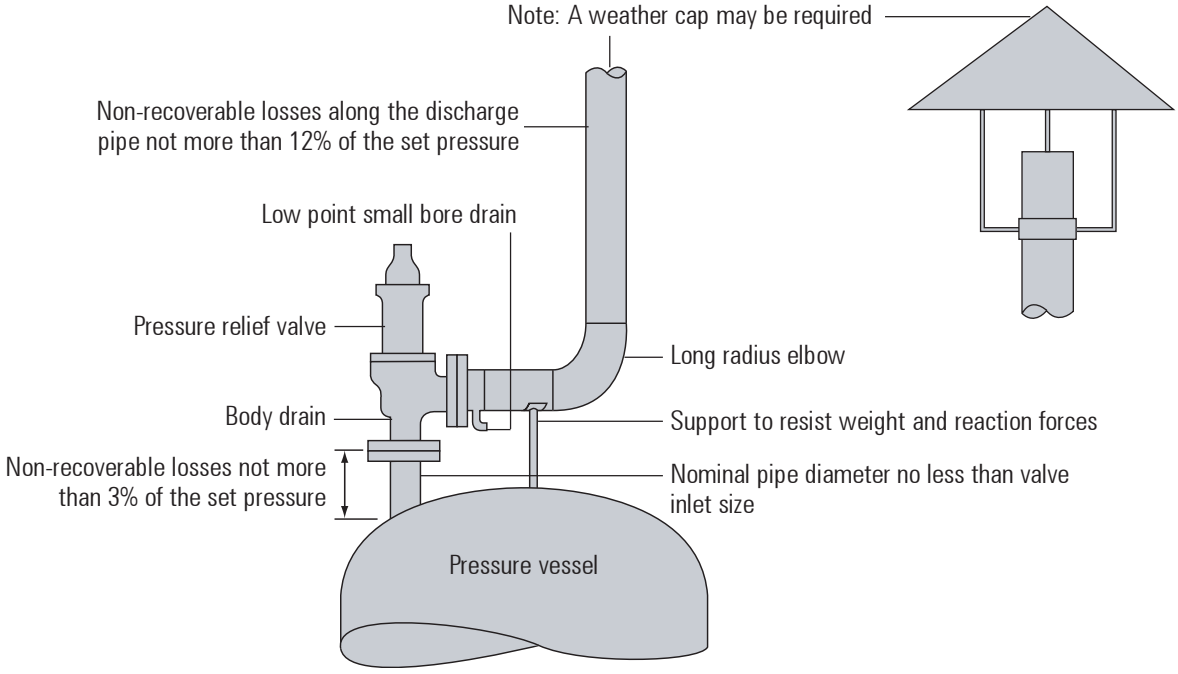


Fig. 9.5.8 A typical safety valve installation with open discharge system

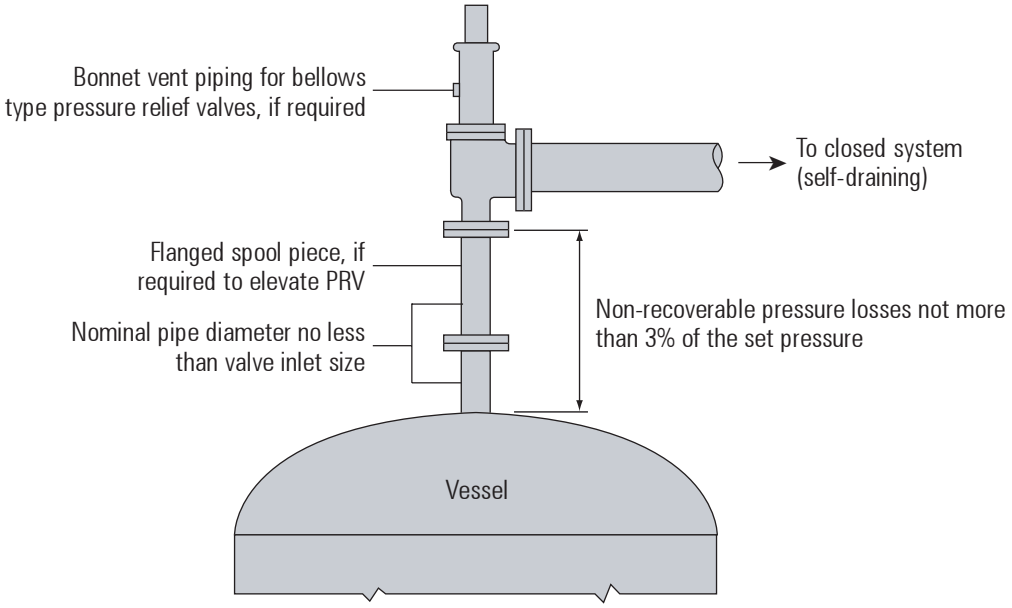


Fig. 9.5.9 A typical safety valve installation with closed discharge system

Changeover valves

Changeover valves (see Figure 9.5.10) permit two valves to be mounted side by side, with one in service and one isolated. This means regular maintenance can be carried out without interruption of service or the vessel being protected. Changeover valves are designed in such a way that when they are operated, the pass area is never restricted.

Changeover valves can also be used to connect safety valve outlets so that the discharge pipework does not have to be duplicated. The action of both inlet and outlet changeover valves has to be limited and synchronised for safety reasons. This is usually by means of a chain drive system linking both handwheels.

Consideration must be made to pressure loss caused by the changeover valve when establishing the safety valve inlet pressure drop, which should be limited to 3% of the set pressure.

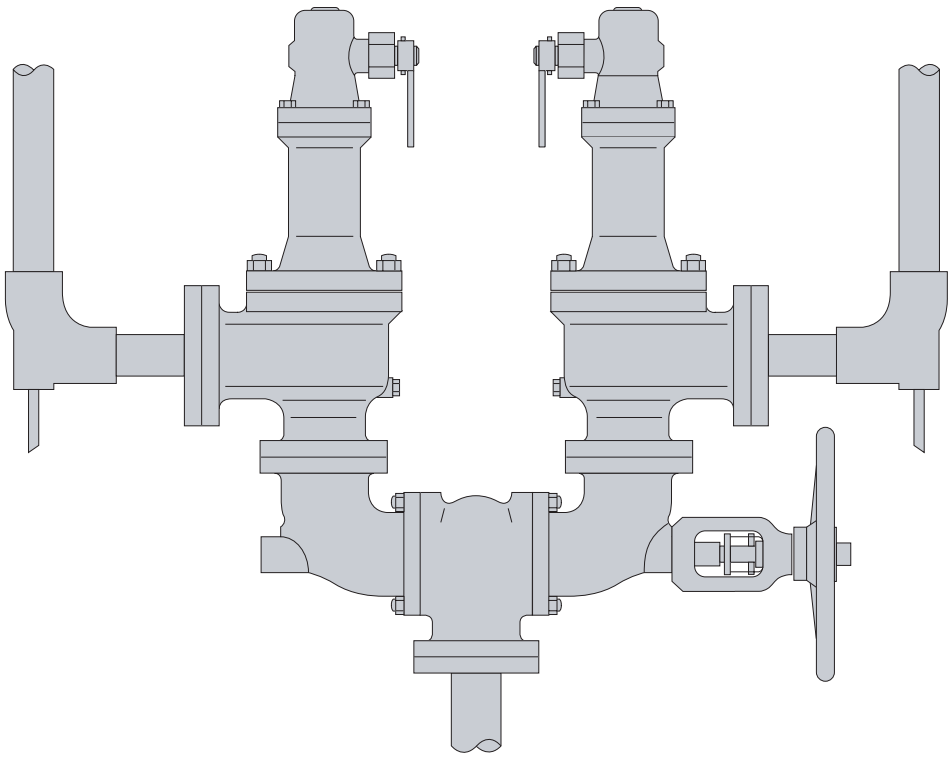


Fig. 9.5.10 Changeover valve

Noise emission

Although discharge from a safety valve should not occur frequently, the noise generated can often be significant. It is therefore necessary to determine the sound power level of safety valves to ensure that relevant health and safety regulation levels are not exceeded.

Assuming a sonic flow nozzle discharge, an approximate value of the sound power level, L_P, in decibels at a flange outlet can be calculated using the formula given in Equation 9.5.3.

$$L_P = 60 \log_{10} u + 10 \log_{10} \dot{m} - 52 \qquad \textbf{Equation 9.5.3}$$

Where:
L_P = Sound power level in dB (A)
$\dot{m}$ = Mass flow (kg/h)
u = Speed of sound in an ideal gas (m/s), $u = \sqrt{\frac{k R_u T}{M}}$
k = Isentropic coefficient of the gas
R_u = Universal gas constant (8 314 J/kmol K)
T = Absolute gas temperature at the safety valve outlet (K)
M = Molar mass (kg/kmol)

The sound pressure level (L) at a distance (R) is calculated from the sound power level (L_P) by using the formula given in Equation 9.5.4.

$$L = L_P - 10 \log_{10}(2 \pi R^2) \qquad \textbf{Equation 9.5.4}$$

Where:
L = Sound pressure level in dB (A)
L_P = Sound power level in dB (A)
R = Distance from the source (m)

There are several ways to reduce noise level, the simplest being to use larger diameter discharge pipes, or to lag the discharge pipe (however, the valve must not be lagged). It is also permissible for a silencer to be used in extreme cases, in which case any backpressure generated must then be taken into account.

EN ISO 4126 uses the following equations:

$$L_P = 20 \log (10^{-3} D_A) - 10 \log v_g + 80 \log u - 53 \qquad \textbf{Equation 9.5.5}$$

Where:
L_P = Sound power level in dB (A)
D_A = Discharge pipe bore (mm)
v_g = Specific volume at relieving pressure and temperature (m^3/kg)
u = Velocity of fluid in the outlet pipe (m/s)

The sound pressure level, L, is calculated using Equation 9.5.4.

Questions

1. Which of the following information is not normally included on the safety valve marking plate?

a| Set pressure ☐
b| Relieving pressure ☐
c| The relevant standard number ☐
d| Information relating to the capacity of the valve ☐

2. Which of the following must be taken into account when designing the inlet pipework for a safety valve?

i The inlet pressure drop must be less than 3%

ii The inlet pressure must be representative of the pressure in the protected apparatus

iii The safety valve should always be installed with the bonnet vertically upwards

a| i only ☐
b| ii only ☐
c| i and ii ☐
d| i, ii and iii ☐

3. Which of the following methods could help overcome safety valve 'chatter' caused by poor inlet pipework design?

i Increase the nominal diameter of the pipes

ii Increase the radius of any corners

iii Increase the pipe length

a| i only ☐
b| ii only ☐
c| i and ii ☐
d| i, ii and iii ☐

4. A safety valve manufactured according to EN ISO 4126 is used to discharge saturated steam along a 50 m long pipeline. Select a suitable nominal diameter for the discharge pipe if the safety valve is set at 7 bar g and has been sized to pass 1 500 kg/h.

a| 40 mm ☐
b| 50 mm ☐
c| 65 mm ☐
d| 80 mm ☐

5. A safety valve discharges 800 kg/h of saturated steam to the atmosphere through a 65 mm diameter vent pipe. If the safety valve has a relieving pressure of 1 bar g, determine the reaction force in newtons (N) acting on the safety valve. (Take k to be 1.135 for saturated steam)

a| 430 N ☐
b| 460 N ☐
c| 490 N ☐
d| 520 N ☐

6. In which of the following situations would it be necessary to install a changeover valve?

a| In a system where the safety valves require frequent maintenance ☐
b| In a steam system for which any downtime is costly ☐
c| To eliminate duplicating outlet pipework on a dual safety valve system ☐
d| All of the above ☐

Answers

1:b, 2: d, 3: c, 4: d, 5: a, 6: d

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Module 9.6

Alternative Plant Protection Devices and Terminology

Alternative Methods of Plant Protection

Although safety valves are by far the most common devices used for plant protection in steam systems, there are several other devices available to protect plant from overpressure conditions. Whilst some of them can be used in place of a safety valve, most have their own unique applications and indeed some devices, such as the bursting disc, may be used to complement the safety valve.

- **Weighted pallet -** This is the simplest type of overpressure protection device, and it is on low-pressure tanks and condensers, for pressure relief, vacuum relief or both.

  A weight is applied to the top of a disc, keeping it closed until the pressure acting on the underside of the pallet equals the weight. Due to the large weights required to keep a pallet closed, this type of valve is designed for low pressure applications below 0.1 bar. For higher set pressures, the weight required would be prohibitive and dangerous if oscillation of the pallet occurred at valve opening.

- **Counterweight safety valve -** Although these have been largely superseded by spring-loaded safety valves, they are still sometimes used for low-pressure applications. The closing force of the safety valve is provided by a weight rather than a spring. As the closing force is provided by a weight, it will remain constant and once the set pressure is reached, the safety valve will open fully.

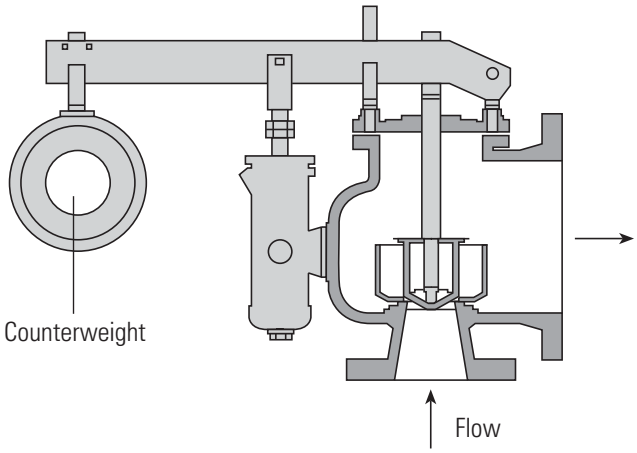


Fig. 9.6.1 A counterweight safety valve

- **Supplementary loaded safety valve -** A supplementary loaded safety valve consists of a conventional safety valve provided with an additional sealing force that is released once the set pressure is reached. One of the main concerns with this type of device is ensuring that the load is suitably released when the set pressure is reached. The BS 6759 standard states that even in the event of the release mechanism failing, the valve must attain its certified discharge capacity within 115% of the set pressure.

  Supplementary loaded safety valves tend only to be used where any leakage of the fluid below set pressure is unacceptable, or on very high pressure systems where maintaining a tight shut-off is otherwise difficult.



Fig. 9.6.2 Typical supplementary loaded safety valves

- **Controlled safety pressure relief systems (CSPRS) -** These are electric or electropneumatic systems, which are not self-acting. When an overpressure situation is detected, a control device acts to correct the situation.

Non-reclosing pressure relief devices

Non-reclosing devices are those which are designed to remain open after operation. A manual means of resetting is usually provided.

- **Bursting or rupture discs -** This consists of an elastomeric membrane or thin metal disk that will burst at a set pressure, relieving any overpressure. Although they can be used by themselves, on many applications, they are used in conjunction with a safety valve.

  A rupture disc can be installed either on the inlet or outlet side of the safety valve. If installed on the inlet, it isolates the contained media from the safety valve. When there is an overpressure situation; the rupture disc bursts allowing the fluid to flow into the safety valve, which will then subsequently lift. This arrangement is used to protect the internals of the safety valve from corrosive fluids.

  Alternatively, if the safety valve discharges into a manifold containing corrosive media, a rupture disc can be installed on the safety valve outlet, preventing any of the fluid from the manifold contacting the internals of the safety valve in normal use.

  Rupture discs can also be installed alongside a safety valve as a secondary relief device.

  Rupture discs are leak tight and low cost, but they require replacing after each operation. Most rupture disc installations contain a mechanism to indicate when the disc has ruptured and that it needs to be replaced. Typically, a pressure gauge is used (see Figure 9.6.3b).

  Explosion panels or explosion rupture discs are similar to rupture discs but are designed for use at higher rates of pressure rise, and for larger capacities.

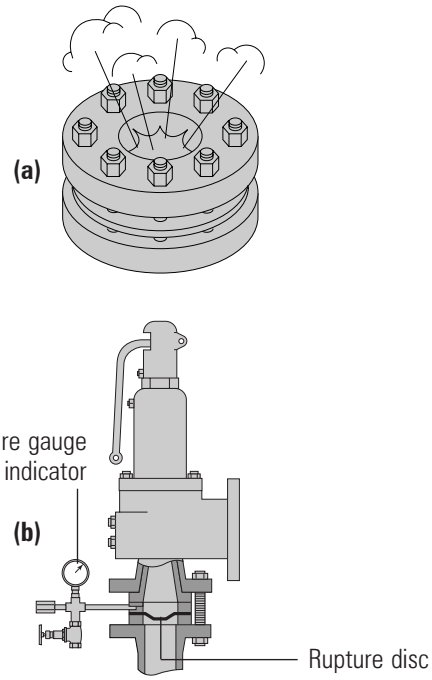


Fig. 9.6.3
A rupture/bursting disc device (a) and a rupture disk installed on the inlet of a safety valve (b)

- **Fusible plug devices -** These consist of a plug with a lower melting point than the maximum operating temperature of the system that it is to protect. In old steam locomotives, this type of device was used to dump the boiler water onto the fire if overtemperature occurred.

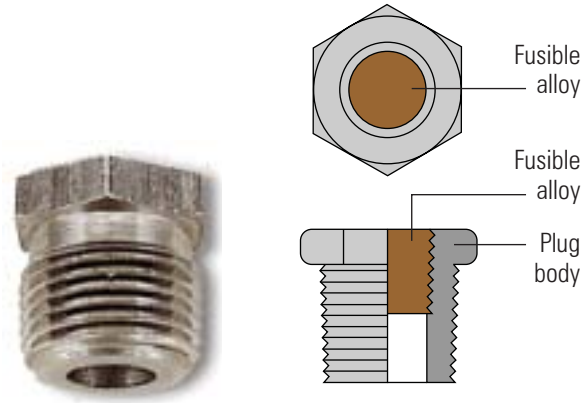


Fig. 9.6.4 An example of a fusible plug device

- **Breaking or shear pin devices -** A breaking pin device is a non-reclosing pressure relief device actuated by inlet static pressure and designed to function by the breakage of a load carrying section of a pin, which supports a pressure-containing member. The force of overpressure forces the pin to buckle and the valve to open. The valve can then be reseated after the pressure is removed and a new pin can be installed. These devices are usually installed on low-pressure applications and large gas distribution systems. They have limited process applications.

Terminology

The following definitions are taken from DIN 3320 but it should be noted that many of the terms and associated definitions used are universal and appear in many other standards. Where commonly used terms are not defined in DIN 3320 then ASME/ANSI PTC25.3 has been used as the source of reference. This list is not exhaustive and is intended as a guide only; it should not be used in place of the relevant current issue standard:

Operating pressure (working pressure) is the gauge pressure existing at normal operating conditions within the system to be protected.

Set pressure is the gauge pressure at which under operating conditions direct loaded safety valves commence to lift.

Test pressure is the gauge pressure at which under test stand conditions (atmospheric backpressure) direct loaded safety valves commence to lift.

Opening pressure is the gauge pressure at which the lift is sufficient to discharge the predetermined flowing capacity. It is equal to the set pressure plus opening pressure difference.

Reseating pressure is the gauge pressure at which the direct loaded safety valve is re-closed.

Built-up backpressure is the gauge pressure built up at the outlet side by blowing.

Superimposed backpressure is the gauge pressure on the outlet side of the closed valve.

Backpressure is the gauge pressure built up on the outlet side during blowing (built-up backpressure + superimposed backpressure).

Accumulation is the increase in pressure over the maximum allowable working gauge pressure of the system to be protected.

Opening pressure difference is the pressure rise over the set pressure necessary for a lift suitable to permit the predetermined flowing capacity.

Reseating pressure difference is the difference between set pressure and reseating pressure.

Functional pressure difference is the sum of opening pressure difference and reseating pressure difference.

Operating pressure difference is the pressure difference between set pressure and operating pressure.

Lift is the travel of the disc away from the closed position.

Commencement of lift (opening) is the first measurable movement of the disc or the perception of discharge noise.

Flow area is the cross sectional area upstream or downstream of the body seat calculated from the minimum diameter which is used to calculate the flow capacity without any deduction for obstructions.

Flow diameter is the minimum geometrical diameter upstream or downstream of the body seat.

Nominal size designation of a safety valve is the nominal size of the inlet.

Theoretical flowing capacity is the calculated mass flow from an orifice having a cross sectional area equal to the flow area of the safety valve without regard to flow losses of the valve.

Actual flowing capacity is the flowing capacity determined by measurement.

Certified flowing capacity is actual flowing capacity reduced by 10%.

Coefficient of discharge is the ratio of actual to the theoretical discharge capacity.

Certified coefficient of discharge is the coefficient of discharge reduced by 10% (also known as derated coefficient of discharge).

The following terms are not defined in DIN 3320 and are taken from ASME/ANSI PTC25.3:

Blowdown (reseating pressure difference) - difference between actual popping pressure and actual reseating pressure, usually expressed as a percentage of set pressure or in pressure units.

Cold differential test pressure the pressure at which a valve is set on a test rig using a test fluid at ambient temperature. This test pressure includes corrections for service conditions e.g. backpressure or high temperatures.

Flow rating pressure is the inlet static pressure at which the relieving capacity of a pressure relief device is measured.

Leak test pressure is the specified inlet static pressure at which a quantitative seat leakage test is performed in accordance with a standard procedure.

Measured relieving capacity is the relieving capacity of a pressure relief device measured at the flow rating pressure.

Rated relieving capacity is that portion of the measured relieving capacity permitted by the applicable code or regulation to be used as a basis for the application of a pressure relieving device.

Overpressure is a pressure increase over the set pressure of a pressure relief valve, usually expressed as a percentage of set pressure.

Popping pressure is the value of increasing static inlet pressure of a pressure relief valve at which there is a measurable lift, or at which the discharge becomes continuous as determined by seeing, feeling or hearing.

Relieving pressure is set pressure plus overpressure.

Simmer is the pressure zone between the set pressure and popping pressure.

Maximum operating pressure is the maximum pressure expected during system operation.

Maximum allowable working pressure (MAWP) is the maximum gauge pressure permissible at the top of a completed vessel in its operating position for a designated temperature.

Maximum allowable accumulated pressure (MAAP) is the maximum allowable working pressure plus the accumulation as established by reference to the applicable codes for operating or fire contingencies.

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Module 10.1

Introduction to Steam Distribution

Introduction to Steam Distribution

The steam distribution system is the essential link between the steam generator and the steam user.

This Module will look at methods of carrying steam from a central source to the point of use. The central source might be a boiler house or the discharge from a co-generation plant. The boilers may burn primary fuel, or be waste heat boilers using exhaust gases from high temperature processes, engines or even incinerators. Whatever the source, an efficient steam distribution system is essential if steam of the right quality and pressure is to be supplied, in the right quantity, to the steam using equipment. Installation and maintenance of the steam system are important issues, and must be considered at the design stage.

Steam system basics

From the outset, an understanding of the basic steam circuit, or 'steam and condensate loop' is required – see Figure 10.1.1. As steam condenses in a process, flow is induced in the supply pipe. Condensate has a very small volume compared to the steam, and this causes a pressure drop, which causes the steam to flow through the pipes.

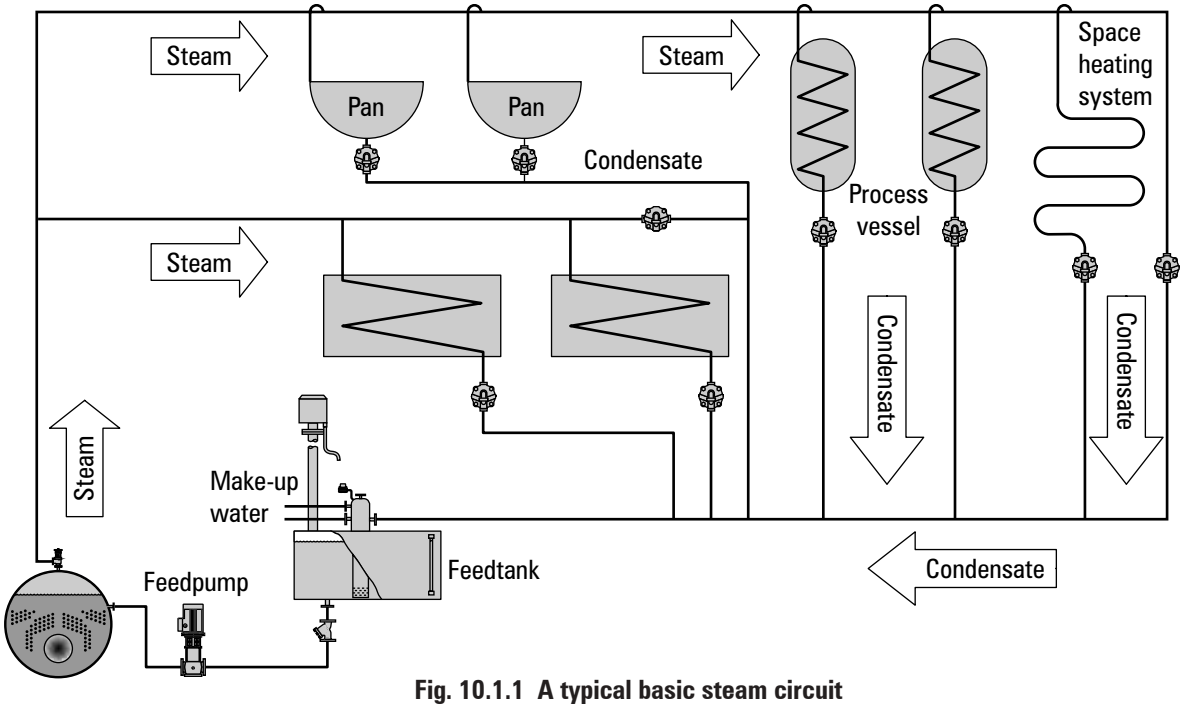


Fig. 10.1.1 A typical basic steam circuit

The steam generated in the boiler must be conveyed through pipework to the point where its heat energy is required. Initially there will be one or more main pipes, or 'steam mains', which carry steam from the boiler in the general direction of the steam using plant. Smaller branch pipes can then carry the steam to the individual pieces of equipment.

When the boiler main isolating valve (commonly called the 'crown' valve) is opened, steam immediately passes from the boiler into and along the steam mains to the points at lower pressure. The pipework is initially cooler than the steam, so heat is transferred from the steam to the pipe. The air surrounding the pipes is also cooler than the steam, so the pipework will begin to transfer heat to the air.

Steam on contact with the cooler pipes will begin to condense immediately. On start-up of the system, the condensing rate will be at its maximum, as this is the time where there is maximum temperature difference between the steam and the pipework. This condensing rate is commonly called the 'starting load'. Once the pipework has warmed up, the temperature difference between the steam and pipework is minimal, but some condensation will occur as the pipework still continues to transfer heat to the surrounding air. This condensing rate is commonly called the 'running load'.

The resulting condensation (condensate) falls to the bottom of the pipe and is carried along by the steam flow and assisted by gravity, due to the gradient in the steam main that should be arranged to fall in the direction of steam flow. The condensate will then have to be drained from various strategic points in the steam main.

When the valve on the steam pipe serving an item of steam using plant is opened, steam flowing from the distribution system enters the plant and again comes into contact with cooler surfaces. The steam then transfers its energy in warming up the equipment and product (starting load), and, when up to temperature, continues to transfer heat to the process (running load).

There is now a continuous supply of steam from the boiler to satisfy the connected load and to maintain this supply more steam must be generated. In order to do this, more water (and fuel to heat this water) is supplied to the boiler to make up for that water which has previously been evaporated into steam.

The condensate formed in both the steam distribution pipework and in the process equipment is a convenient supply of useable hot boiler feedwater. Although it is important to remove this condensate from the steam space, it is a valuable commodity and should not be allowed to run to waste. Returning all condensate to the boiler feedtank closes the basic steam loop, and should be practised wherever practical. The return of condensate to the boiler is discussed further in Block 13, 'Condensate Removal', and Block 14,'Condensate Management'.

The working pressure

The distribution pressure of steam is influenced by a number of factors, but is limited by:

- The maximum safe working pressure of the boiler.
- The minimum pressure required at the plant.

As steam passes through the distribution pipework, it will inevitably lose pressure due to:

- Frictional resistance within the pipework (detailed in Module 10.2).
- Condensation within the pipework as heat is transferred to the environment.

Therefore allowance should be made for this pressure loss when deciding upon the initial distribution pressure.

A kilogram of steam at a higher pressure occupies less volume than at a lower pressure. It follows that, if steam is generated in the boiler at a high pressure and also distributed at a high pressure, the size of the distribution mains will be smaller than that for a low-pressure system for the same heat load. Figure 10.1.2 illustrates this point.

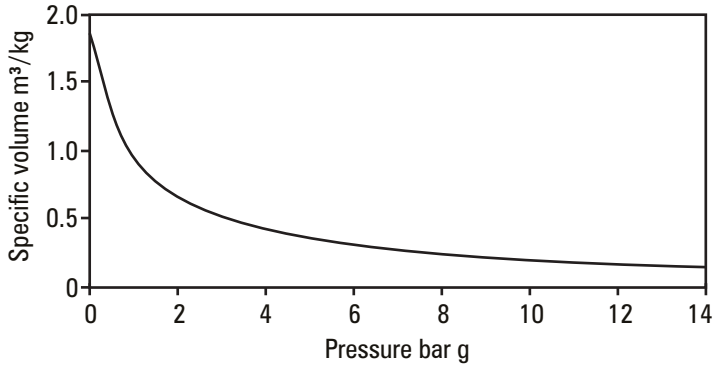


Fig. 10.1.2 Dry saturated steam - pressure/specific volume relationship

Generating and distributing steam at higher pressure offers three important advantages:

- The thermal storage capacity of the boiler is increased, helping it to cope more efficiently with fluctuating loads, minimising the risk of producing wet and dirty steam.
- Smaller bore steam mains are required, resulting in lower capital cost, for materials such as pipes, flanges, supports, insulation and labour.
- Smaller bore steam mains cost less to insulate.

Having distributed at a high pressure, it will be necessary to reduce the steam pressure to each zone or point of use in the system in order to correspond with the maximum pressure required by the application. Local pressure reduction to suit individual plant will also result in drier steam at the point of use. (Module 2.3 provides an explanation of this).

Note: It is sometimes thought that running a steam boiler at a lower pressure than its rated pressure will save fuel. This logic is based on more fuel being needed to raise steam to a higher pressure.

Whilst there is an element of truth in this logic, it should be remembered that it is the connected load, and not the boiler output, which determines the rate at which energy is used. The same amount of energy is used by the load whether the boiler raises steam at 4 bar g, 10 bar g or 100 bar g. Standing losses, flue losses, and running losses are increased by operating at higher pressures, but these losses are reduced by insulation and proper condensate return systems. These losses are marginal when compared to the benefits of distributing steam at high pressure.

Pressure reduction

The common method for reducing pressure at the point where steam is to be used is to use a pressure reducing valve, similar to the one shown in the pressure reducing station Figure 10.1.3.

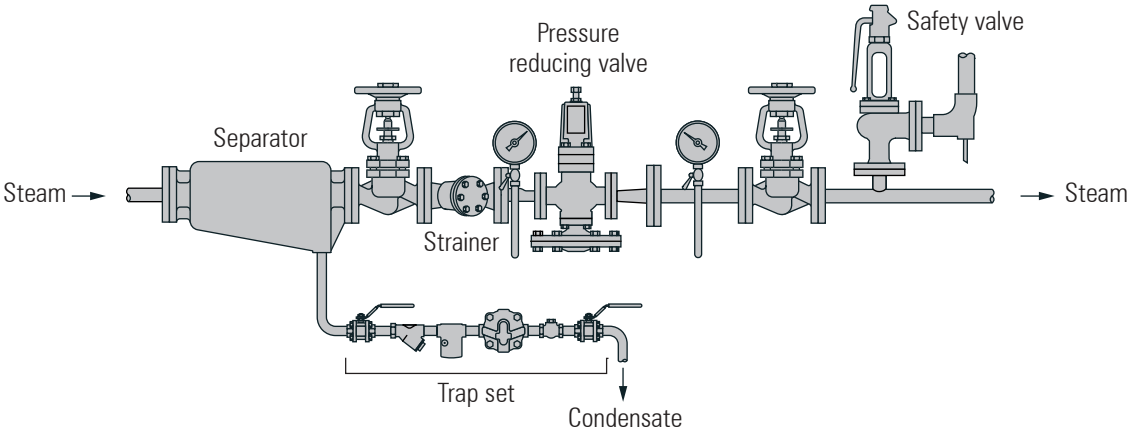


Fig. 10.1.3 Typical pressure reducing valve station

A separator is installed upstream of the reducing valve to remove entrained water from incoming wet steam, thereby ensuring high quality steam to pass through the reducing valve. This is discussed in more detail in Module 9.3 and Module 12.5.

Plant downstream of the pressure reducing valve is protected by a safety valve. If the pressure reducing valve fails, the downstream pressure may rise above the maximum allowable working pressure of the steam using equipment. This, in turn, may permanently damage the equipment, and, more importantly, constitute a danger to personnel.

With a safety valve fitted, any excess pressure is vented through the valve, and will prevent this from happening (safety valves are discussed in Block 9).

Other components included in the pressure reducing valve station are:

- **The primary isolating valve -** To shut the system down for maintenance.
- **The primary pressure gauge -** To monitor the integrity of supply.
- **The strainer -** To keep the system clean.
- **The secondary pressure gauge -** To set and monitor the downstream pressure.
- **The secondary isolating valve -** To assist in setting the downstream pressure on no-load conditions.

Questions

1. Distributing steam at high pressure, instead of low pressure, will have the following effect.

a | Heat losses from the pipes will be less. ☐
b | A lower storage capacity in the high pressure pipes. ☐
c | High pressure small bore steam pipes cost less to install and insulate. ☐
d | The steam pipes will be smaller creating wet steam. ☐

2. A steam pressure reducing valve is fitted to:

a | Prevent the pressure at the plant exceeding its safe working pressure. ☐
b | Help dry the steam supply to the plant. ☐
c | Reduce the flash steam losses as condensate passes through the plant steam traps. ☐
d | Supply the plant with steam at the designed temperature and pressure. ☐

3. The start-up condensate load of a steam main is generally greater than the running load because:

a | The pipework and fittings are cold, so steam is required to heat it up to steam temperature. ☐
b | The steam space within the pipework has to be charged with steam to the desired running pressure. ☐
c | The boiler crown valve or stop valve is opened very slowly and initially there is insufficient pressure to discharge condensate through the steam traps. ☐
d | On initial opening of the crown valve, the steam distribution pressure will be low and the enthalpy of evaporation of low pressure steam is greater than at high pressure so a greater mass of steam will be condensed. ☐

4. The pressure at which steam is supplied to the plant should be dictated by:

a | The boiler operating pressure. ☐
b | The steam distribution pressure. ☐
c | The maximum allowable safe working pressure of the plant. ☐
d | The plant design pressure and temperature. ☐

5. Which of the following results in pressure losses in distribution pipework?

a | Sizing the pipes on low pressure instead of high pressure. ☐
b | Frictional resistance within and heat loss from the pipe and fittings. ☐
c | Sizing the pipes on start-up load of the plant. ☐
d | Large steam users. ☐

6. The steam pipe after a pressure reducing valve is likely to be:

a | Smaller than the upstream pipe because of the smaller volume of low pressure steam. ☐
b | The same size as the connection to the plant. ☐
c | Larger than the upstream pipe because the volume of the low pressure steam is greater. ☐
d | The same size as the upstream pipe because the flowrate through each pipe is the same. ☐

Answers

1: c, 2: d, 3: a, 4: d, 5: b 6: c

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Module 10.2

Pipes and Pipe Sizing

Pipes and Pipe Sizing

Standards and wall thickness

There are a number of piping standards in existence around the world, but arguably the most global are those derived by the American Petroleum Institute (API), where pipes are categorised in schedule numbers.

These schedule numbers bear a relation to the pressure rating of the piping. There are eleven Schedules ranging from the lowest at 5 through 10, 20, 30, 40, 60, 80, 100, 120, 140 to schedule No. 160. For nominal size piping 150 mm and smaller, Schedule 40 (sometimes called 'standard weight') is the lightest that would be specified for steam applications.

Regardless of schedule number, pipes of a particular size all have the same outside diameter (not withstanding manufacturing tolerances). As the schedule number increases, the wall thickness increases, and the actual bore is reduced. For example:

- A 100 mm Schedule 40 pipe has an outside diameter of 114.30 mm, a wall thickness of 6.02 mm, giving a bore of 102.26 mm.
- A 100 mm Schedule 80 pipe has an outside diameter of 114.30 mm, a wall thickness of 8.56 mm, giving a bore of 97.18 mm.

Only Schedules 40 and 80 cover the full range from 15 mm up to 600 mm nominal sizes and are the most commonly used schedule for steam pipe installations.

This Module considers Schedule 40 pipework as covered in BS 1600.

Tables of schedule numbers can be obtained from BS 1600 which are used as a reference for the nominal pipe size and wall thickness in millimetres. Table 10.2.1 compares the actual bore sizes of different sized pipes, for different schedule numbers.

In mainland Europe, pipe is manufactured to DIN standards, and DIN 2448 pipe is included in Table 10.2.1.

Table 10.2.1 Comparison of pipe standards and actual bore diameters.

Nominal size pipe (mm)		15	20	25	32	40	50	65	80	100	150
Bore (mm)	Schedule 40	15.8	21.0	26.6	35.1	40.9	52.5	62.7	77.9	102.3	154.1
	Schedule 80	13.8	18.9	24.3	32.5	38.1	49.2	59.0	73.7	97.2	146.4
	Schedule 160	11.7	15.6	20.7	29.5	34.0	42.8	53.9	66.6	87.3	131.8
	DIN 2448	17.3	22.3	28.5	37.2	43.1	60.3	70.3	82.5	107.1	159.3

In the United Kingdom, piping to EN 10255, (steel tubes and tubulars suitable for screwing to BS 21 threads) is also used in applications where the pipe is screwed rather than flanged. They are commonly referred to as 'Blue Band' and 'Red Band'; this being due to their banded identification marks. The different colours refer to particular grades of pipe:

- Red Band, being heavy grade, is commonly used for steam pipe applications.
- Blue Band, being medium grade, is commonly used for air distribution systems, although it is sometimes used for low-pressure steam systems.

The coloured bands are 50 mm wide, and their positions on the pipe denote its length. Pipes less than 4 metres in length only have a coloured band at one end, while pipes of 4 to 7 metres in length have a coloured band at either end.

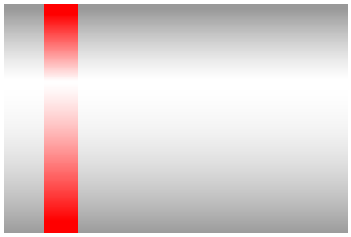


Fig. 10.2.1 Red band, branded pipe, - heavy grade, up to 4 metres in length

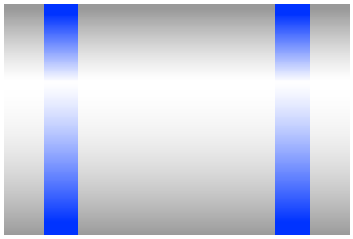


Fig. 10.2.2 Blue band, branded pipe, - medium grade, between 4-7 metres in length

Pipe material

Pipes for steam systems are commonly manufactured from carbon steel to ANSI B 16.9 A106. The same material may be used for condensate lines, although copper tubing is preferred in some industries.

For high temperature superheated steam mains, additional alloying elements, such as chromium and molybdenum, are included to improve tensile strength and creep resistance at high temperatures.

Typically, pipes are supplied in 6 metre lengths.

Pipeline sizing

The objective of any fluid distribution system is to supply the fluid at the correct pressure to the point of use. It follows, therefore, that pressure drop through the distribution system is an important feature.

Liquids

Bernoulli's Theorem (Daniel Bernoulli 1700 - 1782) is discussed in Block 4 - Flowmetering. D'Arcy (D'Arcy Thompson 1860 - 1948) added that for fluid flow to occur, there must be more energy at Point 1 than Point 2 (see Figure 10.2.3). The difference in energy is used to overcome frictional resistance between the pipe and the flowing fluid.

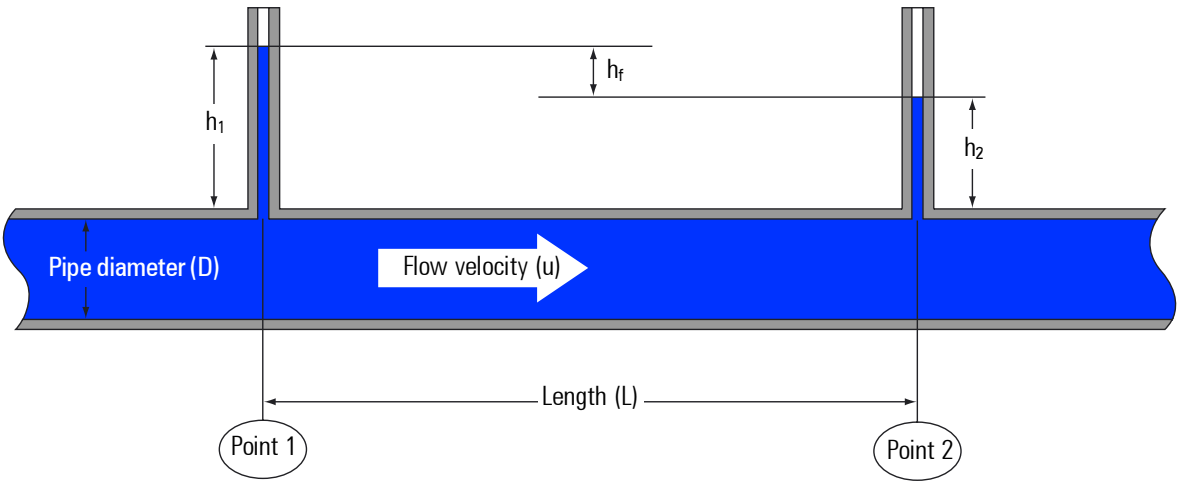


Fig. 10.2.3 Friction in pipes due to the flow of the fluid

Bernoulli relates changes in the total energy of a flowing fluid to energy dissipation expressed either in terms of a head loss h_f (m) or specific energy loss $g\ h_f$ (J/kg). This, in itself, is not very useful without being able to predict the pressure losses that will occur in particular circumstances.

Here, one of the most important mechanisms of energy dissipation within a flowing fluid is introduced, that is, the loss in total mechanical energy due to friction at the wall of a uniform pipe carrying a steady flow of fluid.

The loss in the total energy of fluid flowing through a circular pipe must depend on:

L = The length of the pipe (m)
D = The pipe diameter (m)
u = The mean velocity of the fluid flow (m/s)
μ = The dynamic viscosity of the fluid (kg/m s = Pa s)
ρ = The fluid density (kg/m³)
k_s = The roughness of the pipe wall* (m)

*Since the energy dissipation is associated with shear stress at the pipe wall, the nature of the wall surface will be influential, as a smooth surface will interact with the fluid in a different way than a rough surface.

All these variables are brought together in the D'Arcy-Weisbach equation (often referred to as the D'Arcy equation), and shown as Equation 10.2.1. This equation also introduces a dimensionless term referred to as the friction factor, which relates the absolute pipe roughness to the density, velocity and viscosity of the fluid and the pipe diameter.

The term that relates fluid density, velocity and viscosity and the pipe diameter is called the Reynolds number, named after Osborne Reynolds (1842-1912, of Owens College, Manchester, United Kingdom), who pioneered this technical approach to energy losses in flowing fluids circa 1883.

The D'Arcy equation (Equation 10.2.1):

$$h_f = \frac{4 f L u^2}{2 g D}$$ **Equation 10.2.1**

Where:
h_f = Head loss to friction (m)
f = Friction factor (dimensionless)
L = Length (m)
u = Flow velocity (m/s)
g = Gravitational constant (9.81 m/s²)
D = Pipe diameter (m)

Interesting point
Readers in some parts of the world may recognise the D'Arcy equation in a slightly different form, as shown in Equation 10.2.2. Equation 10.2.2 is similar to Equation 10.2.1 but does not contain the constant 4.

$$h_f = \frac{f L u^2}{2 g D}$$ **Equation 10.2.2**

The reason for the difference is the type of friction factor used. It is essential that the right version of the D'Arcy equation be used with the selected friction factor. Matching the wrong equation to the wrong friction factor will result in a 400% error and it is therefore important that the correct combination of equation and friction factor is utilised. Many textbooks simply do not indicate which friction factors are defined, and a judgement must sometimes be based on the magnitudes quoted.

Equation 10.2.2 tends to be used by those who traditionally work in Imperial units, and still tends to be used by practitioners in the United States and Pacific rim regions even when metric pipe sizes are quoted. Equation 10.2.1 tends to be used by those who traditionally work in SI units and tends more to be used by European practitioners. For the same Reynolds number and relative roughness, the 'Imperial based friction factor' will be exactly four times larger than the 'SI based friction factor'.

Friction factors can be determined either from a Moody chart or, for turbulent flows, can be calculated from Equation 10.2.3, a development of the Colebrook - White formula.

$$\frac{1}{\sqrt{f}} = 3.48 - 1.74 \ln\left(\frac{2 k_s}{D} + \frac{9.3}{R_e \sqrt{f}}\right)$$ **Equation 10.2.3**

Where:
f = Friction factor (Relates to the SI Moody chart)
k_s = Absolute pipe roughness (m)
D = Pipe bore (m)
R_e = Reynolds number (dimensionless)

However, Equation 10.2.3 is difficult to use because the friction factor appears on both sides of the equation, and it is for this reason that manual calculations are likely to be carried out by using the Moody chart.

On an SI style Moody chart, the friction factor scale might typically range from 0.002 to 0.02, whereas on an Imperial style Moody chart, this scale might range from 0.008 to 0.08.

As a general rule, for turbulent flow with Reynolds numbers between 4 000 and 100 000, 'SI based' friction factors will be of the order suggested by Equation 10.2.4, whilst 'Imperial based' friction factors will be of the order suggested by Equation 10.2.5.

'SI based' friction factors

$$f \approx \frac{0.08}{\sqrt[4]{R_e}}$$

Equation 10.2.4

'Imperial based' friction factors

$$f \approx \frac{0.03}{\sqrt[4]{R_e}}$$

Equation 10.2.5

The friction factor used will determine whether the D'Arcy Equation 10.2.1 or 10.2.2 is used. For 'SI based' friction factors, use Equation 10.2.1; for 'Imperial based' friction factors, use Equation 10.2.2.

Example 10.2.1 Water pipe
Determine the velocity, friction factor and the difference in pressure between two points 1 km apart in a 150 mm constant bore horizontal pipework system if the water flowrate is 45 m³/h at 15°C.

$$\text{Velocity (m/s)} = \frac{\text{Volume flowrate (m}^3\text{/s)}}{\text{Cross sectional area (m}^2\text{)}}$$

$$\text{Velocity} = \frac{45 \text{ m}^3\text{/h} \times 4}{3\,600 \text{ s/h} \times \pi \times 0.15^2}$$

$$\textbf{Velocity} = \textbf{0.71 m/s}$$

In essence, the friction factor depends on the Reynolds number (Re) of the flowing liquid and the relative roughness (k_s/d) of the inside of the pipe; the former calculated from Equation 10.2.6, and the latter from Equation 10.2.7.

Reynolds number (Re)

$$R_e = \frac{\rho \, u \, D}{\mu}$$

Equation 10.2.6

Where:
R_e = Reynolds number
ρ = Density of water = 1 000 kg/m³
u = Velocity of water = 0.71 m/s
D = Pipe diameter = 0.15 m
μ = Dynamic viscosity of water (at 15°C) = 1.138×10^{-3} kg/m s (from steam tables)

From Equation 10.2.6: $R_e = \frac{1000 \times 0.71 \times 0.15}{1.138 \times 10^{-3}}$

$$\textbf{R}_e \textbf{ = 93585}$$

The pipe roughness or 'k_S' value (often quoted as 'ε' in some texts) is taken from standard tables, and for 'commercial steel pipe' would generally be taken as 0.000 045 metres.

From this the relative roughness is determined (as this is what the Moody chart requires).

Relative pipe roughness $\left(\frac{k_S}{D}\right)$ = Pipe roughness / Pipe bore	**Equation 10.2.7**

From Equation 10.2.7 relative roughness $\left(\frac{k_S}{D}\right) = \frac{0.000\,045 \text{ m}}{0.15 \text{ m}}$

Relative roughness = 0.0003

The friction factor can now be determined from the Moody chart and the friction head loss calculated from the relevant D'Arcy Equation.

From the European Moody chart (Figure 10.2.4),
Where: $k_S/D = 0.000\,3$ $R_e = 93\,585$: **Friction factor (f) = 0.005**

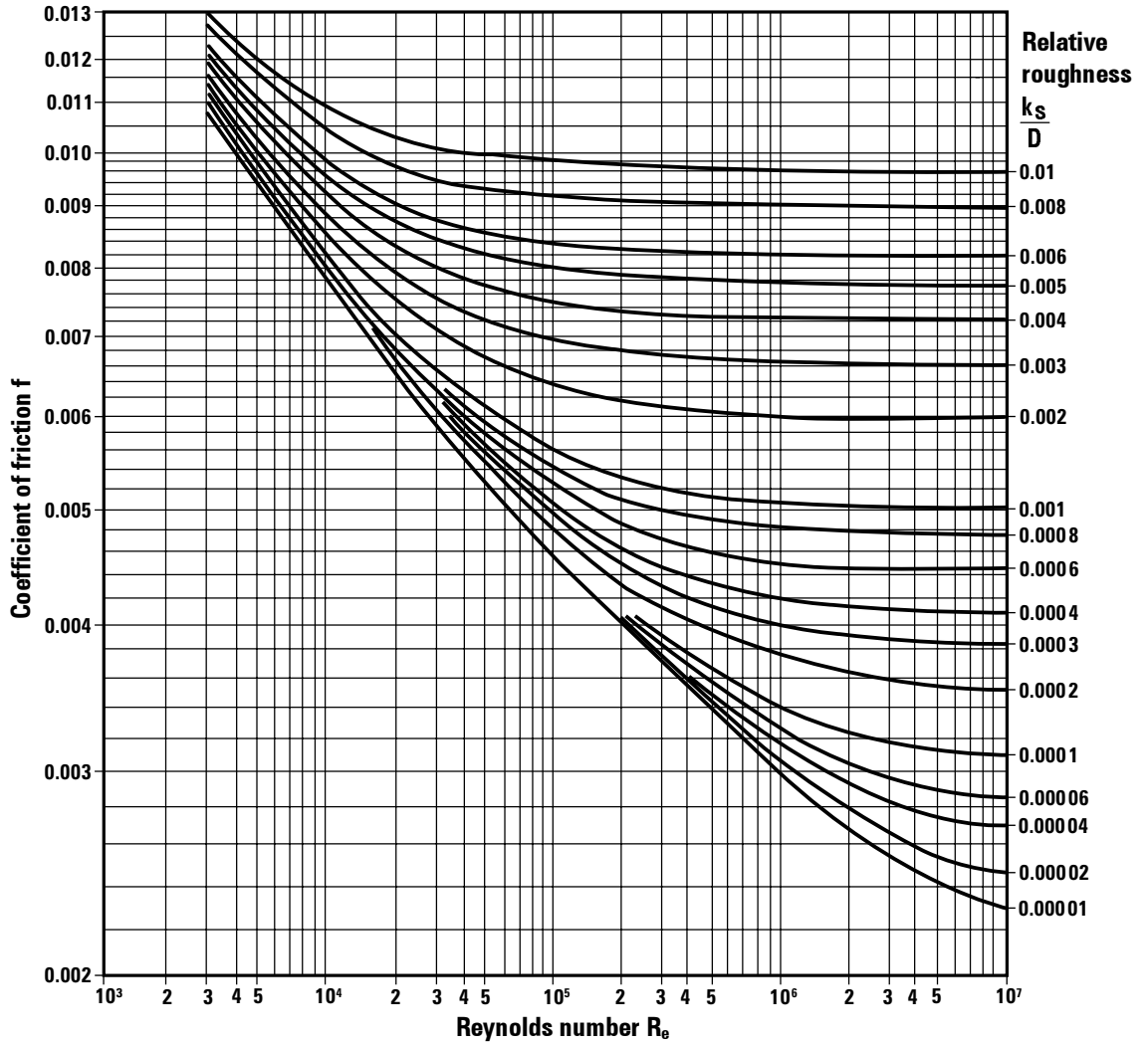


Fig. 10.2.4 'SI based' Moody chart (abridged)

From the European D'Arcy equation (Equation 10.2.1):

$$h_f = \frac{4 f L u^2}{2 g D}$$

$$h_f = \frac{4 \times 0.005 \times 1000 \times 0.71^2}{2 \times 9.81 \times 0.15}$$

h_f = 3.43 metres head loss

From the **USA** / **AUS** Moody chart (Figure 10.2.5),
Where: $k_s/D = 0.0003$ $R_e = 93\,585$ **Friction factor (f) = 0.02**

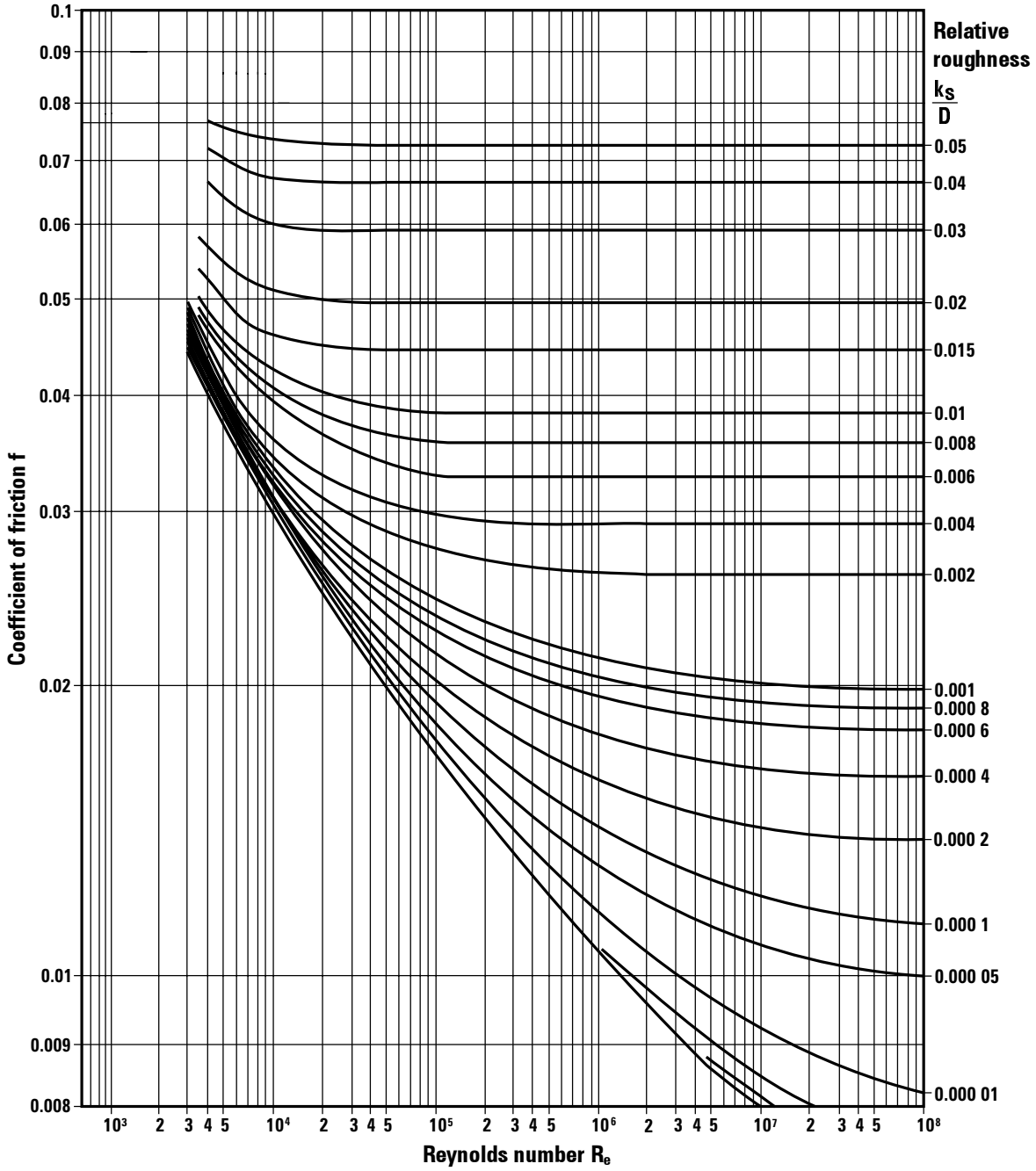


Fig. 10.2.5 'Imperial based' Moody chart (abridged)

From the **USA** / **AUS** D'Arcy equation (Equation 10.2.2):

$$h_f = \frac{f\,L\,u^2}{2\,g\,D}$$

$$h_f = \frac{0.02 \times 1\,000 \times 0.71^2}{2 \times 9.81 \times 0.15}$$

h_f = **3.43 metres head loss**

The same friction head loss is obtained by using the different friction factors and relevant D'Arcy equations.

In practice whether for water pipes or steam pipes, a balance is drawn between pipe size and pressure loss.

Steam

Oversized pipework means:

- Pipes, valves, fittings, etc. will be more expensive than necessary.
- Higher installation costs will be incurred, including support work, insulation, etc.
- For steam pipes a greater volume of condensate will be formed due to the greater heat loss. This, in turn, means that either:
  - More steam trapping is required, or
  - Wet steam is delivered to the point of use.

In a particular example:

- The cost of installing 80 mm steam pipework was found to be 44% higher than the cost of 50 mm pipework, which would have had adequate capacity.
- The heat lost by the insulated pipework was some 21% higher from the 80 mm pipeline than it would have been from the 50 mm pipework. Any non-insulated parts of the 80 mm pipe would lose 50% more heat than the 50 mm pipe, due to the extra heat transfer surface area.

Undersized pipework means:

- A lower pressure might be available at the point of use, which may hinder equipment performance.
- There is a risk of steam starvation due to an excessive pressure drop.
- There is a greater risk of erosion, waterhammer and noise due to the inherent increase in steam velocity.

As previously mentioned, the friction factor (f) can be difficult to determine, and the calculation itself is time consuming especially for turbulent steam flow. As a result, there are numerous graphs, tables and slide rules available for relating steam pipe sizes to flowrates and pressure drops.

One pressure drop sizing method, which has stood the test of time, is the 'pressure factor' method. A table of pressure factor values is used in Equation 10.2.8 to determine the pressure drop for a particular installation.

$$F = \frac{P_1 - P_2}{L}$$ **Equation 10.2.8**

Where:
F = Pressure factor (Used in Table 10.2.6)
P_1 = Factor based on the inlet pressure (Taken from Table 10.2.5)
P_2 = Factor based on the pressure at a distance of L metres (Taken from Table 10.2.5)
L = Equivalent length of pipe (m)

Example 10.2.2

Consider the system shown in Figure 10.2.6, and determine the pipe size required from the boiler to the unit heater branch line. Unit heater steam load = 270 kg/h.

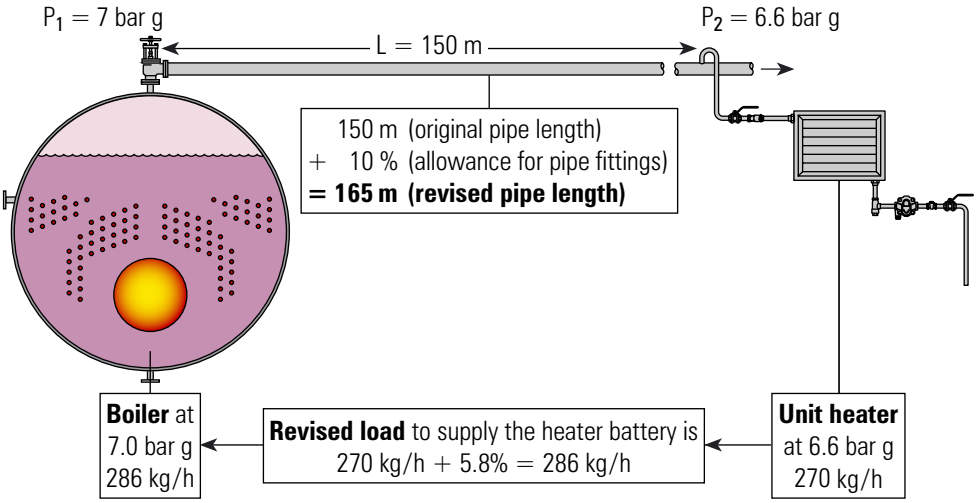


Fig. 10.2.6 System used to illustrate Example 10.2.2

Although the unit heater only requires 270 kg/h, the boiler has to supply more than this due to heat losses from the pipe.

The allowance for pipe fittings

The length of travel from the boiler to the unit heater is known, but an allowance must be included for the additional frictional resistance of the fittings. This is generally expressed in terms of 'equivalent pipe length'. If the size of the pipe is known, the resistance of the fittings can be calculated. As the pipe size is not yet known in this example, an addition to the equivalent length can be used based on experience.

- If the pipe is less than 50 metres long, add an allowance for fittings of 5%.
- If the pipe is over 100 metres long and is a fairly straight run with few fittings, an allowance for fittings of 10% would be made.
- A similar pipe length, but with more fittings, would increase the allowance towards 20%.

In this instance, **revised length** = 150 m + 10% **= 165 m**

The allowance for the heat losses from the pipe

The unit heater requires 270 kg/h of steam; therefore the pipe must carry this quantity plus the quantity of steam condensed by heat losses from the main. As the size of the main is yet to be determined, the true calculations cannot be made, but, assuming that the main is insulated, it may be reasonable to add 3.5% of the steam load per 100 m of the revised length as heat losses.

In this instance, the **additional allowance** = $\frac{165}{100}$ x 3.5% **= 5.8%**

Revised boiler load = 270 kg/h + 5.8% **= 286 kg/h**

From Table 10.2.2 (an extract from the complete pressure factor table, Table 10.2.5, which can be found in the Appendix at the end of this Module) 'F' can be determined by finding the pressure factors P_1 and P_2, and substituting them into Equation 10.2.8.

Table 10.2.2 Extract from pressure factor table (Table 10.2.5)

Pressure bar g	Pressure factor (P)
6.5	49.76
6.6	51.05
6.7	52.36
6.9	55.02
7.0	56.38
7.1	57.75

From the pressure factor table (see Table 10.2.2):

P_1 (7.0 bar g) = 56.38 P_2 (6.6 bar g) = 51.05

Substituting these pressure factors (P_1 and P_2) into Equation 10.2.8 will determine the value for 'F':

$$F = \frac{P_1 - P_2}{L}$$ **Equation 10.2.8**

$$F = \frac{56.38 - 51.05}{165 \text{ m}}$$

$$\mathbf{F = 0.032}$$

Following down the left-hand column of the pipeline capacity and pressure drop factors table (Table 10.2.6 - Extract shown in Table 10.2.3); the nearest two readings around the requirement of 0.032 are 0.030 and 0.040. The next lower factor is always selected; in this case, 0.030.

Table 10.2.3 Extract from pipeline capacity and pressure factor table (Table 10.2.6)

Factor (F)	Pipe size (DN)										
	15	20	25	32	40	50	65	80	100	150	200
	Capacity (kg/h)										
0.025	10.99	33.48	70.73	127.3	209.8	459.7	834.6	1 367	2 970	8 817	19 332
0.030	12.00	36.78	77.23	137.9	229.9	**501.1**	919.4	1 480	3 264	9 792	20 917
0.040	14.46	44.16	93.17	169.2	279.5	600.7	1 093	1 790	3 923	11 622	25 254

Although values can be interpolated, the table does not conform exactly to a straight-line graph, so interpolation cannot be absolutely correct. Also, it is bad practice to size any pipe up to the limit of its capacity, and it is important to have some leeway to allow for the inevitable future changes in design.

From factor 0.030, by following the row of figures to the right it will be seen that:

- A 40 mm pipe will carry 229.9 kg/h.
- A 50 mm pipe will carry 501.1 kg/h.

Since the application requires 286 kg/h, the 50 mm pipe would be selected.

Having sized the pipe using the pressure drop method, the velocity can be checked if required.

$$\text{Steam velocity} = \frac{\text{Volume flow } (m^3/s)}{\text{Cross sectional area of pipe } (m^2)} \text{ m/s}$$

$$\text{Steam velocity} = \frac{\text{Steam flowrate } (kg/h) \times v_g\,(m^3/kg) \times 4}{3\,600 \text{ s/h} \times \pi \times D^2\,(m^2)} \text{ m/s}$$

Where: Steam flowrate = 286 kg/h (revised load)

Specific volume (v_g) = 0.24 m^3/kg (At 7 bar g)

Pipe diameter (D) = 0.05 m (calculated above)

$$\text{Steam velocity} = \frac{286 \text{ kg/h} \times 0.24 \text{ m}^3/\text{kg} \times 4}{3\,600 \text{ s/h} \times \pi \times 0.05^2 \text{ (m}^2\text{)}} \text{ m/s}$$

Steam velocity = 9.7 m/s

Viewed in isolation, this velocity may seem low in comparison with maximum permitted velocities. However, this steam main has been sized to limit pressure drop, and the next smaller pipe size would have given a velocity of over 47 m/s, and a final pressure less than the requirement of 6.6 bar g, which is unacceptable.

As can be seen, this procedure is fairly complex and can be simplified by using the nomogram shown in Figure 10.2.9 (in the Appendix of this Module). The method of use is explained in Example 10.2.3.

Example 10.2.3
Using the data from Example 10.2.2, determine the pressure drop using the nomogram shown in Figure 10.2.7.

$$\text{Inlet pressure} = 7 \text{ bar g}$$

$$\text{Steam flowrate} = 286 \text{ kg/h}$$

$$\text{Minimum allowable } P_2 = 6.6 \text{ bar g}$$

$$\text{Maximum pressure drop per 100 m} = \frac{(P_1 - P_2)}{L} \times 100$$

$$\text{Maximum pressure drop per 100 m} = \frac{(7.0 - 6.6)}{165} \times 100$$

Maximum pressure drop per 100 m = 0.24 bar

Method:

- Select the point on the saturated steam line at 7 bar g, and mark Point **A**.
- From point **A**, draw a horizontal line to the steam flowrate of 286 kg/h, and mark Point **B**.
- From point **B**, draw a vertical line towards the top of the nomogram (Point **C**).
- Draw a horizontal line from 0.24 bar/100 m on the pressure loss scale (Line **DE**).
- The point at which lines **DE** and **BC** cross will indicate the pipe size required. In this case, a 40 mm pipe is too small, and a 50 mm pipe would be used.

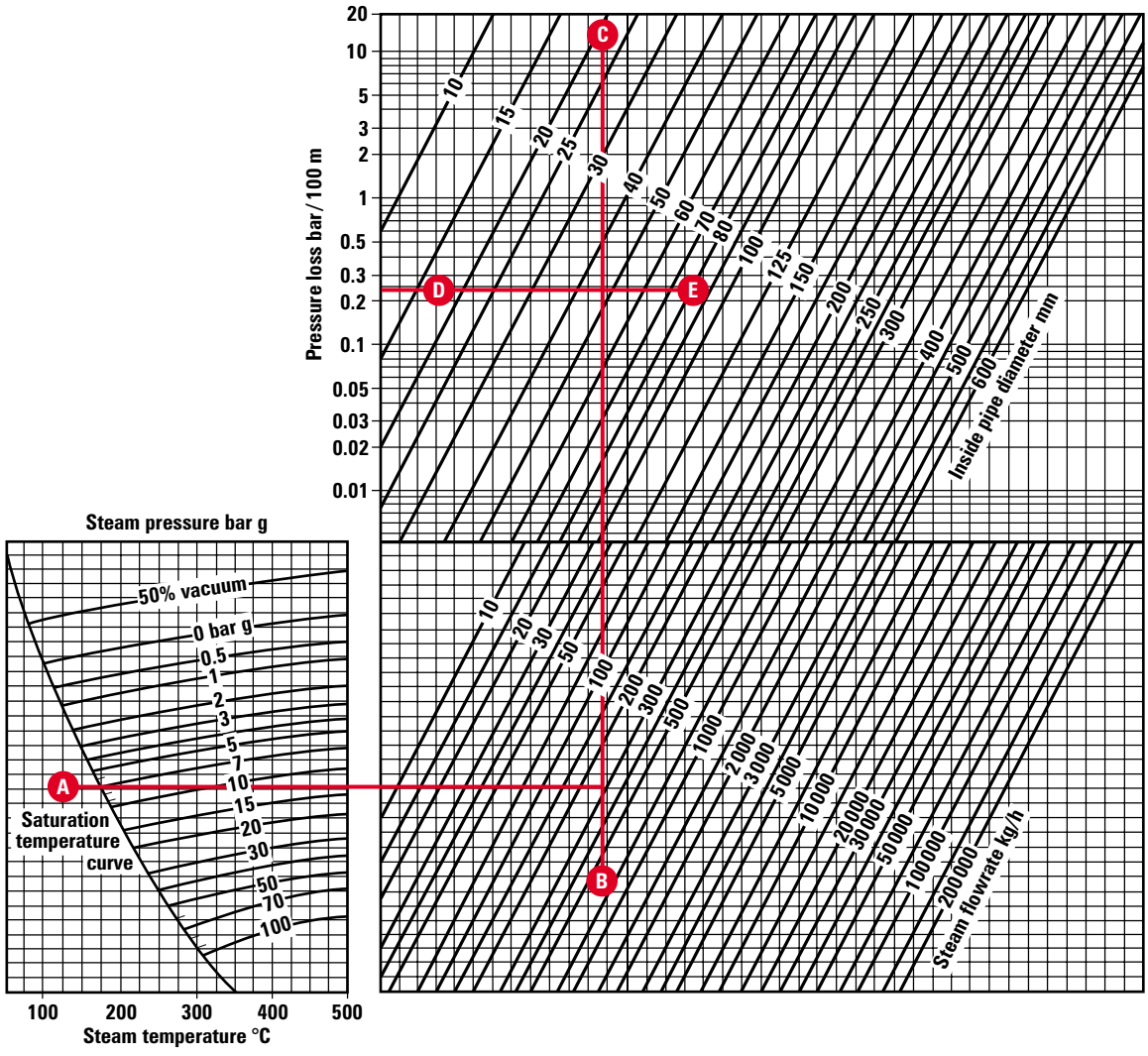


Fig. 10.2.7 Steam pipeline sizing chart - Pressure drop

Sizing pipes on velocity

From the knowledge gained at the beginning of this Module, and particularly the notes regarding the D'Arcy equation (Equation 10.2.1), it is acknowledged that velocity is an important factor in sizing pipes. It follows then, that if a reasonable velocity could be used for a particular fluid flowing through pipes, then velocity could be used as a practical sizing factor. As a general rule, a velocity of 25 to 40 m/s is used when saturated steam is the medium.

40 m/s should be considered a practical limit, as above this, noise and erosion will take place particularly if the steam is wet. Some National standards quote velocities up to 76 m/s for saturated steam. This can only be feasible if; the steam is dry, the pipe is very well insulated, relatively short, straight, horizontal and can supply the required pressure at the point of use.

Even these velocities can be high in terms of their effect on pressure drop. In longer supply lines, it is often necessary to restrict velocities to 15 m/s to avoid high pressure drops. It is recommended that pipelines over 50 m long are always checked for pressure drop, no matter what the velocity.

By using Table 10.2.4 as a guide, it is possible to select pipe sizes from known data; steam pressure, velocity and flowrate.

Table 10.2.4 Saturated steam pipeline capacities in kg/h for different velocities (Schedule 40 pipe)

Pressure bar g	Velocity m/s	Pipe size (nominal) 15	20	25	32	40	50	65	80	100	125	150
		Actual inside pipe diameter Schedule 40 15.80	20.93	26.64	35.04	40.90	52.50	62.70	77.92	102.26	128.20	154.05
		Pipeline capacity kg/h										
0.4	15	9	15	25	43	58	95	136	210	362	569	822
	25	14	25	41	71	97	159	227	350	603	948	1 369
	40	23	40	66	113	154	254	363	561	965	1 517	2 191
0.7	15	10	18	29	51	69	114	163	251	433	681	983
	25	17	30	49	85	115	190	271	419	722	1 135	1 638
	40	28	48	78	136	185	304	434	671	1 155	1 815	2 621
1	15	12	21	34	59	81	133	189	292	503	791	1 142
	25	20	35	57	99	134	221	315	487	839	1 319	1 904
	40	32	56	91	158	215	354	505	779	1342	2 110	3 046
2	15	18	31	50	86	118	194	277	427	735	1 156	1 669
	25	29	51	83	144	196	323	461	712	1 226	1 927	2 782
	40	47	82	133	230	314	517	737	1 139	1 961	3 083	4 451
3	15	23	40	65	113	154	254	362	559	962	1 512	2 183
	25	38	67	109	188	256	423	603	931	1 603	2 520	3 639
	40	61	107	174	301	410	676	964	1 490	2 565	4 032	5 822
4	15	28	50	80	139	190	313	446	689	1 186	1 864	2 691
	25	47	83	134	232	316	521	743	1 148	1 976	3 106	4 485
	40	75	132	215	371	506	833	1 189	1 836	3 162	4 970	7 176
5	15	34	59	96	165	225	371	529	817	1 408	2 213	3 195
	25	56	98	159	276	375	619	882	1 362	2 347	3 688	5 325
	40	90	157	255	441	601	990	1 411	2 180	3 755	5 901	8 521
6	15	39	68	111	191	261	430	613	947	1 631	2 563	3 700
	25	65	114	184	319	435	716	1 022	1 578	2 718	4 271	6 167
	40	104	182	295	511	696	1 146	1 635	2 525	4 348	6 834	9 867
7	15	44	77	125	217	296	487	695	1 073	1 848	2 904	4 194
	25	74	129	209	362	493	812	1 158	1 788	3 080	4 841	6 989
	40	118	206	334	579	788	1 299	1 853	2 861	4 928	7 745	11 183
8	15	49	86	140	242	330	544	775	1 198	2 063	3 242	4 681
	25	82	144	233	404	550	906	1 292	1 996	3 438	5 403	7 802
	40	131	230	373	646	880	1 450	2 068	3 194	5 501	8 645	12 484
10	15	60	105	170	294	401	660	942	1 455	2 506	3 938	5 686
	25	100	175	283	490	668	1 101	1 570	2 425	4 176	6 563	9 477
	40	160	280	453	785	1 069	1 761	2 512	3 880	6 682	10 502	15 164
14	15	80	141	228	394	537	886	1 263	1 951	3 360	5 281	7 625
	25	134	235	380	657	896	1 476	2 105	3 251	5 600	8 801	12 708
	40	214	375	608	1 052	1 433	2 362	3 368	5 202	8 960	14 082	20 333

Alternatively the pipe size can be calculated arithmetically. The following information is required, and the procedure used for the calculation is outlined below.

Information required to calculate the required pipe size:

u = Flow velocity (m/s)

v_g = Specific volume (m^3/kg)

$\dot{m}_s$ = Mass flowrate (kg/s)

$\dot{V}$ = Volumetric flowrate (m^3/s) = $\dot{m}_s \times v_g$

From this information, the cross sectional area (A) of the pipe can be calculated:

$$\text{Cross sectional area (A)} = \frac{\text{Volume flowrate } (\dot{V})}{\text{Flow velocity } (u)}$$

$$\text{i.e.} \qquad \frac{\pi \times D^2}{4} = \frac{\dot{V}}{u}$$

Rearranging the formula to give the diameter of the pipe (D) in metres:

$$D^2 = \frac{4 \times \dot{V}}{\pi \times u}$$

$$D = \sqrt{\frac{4 \times \dot{V}}{\pi \times u}}$$

Example 10.2.4

A process requires 5 000 kg/h of dry saturated steam at 7 bar g. For the flow velocity not to exceed 25 m/s, determine the pipe size.

Where:

$$\text{Flow velocity (u)} = 25 \text{ m/s}$$

$$\text{Specific volume at 7 bar g } (v_g) = 0.24 \text{ m}^3/\text{kg}$$

$$\text{Mass flowrate } (\dot{m}) = 5\,000 \text{ kg/h or } 1.389 \text{ kg/s}$$

$$\text{Volumetric flowrate} = \dot{m} \times v_g$$

$$\text{Volumetric flowrate} = 1.389 \text{ kg/s} \times 0.24 \text{ m}^3/\text{kg}$$

$$\textbf{Volumetric flowrate} = \mathbf{0.333\ m^3/s}$$

Therefore, using:

$$\text{Cross sectional area (A)} = \frac{\text{Volumetric flowrate } (\dot{V})}{\text{Flow velocity } (u)}$$

$$\frac{\pi \times \mathbf{D^2}}{4} = \frac{\dot{V}}{u}$$

$$\mathbf{D^2} = \frac{4 \times \dot{V}}{\pi \times u}$$

$$\mathbf{D} = \sqrt{\frac{4 \times \dot{V}}{\pi \times u}}$$

$$\text{Pipe diameter } (\mathbf{D}) = \sqrt{\frac{4 \times 0.333}{\pi \times 25}}$$

$$\textbf{Pipe diameter (D)} = \mathbf{0.130\ m \text{ or } 130\ mm}$$

Since the steam velocity must not exceed 25 m/s, the pipe size must be at least 130 mm; the nearest commercially available size, 150 mm, would be selected.

Again, a nomogram has been created to simplify this process, see Figure 10.2.8.

Example 10.2.5
Using the information from Example 10.2.4, use Figure 10.2.8 to determine the minimum acceptable pipe size

Inlet pressure = 7 bar g
Steam flowrate = 5 000 kg/h
Maximum velocity = 25 m/s

Method:

- Draw a horizontal line from the saturation temperature line at 7 bar g (Point **A**) on the pressure scale to the steam mass flowrate of 5 000 kg/h (Point **B**).
- From point **B**, draw a vertical line to the steam velocity of 25 m/s (Point **C**). From point **C**, draw a horizontal line across the pipe diameter scale (Point **D**).
- A pipe with a bore of 130 mm is required; the nearest commercially available size, 150 mm, would be selected.

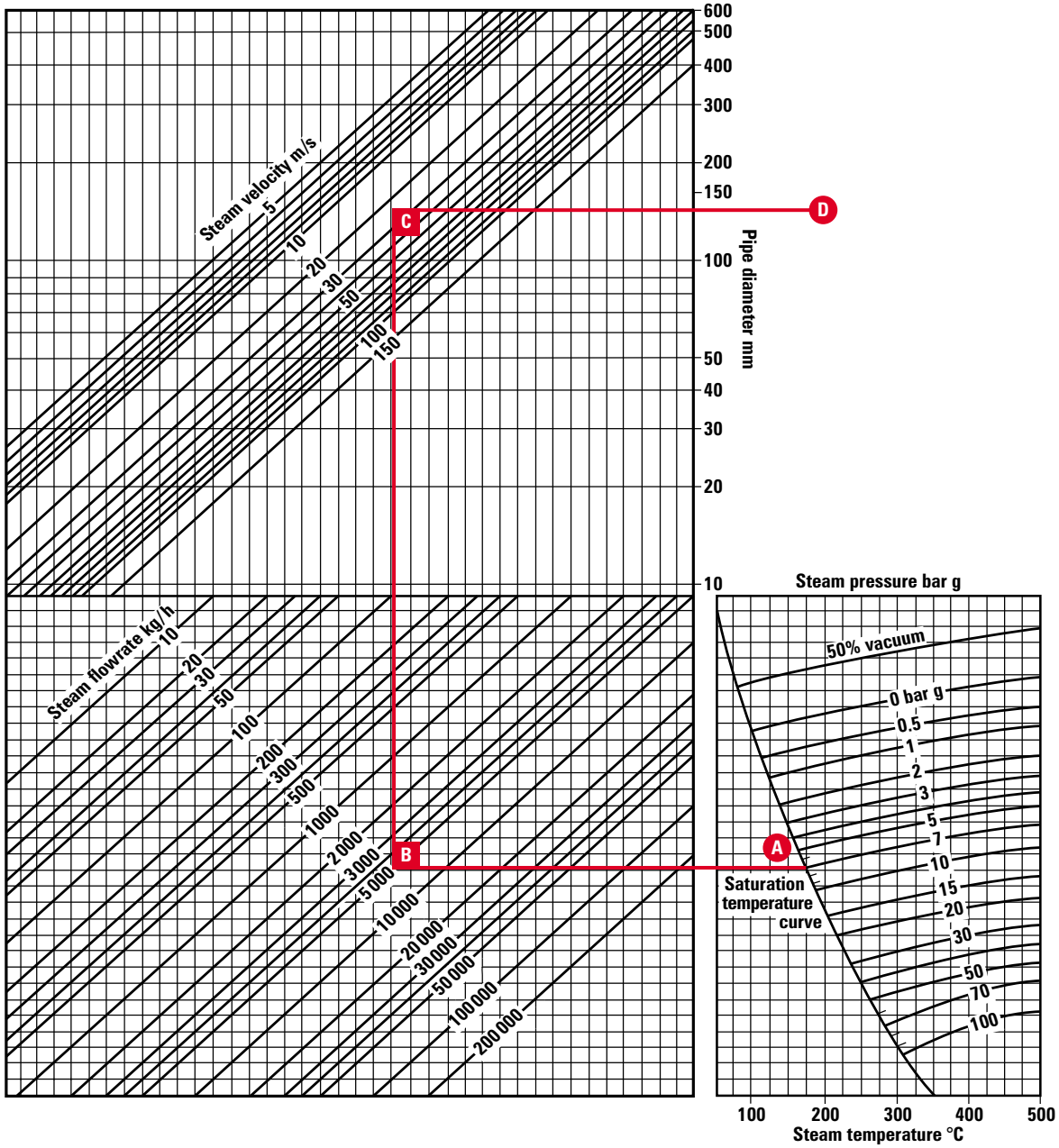


Fig. 10.2.8 Steam pipeline sizing chart - Velocity

Sizing pipes for superheated steam duty

Superheated steam can be considered as a dry gas and therefore carries no moisture. Consequently there is no chance of pipe erosion due to suspended water droplets, and steam velocities can be as high as 50 to 70 m/s if the pressure drop permits this. The nomograms in Figures 10.2.9 and 10.2.10 can also be used for superheated steam applications.

Example 10.2.6

Utilising the waste heat from a process, a boiler/superheater generates 30 t/h of superheated steam at 50 bar g and 450°C for export to a neighbouring power station. If the velocity is not to exceed 50 m/s, determine:

1. The pipe size based on velocity (use Figure 10.2.10).

2. The pressure drop if the pipe length, including allowances, is 200 m (use Figure 10.2.9).

Part 1

- Using Figure 10.2.8, draw a vertical line from 450°C on the temperature axis until it intersects the 50 bar line (Point **A**).
- From point **A**, project a horizontal line to the left until it intersects the steam 'mass flowrate' scale of 30 000 kg/h (30 t/h) (Point **B**).
- From point **B**, project a line vertically upwards until it intersects 50 m/s on the 'steam velocity' scale (Point **C**).
- From Point **C**, project a horizontal line to the right until it intersects the 'inside pipe diameter' scale.

The 'inside pipe diameter' scale recommends a pipe with an inside diameter of about 120 mm. From Table 10.2.1 and assuming that the pipe will be Schedule 80 pipe, the nearest size would be 150 mm, which has a bore of 146.4 mm.

Part 2

- Using Figure 10.2.7, draw a vertical line from 450°C on the temperature axis until it intersects the 50 bar line (Point **A**).
- From point **A**, project a horizontal line to the right until it intersects the 'steam mass flowrate' scale of 30 000 kg/h (30 t/h) (Point **B**).
- From point **B**, project a line vertically upwards until it intersects the 'inside pipe diameter' scale of (approximately) 146 mm (Point **C**).
- From Point **C**, project a horizontal line to the left until it intersects the 'pressure loss bar/100 m' scale (Point **D**).

The 'pressure loss bar/100 m' scale reads about 0.9 bar/100 m. The pipe length in the example is 200 m, so the pressure drop is:

$$\text{Pressure drop} = \frac{200 \text{ m}}{100 \text{ m}} \times 0.9 \text{ bar} = 1.8 \text{ bar}$$

This pressure drop must be acceptable at the process plant.

Using formulae to establish steam flowrate on pressure drop

Empirical formulae exist for those who prefer to use them. Equations 10.2.9 and 10.2.10 are shown below. These have been tried and tested over many years, and which appear to give results close to the pressure factor method. The advantage of using these formulae is that they can be programmed into a scientific calculator, or a spreadsheet, and consequently used without the need to look up tables and charts. Equation 10.2.10 requires the specific volume of steam to be known, which means it is necessary to look up this value from a steam table. Also, Equation 10.2.10 should be restricted to a maximum pipe length of 200 metres.

Pressure drop formula 1

$$\frac{(P_1)^{1.9375} - (P_2)^{1.9375}}{L} = \frac{\dot{m}^{1.853}}{0.011\,D^{4.987}}$$ **Equation 10.2.9**

Where:
P_1 = Upstream pressure (bar a)
P_2 = Downstream pressure (bar a)
L = Length of pipe (m)
$\dot{m}$ = Mass flowrate (kg/h)
D = Pipe diameter (mm)

Pressure drop formula 2 (Maximum pipe length: 200 metres)

$$\Delta P = \frac{L\, v_g\, \dot{m}^2}{0.08\, D^5}$$ **Equation 10.2.10**

Where:
ΔP = Pressure drop (bar)
L = Length of pipe (m)
v_g = Specific volume of steam (m^3/kg)
$\dot{m}$ = Mass flowrate (kg/h)
D = Pipe diameter (mm)

Summary

- The selection of piping material and the wall thickness required for a particular installation is stipulated in standards such as EN 45510 and ASME 31.1.
- Selecting the appropriate pipe size (nominal bore) for a particular application is based on accurately identifying pressure and flowrate. The pipe size may be selected on the basis of:
  - Velocity (usually pipes less than 50 m in length).
  - Pressure drop (as a general rule, the pressure drop should not normally exceed 0.1 bar/50 m.

Appendix

Table 10.2.5 Pressure drop factor (F) table

Pressure bar a	Pressure factor (F)
0.05	0.0301
0.10	0.0115
0.15	0.0253
0.20	0.0442
0.25	0.0681
0.30	0.0970
0.35	0.1308
0.40	0.1694
0.45	0.2128
0.50	0.2610
0.55	0.3140
0.60	0.3716
0.65	0.4340
0.70	0.5010
0.75	0.5727
0.80	0.6489
0.85	0.7298
0.90	0.8153
0.95	0.9053
1.013	1.0250

Pressure bar g	Pressure factor (F)
0	1.025
0.05	1.126
0.10	1.230
0.15	1.339
0.20	1.453
0.25	1.572
0.30	1.694
0.35	1.822
0.40	1.953
0.45	2.090
0.50	2.230
0.55	2.375
0.60	2.525
0.65	2.679
0.70	2.837
0.75	2.999
0.80	3.166
0.85	3.338
0.90	3.514
0.95	3.694
1.00	3.878
1.05	4.067
1.10	4.260
1.15	4.458
1.20	4.660
1.25	4.866
1.30	5.076
1.35	5.291
1.40	5.510
1.45	5.734
1.50	5.961
1.55	6.193
1.60	6.429
1.65	6.670
1.70	6.915
1.75	7.164
1.80	7.417
1.85	7.675
1.90	7.937
1.95	8.203
2.00	8.473

Pressure bar g	Pressure factor (F)
2.05	8.748
2.10	9.026
2.15	9.309
2.20	9.597
2.25	9.888
2.30	10.18
2.35	10.48
2.40	10.79
2.45	11.40
2.50	11.41
2.55	11.72
2.60	12.05
2.65	12.37
2.70	12.70
2.75	13.03
2.80	13.37
2.85	13.71
2.90	14.06
2.95	14.41
3.00	14.76
3.10	15.48
3.20	16.22
3.30	16.98
3.40	17.75
3.50	18.54
3.60	19.34
3.70	20.16
3.80	21.00
3.90	21.85
4.00	22.72
4.10	23.61
4.20	24.51
4.30	25.43
4.40	26.36
4.50	27.32
4.60	28.28
4.70	29.27
4.80	30.27
4.90	31.29
5.00	32.32
5.10	33.37
5.20	34.44
5.30	35.52
5.40	36.62
5.50	37.73
5.60	38.86
5.70	40.01
5.80	41.17
5.90	42.35
6.00	43.54
6.10	44.76
6.20	45.98
6.30	47.23
6.40	48.48
6.50	49.76
6.60	51.05
6.70	52.36
6.80	53.68
6.90	55.02
7.00	56.38
7.10	57.75
7.20	59.13
7.30	60.54
7.40	61.96
7.50	63.39

Pressure bar g	Pressure factor (F)
7.60	64.84
7.70	66.31
7.80	67.79
7.90	69.29
8.00	70.80
8.10	72.33
8.20	73.88
8.30	75.44
8.40	77.02
8.50	78.61
8.60	80.22
8.70	81.84
8.80	83.49
8.90	85.14
9.00	86.81
9.10	88.50
9.20	90.20
9.30	91.92
9.40	93.66
9.50	95.41
9.60	97.18
9.70	98.96
9.80	100.75
9.90	102.57
10.00	104.40
10.20	108.10
10.40	111.87
10.60	115.70
10.80	119.59
11.00	123.54
11.20	127.56
11.40	131.64
11.60	135.78
11.80	139.98
12.00	144.25
12.20	148.57
12.40	152.96
12.60	157.41
12.80	161.92
13.00	166.50
13.20	171.13
13.40	175.83
13.60	180.58
13.80	185.40
14.00	190.29
14.20	195.23
14.40	200.23
14.60	205.30
14.80	210.42
15.00	215.61
15.20	220.86
15.40	226.17
15.60	231.50
15.80	236.97
16.00	242.46
16.20	248.01
16.40	253.62
16.60	259.30
16.80	265.03
17.00	270.83
17.20	276.69
17.40	282.60
17.60	288.58
17.80	294.52
18.00	300.72

Table 10.2.6 Pipeline capacity and pressure factor table

Factor F	Pipe size (mm)												
	15	20	25	32	40	50	65	80	100	150	200	250	300
	Capacity (kg/h)												
0.00016						30.40	55.41	90.72	199.1	598.2	1 275	2 329	3 800
0.00020					16.18	34.32	62.77	103.0	225.6	662.0	1 437	2 623	4 276
0.00025				10.84	17.92	38.19	69.31	113.2	249.9	735.5	1 678	2 904	4 715
0.00030				11.95	19.31	41.83	75.85	124.1	271.2	804.5	1 733	3 172	5 149
0.00035			6.86	12.44	20.59	43.76	80.24	130.0	285.3	845.3	1 823	3 346	5 530
0.00045		3.62	7.94	14.56	23.39	50.75	92.68	150.9	333.2	979.7	2 118	3 884	6 267
0.00055		4.04	8.99	16.18	26.52	57.09	103.8	170.8	373.1	1 101	2 382	4 338	7 057
0.00065		4.46	9.56	17.76	29.14	62.38	113.8	186.7	409.8	1 207	2 595	4 781	7 741
0.00075		4.87	10.57	19.31	31.72	68.04	124.1	203.2	445.9	1 315	2 836	5 172	8 367
0.00085		5.52	11.98	21.88	35.95	77.11	140.7	230.2	505.4	1 490	3 215	5 861	9 482
0.00100	1.96	5.84	12.75	23.50	38.25	81.89	148.6	245.2	539.4	1 579	3 383	6 228	10 052
0.00125	2.10	6.26	13.57	24.96	40.72	87.57	159.8	261.8	577.9	1 699	3 634	6 655	10 639
0.00150	2.39	7.35	15.17	28.04	45.97	98.84	179.3	295.1	652.8	1 908	4 091	7 493	11 999
0.00175	2.48	7.51	16.30	29.61	49.34	103.4	188.8	311.1	686.5	2 017	4 291	7 852	13 087
0.0020	2.84	8.58	18.63	33.83	56.39	118.2	215.8	355.5	784.6	2 305	4 904	8 974	14 956
0.0025	3.16	9.48	20.75	37.25	61.30	132.0	240.5	391.3	881.7	2 456	5 422	10 090	16 503
0.0030	3.44	10.34	22.5	40.45	66.66	143.4	262.0	429.8	924.4	2 767	6 068	11 033	18 021
0.0040	4.17	12.50	26.97	48.55	80.91	173.1	313.8	514.9	1 128	3 330	7 208	13 240	21 625
0.0050	4.71	14.12	30.40	54.92	90.23	196.1	354.0	578.6	1 275	3 727	8 189	14 858	24 469
0.0060	5.25	15.69	35.80	60.31	99.05	215.8	392.3	647.3	1 412	4 148	9 072	16 476	26 970
0.0080	6.08	18.34	39.23	70.12	116.2	251.5	456.0	750.3	1 648	4 879	10 543	19 173	31 384
0.0100	6.86	20.64	44.13	79.44	130.4	283.9	514.9	845.9	1 863	5 492	11 867	21 576	35 307
0.0125	7.35	22.20	47.28	81.00	140.1	302.1	547.3	901.9	1 983	5 867	12 697	23 074	37 785
0.0150	8.27	25.00	53.33	95.62	157.2	342.0	620.6	1 020	2 230	6 620	14 251	25 974	42 616
0.0175	8.58	26.39	55.78	100.4	165.6	360.4	665.1	1 073	2 360	6 994	15 017	27 461	44 194
0.0200	9.80	30.16	63.75	114.7	189.3	411.9	760.1	1 226	2 697	7 993	17 163	31 384	50 508
0.0250	10.99	33.48	70.73	127.3	209.8	459.7	834.6	1 367	2 970	8 817	19 332	34 750	56 581
0.0300	12.00	36.78	77.23	137.9	229.9	501.1	919.4	1 480	3 264	9 792	20 917	37 697	62 522
0.0400	14.46	44.16	93.17	169.2	279.5	600.7	1 093	1 790	3 923	11 622	25 254	45 604	75 026
0.0500	16.43	49.53	104.4	191.2	313.8	676.7	1 231	2 020	4 413	13 044	28 441	51 489	85 324
0.060	18.14	52.96	115.7	210.8	343.2	750.3	1 373	2 231	4 855	14 368	31 384	57 373	
0.080	21.08	62.28	134.8	245.2	402.1	872.8	1 594	2 599	5 688	16 672	36 532		
0.100	24.03	70.12	152.0	277.0	456.0	980.7	1 804	2 942	6 424	18 879			
0.120	25.99	77.48	167.7	306.5	500.2	1 079	1 986	3 236	7 110	20 841			
0.150	28.50	84.13	183.9	334.2	551.7	1 195	2 161	3 494	7 769				
0.200	34.32	102.0	220.7	402.1	622.0	1 427	2 599	4 217	9 317				
0.250	37.72	112.7	245.2	447.9	735.5	1 565	2 876	4 668					
0.300	41.37	122.7	266.6	487.3	804.5	1 710	3 126	5 057					
0.350	43.34	128.7	283.2	514.9	841.0	1 802	3 261						
0.400	49.93	147.1	323.6	588.4	961.1	2 059	3 727						
0.450	50.31	150.0	326.6	600.2	979.9	2 083							
0.500	55.90	166.7	362.9	666.9	1 089	2 314							
0.600	62.28	185.3	402.1	735.5	1 201								
0.700	63.07	188.8	407.6	750.9									
0.800	72.08	215.8	465.8	858.1									
0.900	73.28	218.4	476.6										

Fig. 10.2.9 Steam pipeline sizing chart - Pressure drop

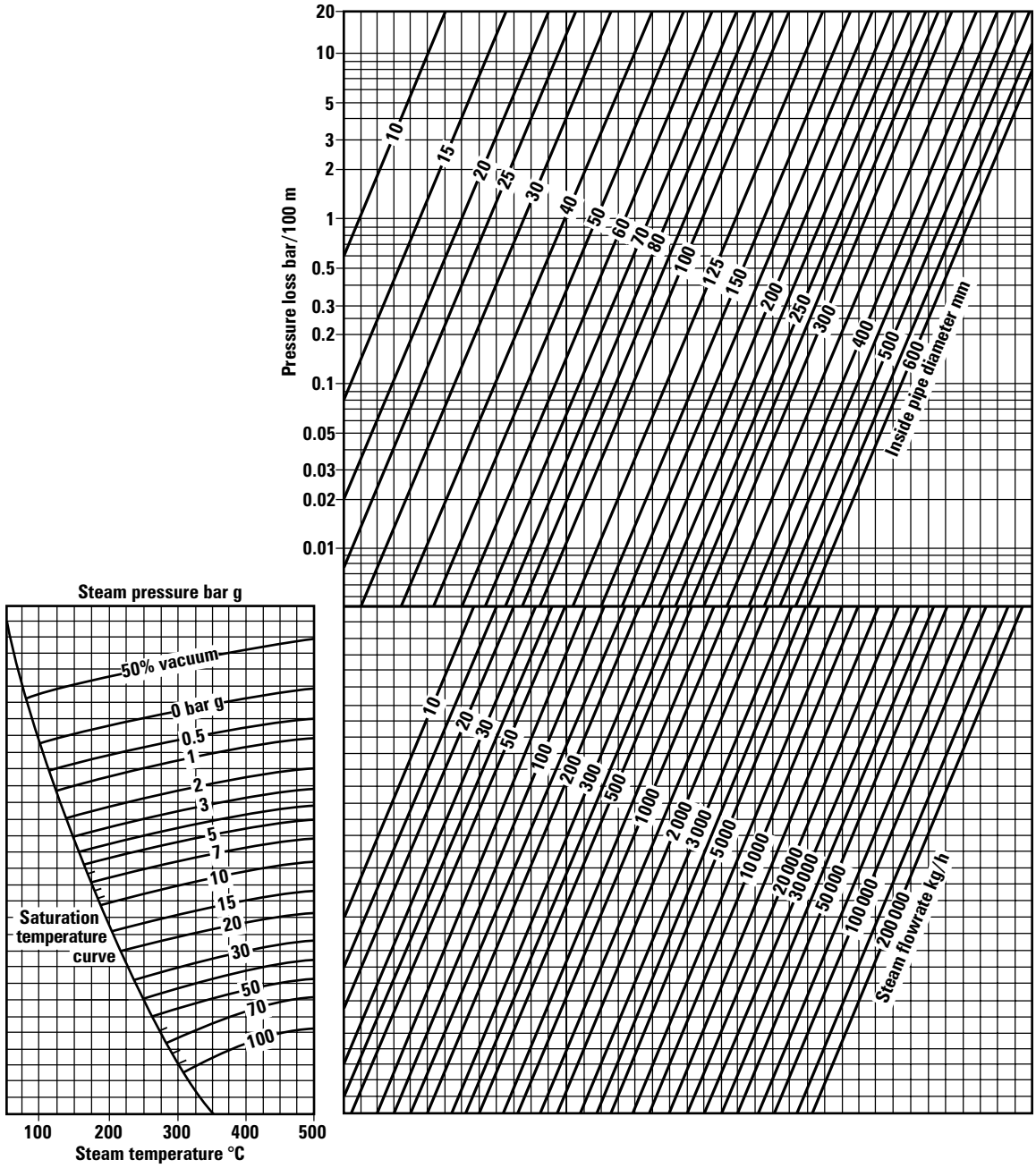
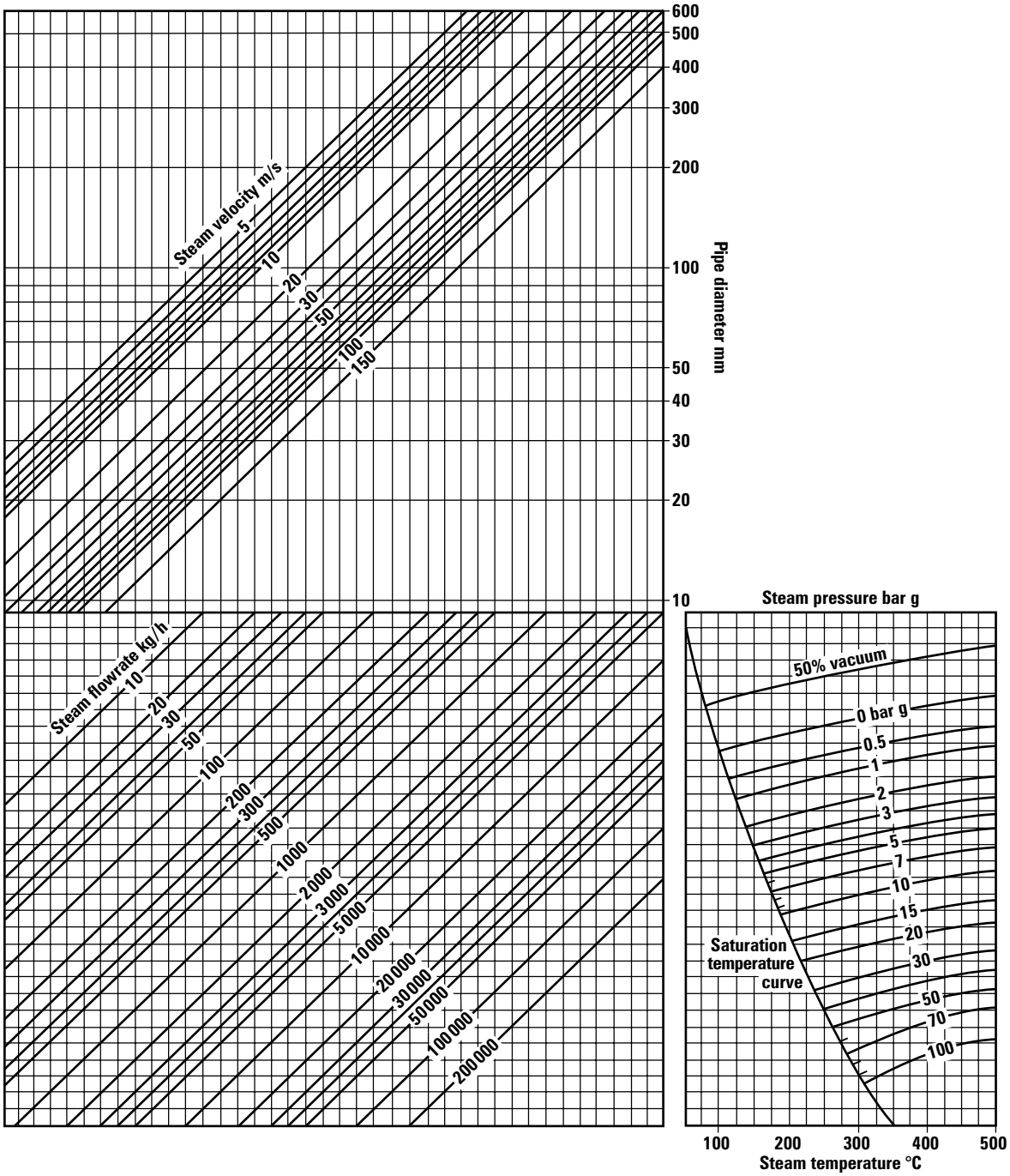


Fig. 10.2.10 Steam pipeline sizing chart - Velocity



Questions

1. A boiler is operated at 10 bar g and is required to supply 500 kg/h of saturated steam at 9.8 bar g to equipment 110 m away. The pipe run is torturous and contains many fittings adding 20% to the equivalent length. What size pipe should be selected?

a | 100 mm nominal bore ☐
b | 80 mm nominal bore ☐
c | 50 mm nominal bore ☐
d | 65 mm nominal bore ☐

2. A 100 mm steam pipe has been selected for a particular steam flowrate with 8.3 bar g at the inlet and 7.7 bar g at the end of the run. Calculations show that, for this flowrate and size of pipe, the pressure at the end of the run will actually be 7.9 bar g. Which of the following is true?

a | The steam velocity is higher than expected, and could cause noise ☐
b | The pipe has some additional spare capacity for future additional loads ☐
c | The resistance to flow is higher than expected ☐
d | A larger pipe is required ☐

3. A 40 m long 5 bar g saturated steam pipe is to be sized to carry 850 kg/h of steam. Should the pipe be sized on velocity or pressure drop?

a | Pressure drop to limit the steam velocity ☐
b | On a velocity over 40 m/s ☐
c | On a velocity of about 25 m/s ☐
d | Either, provided the steam velocity does not exceed, approximately 5 m/s ☐

4. A 40 m pipe incorporating a number of bends and fittings is to be sized by the velocity method to handle 1 200 kg/h of saturated steam at 4 bar g. What size pipe is required?

a | 100 mm ☐
b | 80 mm ☐
c | 125 mm ☐
d | The pipe should be sized on pressure drop, and not by velocity ☐

5. A straight run of pipe 30 m long and carrying saturated steam at 10 bar g is to be sized by the velocity method to pass 20 000 kg/h. What size pipe is required?

a | 175 mm ☐
b | 150 mm ☐
c | 200 mm ☐
d | 250 mm ☐

6. From the following, what is the effect of sizing a 100 m long, 8 bar g steam pipe by the velocity method?

a | Sizing by velocity takes no account of pressure drop along the pipe ☐
b | If the velocity is more than 40 m/s, the pressure drop along the pipe may be very small and in practice a small pipe may be used ☐
c | If a low velocity is selected, the chosen pipe will probably be undersized resulting in steam starvation at the plant ☐
d | Over a length of 100 m, the noise of steam flow can be unacceptable ☐

Answers

1: d, 2: b, 3: c, 4: a, 5: d, 6: a

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Module 10.3

Steam Mains and Drainage

Steam Mains and Drainage

Throughout the length of a hot steam main, an amount of heat will be transferred to the environment, and this will depend on the parameters identified in Block 2 - 'Steam Engineering and Heat Transfer', and brought together in Equation 2.5.1.

$$\dot{Q} = k A \frac{\Delta T}{\varkappa} \qquad \textbf{Equation 2.5.1}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W)
k = Thermal conductivity of the material (W/m K or W/m °C)
A = Heat transfer area (m²)
ΔT = Temperature difference across the material (K or °C)
$\varkappa$ = Material thickness (m)

With steam systems, this loss of energy represents inefficiency, and thus pipes are insulated to limit these losses. Whatever the quality or thickness of insulation, there will always be a level of heat loss, and this will cause steam to condense along the length of the main.

The effect of insulation is discussed in Module 10.5. This Module will concentrate on disposal of the inevitable condensate, which, unless removed, will accumulate and lead to problems such as corrosion, erosion, and waterhammer.

In addition, the steam will become wet as it picks up water droplets, which reduces its heat transfer potential. If water is allowed to accumulate, the overall effective cross sectional area of the pipe is reduced, and steam velocity can increase above the recommended limits.

Piping layout

The subject of drainage from steam lines is covered in the European Standard EN 45510, Section 10.1.14.

EN 45510 states that, whenever possible, the main should be installed with a fall of not less than 1:100 (1 m fall for every 100 m run), in the direction of the steam flow. This slope will ensure that gravity, as well as the flow of steam, will assist in moving the condensate towards drain points where the condensate may be safely and effectively removed (See Figure 10.3.1).

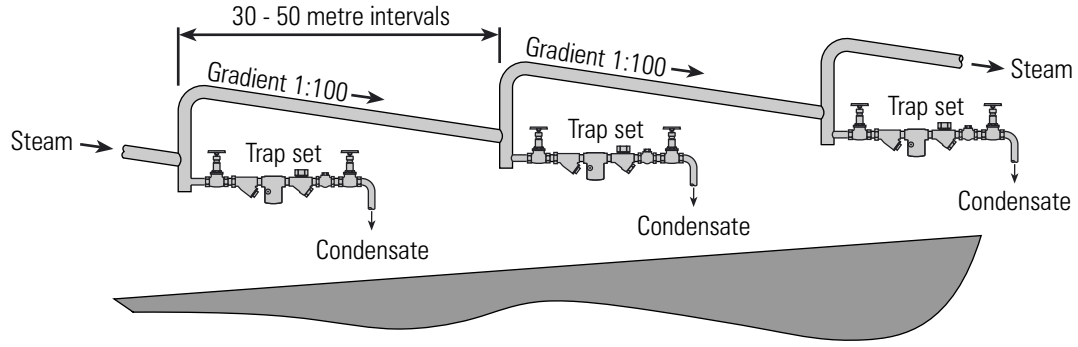


Fig. 10.3.1 Typical steam main installation

Drain points

The drain point must ensure that the condensate can reach the steam trap. Careful consideration must therefore be given to the design and location of drain points.

Consideration must also be given to condensate remaining in a steam main at shutdown, when steam flow ceases. Gravity will ensure that the water (condensate) will run along sloping pipework and collect at low points in the system. Steam traps should therefore be fitted to these low points.

The amount of condensate formed in a large steam main under start-up conditions is sufficient to require the provision of drain points at intervals of 30 m to 50 m, as well as natural low points such as at the bottom of rising pipework.

In normal operation, steam may flow along the main at speeds of up to 145 km/h, dragging condensate along with it. Figure 10.3.2 shows a 15 mm drain pipe connected directly to the bottom of a main.

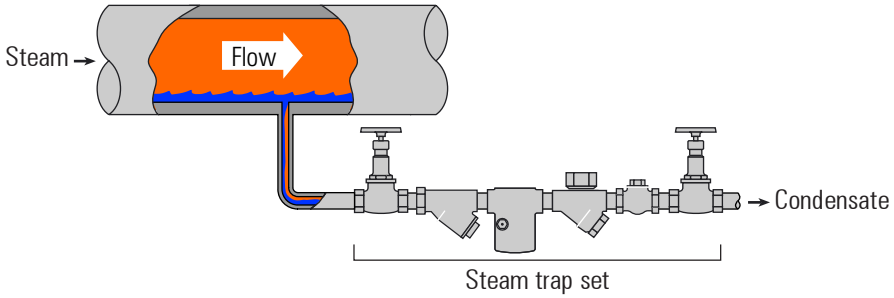


Fig. 10.3.2 Trap pocket too small

Although the 15 mm pipe has sufficient capacity, it is unlikely to capture much of the condensate moving along the main at high speed. This arrangement will be ineffective.

A more reliable solution for the removal of condensate is shown in Figure 10.3.3. The trap line should be at least 25 to 30 mm from the bottom of the pocket for steam mains up to 100 mm, and at least 50 mm for larger mains. This allows a space below for any dirt and scale to settle.

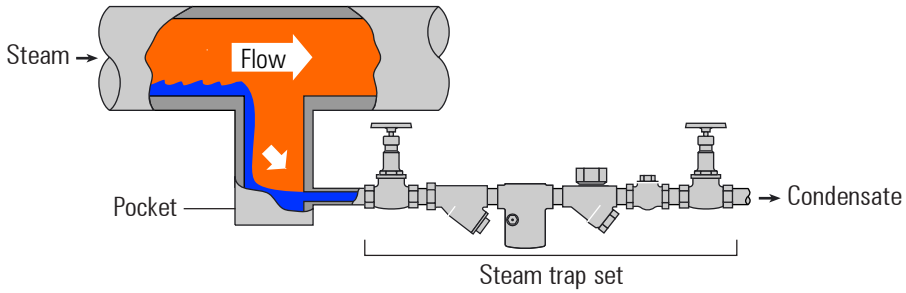


Fig. 10.3.3 Trap pocket properly sized

The bottom of the pocket may be fitted with a removable flange or blowdown valve for cleaning purposes.

Recommended drain pocket dimensions are shown in Table 10.3.1 and in Figure 10.3.4.

Table 10.3.1 Recomended drain pocket dimensions

Mains diameter - D	Pocket diameter - d_1	Pocket depth - d_2
Up to 100 mm nb	$d_1 = D$	Minimum $d_2 = 100$ mm
125 - 200 mm nb	$d_1 = 100$ mm	Minimum $d_2 = 150$ mm
250 mm and above	$d_1 \geq D/2$	Minimum $d_2 = D$

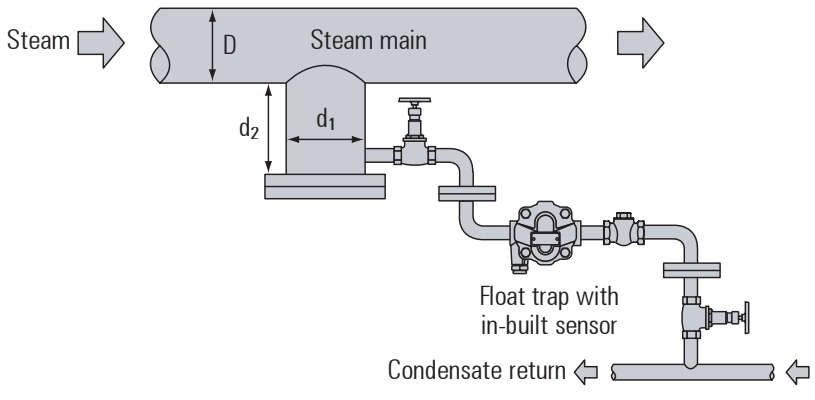


Fig. 10.3.4

Waterhammer and its effects

Waterhammer is the noise caused by slugs of condensate colliding at high velocity into pipework fittings, plant, and equipment. This has a number of implications:

- Because the condensate velocity is higher than normal, the dissipation of kinetic energy is higher than would normally be expected.
- Water is dense and incompressible, so the 'cushioning' effect experienced when gases encounter obstructions is absent.
- The energy in the water is dissipated against the obstructions in the piping system such as valves and fittings.

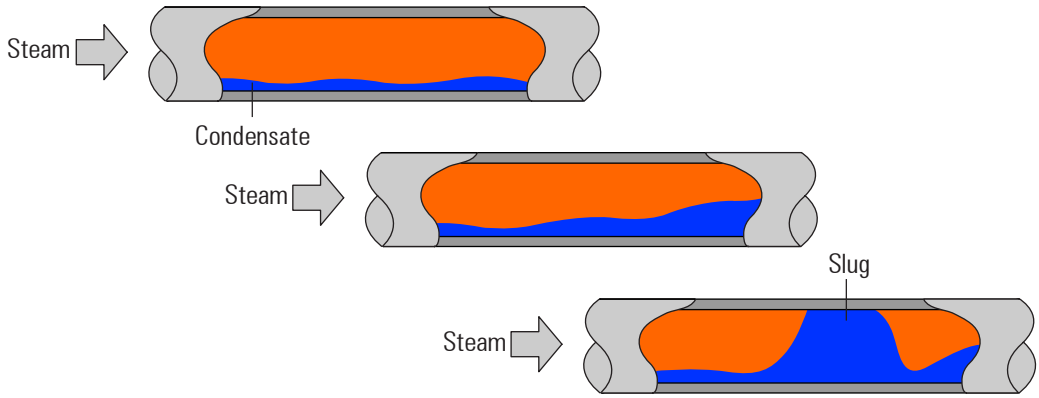


Fig. 10.3.5 Formation of a 'solid' slug of water

Indications of waterhammer include a banging noise, and perhaps movement of the pipe.

In severe cases, waterhammer may fracture pipeline equipment with almost explosive effect, with consequent loss of live steam at the fracture, leading to an extremely hazardous situation.

Good engineering design, installation and maintenance will avoid waterhammer; this is far better practice than attempting to contain it by choice of materials and pressure ratings of equipment.

Commonly, sources of waterhammer occur at the low points in the pipework (See Figure 10.3.6). Such areas are due to:

- Sagging in the line, perhaps due to failure of supports.
- Incorrect use of concentric reducers (see Figure 10.3.7) - Always use eccentric reducers with the flat at the bottom.
- Incorrect strainer installation - They should be fitted with the basket on the side.
- Inadequate drainage of steam lines.
- Incorrect operation - Opening valves too quickly at start-up when pipes are cold.

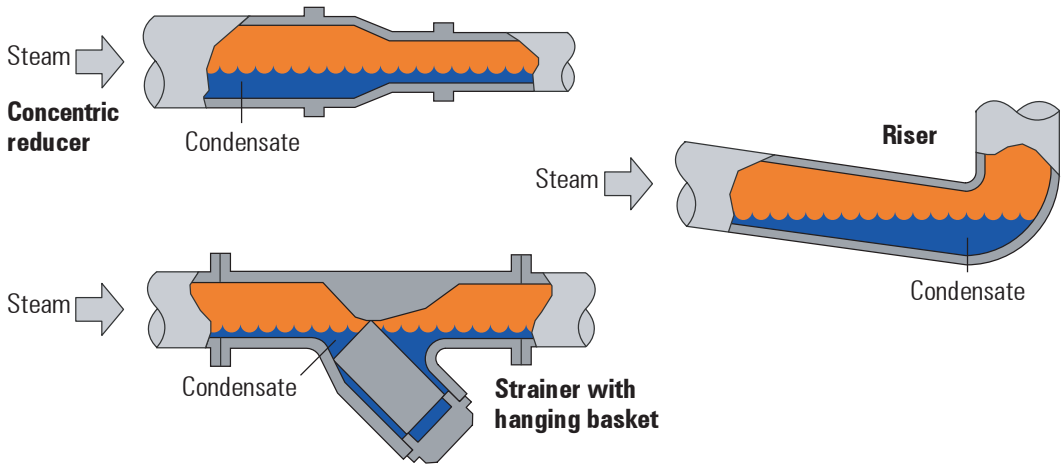


Fig. 10.3.6 Potential sources of waterhammer

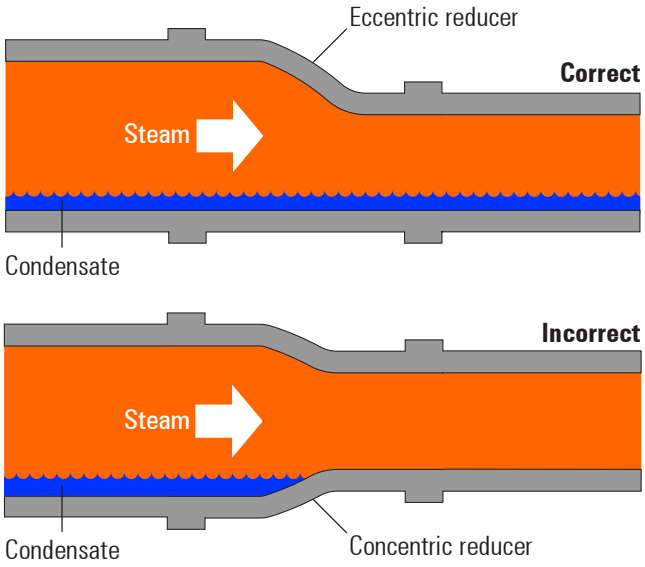


Fig. 10.3.7 Eccentric and concentric pipe reducers

To summarise, the possibility of waterhammer is minimised by:

- Installing steam lines with a gradual fall in the direction of flow, and with drain points installed at regular intervals and at low points.
- Installing check valves after all steam traps which would otherwise allow condensate to run back into the steam line or plant during shutdown.
- Opening isolation valves slowly to allow any condensate which may be lying in the system to flow gently through the drain traps, before it is picked up by high velocity steam. This is especially important at start-up.

Branch lines

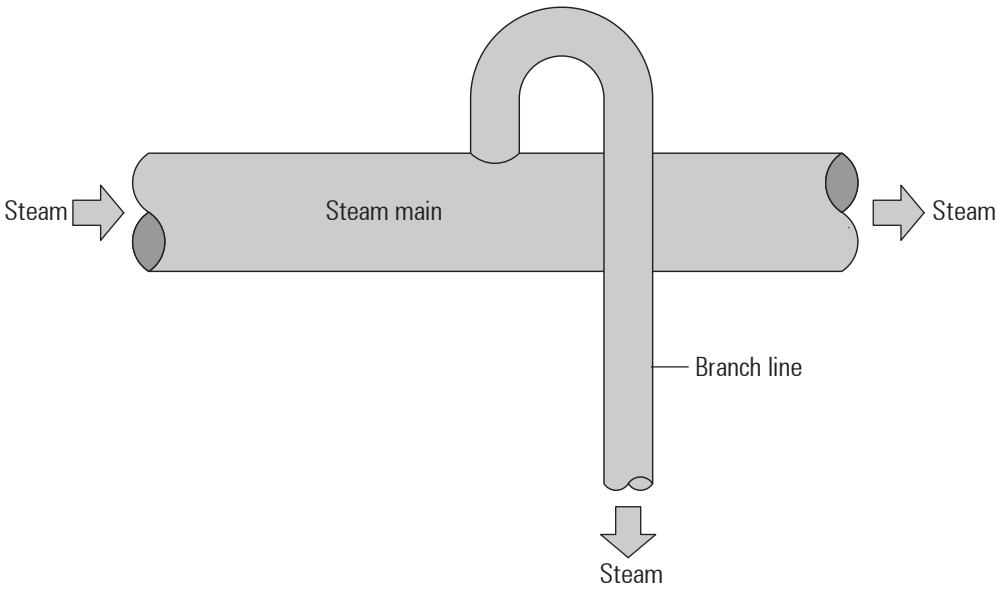


Fig. 10.3.8 Branch line

Branch lines are normally much shorter than steam mains. As a general rule, therefore, provided the branch line is not more than 10 metres in length, and the pressure in the main is adequate, it is possible to size the pipe on a velocity of 25 to 40 m/s, and not to worry about the pressure drop.

Table 10.2.4 'Saturated steam pipeline capacities for different velocities' in Module 10.2 will prove useful in this exercise.

Branch line connections

Branch line connections taken from the top of the main carry the driest steam (Figure 10.3.8). If connections are taken from the side, or even worse from the bottom (as in Figure 10.3.9 (a)), they can accept the condensate and debris from the steam main. The result is very wet and dirty steam reaching the equipment, which will affect performance in both the short and long term.

The valve in Figure 10.3.9 (b) should be positioned as near to the off-take as possible to minimise condensate lying in the branch line, if the plant is likely to be shutdown for any extended periods.

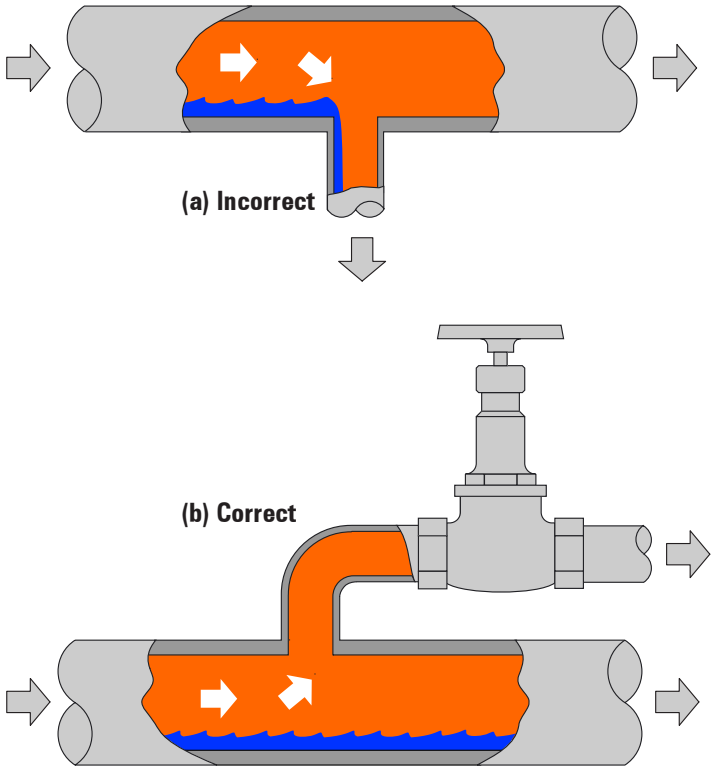


Fig. 10.3.9 Steam off-take

Drop leg

Low points will also occur in branch lines. The most common is a drop leg close to an isolating valve or a control valve (Figure 10.3.10). Condensate can accumulate on the upstream side of the closed valve, and then be propelled forward with the steam when the valve opens again - consequently a drain point with a steam trap set is good practice just prior to the strainer and control valve.

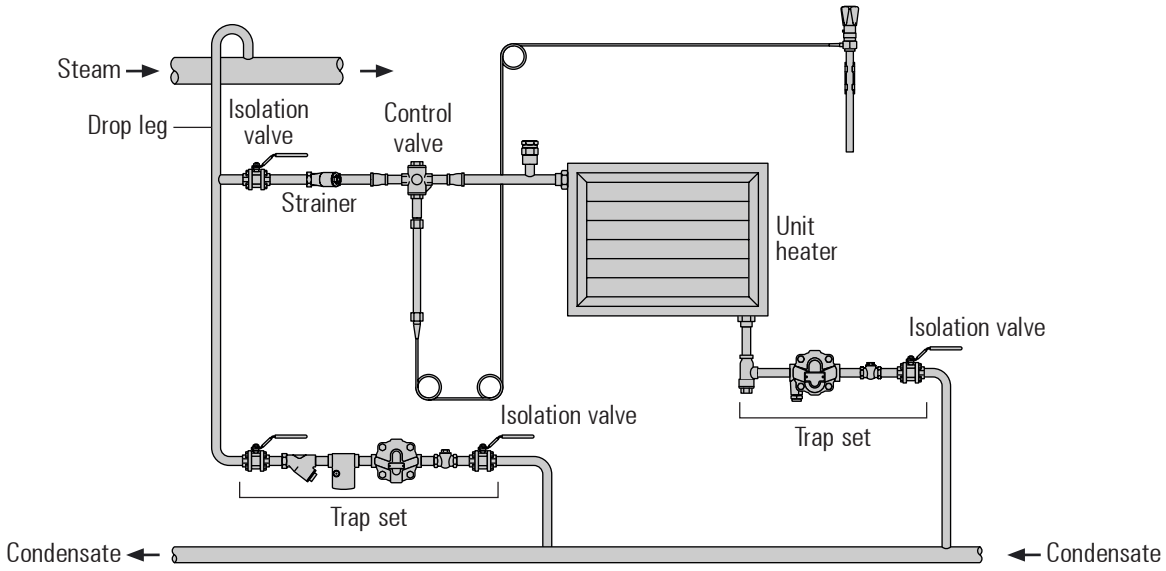


Fig. 10.3.10 Diagram of a drop leg supplying a unit heater

Rising ground and drainage

There are many occasions when a steam main must run across rising ground, or applications where the contours of the site make it impractical to lay the pipe with the 1:100 fall proposed earlier. In these situations, the condensate must be encouraged to run downhill and against the steam flow. Good practice is to size the pipe on a low steam velocity of not more than 15 m/s, to run the line at a slope of no less than 1:40, and install the drain points at not more than 15 metre intervals (see Figure 10.3.11).

The objective is to prevent the condensate film on the bottom of the pipe increasing in thickness to the point where droplets can be picked up by the steam flow.

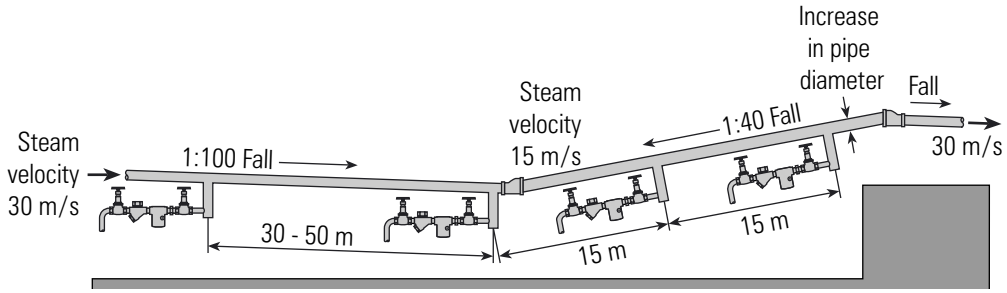


Fig. 10.3.11 Reverse gradient on steam main

Steam separators

Modern packaged steam boilers have a large evaporating capacity for their size and have limited capacity to cope with rapidly changing loads. In addition, as discussed in Block 3 'The Boiler House', other circumstances, such as . . .

- Incorrect chemical feedwater treatment and/or TDS control
- Transient peak loads in other parts of the plant

. . . can cause priming and carryover of boiler water into the steam mains.

Separators, as shown by the cut section in Figure 10.3.12, may be installed to remove this water.

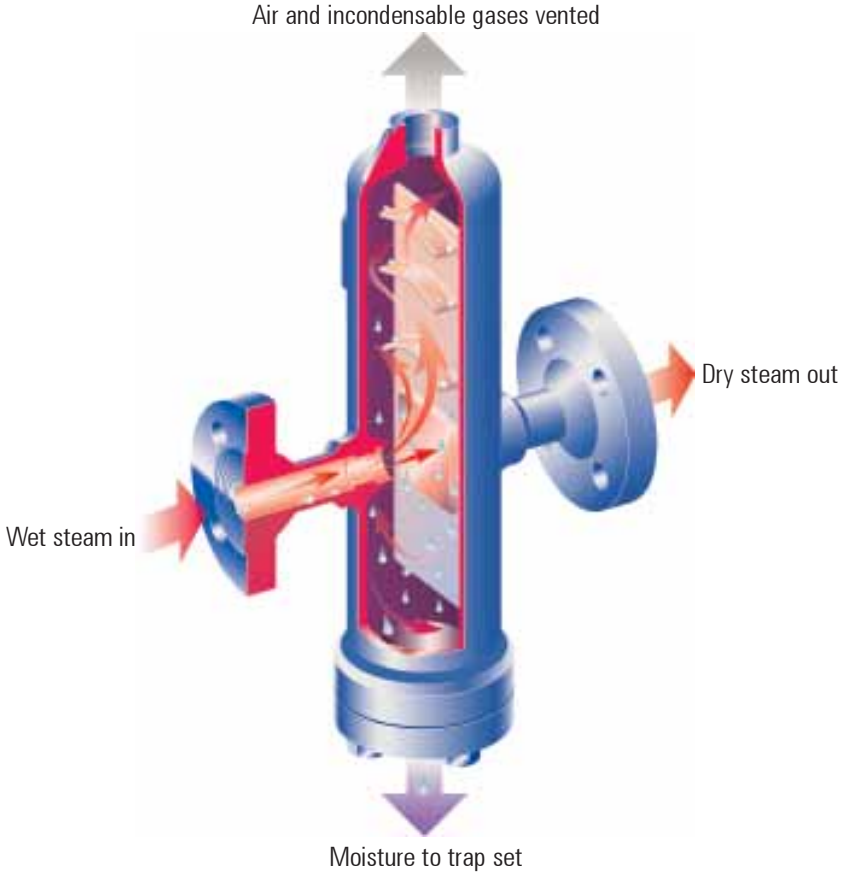


Fig. 10.3.12 Cut section through a separator

As a general rule, providing the velocities in the pipework are within reasonable limits, separators will be line sized. (Separators are discussed in detail in Module 12.5)

A separator will remove both droplets of water from pipe walls and suspended mist entrained in the steam itself. The presence and effect of waterhammer can be eradicated by fitting a separator in a steam main, and can often be less expensive than increasing the pipe size and fabricating drain pockets.

A separator is recommended before control valves and flowmeters. It is also wise to fit a separator where a steam main enters a building from outside. This will ensure that any condensate produced in the external distribution system is removed and the building always receives dry steam. This is equally important where steam usage in the building is monitored and charged for.

Strainers

When new pipework is installed, it is not uncommon for fragments of casting sand, packing, jointing, swarf, welding rods and even nuts and bolts to be accidentally deposited inside the pipe. In the case of older pipework, there will be rust, and in hard water districts, a carbonate deposit. Occasionally, pieces will break loose and pass along the pipework with the steam to rest inside a piece of steam using equipment. This may, for example, prevent a valve from opening/closing correctly. Steam using equipment may also suffer permanent damage through wiredrawing - the cutting action of high velocity steam and water passing through a partly open valve. Once wiredrawing has occurred, the valve will never give a tight shut-off, even if the dirt is removed.

It is therefore wise to fit a line-size strainer in front of every steam trap, flowmeter, reducing valve and regulating valve. The illustration shown in Figure 10.3.13 shows a cut section through a typical strainer.

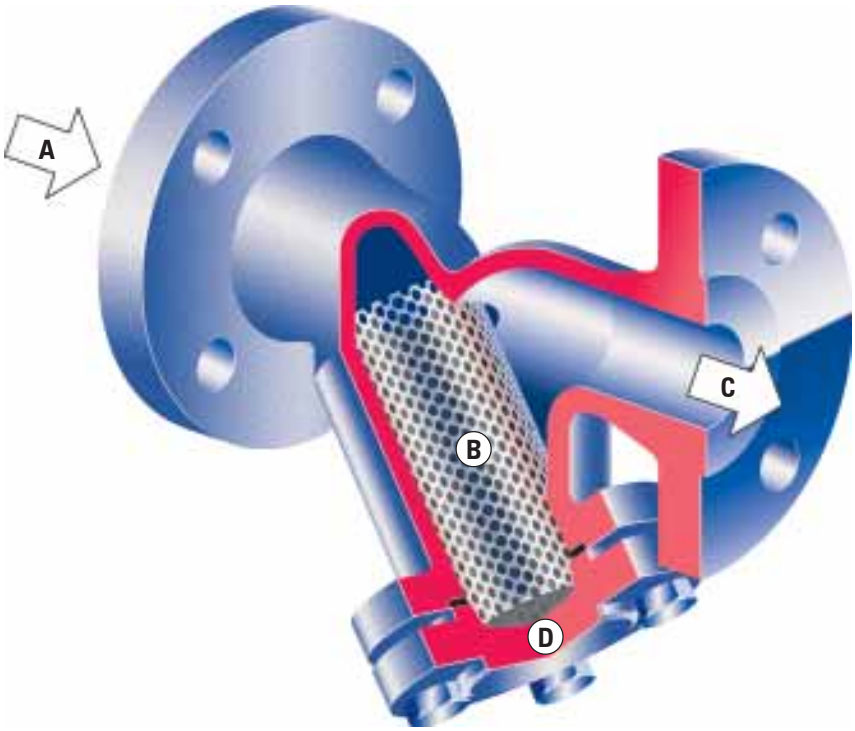


Fig. 10.3.13 Cut section through a Y-type strainer.

Steam flows from the inlet '**A**' through the perforated screen '**B**' to the outlet '**C**'. While steam and water will pass readily through the screen, dirt cannot. The cap '**D**', can be removed, allowing the screen to be withdrawn and cleaned at regular intervals. A blowdown valve can also be fitted to cap '**D**' to facilitate regular cleaning.

Strainers can however, be a source of wet steam as previously mentioned. To avoid this situation, strainers should always be installed in steam lines with their baskets to the side.

Strainers and screen details are discussed in Module 12.4.

How to drain steam mains

Steam traps are the most effective and efficient method of draining condensate from a steam distribution system.

The steam traps selected must suit the system in terms of:

- Pressure rating
- Capacity
- Suitability

Pressure rating
Pressure rating is easily dealt with; the maximum possible working pressure at the steam trap will either be known or should be established.

Capacity
Capacity, that is, the quantity of condensate to be discharged, which needs to be divided into two categories; warm-up load and running load.

Warm-up load - In the first instance, the pipework needs to be brought up to operating temperature. This can be determined by calculation, knowing the mass and specific heat of the pipework and fittings. Alternatively, Table 10.3.2 may be used.

- The table shows the amount of condensate generated when bringing 50 m of steam main up to working temperature; 50 m being the maximum recommended distance between trapping points.
- The values shown are in kilograms. To determine the average condensing rate, the time taken for the process must be considered. For example, if the warm-up process required 50 kg of steam, and was to take 20 minutes, then the average condensing rate would be:

$$\text{Average condensing rate} = \frac{60 \text{ minutes}}{20 \text{ minutes}} \times 50 \text{ kg}$$

$$\textbf{Average condensing rate} = \textbf{150 kg/h}$$

- When using these capacities to size a steam trap, it is worth remembering that the initial pressure in the main will be little more than atmospheric when the warm-up process begins. However, the condensate loads will still generally be well within the capacity of a DN15 'low capacity' steam trap. Only in rare applications at very high pressures (above 70 bar g), combined with large pipe sizes, will greater trap capacity be needed.

Running load - Once the steam main is up to operating temperature, the rate of condensation is mainly a function of the pipe size and the quality and thickness of the insulation.

For accurate means of calculating running losses from steam mains, refer to Module 2.12 'Steam consumption of pipes and air heaters'. Alternatively, for quick approximations of running load, Table 10.3.3 can be used which shows typical amounts of steam condensed each hour per 50 m of insulated steam main at various pressures.

Table 10.3.2 Amount of steam condensed to warm-up 50 m of schedule 40 pipe (kg)

Note: Figures are based on an ambient temperature of 20°C, and an insulation efficiency of 80%

Steam pressure bar g	Steam main size (mm)														-18°C correction factor
	50	65	80	100	125	150	200	250	300	350	400	450	500	600	
1	5	9	11	16	22	28	44	60	79	94	123	155	182	254	1.39
2	6	10	13	19	25	33	49	69	92	108	142	179	210	296	1.35
3	7	11	14	20	25	36	54	79	101	120	156	197	232	324	1.32
4	8	12	16	22	30	39	59	83	110	131	170	215	254	353	1.29
5	8	13	17	24	33	42	63	70	119	142	185	233	275	382	1.28
6	9	13	18	25	34	43	66	93	124	147	198	242	285	396	1.27
7	9	14	18	26	35	45	68	97	128	151	197	250	294	410	1.26
8	9	14	19	27	37	47	71	101	134	158	207	261	307	428	1.25
9	10	15	20	28	38	50	74	105	139	164	216	272	320	436	1.24
10	10	16	20	29	40	51	77	109	144	171	224	282	332	463	1.24
12	10	17	22	31	42	54	84	115	152	180	236	298	350	488	1.23
14	11	17	23	32	44	57	85	120	160	189	247	311	366	510	1.22
16	12	19	24	35	47	61	91	128	172	203	265	334	393	548	1.21
18	17	23	31	45	62	84	127	187	355	305	393	492	596	708	1.21
20	17	26	35	51	71	97	148	220	302	362	465	582	712	806	1.20
25	19	29	39	56	78	108	164	243	333	400	533	642	786	978	1.19
30	21	32	41	62	86	117	179	265	364	437	571	702	859	1 150	1.18
40	22	34	46	67	93	127	194	287	395	473	608	762	834	1 322	1.16
50	24	37	50	73	101	139	212	214	432	518	665	834	1 020	1 450	1.15
60	27	41	54	79	135	181	305	445	626	752	960	1 218	1 480	2 140	1.15
70	29	44	59	86	156	208	346	510	717	861	1 100	1 396	1 694	2 455	1.15
80	32	49	65	95	172	232	386	568	800	960	1 220	1 550	1 890	2 730	1.14
90	34	51	69	100	181	245	409	598	842	1 011	1 288	1 635	1 990	2 880	1.14
100	35	54	72	106	190	257	427	628	884	1 062	1 355	1 720	2 690	3 030	1.14
120	42	64	86	126	227	305	508	748	1 052	1 265	1 610	2 050	2 490	3 600	1.13

Table 10.3.3 Condensing rate of steam in 50 m of schedule 40 pipe - at working temperature (kg/h)

Note: Figures are based on an ambient temperature of 20°C, and an insulation efficiency of 80%

Steam pressure bar g	Steam main size (mm)														-18°C correction factor
	50	65	80	100	125	150	200	250	300	350	400	450	500	600	
1	5	5	7	9	10	13	16	19	23	25	28	31	35	41	1.54
2	5	6	8	10	12	14	18	22	26	28	32	35	39	46	1.50
3	6	7	9	11	14	16	20	25	30	32	37	40	45	54	1.48
4	7	9	10	12	16	18	23	28	33	37	42	46	51	61	1.45
5	7	9	11	13	17	20	24	30	36	40	46	49	55	66	1.43
6	8	10	11	14	18	21	26	33	39	43	49	53	59	71	1.42
7	8	10	12	15	19	23	28	35	42	46	52	56	63	76	1.41
8	9	11	14	16	20	24	30	37	44	49	57	61	68	82	1.40
9	9	11	14	17	21	25	32	39	47	52	60	64	72	88	1.39
10	10	12	15	17	21	25	33	41	49	54	62	67	75	90	1.38
12	11	13	16	18	23	26	36	45	53	59	67	73	81	97	1.38
14	12	14	17	20	26	30	39	49	58	64	73	79	93	106	1.37
16	12	15	18	23	29	34	42	52	62	68	78	85	95	114	1.36
18	14	16	19	24	30	36	44	55	66	72	82	90	100	120	1.36
20	15	17	21	25	31	37	46	58	69	76	86	94	105	125	1.35
25	15	19	23	28	35	42	52	66	78	86	97	106	119	141	1.34
30	17	21	25	31	39	47	58	73	87	96	108	118	132	157	1.33
40	20	25	30	38	46	56	70	87	104	114	130	142	158	189	1.31
50	24	29	34	44	54	65	82	102	121	133	151	165	184	220	1.29
60	27	32	39	50	62	74	95	119	140	155	177	199	222	265	1.28
70	29	35	43	56	70	82	106	133	157	173	198	222	248	296	1.27
80	34	42	51	66	81	97	126	156	187	205	234	263	293	350	1.26
90	38	46	56	72	89	106	134	171	204	224	265	287	320	284	1.26
100	41	50	61	78	96	114	149	186	220	242	277	311	347	416	1.25
120	52	63	77	99	122	145	189	236	280	308	352	395	440	527	1.22

Suitability

A mains drain trap should consider the following constraints:

- **Discharge temperature -** The steam trap should discharge at, or very close to saturation temperature, unless cooling legs are used between the drain point and the trap. This means that the choice is a mechanical type trap (such as a float, inverted bucket type, or thermodynamic traps).
- **Frost damage -** Where the steam main is located outside a building and there is a possibility of sub-zero ambient temperature, the thermodynamic steam trap is ideal, as it not damaged by frost. Even if the installation causes water to be left in the trap at shutdown and freezing occurs, the thermodynamic trap may be thawed out without suffering damage when brought back into use.
- **Waterhammer -** In the past, on poorly laid out installations where waterhammer was a common occurrence, float traps were not always ideal due to their susceptibility to float damage. Contemporary design and manufacturing techniques now produce extremely robust units for mains drainage purposes. Float traps are certainly the first choice for proprietary separators as high capacities are readily achieved, and they are able to respond quickly to rapid load increases.

Steam traps used to drain condensate from steam mains, are shown in Figure 10.3.14. The thermostatic trap is included because it is ideal where there is no choice but to discharge condensate into a flooded return pipe.

The subject of steam trapping is dealt with in detail in the Block 11, 'Steam Trapping'.



Ball float type **Thermodynamic type** **Thermostatic type** **Inverted bucket type**

Fig. 10.3.14 Steam traps suitable for steam mains drainage

Steam leaks

Steam leaking from pipework is often ignored. Leaks can be costly in both the economic and environmental sense and therefore need prompt attention to ensure the steam system is working at its optimum efficiency with a minimum impact on the environment.

Figure 10.3.15 illustrates the steam loss for various sizes of hole at various pressures. This loss can be readily translated into a fuel saving based on the annual hours of operation.

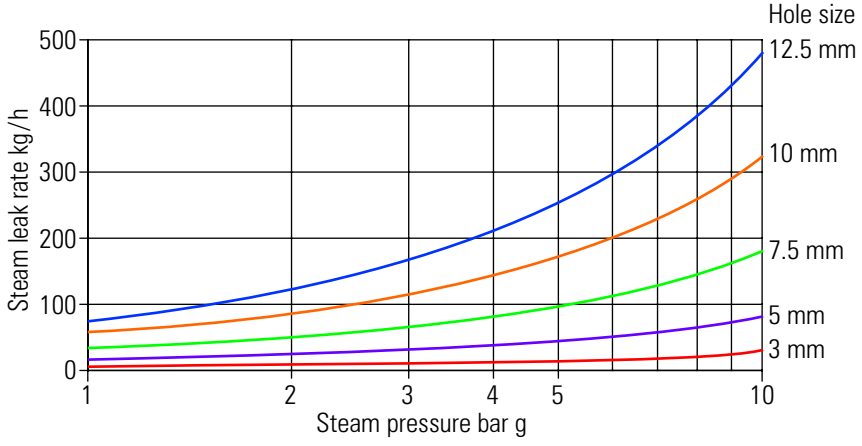


Fig. 10.3.15 Steam leakage rate through holes

Summary

Proper pipe alignment and drainage means observing a few simple rules:

- Steam lines should be arranged to fall in the direction of flow, at not less than 100 mm per 10 metres of pipe (1:100). Steam lines rising in the direction of flow should slope at not less than 250 mm per 10 metres of pipe (1:40).
- Steam lines should be drained at regular intervals of 30 - 50 m and at any low points in the system.
- Where drainage has to be provided in straight lengths of pipe, then a large bore pocket should be used to collect condensate.
- If strainers are to be fitted, then they should be fitted on their sides.
- Branch connections should always be taken from the top of the main from where the driest steam is taken.
- Separators should be considered before any piece of steam using equipment ensuring that dry steam is used.
- Traps selected should be robust enough to avoid waterhammer damage and frost damage.

Questions

1. Which of the following is true of wet steam?

a| It can cause waterhammer if allowed to build up ☐

b| It can corrode pipes if allowed to continue ☐

c| It causes erosion of bends ☐

d| All of the above ☐

2. What is the effect of installing a steam main horizontally level?

a| None, provided the pipe is drained at 30 - 50 m intervals ☐

b| Complete drainage will be less effective, and waterhammer could result ☐

c| Larger diameter drain points should be fitted ☐

d| Condensate will not reach the drain points ☐

3. Steam pipeline strainers should be fitted with their baskets on the side to:

a| Prevent condensate filling the body and being carried over to the equipment being protected ☐

b| Provide a greater screening area ☐

c| Extend the periods between cleaning the strainer ☐

d| Provide more effective removal of the debris ☐

4. Using the velocity method, what size pipe is required to carry 500 kg/h of steam at 6 bar g over a 40 m run with a rising slope? (The specific volume of steam at 6 bar g is 0.272 m^3/kg

a| 40 mm ☐

b| 80 mm ☐

c| 50 mm ☐

d| 65 mm ☐

5. A correctly sized pilot operated reducing valve has been installed in a pressure reducing station supplying an autoclave, as shown in Figure 10.3.16. What is wrong with the installation?

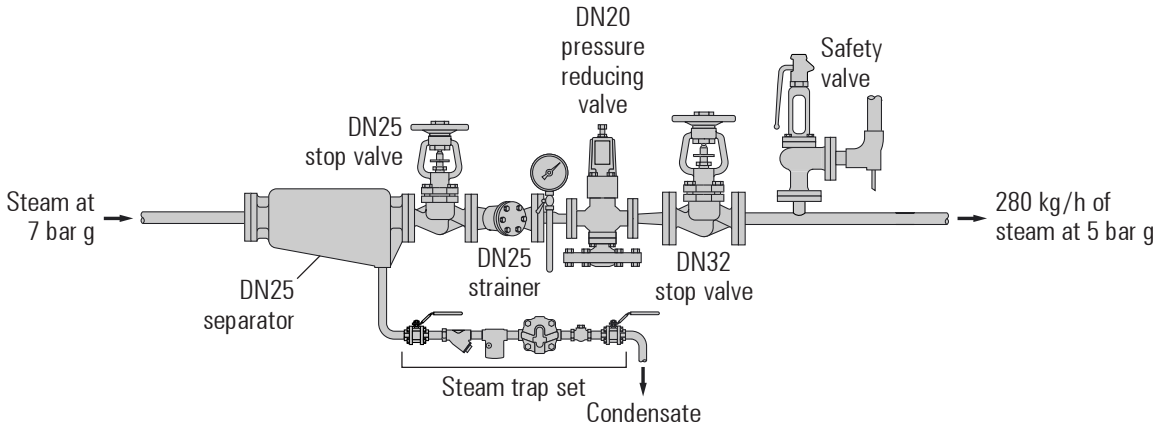


Fig. 10.3.16

a| The pipe after the PRV is at a lower pressure, and steam has a higher volume, so the pipe should be larger than 32 mm ☐

b| The upstream strainer and isolation valve should be the same size as the reducing valve ☐

c| The separator should be one size larger than the pipework to avoid excessive pressure drop ☐

d| There is no downstream pressure gauge before the DN32 stop valve ☐

6. As a minimum, horizontal runs of 150 mm steam main should be drained at intervals of:

a| Every 15 metres via 100 mm bore drain pockets, 100 mm deep ☐

b| Every 30 - 50 metres via 150 mm bore drain pockets, 100 mm deep ☐

c| Every 15 metres via 100 mm bore drain pockets, 150 mm deep ☐

d| Every 30 - 50 metres via 100 mm bore drain pockets, 150 mm deep ☐

Answers

1: d, 2: b, 3: a, 4: d, 5: d, 6: d

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Module 10.4

Pipe Expansion and Support

Pipe Expansion and Support

Allowance for expansion

All pipes will be installed at ambient temperature. Pipes carrying hot fluids such as water or steam operate at higher temperatures.

It follows that they expand, especially in length, with an increase from ambient to working temperatures. This will create stress upon certain areas within the distribution system, such as pipe joints, which, in the extreme, could fracture. The amount of the expansion is readily calculated using Equation 10.4.1, or read from an appropriate chart such as Figure 10.4.1.

$$\text{Expansion (mm)} = L\,\Delta T\,\alpha \qquad \textbf{Equation 10.4.1}$$

Where:
L = Length of pipe between anchors (m)
ΔT = Temperature difference between ambient temperature and operating temperatures (°C)
α = Expansion coefficient (mm/m °C) x 10^{-3}

Table 10.4.1 Expansion coefficients (α) (mm/m °C x 10^{-3})

Material	Temperature range (°C)							
	< 0	0 - 100	0 - 200	0 - 300	0 - 400	0 - 500	0 - 600	0 - 700
Carbon steel 0.1% - 0.2% C	12.8	13.9	14.9	15.8	16.6	17.3	17.9	-
Alloy steel 1% Cr 0.5% Mo	13.7	14.5	15.2	15.8	16.4	17.0	17.6	-
Stainless steel 18% Cr 8% Ni	9.4	20.0	20.9	21.2	21.8	22.3	22.7	23.0

Example 10.4.1
A 30 m length of carbon steel pipe is to be used to transport steam at 4 bar g (152°C). If the pipe is installed at 10°C, determine the expansion using Equation 10.4.1.

$$\text{Expansion (mm)} = L\,\Delta T\,\alpha$$

Where:

$$L = 30 \text{ m}$$

$$\Delta T = 152°C - 10°C$$

$$\Delta T = 142°C$$

$$\alpha \text{ in the range } 0 - 200 = 14.9 \times 10^{-3} \text{ mm/m °C for carbon steel pipe}$$

$$\text{Expansion} = 30 \text{ m} \times 142°C \times 14.9 \times 10^{-3} \text{ mm/m °C}$$

$$\textbf{Expansion = 63.5 mm}$$

Alternatively, the chart in Figure 10.4.1 can be used for finding the approximate expansion of a variety of steel pipe lengths - see Example 10.4.2 for explanation of use.

Example 10.4.2
Using Figure 10.4.1. Find the approximate expansion from 15°C, of 100 metres of carbon steel pipework used to distribute steam at 265°C.

Temperature difference is 265 - 15°C = 250°C.

Where the diagonal temperature difference line of 250°C cuts the horizontal pipe length line at 100 m, drop a vertical line down. For this example an approximate expansion of 330 mm is indicated.

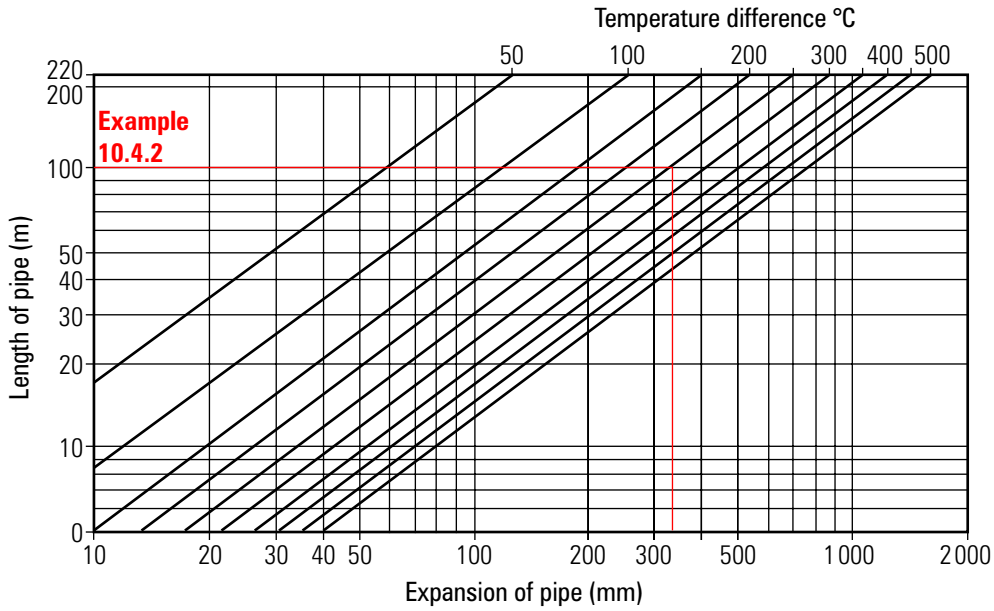


Fig. 10.4.1 A chart showing the expansion in various steel pipe lengths at various temperature differences

Table 10.4.2 Temperature of saturated steam

bar g	1	2	3	4	5	7.5	10	15	20	25	30
°C	120	134	144	152	159	173	184	201	215	226	236

Pipework flexibility

The pipework system must be sufficiently flexible to accommodate the movements of the components as they expand. In many cases the flexibility of the pipework system, due to the length of the pipe and number of bends and supports, means that no undue stresses are imposed. In other installations, however, it will be necessary to incorporate some means of achieving this required flexibility.

An example on a typical steam system is the discharge of condensate from a steam mains drain trap into the condensate return line that runs along the steam line (Figure 10.4.2). Here, the difference between the expansions of the two pipework systems must be taken into account. The steam main will be operating at a higher temperature than that of the condensate main, and the two connection points will move relative to each other during system warm-up.

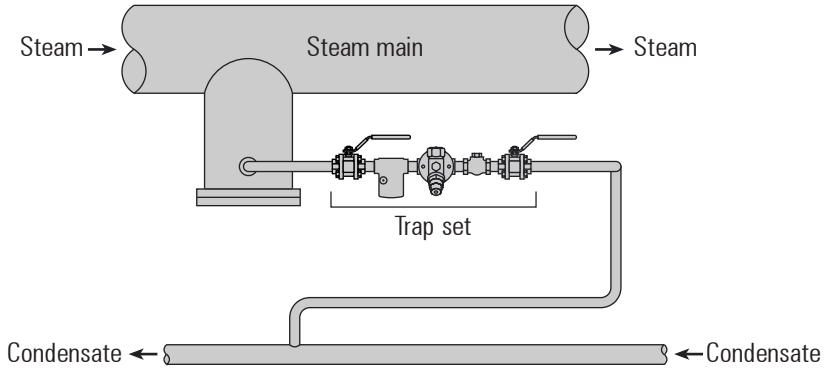


Fig. 10.4.2 Flexibility in connection to condensate return line

The amount of movement to be taken up by the piping and any device incorporated in it can be reduced by 'cold draw'. The total amount of expansion is first calculated for each section between fixed anchor points. The pipes are left short by half of this amount, and stretched cold by pulling up bolts at a flanged joint, so that at ambient temperature, the system is stressed in one direction. When warmed through half of the total temperature rise, the piping is unstressed. At working temperature and having fully expanded, the piping is stressed in the opposite direction. The effect is that instead of being stressed from 0 F to +1 F units of force, the piping is stressed from -½ F to + ½ F units of force.

In practical terms, the pipework is assembled cold with a spacer piece, of length equal to half the expansion, between two flanges. When the pipework is fully installed and anchored at both ends, the spacer is removed and the joint pulled up tight (see Figure 10.4.3).

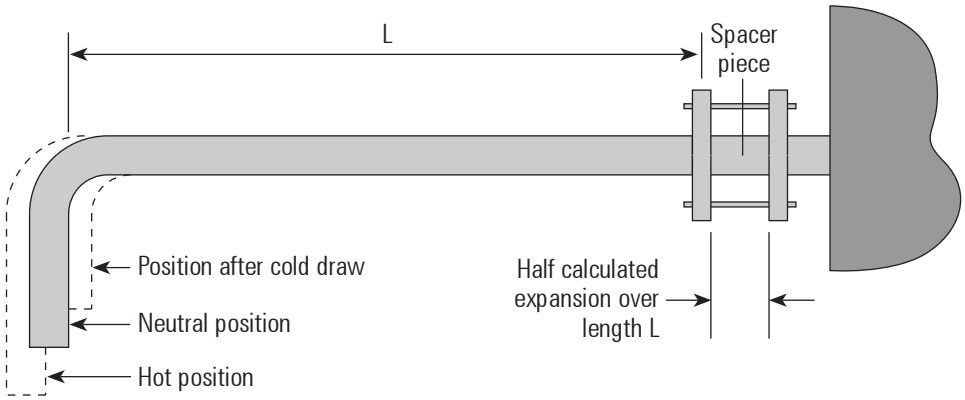


Fig. 10.4.3 Use of spacer for expansion when pipework is installed

The remaining part of the expansion, if not accepted by the natural flexibility of the pipework will call for the use of an expansion fitting.

In practice, pipework expansion and support can be classified into three areas as shown in Figure 10.4.4.

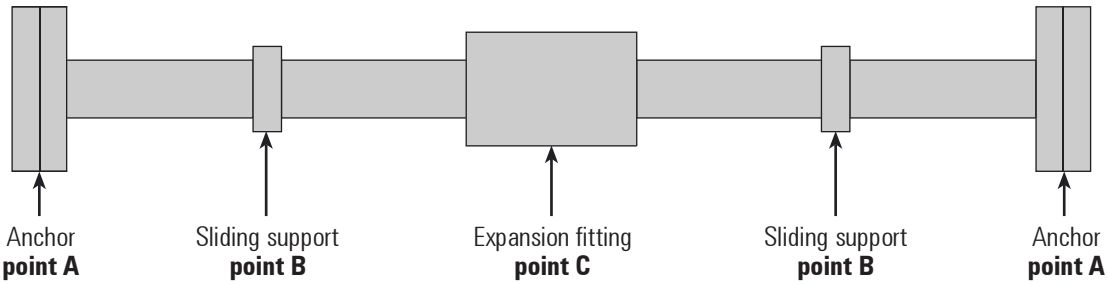


Fig. 10.4.4 Diagram of pipeline with fixed point, variable anchor point and expansion fitting

The fixed or 'anchor' points '**A**' provide a datum position from which expansion takes place.

The sliding support points '**B**' allow free movement for expansion of the pipework, while keeping the pipeline in alignment.

The expansion device at point '**C**' is to accommodate the expansion and contraction of the pipe.

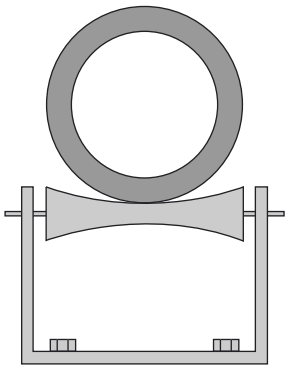


Fig. 10.4.5 Chair and roller

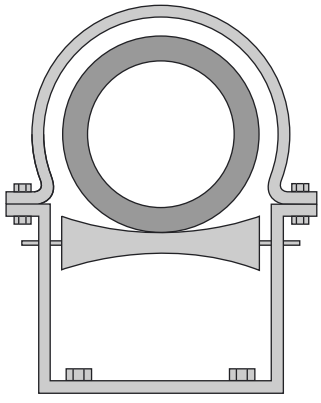


Fig. 10.4.6 Chair roller and saddle

Roller supports (Figure 10.4.5 and 10.4.6) are ideal methods for supporting pipes, at the same time allowing them to move in two directions. For steel pipework, the rollers should be manufactured from ferrous material. For copper pipework, they should be manufactured from non-ferrous material. It is good practice for pipework supported on rollers to be fitted with a pipe saddle bolted to a support bracket at not more than distances of 6 metres to keep the pipework in alignment during any expansion and contraction.

Where two pipes are to be supported one below the other, it is poor practice to carry the bottom pipe from the top pipe using a pipe clip. This will cause extra stress to be added to the top pipe whose thickness has been sized to take only the stress of its working pressure.

All pipe supports should be specifically designed to suit the outside diameter of the pipe concerned.

Expansion fittings

The expansion fitting ('**C**' Figure 10.4.4) is one method of accommodating expansion. These fittings are placed within a line, and are designed to accommodate the expansion, without the total length of the line changing. They are commonly called expansion bellows, due to the bellows construction of the expansion sleeve.

Other expansion fittings can be made from the pipework itself. This can be a cheaper way to solve the problem, but more space is needed to accommodate the pipe.

Full loop

This is simply one complete turn of the pipe and, on steam pipework, should preferably be fitted in a horizontal rather than a vertical position to prevent condensate accumulating on the upstream side.

The downstream side passes below the upstream side and great care must be taken that it is not fitted the wrong way round, as condensate can accumulate in the bottom. When full loops are to be fitted in a confined space, care must be taken to specify that wrong-handed loops are not supplied.

The full loop does not produce a force in opposition to the expanding pipework as in some other types, but with steam pressure inside the loop, there is a slight tendency to unwind, which puts an additional stress on the flanges.

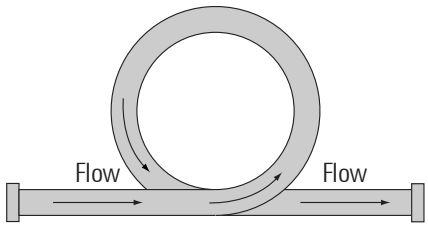


Fig. 10.4.7 Full loop

This design is used rarely today due to the space taken up by the pipework, and proprietary expansion bellows are now more readily available. However large steam users such as power stations or establishments with large outside distribution systems still tend to use full loop type expansion devices, as space is usually available and the cost is relatively low.

Horseshoe or lyre loop

When space is available this type is sometimes used. It is best fitted horizontally so that the loop and the main are on the same plane. Pressure does not tend to blow the ends of the loop apart, but there is a very slight straightening out effect. This is due to the design but causes no misalignment of the flanges.

If any of these arrangements are fitted with the loop vertically above the pipe then a drain point must be provided on the upstream side as depicted in Figure 10.4.8.

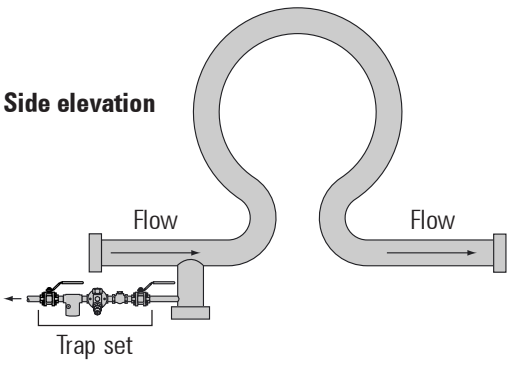


Fig. 10.4.8 Horseshoe or lyre loop

Expansion loops

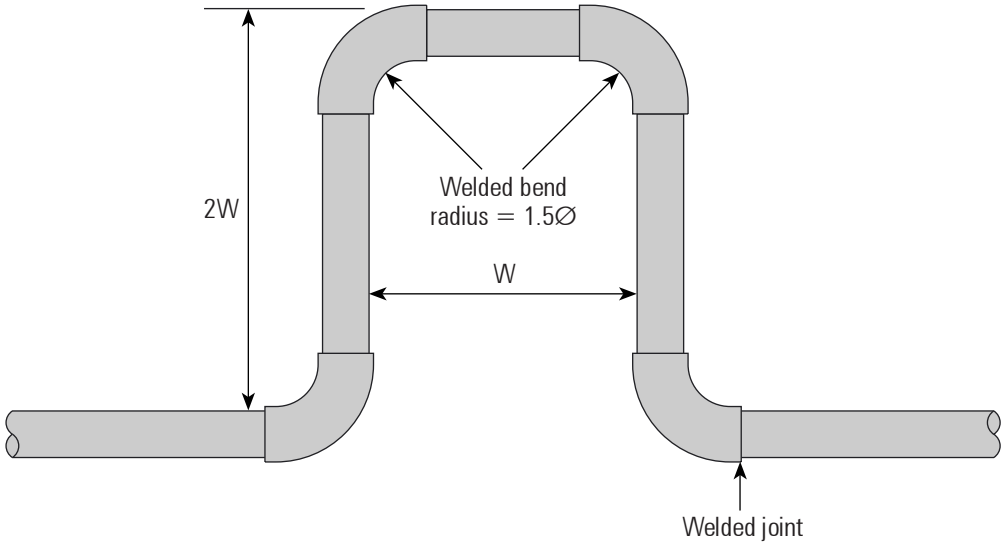


Fig. 10.4.9 Expansion loop

The expansion loop can be fabricated from lengths of straight pipes and elbows welded at the joints (Figure 10.4.9). An indication of the expansion of pipe that can be accommodated by these assemblies is shown in Figure 10.4.10.

It can be seen from Figure 10.4.9 that the depth of the loop should be twice the width, and the width is determined from Figure 10.4.10, knowing the total amount of expansion expected from the pipes either side of the loop.

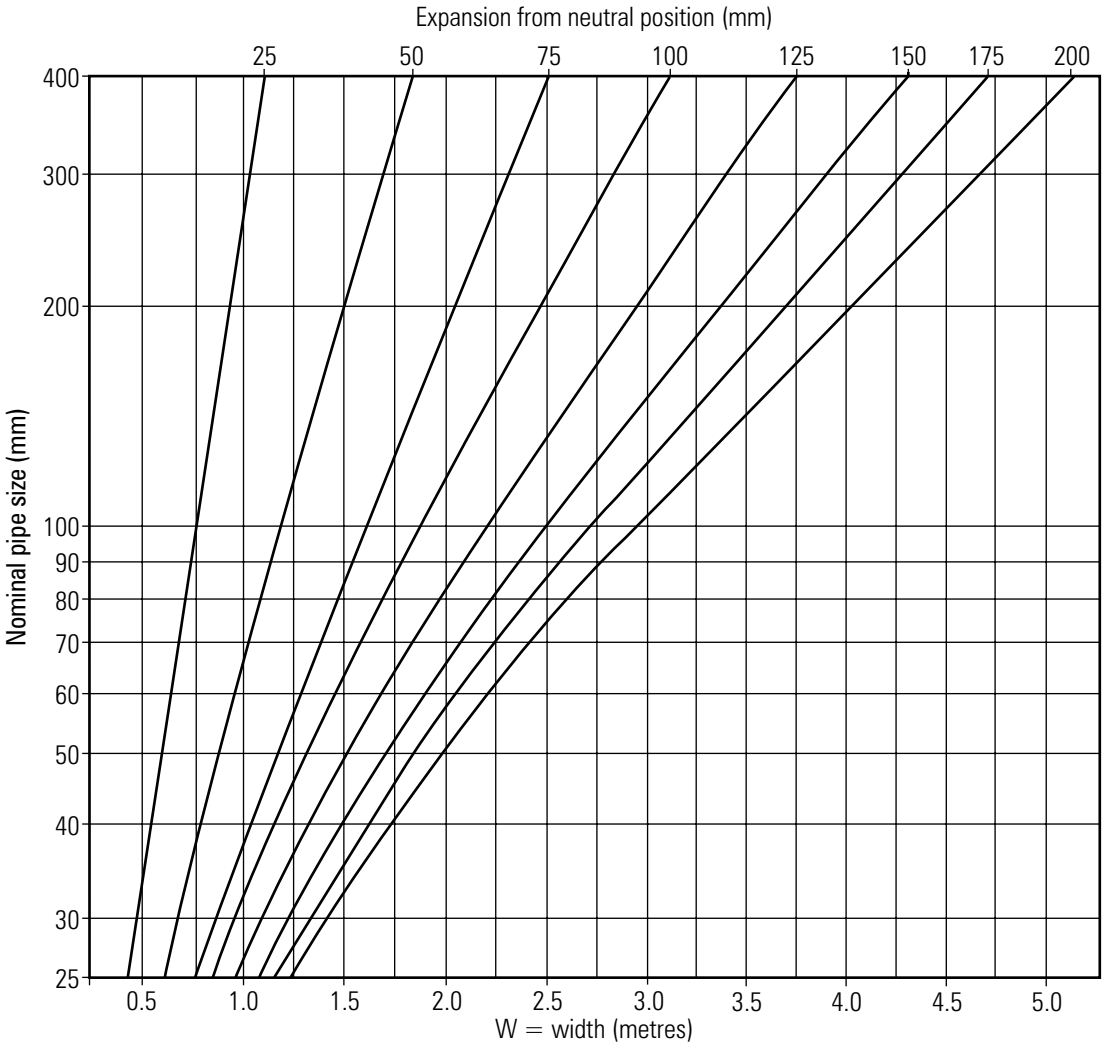


Fig. 10.4.10 Expansion loop capacity for carbon steel pipes

Sliding joint

These are sometimes used because they take up little room, but it is essential that the pipeline is rigidly anchored and guided in strict accordance with the manufacturers' instructions; otherwise steam pressure acting on the cross sectional area of the sleeve part of the joint tends to blow the joint apart in opposition to the forces produced by the expanding pipework (see Figure 10.4.11). Misalignment will cause the sliding sleeve to bend, while regular maintenance of the gland packing may also be needed.

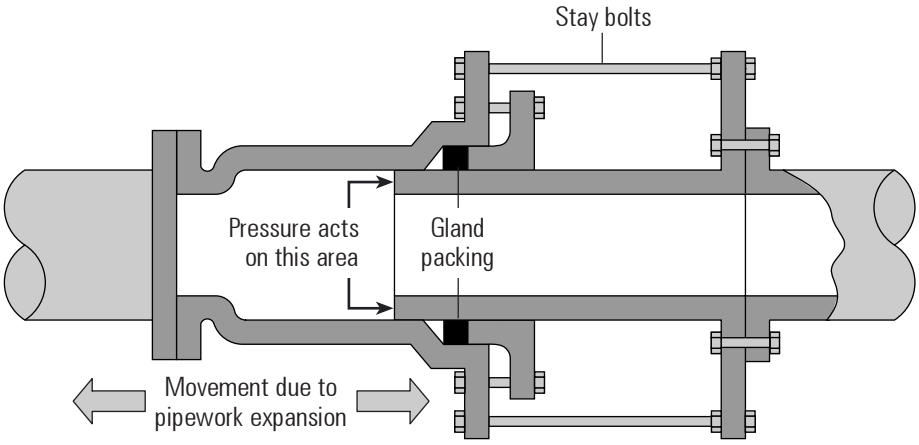


Fig. 10.4.11 Sliding joint

Expansion bellows

An expansion bellows, Figures 10.4.12, has the advantage that it requires no packing (as does the sliding joint type). But it does have the same disadvantages as the sliding joint in that pressure inside tends to extend the fitting, consequently, anchors and guides must be able to withstand this force.

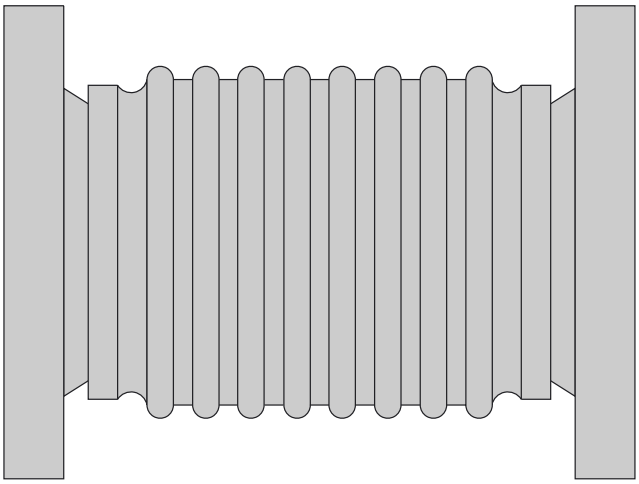


Fig. 10.4.12 Simple expansion bellows

Bellows may incorporate limit rods, which limit over-compression and over-extension of the element. These may have little function under normal operating conditions, as most simple bellows assemblies are able to withstand small lateral and angular movement. However, in the event of anchor failure, they behave as tie rods and contain the pressure thrust forces, preventing damage to the unit whilst reducing the possibility of further damage to piping, equipment and personnel (Figure 10.4.13 (b)).

Where larger forces are expected, some form of additional mechanical reinforcement should be built into the device, such as hinged stay bars (Figure 10.4.13 (c)).

There is invariably more than one way to accommodate the relative movement between two laterally displaced pipes depending upon the relative positions of bellows anchors and guides. In terms of preference, axial displacement is better than angular, which in turn, is better than lateral. Angular and lateral movement should be avoided wherever possible.

Figure 10.4.13 (a), (b), and (c) give a rough indication of the effects of these movements, but, under all circumstances, it is highly recommended that expert advice is sought from the bellows' manufacturer regarding any installation of expansion bellows.

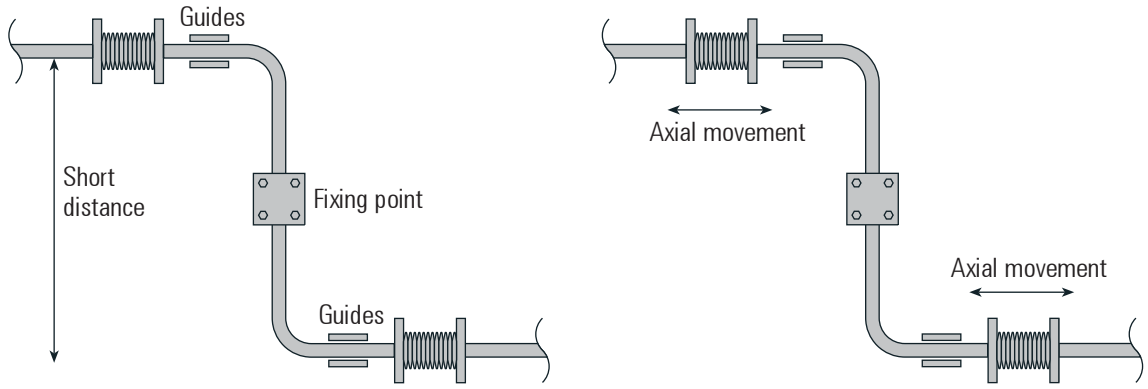


Fig. 10.4.13 (a) Axial movement of bellows

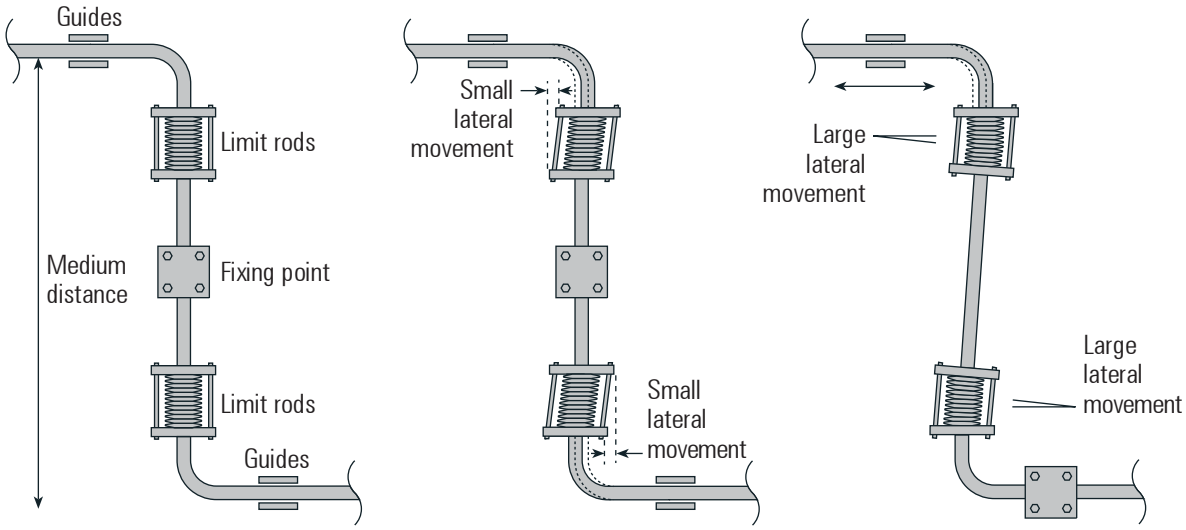


Fig. 10.4.13 (b) Lateral and angular movement of bellows

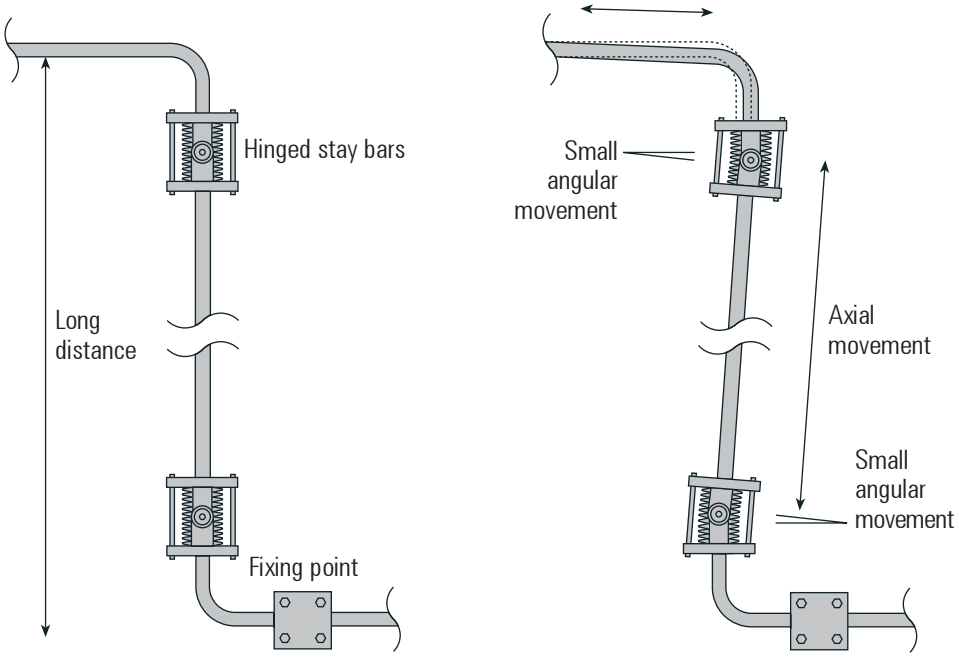


Fig. 10.4.13 (c) Angular and axial movement of bellows

Pipe support spacing

The frequency of pipe supports will vary according to the bore of the pipe; the actual pipe material (i.e. steel or copper); and whether the pipe is horizontal or vertical.

Some practical points worthy of consideration are as follows:

- Pipe supports should be provided at intervals not greater than shown in Table 10.4.3, and run along those parts of buildings and structures where appropriate supports may be mounted.
- Where two or more pipes are supported on a common bracket, the spacing between the supports should be that for the smallest pipe.
- When an appreciable movement will occur, i.e. where straight pipes are greater than 15 metres in length, the supports should be of the roller type as outlined previously.
- Vertical pipes should be adequately supported at the base, to withstand the total weight of the vertical pipe and the fluid within it. Branches from vertical pipes must not be used as a means of support for the pipe, because this will place undue strain upon the tee joint.
- All pipe supports should be specifically designed to suit the outside diameter of the pipe concerned. The use of oversized pipe brackets is not good practice.

Table 10.4.3 can be used as a guide when calculating the distance between pipe supports for steel and copper pipework.

Table 10.4.3 Recommended support for pipework

Nominal pipe size (mm)		Interval of horizontal run (metre)		Interval of vertical run (metre)	
Steel bore	Copper outside diameter	Mild steel	Copper	Mild steel	Copper
	15		1.2	2.4	1.8
15		1.8		3.0	
20	22	2.4	1.2	3.0	1.8
25	28	2.4	0.5	3.0	2.4
32	35	2.4	1.8	3.7	3.0
40	42	2.4	1.8	3.7	3.0
50	54	2.4	1.8	4.6	3.0
65	67	3.0	2.4	4.6	3.7
80	76	3.0	2.4	4.6	3.7
100	108	3.0	2.4	5.5	3.7
125	133	3.7	3.0	5.5	3.7
150	159	4.5	3.7	5.5	
200		6.0		8.5	
250		6.5		9.0	
300		7.0		10.0	

The subject of pipe supports is covered comprehensively in the European standard EN 13480, Part3.

Questions

1. A DN100 Schedule 40 pipe carries steam at 10 bar g over a length of 80 m. If the pipe is installed at 10°C, using Equation 10.4.1 and Table 10.4.1, by how much will it expand?

a | 291 mm ☐
b | 196 mm ☐
c | 352 mm ☐
d | 207 mm ☐

2. If the expansion of a pipe from installation to working temperature was 352 mm, what length of spacer would be used in 'cold drawing' the pipe being installed?

a | 352 mm ☐
b | 704 mm ☐
c | 176 mm ☐
d | 88 mm ☐

3. A 100 m run of 80 mm pipe at 15 bar g is supported at its ends and three intermediate points. It is trapped at intervals of 40 m. Noise and vibration often occurs at start-up. What do you think is required to put things right?

a | Fit more supports at 3 m intervals ☐
b | Check that the steam traps are removing condensate properly ☐
c | Check that the steam main isolating valve is opened slowly ☐
d | All of the above ☐

4. A 150 mm steam pipe is to incorporate a fabricated expansion loop to take up 125 mm of expansion. Using Figures 10.4.9 and 10.4.10, what should be the width and length of the loop?

a | Width : 2.6 m; Depth : 5.2 m ☐
b | Width : 5.2 m; Depth : 2.6 m ☐
c | Width : 5.2 m; Depth : 10.4 m ☐
d | Width : 1.3 m; Depth : 2.6 m ☐

5. What is one advantage of a bellows expansion fitting over a horseshoe loop?

a | It is less expensive ☐
b | Its operating movement can be observed ☐
c | Fewer pipe supports are required ☐
d | It takes up less space ☐

6. Condensate from a heater battery operating at 3.8 bar g returns to a vented pump set from where it is pumped through a carbon steel pipe to an atmospheric boiler feedtank which is 85 m away. Using the chart in Figure 10.4.1, what will be the approximate pipe expansion from an ambient temperature of 0°C?

a | 130 mm ☐
b | 200 mm ☐
c | 160 mm ☐
d | 100 mm ☐

Answers

1: d, 2: c, 3: d, 4: a, 5: d, 6: d

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Module 10.5

Air Venting, Heat Losses and a Summary of Various Pipe Related Standards

Air Venting, Heat Losses and a Summary of Various Pipe Related Standards

Air venting

When steam is first admitted to a pipe after a period of shutdown, the pipe is full of air. Further amounts of air and other non-condensable gases will enter with the steam, although the proportions of these gases are normally very small compared with the steam. When the steam condenses, these gases will accumulate in pipes and heat exchangers. Precautions should be taken to discharge them. The consequence of not removing air is a lengthy warming up period, and a reduction in plant efficiency and process performance.

Air in a steam system will also affect the system temperature. Air will exert its own pressure within the system, and will be added to the pressure of the steam to give a total pressure. Therefore, the actual steam pressure and temperature of the steam/air mixture will be lower than that suggested by a pressure gauge.

Of more importance is the effect air has upon heat transfer. A layer of air only 1 mm thick can offer the same resistance to heat as a layer of water 25 μm thick, a layer of iron 2 mm thick or a layer of copper 15 mm thick. It is very important therefore to remove air from any steam system.

Automatic air vents for steam systems (which operate on the same principle as thermostatic steam traps) should be fitted above the condensate level so that only air or steam/air mixtures can reach them. The best location for them is at the end of the steam mains as shown in Figure 10.5.1.

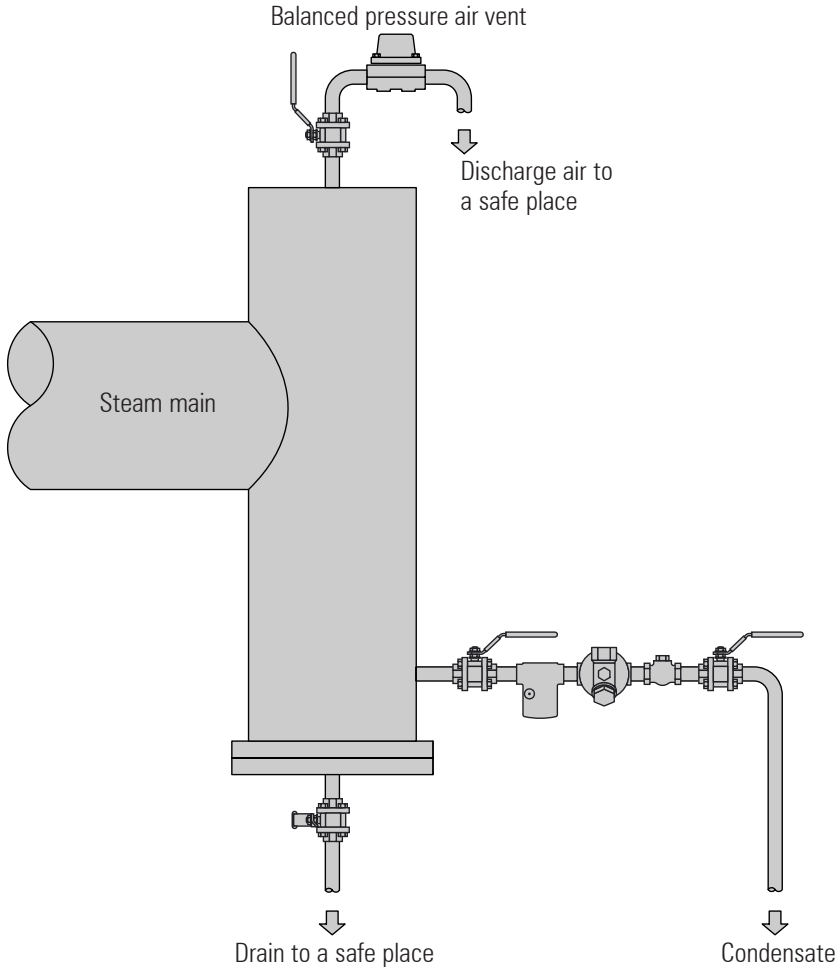


Fig. 10.5.1 Draining and venting at the end of a steam main

The discharge from an air vent must be piped to a safe place. In practice, a condensate line falling towards a vented receiver can accept the discharge from an air vent.

In addition to air venting at the end of a main, air vents should also be fitted:

- In parallel with an inverted bucket trap or, in some instances, a thermodynamic trap. These traps are sometimes slow to vent air on start-up.
- In awkward steam spaces (such as at the opposite side to where steam enters a jacketed pan).
- Where there is a large steam space (such as an autoclave), and a steam/air mixture could affect the process quality.

Reduction of heat losses

Even when a steam main has warmed up, steam will continue condensing as heat is lost by radiation. The condensing rate will depend upon the steam temperature, the ambient temperature, and the efficiency of the pipe insulation.

For a steam distribution system to be efficient, appropriate steps should be taken to ensure that heat losses are reduced to the economic minimum. The most economical thickness of insulation will depend upon several factors:

- Installation cost.
- The heat carried by the steam.
- Size of the pipework.
- Pipework temperature.

When insulating external pipework, dampness and wind speed must be taken into account.

The effectiveness of most insulation materials depends on minute air cells which are held in a matrix of inert material such as mineral wool, fibreglass or calcium silicate. Typical installations use aluminium clad fibreglass, aluminium clad mineral wool and calcium silicate. It is important that insulating material is not crushed or allowed to waterlog. Adequate mechanical protection and waterproofing are essential, especially in outdoor locations.

The heat loss from a steam pipe to water, or to wet insulation, can be as much as 50 times greater than from the same pipe to air. Particular care should be taken to protect steam lines, running through waterlogged ground, or in ducts, which may be subjected to flooding. The same applies to protecting the lagging from damage by ladders etc., to avoid the ingress of rainwater.

It is important to insulate all hot parts of the system with the exception of safety valves. This includes all flanged joints on the mains, and also the valves and other fittings. It was, at one time, common to cut back the insulation at each side of a flanged joint, to leave access to the bolts for maintenance purposes. This is equivalent to leaving about 0.5 m of bare pipe.

Fortunately, prefabricated insulating covers for flanged joints and valves are now more widely available. These are usually provided with fasteners so that they can readily be detached to provide access for maintenance purposes.

Calculation of heat transfer

The calculation of heat losses from pipes can be very complex and time consuming, and assume that obscure data concerning pipe wall thickness, heat transfer coefficients and various derived constants are easily available, which, usually, they are not.

The derivations of these formulae are outside the scope of this Module, but further information can be readily found in any good thermodynamics textbook. To add to this, an abundance of contemporary computer software is available for the discerning engineer.

This being so, pipe heat losses can easily be found by reference to Table 10.5.1 and a simple equation (Equation 2.12.2).

The table assumes ambient conditions of between 10 - 21°C, and considers heat losses from bare horizontal pipes of different sizes with steam contained at various pressures.

Table 10.5.1 Heat emission from pipes

Note: Heat emission from bare horizontal pipes with ambient temperatures between 10°C and 20°C and still air conditions

Temperature difference steam to air °C	Pipe size (DN)									
	15	20	25	32	40	50	65	80	100	150
	W/m									
60	60	72	88	111	125	145	172	210	250	351
70	72	87	106	132	147	177	209	253	311	432
80	86	104	125	155	171	212	248	298	376	519
90	100	121	146	180	196	248	291	347	443	610
100	116	140	169	207	223	287	336	400	514	706
110	132	160	193	237	251	328	385	457	587	807
120	149	181	219	268	282	371	436	517	664	914
130	168	203	247	301	313	417	490	581	743	1 025
140	187	226	276	337	347	464	547	649	825	1 142
150	208	250	306	374	382	514	607	720	911	1 263
160	229	276	338	413	418	566	670	794	999	1 390
170	251	302	372	455	457	620	736	873	1 090	1 521
180	275	330	407	499	497	676	805	955	1 184	1 658
190	299	359	444	544	538	735	877	1 041	1 281	1 800
200	325	389	483	592	582	795	951	1 130	1 381	1 947

Other factors can be included in the equation, for instance, if a pipe is lagged with insulation providing a reduction in heat losses to 10% of the uninsulated pipe, then it is multiplied by a factor of 0.1.

$$m_s = \frac{3.6\ Q\ L\ f}{h_{fg}} \qquad \textbf{Equation 2.12.2}$$

Where:
m_s = Rate of condensation (kg/h)
Q = Heat emission (W/m) (from Table 10.5.1)
L = Effective length of pipe, allowing for flanges and fittings (m)
f = Insulation factor. e.g.: 1 for bare pipes, 0.1 for good insulation
h_{fg} = Specific enthalpy of evaporation (kJ/kg)

Equivalent lengths:
Pair of mating flanges 0.5 m
Line size valve 1.0 m

Example 10.5.1
50 m of 100 mm pipe has 8 pairs of flanges and two valves, and carries saturated steam at 7 bar g. Ambient temperature is 10°C, and the insulation efficiency is given as 0.1

With reference to Table 10.5.1 and the application of Equation 10.5.1: determine the quantity of steam that will be condensed per hour:

Part 1 - Without insulation.

Part 2 - With the pipe insulated, but the valves and flanges are left without insulation.

Part 3 - Completely insulated.

Equivalent length of fittings:

(8 pairs of flanges @ 0.5 m) + (2 valves @ 1.0 m) = 6.0 m of pipe

Saturated steam at 7 bar g:

Steam temperature = 170°C
Temperature difference (pipe to ambient temperature) = 170°C - 10°C = 160°C
Enthalpy of evaporation (h_{fg}) = 2 048 kJ/kg
Heat loss per metre of 100 mm pipe (from Table 10.5.1) = 999 W/m

Part 1 - Without insulation:

$$m_s = \frac{3.6\ Q\ L\ f}{h_{fg}} \qquad \textbf{Equation 2.12.2}$$

$$m_s = \frac{3.6\ Q\ L\ f}{h_{fg}}$$

$$m_s = \frac{3.6 \times 999 \times (50 + 6) \times 1}{2\,048}$$

Condensing rate = 98.3 kg/h

Part 2 - Pipe insulated, but without insulation on the valves and flanges:

Consider the two elements separately:

Insulated pipe :

$$m_s = \frac{3.6\ Q\ L\ f}{h_{fg}}$$

$$m_s = \frac{3.6 \times 999 \times 50 \times 0.1}{2\,048}$$

Heat loss from pipes m_s = 8.78 kg/h

Uninsulated fittings :

$$m_s = \frac{3.6\ Q\ L\ f}{h_{fg}}$$

$$m_s = \frac{3.6 \times 999 \times 6 \times 1}{2\,048}$$

Heat loss from fittings m_s = 10.54 kg/h

Total condensing rate = heat loss from pipes + heat loss from fittings

Total condensing rate = 8.78 kg/h + 10.54 kg/h = 19.32 kg/h

Part 3 - Pipe and fittings insulated:

$$m_s = \frac{3.6\ Q\ L\ f}{h_{fg}}$$

$$m_s = \frac{3.6 \times 999 \times (50 + 6) \times 0.1}{2\,048}$$

Condensing rate = 9.83 kg/h

Relevant UK and International Standards

Symbols have been used to indicate, technically equivalent standards (=), and related standards (≠) respectively.

Table 10.5.2

BS 10	Specification for flanges and bolting for pipes, valves and fittings.
BS 21 = ISO 7/1 ≠ ISO 7/2	Specification for pipe threads for tubes and fittings where pressure tight joints are made on the threads.
EN 13480	Specification for metallic industrial piping.
BS 1306	Specification for copper and copper alloy piping systems.
EN 10255	Specification for screwed and socketed tubes and tubulars and for plain end steel tubes suitable for welding and screwing to BS 21 pipe threads.
BS 1560	Circular flanges for pipes, valves and fittings (Class designated): - Part 3, Section 3.1 - Specification for steel flanges (≠ ISO 7005). - Part 3, Section 3.2 - Specification for cast iron flanges (≠ ISO 7005-2). - Part 3, Section 3.3 - Specification for copper alloy and composite flanges (≠ ISO 7005-3).
BS 1600	Dimensions of steel pipe for the petroleum industry.
EN 10253-1	Specification for butt welding pipe fittings for pressure purposes.
BS 1710	Specification for identification of pipelines.
BS 2779= ISO 228/1, ISO 228/2	Specification for pipe threads for tubes and fittings where pressure tight joints are not made onthe threads.
EN 10220	Specification for dimensions and masses per unit length of welded and seamless steel pipes and tubes for pressure purposes.
BS 3601	Specification for steel pipes and tubes with specified room temperature properties for pressure purposes.
EN 10216-2 EN 10217-2/3/5	Specification for steel pipes and tubes for pressure purposes: carbon and carbon manganese steel with specified elevated temperature properties.
EN 10216-4 EN 10217-4	Specification for carbon and alloy steel pipes and tubes with specified low temperature properties for pressure purposes.
EN 10216-2 EN 10217-2 BS 3604-2	Steel pipes and tubes for pressure purposes: ferritic alloy steel with specified elevated temperature properties.
BS 3605-1/2	Austenitic stainless steel pipes and tubes for pressure purposes.
BS 3799	Specification for steel pipe fittings, screwed and socket welded for the petroleum industry.
BS 3974	Specification for pipe supports.
EN 1092-1 EN 1092-2	3.1 - Specification for steel flanges; 3.2 - Specification for cast iron flanges (≠ ISO 7005-2);
BS 4504	3.3 - Specification for copper alloy and composite flanges (≠ ISO 7005-3).

Summary

To summarise the 'Steam Distribution' Block of *The Steam and Condensate Loop*, the following checklist may be used to ensure that a steam distribution system will operate efficiently and effectively:

- Are steam mains properly sized?
- Are steam mains properly laid out?
- Are steam mains adequately drained?
- Are steam mains adequately air vented?
- Is adequate provision made for expansion?
- Can separators be used to improve steam quality?
- Are there leaking joints, glands or safety valves and why?
- Can redundant piping be blanked off or removed?
- Is the system effectively insulated?

Questions

1. As a general rule, where should air vents be fitted in a steam system?

a | At the highest points ☐

b | On a bypass around a steam trap ☐

c | At points where air is driven by the incoming steam ☐

d | Around all steam traps or points adjacent to them ☐

2. On what principle do automatic air vents operate?

a | They sense the difference in pressure between steam pressure and water pressure in a steam/air mixture ☐

b | They are temperature sensitive and remain open until steam at any pressure reaches them ☐

c | They remain open until the air passing through them reaches steam temperature ☐

d | They remain open until steam at saturation temperature reaches them. They will then close and will remain closed until, the temperature drops by approximately 12°C. ☐

3. From the following, what is the effect of air in a steam and condensate system?

a | Erosion of pipes ☐

b | Reduced heat output from the plant ☐

c | The steam traps will close as they would on sensing steam ☐

d | The air will prevent steam and condensate reaching the traps ☐

4. The surface cladding of insulation on a steam main is damaged and allows rain to enter the lagging. What is the effect?

a | No significant effect ☐

b | Less condensation will occur in the pipe because the heat transfer rate through water is less than the heat transfer rate through air ☐

c | The water will be evaporated and the steam formed will destroy the insulation ☐

d | Heat losses will increase because the heat transfer rate to water is much greater than to air ☐

5. A 75 m long, 150 mm steam main operates at 10 bar g. The main runs outside and the insulation is claimed to be 80% efficient. Approximately how much steam will be condensed in meeting heat losses from the pipe?

a | 200 kg/h ☐

b | 40 kg/h ☐

c | 97 kg/h ☐

d | 28 kg/h ☐

6. If, in Question 5, the insulation was 90% efficient, what would the heat losses now be?

a | 180 kg/h ☐

b | 20 kg/h ☐

c | 194 kg/h ☐

d | 14 kg/h ☐

Answers

1: c, 2: d, 3: b, 4: d, 5: b, 6: b

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Module 11.1

Introduction - Why Steam Traps?

Introduction

Throughout the history of steam utilisation, Spirax Sarco has been at the forefront of improving the efficiency of steam plant. Since 1935, the Spirax Sarco range of products has widened considerably and is now specified worldwide on the many types of plant employing steam. Today, there are few manufacturing processes that do not rely upon steam to provide an end product.

The steam trap is an essential part of any steam system. It is the important link between good steam and condensate management, retaining steam within the process for maximum utilisation of heat, but releasing condensate and incondensable gases at the appropriate time.

Although it is tempting to look at steam traps in isolation, it is their effect on the steam system as a whole that is often not appreciated. The following questions become important:

- Does the plant come quickly up to temperature or is it slow to respond, and its performance less than it should be?
- Is the system trouble free, or does inadequate steam trapping permit waterhammer, corrosion and leakage, and high maintenance costs?
- Does the design of the system have a negative effect on the life and efficiency of the steam traps?

It is often true that if an inappropriate steam trap is selected for a particular application, no ill effects are noticed. Sometimes, steam traps are even shut-off completely without any apparent problems, for example on a steam main, where incomplete drainage of condensate from one drain point often means that the remainder is simply carried on to the next. This could well be a problem if the next drain point is blocked or has been shut-off too!

The observant engineer may recognise that wear and tear of control valves, leakage and reduced plant output, can all be remedied by paying proper attention to steam trapping. It is natural for any mechanism to suffer from wear, and steam traps are no exception. When steam traps fail open, a certain amount of steam can be passed into the condensate system, although it is often a smaller quantity than might be expected. Fortunately, rapid means of detecting and rectifying such failures are now available to the steam user.

Why steam traps?

'The duty of a steam trap is to discharge condensate while not permitting the escape of live steam'

No steam system is complete without that crucial component 'the steam trap' (or trap). This is the most important link in the condensate loop because it connects steam usage with condensate return.

A steam trap quite literally 'purges' condensate, (as well as air and other incondensable gases), out of the system, allowing steam to reach its destination in as dry a state/condition as possible to perform its task efficiently and economically.

The quantity of condensate a steam trap has to deal with may vary considerably. It may have to discharge condensate at steam temperature (i.e. as soon as it forms in the steam space) or it may be required to discharge below steam temperature, giving up some of its 'sensible heat' in the process.

The pressures at which steam traps can operate may be anywhere from vacuum to well over a hundred bar. To suit these varied conditions there are many different types, each having their own advantages and disadvantages. Experience shows that steam traps work most efficiently when their characteristics are matched to that of the application. It is imperative that the correct trap is selected to carry out a given function under given conditions. At first sight it may not seem obvious what these conditions are. They may involve variations in operating pressure, heat load or condensate pressure. Steam traps may be subjected to extremes of temperature or even waterhammer. They may need to be resistant to corrosion or dirt. Whatever the conditions, correct steam trap selection is important to system efficiency.

It will become clear that one type of steam trap can not possibly be the correct choice for all applications

Considerations for steam trap selection

Air venting

At 'start-up', i.e. the beginning of the process, the heater space is filled with air, which unless displaced, will reduce heat transfer and increase the warm-up time. Start-up times increase and plant efficiency falls. It is preferable to purge air as quickly as possible before it has a chance to mix with the incoming steam. Should the air and steam be mixed together they can only be separated by condensing the steam to leave the air, which must then be vented to a safe place. Separate air vents may be required on larger or more awkward steam spaces, but in most cases air in the system is discharged through the steam traps. Here thermostatic traps have a clear advantage over some types of trap since they are fully open at start-up. Float traps with inbuilt thermostatic air vents are especially useful, while many thermodynamic traps are also quite capable of handling moderate amounts of air. However, the small hole in fixed orifice condensate outlets and the bleed hole in inverted bucket traps both vent air slowly. This could increase production times, warm-up times, and corrosion.

Condensate removal

Having vented the air, the trap must then pass the condensate but not the steam. Leakage of steam at this point is inefficient and uneconomical. The steam trap has to allow condensate to pass whilst trapping the steam in the process. If good heat transfer is critical to the process, then condensate must be discharged immediately and at steam temperature. Waterlogging is one of the main causes of inefficient steam plant as a result of incorrect steam trap selection.

Plant performance

When the basic requirements of removing air and condensate have been considered, attention may be turned to 'plant performance'. Simply put, unless specifically designed to waterlog, for a heat exchanger to operate at its best performance, the steam space must be filled with clean dry steam. The type of steam trap will influence this. For instance, thermostatic traps retain condensate until cooled to below saturation temperature. Should this condensate remain in the steam space, it would reduce the heat transfer area and the heater performance. The discharge of condensate at the lowest possible temperature may seem very attractive, but generally most applications require condensate to be removed from the steam space at steam temperature. This needs a steam trap with different operating properties to the thermostatic type, and this usually means either a mechanical or thermodynamic type trap.

Before choosing a particular steam trap it is necessary to consider the needs of the process. This will usually decide the type of trap required. The way in which the process is connected to the steam and condensate system may then decide the type of trap preferred to do the best job under the circumstances. Once chosen, it is necessary to size the steam trap. This will be determined by the system conditions and such process parameters as:

- Maximum steam and condensate pressures.
- Operating steam and condensate pressures.
- Temperatures and flowrates.
- Whether the process is temperature controlled.

These parameters will be discussed further in subsequent Modules within this Block.

Reliability

Experience has shown that 'good steam trapping' is synonymous with reliability, i.e. optimum performance with the minimum of attention.

Causes of unreliability are often associated with the following:

- Corrosion, due to the condition of the condensate. This can be countered by using particular materials of construction, and good feedwater conditioning.
- Waterhammer, often due to a lift after the steam trap, sometimes overlooked at the design stage and often the cause of unnecessary damage to otherwise reliable steam traps.
- Dirt, accumulating from a system where water treatment compound is carried over from the boiler, or where pipe debris is allowed to interfere with trap operation.

The primary task of a steam trap is the proper removal of condensate and air and this requires a clear understanding of how steam traps operate.

Flash steam

An effect caused by passing hot condensate from a high pressure system to a low pressure system is the naturally occurring phenomenon of flash steam. This can confuse the observer regarding the condition of the steam trap.

Consider the enthalpy of freshly formed condensate at steam pressure and temperature (obtainable from steam tables). For example, at a pressure of 7 bar g, condensate will contain 721 kJ/kg at a temperature of 170.5°C. If this condensate is discharged to atmosphere, it can only exist as water at 100°C, containing 419 kJ/kg of enthalpy of saturated water. The surplus enthalpy content of 721 - 419 i.e. 302 kJ/kg, will boil off a proportion of the water, producing a quantity of steam at atmospheric pressure.

The low pressure steam produced is usually referred to as 'flash steam'. The amount of 'flash' steam released can be calculated as follows:

$$\text{Flash steam produced} = \frac{\text{Excess enthalpy kJ/kg}}{\text{Specific enthalpy of evaporation at lower pressure}}$$

$$= \frac{302.0 \text{ kJ/kg}}{2\,257.0 \text{ kJ/kg}}$$

$$= 0.134 \text{ kg of steam per kg of condensate}$$

or

13.4%

If the trap were discharging 500 kg/h of condensate at 7 bar g to atmosphere, the amount of flash steam generated would be 500 x 0.134 = 67 kg/h, equivalent to approximately 38 kW of energy loss!

This represents quite a substantial quantity of useful energy, which is all too often lost from the heat balance of the steam and condensate loop, and offers a simple opportunity to increase system efficiency if it can be captured and used.

How steam traps operate

There are three basic types of steam trap into which all variations fall, all three are classified by International Standard ISO 6704:1982.

Types of steam trap:

- **Thermostatic (operated by changes in fluid temperature) -** The temperature of saturated steam is determined by its pressure. In the steam space, steam gives up its enthalpy of evaporation (heat), producing condensate at steam temperature. As a result of any further heat loss, the temperature of the condensate will fall. A thermostatic trap will pass condensate when this lower temperature is sensed. As steam reaches the trap, the temperature increases and the trap closes.
- **Mechanical (operated by changes in fluid density) -** This range of steam traps operates by sensing the difference in density between steam and condensate. These steam traps include 'ball float traps' and 'inverted bucket traps'. In the 'ball float trap', the ball rises in the presence of condensate, opening a valve which passes the denser condensate. With the 'inverted bucket trap', the inverted bucket floats when steam reaches the trap and rises to shut the valve. Both are essentially 'mechanical' in their method of operation.
- **Thermodynamic (operated by changes in fluid dynamics) -** Thermodynamic steam traps rely partly on the formation of flash steam from condensate. This group includes 'thermodynamic', 'disc', 'impulse' and 'labyrinth' steam traps.

Also loosely included in this type are 'fixed orifice traps', which cannot be clearly defined as automatic devices as they are simply a fixed diameter hole set to pass a calculated amount of condensate under one set of conditions.

All rely on the fact that hot condensate, released under dynamic pressure, will flash-off to give a mixture of steam and water.

The following Modules include reference to these steam traps.

International and European Standards relating to steam traps

ISO 6552 : 1980
(BS 6023 : 1981)
Glossary of technical terms for automatic steam traps

ISO 6553 : 1980
CEN 26553 : 1991 (Replaces BS 6024 : 1981)
Marking of automatic steam traps

ISO 6554 : 1980
CEN 26554 : 1991 (Replaces BS 6026 : 1981)
Face-to-face dimensions for flanged automatic steam traps

ISO 6704 : 1982
CEN 26704 : 1991 (Replaces BS 6022 : 1983)
Classification of automatic steam traps

ISO 6948 :1981
CEN 26948 : 1991 (Replaces BS 6025 : 1982)
Production and performance characteristic tests for automatic steam traps

ISO 7841 : 1988
CEN 27841 : 1991 (Replaces BS 6027 : 1990)
Methods for determination of steam loss of automatic steam traps

ISO 7842 : 1988
CEN 27842 : 1991 (Replaces BS 6028 : 1990)
Methods for determination of discharge capacity of automatic steam traps

Questions

1. Are steam traps required to pass air?

a | Steam traps should not pass air under any circumstances ☐
b | Only when the trap has passed all the condensate ☐
c | Air should be removed as soon as it reaches the trap ☐
d | Only on high pressure steam systems ☐

2. How is flash steam produced?

a | From condensate passing from high to low pressure systems ☐
b | From saturated steam ☐
c | From superheated steam ☐
d | From steam mixed with high temperature air ☐

3. Should steam traps match the application?

a | Any steam trap (if properly sized) can be fitted to any application ☐
b | Only if fitted to a heat exchanger ☐
c | Only on high pressure steam systems ☐
d | Yes ☐

4. Unless they are designed to flood, what is important when removing condensate from heat exchangers?

a | Condensate is allowed to sub-cool before reaching the trap ☐
b | Condensate is removed at steam temperature ☐
c | Condensate should back-up into the steam space ☐
d | That the trap is fitted level with or above the heater outlet ☐

5. Can temperature controlled applications be trapped?

a | Traps should not be fitted under any circumstances ☐
b | Only if there is no lift after the trap ☐
c | If the pressure on the trap is always higher than the backpressure ☐
d | Pumps should always be fitted to remove condensate ☐

6. What are the main considerations for steam trap selection?

a | Price ☐
b | Air venting, plant performance, flow capacity and reliability ☐
c | Connections ☐
d | The trap must be the same size as the condensate drain line ☐

Answers

1: c, 2: a, 3: d, 4: b, 5: c, 6: b

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Module 11.2

Thermostatic Steam Traps

Thermostatic Steam Traps

Liquid expansion steam trap

This is one of the simplest thermostatic traps and is shown in Figure 11.2.1. An oil filled element expands when heated to close the valve against the seat. The adjustment allows the temperature of the trap discharge to be altered between 60°C and 100°C, which makes it ideally suited as a device to get rid of large quantities of air and cold condensate at start-up.

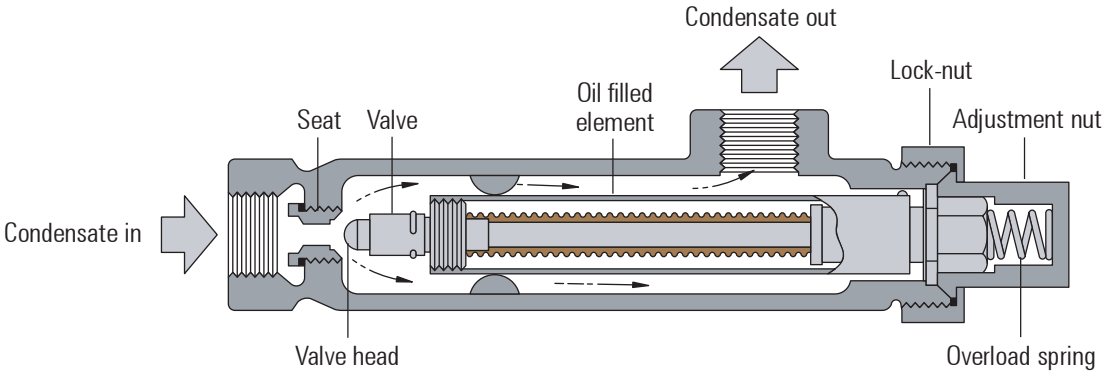


Fig. 11.2.1 Liquid expansion steam trap

As discussed in Module 2.2, the temperature of saturated steam varies with pressure. Figure 11.2.2 shows the saturation curve for steam, together with the fixed temperature response line (X - X) of the liquid expansion trap, set at 90°C.

It can be seen from Figure 11.2.2 that when the pressure is at pressure P_1, condensate would have to cool by only a small amount (ΔT_1), and trapping would be acceptable. However, if pressure is increased to P_2 then condensate has to cool more (ΔT_2) to pass through the steam trap. This cooling can only occur in the pipe between the process and trap, and if the trap discharge temperature remains constant, the process will waterlog.

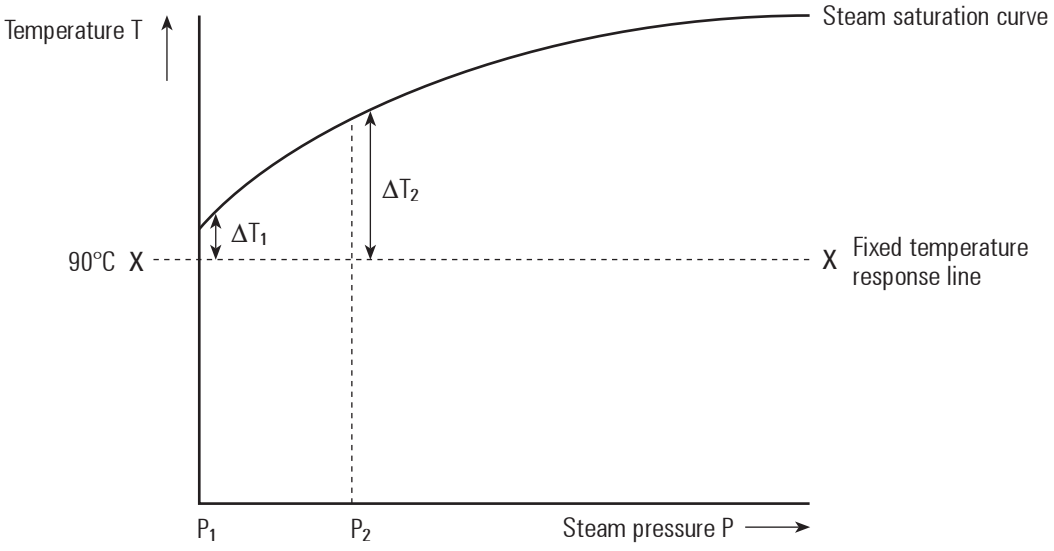


Fig. 11.2.2 Response of a liquid expansion steam trap X - X

Typical application

Because of its fixed temperature discharge characteristic, the liquid expansion trap may be usefully employed as a 'shutdown drain trap'. Here, its outlet must always point upwards, as illustrated in Figure 11.2.3, to enable continuous immersion of the oil filled element. As the trap can only discharge between 60°C - 100°C it will only normally open during start-up. It can be installed alongside a mains drain trap which would normally be piped to a condensate return line.

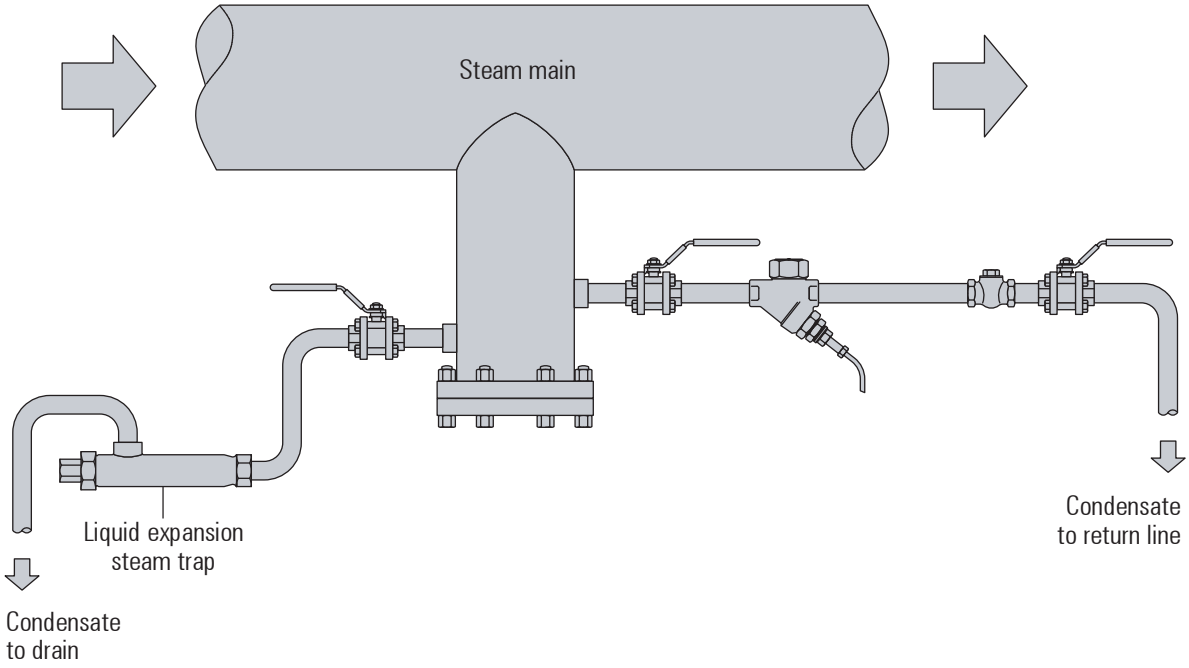


Fig. 11.2.3 Installation of a liquid expansion steam trap

Advantages of the liquid expansion steam trap:

- Liquid expansion traps can be adjusted to discharge at low temperatures, giving an excellent 'cold drain' facility.
- Like the balanced pressure trap, the liquid expansion trap is fully open when cold, giving good air discharge and maximum condensate capacity on 'start-up' loads.
- The liquid expansion trap can be used as a start-up drain trap on low pressure superheated steam mains where a long cooling leg is guaranteed to flood with cooler condensate. It is able to withstand vibration and waterhammer conditions.

Disadvantages of the liquid expansion steam trap:

- The flexible tubing of the element can be destroyed by corrosive condensate or superheat.
- Since the liquid expansion trap discharges condensate at a temperature of 100°C or below, it should never be used on applications which demand immediate removal of condensate from the steam space.
- If the trap is to be subjected to freezing conditions the trap and its associated pipework must be well insulated.
- The liquid expansion trap is not normally a trapping solution on its own, as it usually requires another steam trap to operate in parallel. However, it can often be used where start-up rate is not an important consideration, such as when draining small tank heating coils.

Balanced pressure steam trap

A large improvement on the liquid expansion trap is the balanced pressure trap, shown in Figure 11.2.4. Its operating temperature is affected by the surrounding steam pressure. The operating element is a capsule containing a special liquid and water mixture with a boiling point below that of water. In the cold conditions that exist at start-up, the capsule is relaxed. The valve is off its seat and is wide open, allowing unrestricted removal of air. This is a feature of all balanced pressure traps and explains why they are well suited to air venting.

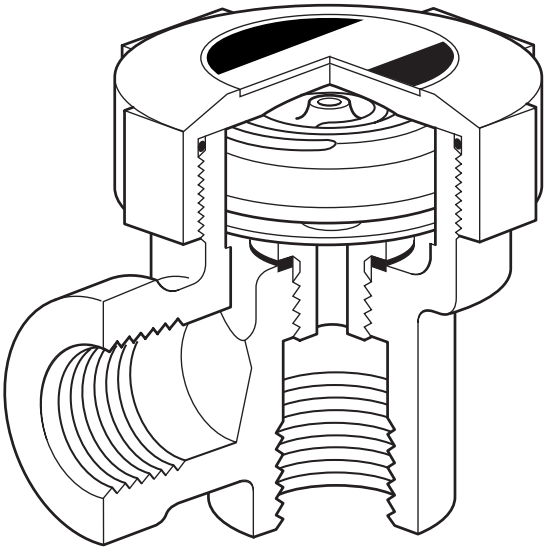


Fig. 11.2.4 Balanced pressure steam trap with replaceable capsule

As condensate passes through the balanced pressure steam trap, heat is transferred to the liquid in the capsule. The liquid vaporises before steam reaches the trap. The vapour pressure within the capsule causes it to expand and the valve shuts. Heat loss from the trap then cools the water surrounding the capsule, the vapour condenses and the capsule contracts, opening the valve and releasing condensate until steam approaches again and the cycle repeats (Figure 11.2.5).

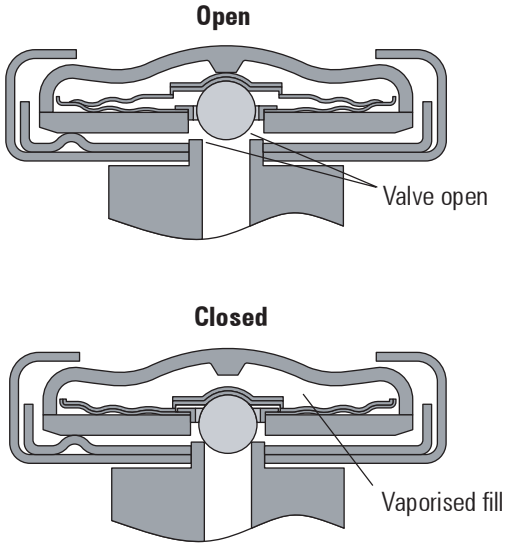


Fig. 11.2.5 Operation of balanced pressure steam trap capsule

The differential below steam temperature at which the trap operates is governed by the concentration of the liquid mixture in the capsule. The 'thin-walled' element gives a rapid response to changes in pressure and temperature. The result is the response line as illustrated in Figure 11.2.6.

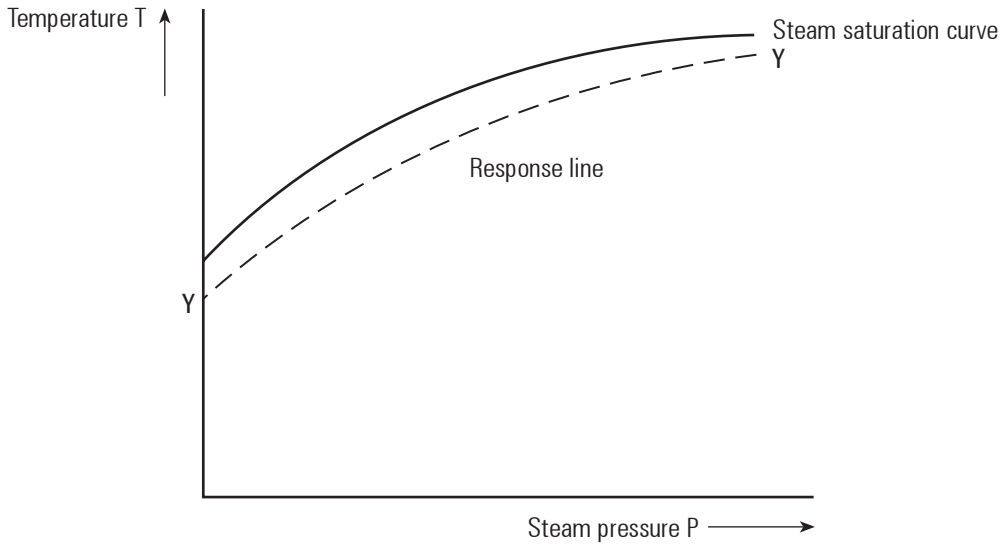


Fig. 11.2.6 Typical response of a balanced pressure steam trap Y - Y

Early bellows type elements of non-ferrous construction were susceptible to damage by waterhammer. The introduction of stainless steel elements improved reliability considerably. Figure 11.2.7 shows an exploded view of a modern balanced pressure steam trap arrangement that has considerable resistance to damage from waterhammer, superheat and corrosion.

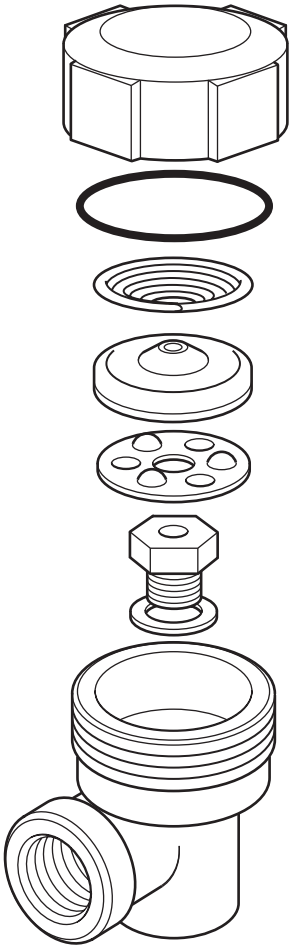


Fig. 11.2.7 Typical balanced pressure capsule arrangement

Advantages of the balanced pressure steam trap:

- Small, light and has a large capacity for its size.
- The valve is fully open on start-up, allowing air and other non-condensable gases to be discharged freely and giving maximum condensate removal when the load is greatest.
- This type of trap is unlikely to freeze when working in an exposed position (unless there is a rise in the condensate pipe after the trap, which would allow water to run back and flood the trap when the steam is off).
- The modern balanced pressure trap automatically adjusts itself to variations of steam pressure up to its maximum operating pressure. It will also tolerate up to 70°C of superheat.
- Trap maintenance is simple. The capsule and valve seat are easily removed, and replacements can be fitted in a few minutes without removing the trap from the line.

Disadvantages of the balanced pressure steam trap:

- The older style balanced pressure steam traps had bellows which were susceptible to damage by waterhammer or corrosive condensate. Welded stainless steel capsules introduced more recently, are better able to tolerate such conditions.
- In common with all other thermostatic traps, the balanced pressure type does not open until the condensate temperature has dropped below steam temperature (the exact temperature difference being determined by the fluid used to fill the element). This is clearly a disadvantage if the steam trap is chosen for an application in which waterlogging of the steam space can not be tolerated, for example; mains drainage, heat exchangers, critical tracing.

Bimetallic steam trap

As the name implies, bimetallic steam traps are constructed using two strips of dissimilar metals welded together into one element. The element deflects when heated. (Figure 11.2.8):

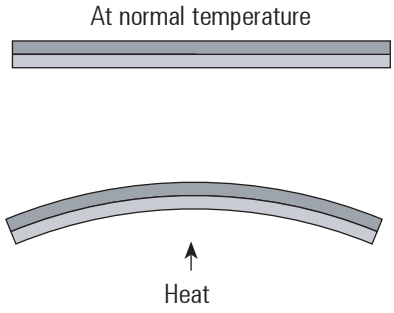


Fig. 11.2.8 Simple bimetallic element

There are two important points to consider regarding this simple element:

- Operation of the steam trap takes place at a certain fixed temperature, which may not satisfy the requirements of a steam system possibly operating at varying pressures and temperatures (see Figure 11.2.9).
- Because the power exerted by a single bimetal strip is small, a large mass would have be used which would be slow to react to temperature changes in the steam system.

The performance of any steam trap can be measured by its response to the steam saturation curve. The ideal response would closely follow the curve and be just below it. A simple bimetal element tends to react to temperature changes in a linear fashion.

Figure 11.2.9 shows the straight line characteristic of a simple bimetal element relative to the steam saturation curve. As steam pressure increases above P_1, the difference between steam saturation temperature and trap operating temperature would increase. Waterlogging increases with system pressure, highlighting the trap's inability to respond to changing pressure conditions.

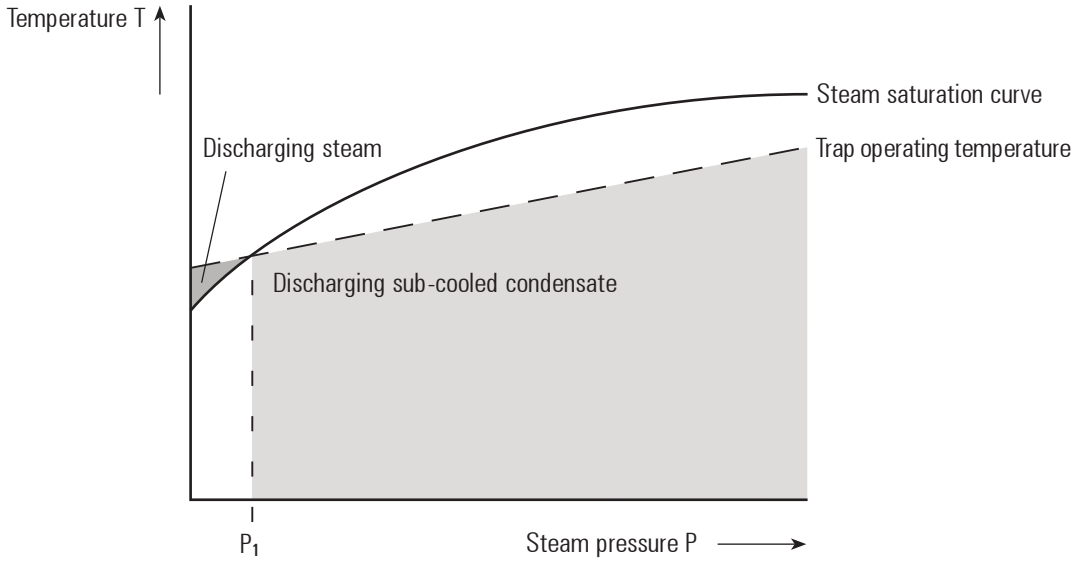


Fig. 11.2.9 Typical response of a single element bimetal steam trap

It needs to be noted that at pressures below P_1, the steam trap operating temperature is actually above the saturation temperature. This would cause the steam trap to pass steam at these lower pressures. It may be possible to ensure the steam trap is adjusted during manufacture to ensure that this portion of the saturation curve is always above the operating line. However, due to the linear action of the element, the difference between the two would increase even more with system pressure, increasing the waterlogging effect.

Clearly, this is not a satisfactory operation for any steam trap, and various attempts have been made by manufacturers to improve upon the situation. Some use combinations of two different sets of bimetal leaves in a single stack, which operate at different temperatures (Figure 11.2.10).

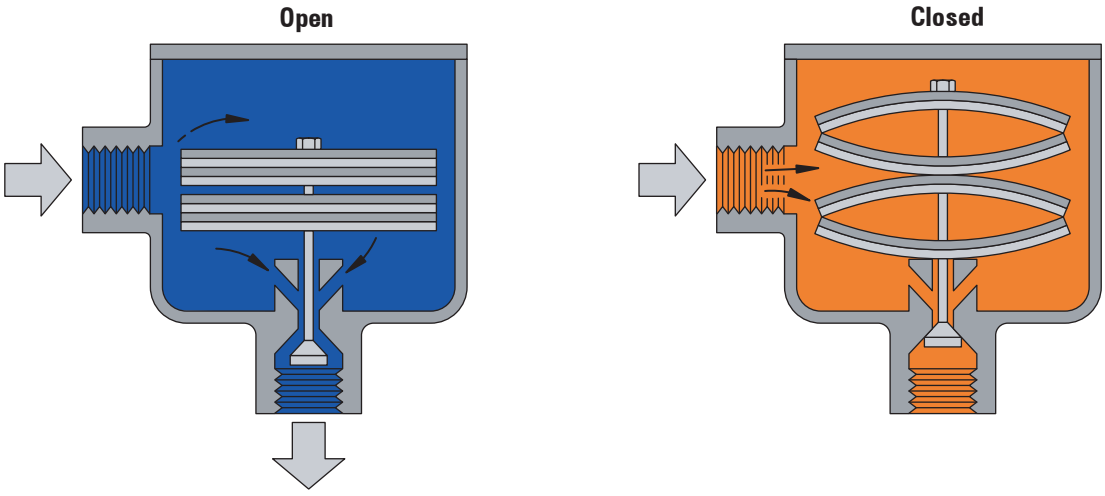


Fig. 11.2.10 Operation of a bimetel steam trap with two leaf element

The typical result is the split response line similar to that shown in Figure 11.2.11. This is an improvement on Figure 11.2.9, but still does not exactly follow the saturation curve. One set of bimetal leaves deflect to give the response P_1 to P_2. At a higher temperature a second set of bimetal leaves contributes to give response P_2 to P_3. Clearly, although an improvement from the former design, this is still unsatisfactory in terms of following the saturation curve.

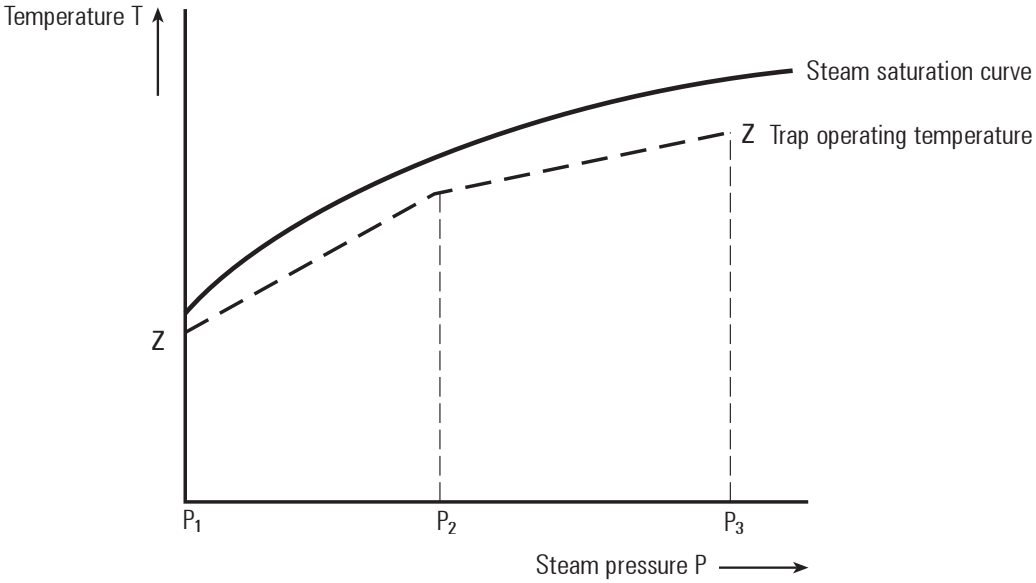


Fig. 11.2.11 Typical response of a two leaf element Z - Z

A more innovative design is the disc spring thermostatic element shown in Figure 11.2.12. The thermostatic element is made up of a set of bimetal discs. These discs, if acting directly between the valve stem and the seat (as with some thermostatic steam traps), cause the discharge temperature of the condensate to change linearly with changing pressure (curve 'A', Figure 11.2.13). By incorporating a spring washer between the discs and a recess in the seat, this absorbs some of the bimetal expansion at low pressure so that a greater temperature change must occur with changing pressure. The spring washer shape is preferred over a coil spring because it develops force in an exponentially increasing rate, rather than in a linear rate. This effect takes place up to 15 bar g until the spring is deflected to the bottom of the recess, and means that the discharge temperature of the condensate will follow the steam saturation curve more accurately (curve 'B', Figure 11.2.13). Discharge rates are also improved by the dynamic clack which tends to produce a blast discharge.

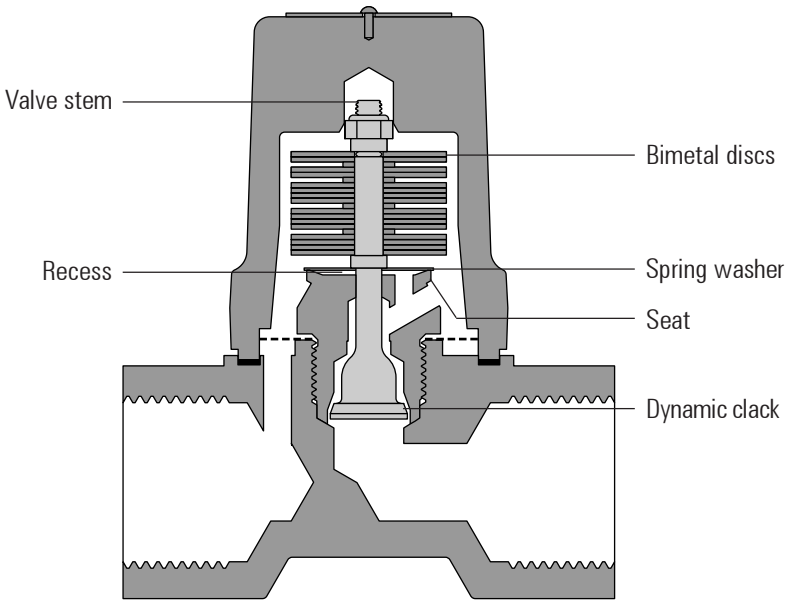


Fig. 11.2.12 Multi-cross elements as used in the Spirax Sarco SM range of bimetallic steam traps

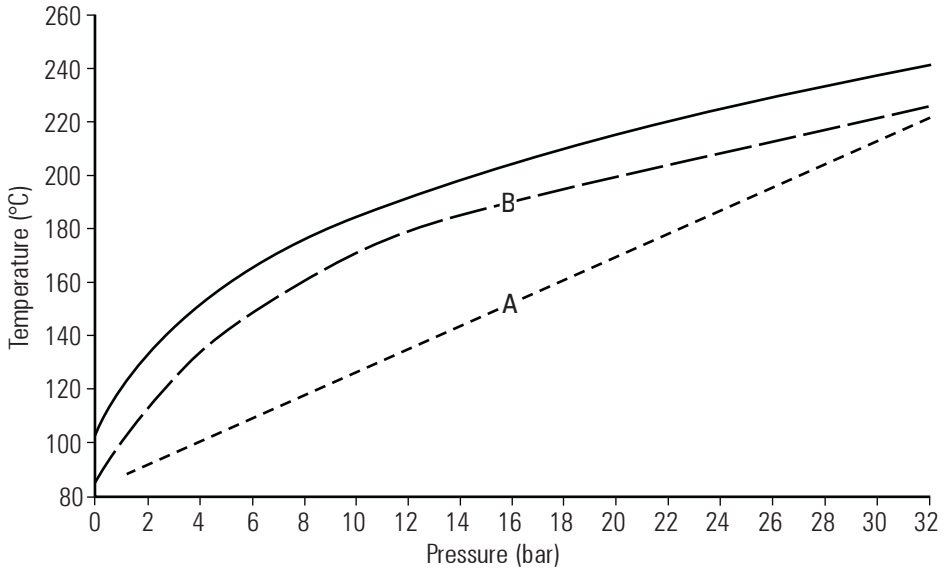


Fig. 11.2.13 Comparing the operating temperatures of single leaf and multi-leaf bimetallic traps

Advantages of the bimetallic steam trap:

- Bimetallic steam traps are usually compact, yet can have a large condensate capacity.
- The valve is wide open when the steam trap is cold, giving good air venting capability and maximum condensate discharge capacity under 'start-up' conditions.
- As condensate tends to drain freely from the outlet, this type of steam trap will not freeze up when working in an exposed position. The bodies of some bimetallic steam traps are designed in such a way that they will not receive any damage even if freezing does occur.
- Bimetallic steam traps are usually able to withstand waterhammer, corrosive condensate, and high steam pressures.
- The bimetal elements can work over a wide range of steam pressures without any need for a change in the size of the valve orifice.
- If the valve is on the downstream side of the seat, it will tend to resist reverse flow through the steam trap. However, if there is any possibility of reverse flow, a separate check valve should be fitted downstream of the trap.
- As condensate is discharged at varying temperatures below saturation temperature and, provided waterlogging of the steam space can be tolerated, some of the enthalpy of saturated water can be transferred to the plant. This extracts the maximum energy from the condensate before it drains to waste, and explains why these traps are used on tracer lines where condensate is often dumped to waste.
- Maintenance of this type of steam trap presents few problems, as the internals can be replaced without removing the trap body from the line.
- The flash steam produced whenever condensate is discharged from a higher to a lower pressure will tend to cause an increase in backpressure in the condensate line. The cooling leg allows the condensate to cool down, producing less flash steam in the condensate line and thus helping to reduce the backpressure.

Disadvantages of the bimetallic steam trap:

- As condensate is discharged below steam temperature, waterlogging of the steam space will occur unless the steam trap is fitted at the end of a long cooling leg, typically 1 - 3 m of unlagged pipe (see Fig. 11.2.14). Bimetallic steam traps are not suitable for fitting to process plants where immediate condensate removal is vital for maximum output to be achieved. This is particularly relevant on temperature controlled plants.
- Some bimetallic steam traps are vulnerable to blockage from pipe dirt due to low internal flow velocities. However, some bimetallic traps have specially shaped valve trims that capture the discharge energy to open the valve more. These tend to give an intermittent blast discharge characteristic rather than a continual dribble discharge, and as such tend to be self-cleaning. These valve trims are sometimes referred to as dynamic clacks.
- If the bimetallic steam trap has to discharge against a significant backpressure, the condensate must cool to a lower temperature than is normally required before the valve will open. A 50% backpressure may cause up to a 50°C drop in discharge temperature. It may be necessary to increase the length of cooling leg to meet this condition.
- Bimetallic steam traps do not respond quickly to changes in load or pressure because the element is slow to react.

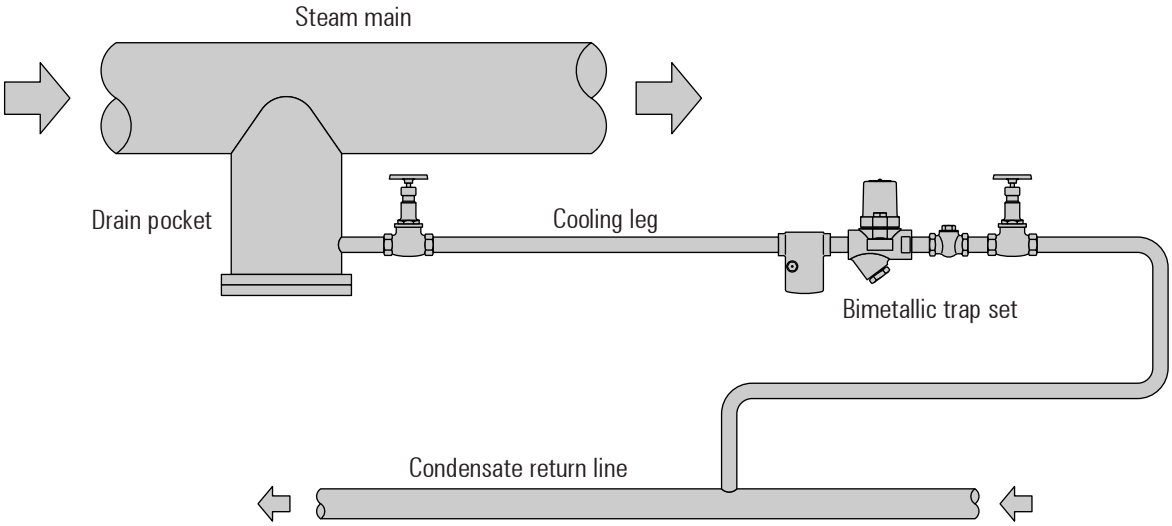


Fig. 11.2.14 Bimetallic steam trap with cooling leg

Questions

1. What is a characteristic feature of thermostatic steam traps?

a| They pass condensate at steam temperature ☐
b| They operate by holding back condensate until it has cooled ☐
c| They cannot be fitted outside ☐
d| They can only be fitted on low pressure steam systems ☐

2. Where can a liquid expansion trap be fitted?

a| Where condensate has to be removed just below steam temperature ☐
b| Where it is necessary to discharge condensate above 100°C ☐
c| Where it is necessary to remove cool condensate at start-up ☐
d| On a superheated steam main ☐

3. Where would a bimetallic steam trap not be fitted?

a| Outside where freezing can occur ☐
b| Where waterhammer is likely to occur ☐
c| Where condensate is to be discharged below steam temperature ☐
d| Where condensate has to be discharged at steam temperature ☐

4. Balance pressure traps are what type of steam trap?

a| Thermodynamic ☐
b| Mechanical ☐
c| Thermostatic ☐
d| They do not belong to any one specific type of trap family ☐

5. What can a balanced pressure trap do that a bimetallic cannot?

a| It can withstand moderate degrees of superheat ☐
b| It follows the steam saturation curve better than a bimetallic trap ☐
c| It can be fitted where waterlogging can be tolerated ☐
d| It can discharge large quantities of air at start-up ☐

6. What is the effect of increasing backpressure on a bimetallic trap?

a| It reduces the temperature of the condensate released by the trap ☐
b| It allows the trap to be used on a higher steam pressure ☐
c| It increases the differential pressure across the trap ☐
d| The trap must be the same size as the condensate drain line ☐

Answers

1: b, 2: c, 3: d, 4: c, 5: b, 6: a

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Module 11.3

Mechanical Steam Traps

Mechanical Steam Traps

Ball float steam trap

The ball float type trap operates by sensing the difference in density between steam and condensate. In the case of the trap shown in Figure 11.3.1, condensate reaching the trap will cause the ball float to rise, lifting the valve off its seat and releasing condensate. As can be seen, the valve is always flooded and neither steam nor air will pass through it, so early traps of this kind were vented using a manually operated cock at the top of the body. Modern traps use a thermostatic air vent, as shown in Figure 11.3.2. This allows the initial air to pass whilst the trap is also handling condensate.

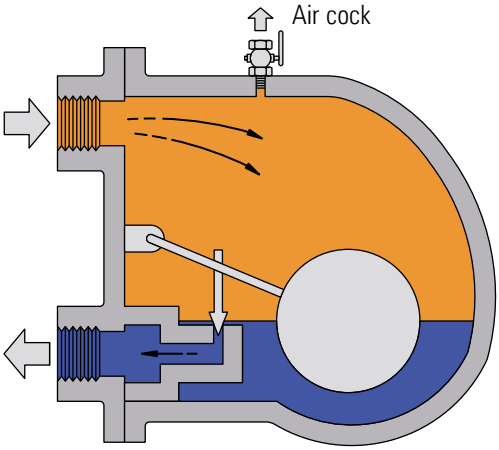


Fig. 11.3.1 Float trap with air cock

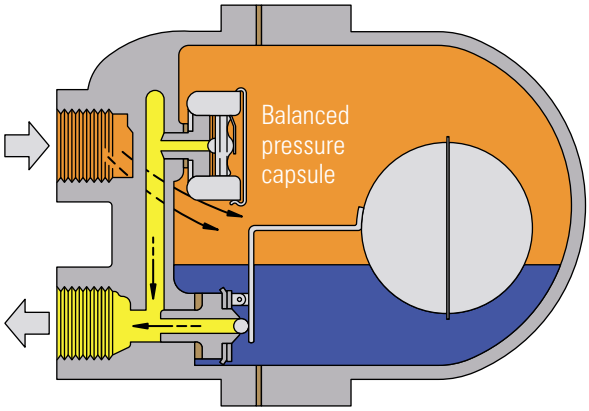


Fig. 11.3.2 Float trap with thermostatic air vent

The automatic air vent uses the same balanced pressure capsule element as a thermostatic steam trap, and is located in the steam space above the condensate level. After releasing the initial air, it remains closed until air or other non-condensable gases accumulate during normal running and cause it to open by reducing the temperature of the air/steam mixture. The thermostatic air vent offers the added benefit of significantly increasing condensate capacity on cold start-up.

In the past, the thermostatic air vent was a point of weakness if waterhammer was present in the system. Even the ball could be damaged if the waterhammer was severe. However, in modern float traps the air vent is a compact, very robust, all stainless steel capsule, and the modern welding techniques used on the ball makes the complete float-thermostatic steam trap very robust and reliable in waterhammer situations.

In many ways the float-thermostatic trap is the closest to an ideal steam trap. It will discharge condensate as soon as it is formed, regardless of changes in steam pressure.

Advantages of the float-thermostatic steam trap

- The trap continuously discharges condensate at steam temperature. This makes it the first choice for applications where the rate of heat transfer is high for the area of heating surface available.
- It is able to handle heavy or light condensate loads equally well and is not affected by wide and sudden fluctuations of pressure or flowrate.
- As long as an automatic air vent is fitted, the trap is able to discharge air freely.
- It has a large capacity for its size.
- The versions which have a steam lock release valve are the only type of trap entirely suitable for use where steam locking can occur.
- It is resistant to waterhammer.

Disadvantages of the float-thermostatic steam trap

- Although less susceptible than the inverted bucket trap, the float type trap can be damaged by severe freezing and the body should be well lagged, and/or complemented with a small supplementary thermostatic drain trap, if it is to be fitted in an exposed position.
- As with all mechanical type traps, different internals are required to allow operation over varying pressure ranges. Traps operating on higher differential pressures have smaller orifices to balance the bouyancy of the float.

Inverted bucket steam trap

The inverted bucket steam trap is shown in Figure 11.3.3. As its name implies, the mechanism consists of an inverted bucket which is attached by a lever to a valve. An essential part of the trap is the small air vent hole in the top of the bucket. Figure 11.3.3 shows the method of operation. In (i) the bucket hangs down, pulling the valve off its seat. Condensate flows under the bottom of the bucket filling the body and flowing away through the outlet. In (ii) the arrival of steam causes the bucket to become buoyant, it then rises and shuts the outlet. In (iii) the trap remains shut until the steam in the bucket has condensed or bubbled through the vent hole to the top of the trap body. It will then sink, pulling the main valve off its seat. Accumulated condensate is released and the cycle is repeated.

In (ii), air reaching the trap at start-up will also give the bucket buoyancy and close the valve. The bucket vent hole is essential to allow air to escape into the top of the trap for eventual discharge through the main valve seat. The hole, and the pressure differential, are small so the trap is relatively slow at venting air. At the same time it must pass (and therefore waste) a certain amount of steam for the trap to operate once the air has cleared. A parallel air vent fitted outside the trap will reduce start-up times.

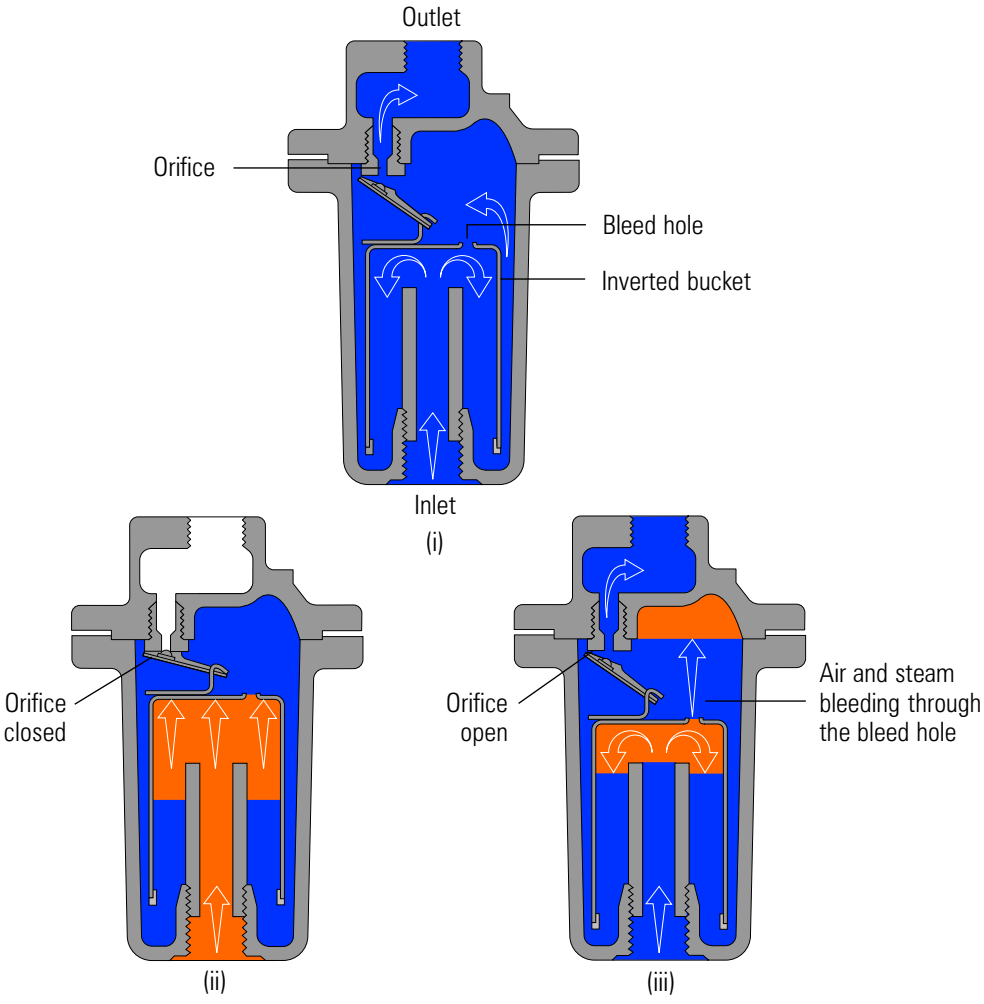


Fig. 11.3.3 Operation of an inverted bucket steam trap

Advantages of the inverted bucket steam trap

- The inverted bucket steam trap can be made to withstand high pressures.
- Like a float-thermostatic steam trap, it has a good tolerance to waterhammer conditions.
- Can be used on superheated steam lines with the addition of a check valve on the inlet.
- Failure mode is usually open, so it's safer on those applications that require this feature, for example turbine drains.

Disadvantages of the inverted bucket steam trap

- The small size of the hole in the top of the bucket means that this type of trap can only discharge air very slowly. The hole cannot be enlarged, as steam would pass through too quickly during normal operation.
- There should always be enough water in the trap body to act as a seal around the lip of the bucket. If the trap loses this water seal, steam can be wasted through the outlet valve. This can often happen on applications where there is a sudden drop in steam pressure, causing some of the condensate in the trap body to 'flash' into steam. The bucket loses its buoyancy and sinks, allowing live steam to pass through the trap orifice. Only if sufficient condensate reaches the trap will the water seal form again, and prevent steam wastage.
- If an inverted bucket trap is used on an application where pressure fluctuation of the plant can be expected, a check valve should be fitted on the inlet line in front of the trap. Steam and water are free to flow in the direction indicated, while reverse flow is impossible as the check valve would be forced onto its seat.
- The higher temperature of superheated steam is likely to cause an inverted bucket trap to lose its water seal. A check valve in front of the trap should be regarded as essential under such conditions. Some inverted bucket traps are manufactured with an integral check valve as standard.
- The inverted bucket trap is likely to suffer damage from freezing if installed in an exposed position with sub-zero ambient conditions. As with other types of mechanical traps, suitable lagging can overcome this problem if conditions are not too severe. If ambient conditions well below zero are to be expected, then it may be prudent to consider a more robust type of trap to do the job. In the case of mains drainage, a thermodynamic trap would be the first choice.

Questions

1. Name one characteristic feature of mechanical steam traps

a| They pass condensate at steam temperature ☐
b| They operate by sensing condensate temperature ☐
c| They can be fitted into any position ☐
d| They are not effected by increasing backpressure ☐

2. Why is a float trap better at venting air than an inverted bucket trap?

a| A float can quickly adjust to the presence of air ☐
b| A float trap is fitted with an automatic air vent ☐
c| A float trap does not vent air better than a bucket trap ☐
d| The air vent orifice is adjustable on a float trap ☐

3. What added benefit does the automatic air vent offer to a float trap?

a| It stops the trap from freezing in cold weather ☐
b| The trap can be used on larger backpressures ☐
c| It significantly increases the cold start-up capacity of the trap ☐
d| The condensate orifice can be the same size for all pressure ranges ☐

4. What advantage does a bucket trap have over a float trap?

a| It is able to withstand waterhammer ☐
b| It can be used on higher pressures ☐
c| It can discharge air freely ☐
d| It cannot lose its water seal ☐

5. A heat exchanger is designed to operate without waterlogging of the steam space. What is the usual choice of trap for its drainage?

a| Thermostatic trap ☐
b| Inverted bucket trap ☐
c| Thermodynamic trap ☐
d| Float trap with thermostatic air vent ☐

6. Which is the best trap to use when steam locking can occur?

a| An inverted bucket trap with an internal check valve mechanism ☐
b| A balanced pressure steam trap ☐
c| A float trap with automatic air vent ☐
d| A float trap with steam lock release mechanism ☐

Answers

1: a, 2: b, 3: c, 4: b, 5: d, 6: d

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Module 11.4

Thermodynamic Steam Traps

Thermodynamic Steam Traps

Traditional thermodynamic steam trap

The thermodynamic trap is an extremely robust steam trap with a simple mode of operation. The trap operates by means of the dynamic effect of flash steam as it passes through the trap, as depicted in Figure 11.4.1. The only moving part is the disc above the flat face inside the control chamber or cap.

On start-up, incoming pressure raises the disc, and cool condensate plus air is immediately discharged from the inner ring, under the disc, and out through three peripheral outlets (only 2 shown, Figure 11.4.1, i).

Hot condensate flowing through the inlet passage into the chamber under the disc drops in pressure and releases flash steam moving at high velocity. This high velocity creates a low pressure area under the disc, drawing it towards its seat (Figure 11.4.1, ii).

At the same time, the flash steam pressure builds up inside the chamber above the disc, forcing it down against the incoming condensate until it seats on the inner and outer rings. At this point, the flash steam is trapped in the upper chamber, and the pressure above the disc equals the pressure being applied to the underside of the disc from the inner ring. However, the top of the disc is subject to a greater force than the underside, as it has a greater surface area.

Eventually the trapped pressure in the upper chamber falls as the flash steam condenses. The disc is raised by the now higher condensate pressure and the cycle repeats (Figure 11.4.1, iv).

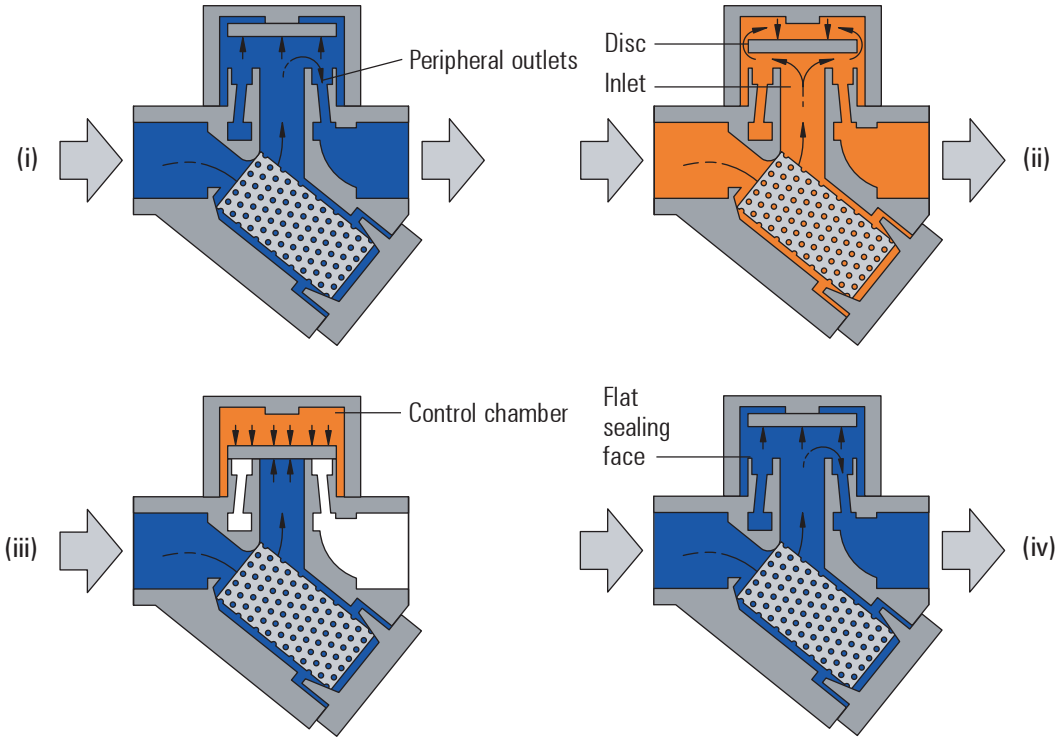


Fig. 11.4.1 Operation of a thermodynamic steam trap

The rate of operation depends on steam temperature and ambient conditions. Most traps will stay closed for between 20 and 40 seconds. If the trap opens too frequently, perhaps due to a cold, wet, and windy location, the rate of opening can be slowed by simply fitting an insulating cover onto the top of the trap.

Advantages of the thermodynamic steam trap

- Thermodynamic traps can operate across their entire working range without any adjustment or change of internals.
- They are compact, simple, lightweight and have a large condensate capacity for their size.
- Thermodynamic traps can be used on high pressure and superheated steam and are not affected by waterhammer or vibration. The all stainless steel construction offers a high degree of resistance to corrosive condensate.
- Thermodynamic traps are not damaged by freezing and are unlikely to freeze if installed with the disc in a vertical plane and discharging freely to atmosphere. However, operation in this position may result in wear of the disc edge.
- As the disc is the only moving part, maintenance can easily be carried out without removing the trap from the line.
- The audible 'click' which occurs as the trap opens and closes makes trap testing very straightforward.

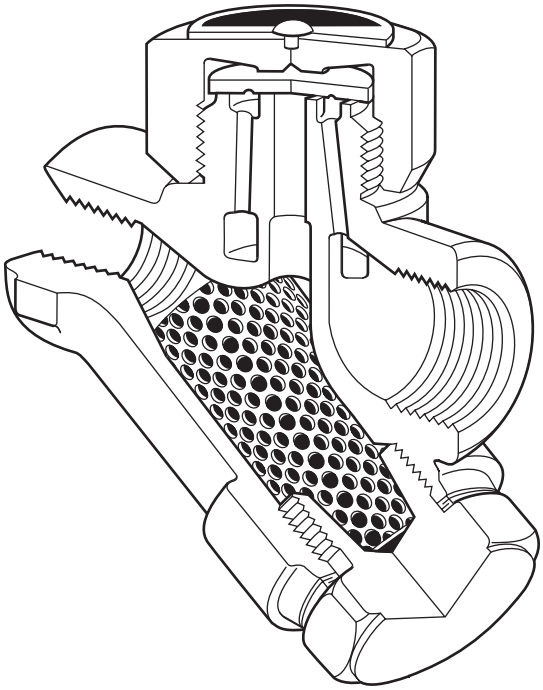


Fig. 11.4.2 Thermodynamic steam trap

Disadvantages of the thermodynamic steam trap

- Thermodynamic steam traps will not work positively on very low differential pressures, as the velocity of flow across the underside of the disc is insufficient for lower pressure to occur. They are subjected to a minimum inlet pressure (typically 0.25 bar g) but can withstand a maximum backpressure of 80% of the inlet pressure.
- Thermodynamic traps can discharge a large amount of air on 'start-up' if the inlet pressure builds up slowly. However, rapid pressure build-up will cause high velocity air to shut the trap in the same way as steam, and it will 'air-bind'. In this case a separate thermostatic air vent can be fitted in parallel with the trap. Modern thermodynamic steam traps can have an inbuilt anti-air-binding disc which prevents air pressure building up on top of the disc and allows air to escape, (Figure 11.4.3).
- The discharge of the trap can be noisy and this factor may prohibit the use of a thermodynamic trap in some locations, e.g. outside a hospital ward or operating theatre. If this is a problem, it can easily be fitted with a diffuser which considerably reduces the discharge noise.
- Care should be taken not to oversize a thermodynamic trap as this can increase cycle times and induce wear. Mains drainage applications often only need to be fitted with low capacity versions, providing proper consideration is given to siting the drain pockets correctly.

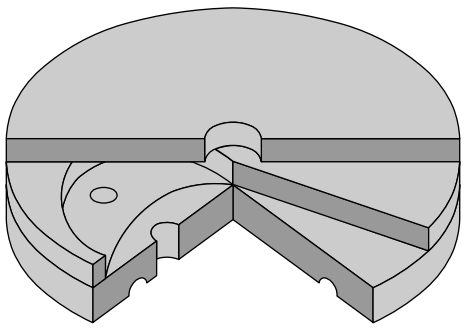


Fig. 11.4.3 Anti-air-binding disc

Impulse steam trap

The impulse trap (as shown in Figure 11.4.4) consists of a hollow piston (**A**) with a piston disc (**B**) working inside a tapered piston (**C**) which acts as a guide. At 'start-up' the main valve (**D**) rests on the seat (**E**) leaving a passage of flow through the clearance between piston and cylinder and hole (**F**) at the top of the piston. Increasing flow of air and condensate will act on the piston disc and lift the main valve off its seat to give increased flow. Some condensate will also flow through the gap between the piston and disc, through **E** and away to the trap outlet.

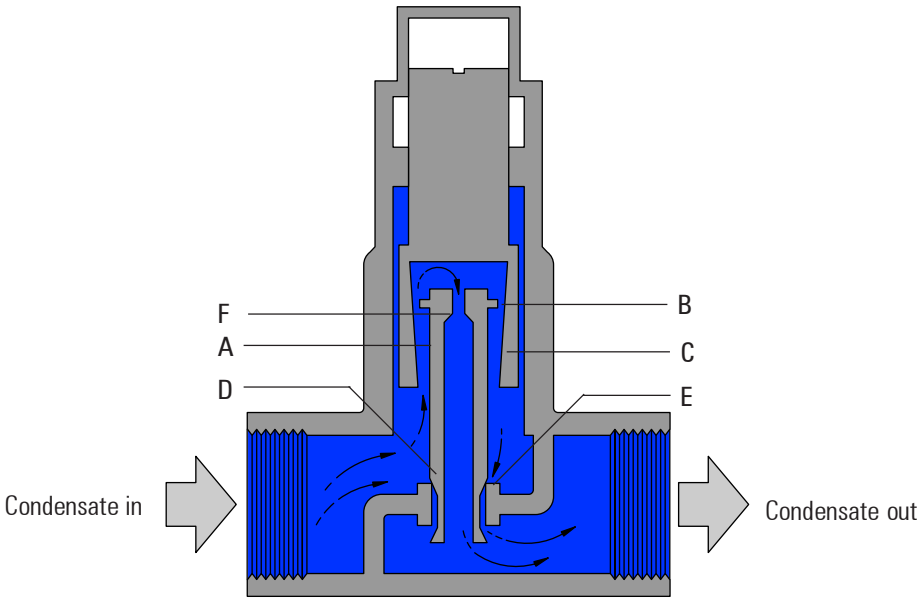


Fig. 11.4.4 Impulse steam trap

As the condensate approaches steam temperature some of it flashes to steam as it passes through the gap. Although this is bled away through hole **F** it does create an intermediate pressure over the piston, which effectively positions the main valve to meet the load. The trap can be adjusted by moving the position of piston (**B**) relative to the seat, but the trap is affected by significant backpressure. It has a substantial capacity, bearing in mind its small size. Conversely, the trap is unable to give complete shut-off and will pass steam on very light loads. The main problem however is the fine clearance between the piston and cylinder. This is readily affected by the dirt normally found in a steam system. The use of impulse traps is relatively limited so they are not considered in some subsequent sections of this Module.

Advantages of the impulse steam trap

- Impulse traps have a substantial condensate handling capacity for their size.
- They will work over a wide range of steam pressures without any change in valve size and can be used on high pressure and superheated steam.
- They are good at venting air and cannot 'air-bind'.

Disadvantages of the impulse steam trap

- Impulse traps cannot give a dead tight shut-off and will blow steam on very light loads.
- They are easily affected by any dirt which enters the trap body due to the extremely small clearance between the piston and the cylinder.
- The traps can pulsate on light load causing noise, waterhammer and even mechanical damage to the valve itself.
- They will not work against a backpressure which exceeds 40% of the inlet pressure.

Labyrinth steam trap

A simple form of the labyrinth trap is shown in Figure 11.4.5. It consists of a series of baffles which can be adjusted by means of a handwheel. Hot condensate passing between the first baffle and the trap body is subject to a drop in pressure and some of it 'flashes' to steam. The space around the next baffle has to cope with an increased volume of hot condensate and prevents the escape of live steam. The baffle plates can be moved either in or out using the handwheel, which alters their position relative to the body, effectively altering the overall size of the orifice.

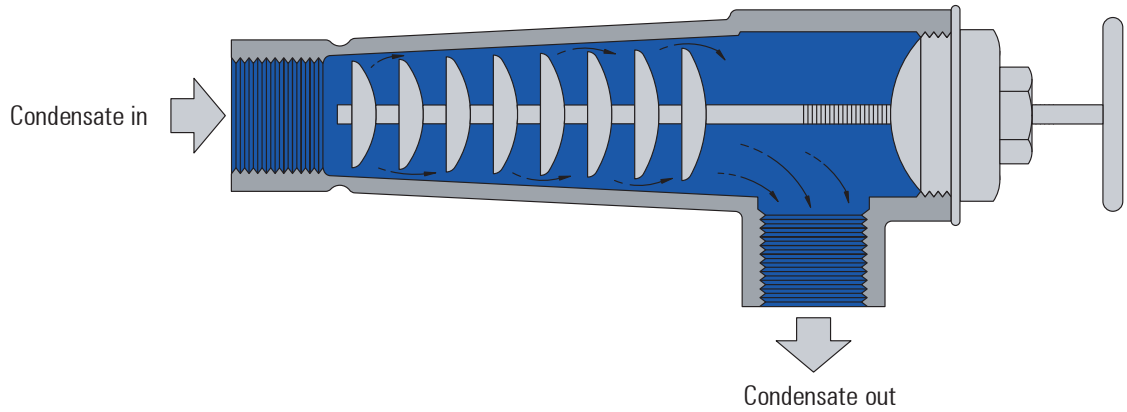


Fig. 11.4.5 Labyrinth steam trap

Advantages of the labyrinth steam trap

- This type of trap is comparatively small in relation to its capacity and there is little potential for mechanical failure since there are no automatic parts.

Disadvantages of the labyrinth steam trap

- The labyrinth trap has to be adjusted manually whenever there is a significant variation in either steam pressure or condensate load. If the setting is not right for the prevailing conditions, steam wastage or waterlogging of the steam space will occur (like a fixed orifice trap).

Fixed orifice traps

These are devices containing a hole of predetermined diameter to allow a calculated amount of condensate to flow under specific pressure conditions. In practice, condensate loads and steam pressures can vary considerably. For instance, start-up and running loads can differ considerably along with steam pressure which will change due to the actions of temperature controls. These varying conditions can result in the fixed orifice either holding back condensate in the process or passing live steam, which can affect plant performance and compromise safety.

Fixed orifices are often sized on running conditions, so that they hold back enough condensate and do not pass steam. If this is so, at start-up, they are undersized to a greater degree and the steam space stands a good chance of waterlogging.

The alternative is to size them so as not to waterlog during start-up. The hole is then effectively oversized for running conditions, and the device will pass steam. The size of hole is usually a compromise between the two conditions, such that, at some points in between, the hole is correctly sized.

Corrosion and service life of plant

Continual waterlogging significantly increases the risk of corrosion in the steam space. It is not unusual to find that after fitting fixed orifice traps, plant service life is reduced below that which may be expected with proper steam traps.

A proper steam trap should be able to achieve just sufficient capacity at all pressures and flowrates present in the application. It can then pass hot condensate without leaking steam under any condition. To achieve this, the size of the hole must vary in the trap. It must be large enough to meet the worst condition, and then have some means of reducing the effective orifice flow area when the capacity becomes too great. This exactly describes the operation of a steam trap.

Advantages of a fixed orifice trap

- Can be used successfully when pressures and loads are constant.
- There are no moving parts

Disadvantages of a fixed orifice trap

- If sized on running load, fixed orifice traps will waterlog on start-up, reducing plant performance over this period, increasing start-up times and the risk of corrosion.
- If sized on start-up load, fixed orifice traps will waste steam when the plant is running, effectively increasing running costs.
- Fixed orifice traps often block with dirt due to the small size of orifice.
- The cost of replacing a heat exchanger due to corrosion will be far higher than the cost of replacing the fixed orifice trap with a steam trap.

Note: Fixed orifice traps are not recommended for draining condensate from any application susceptible to varying load conditions.

Questions

1. Name one particular feature of thermodynamic steam traps?

a| They are difficult to maintain ☐
b| They operate by sensing condensate temperature ☐
c| They cannot be fitted upside-down ☐
d| They cannot be damaged by freezing ☐

2. At what typical application does a thermodynamic trap excel?

a| Draining any type of heat exchanger ☐
b| Draining steam mains ☐
c| Draining bulk oil storage tanks ☐
d| Draining any temperature controlled application ☐

3. What effect does air have on a thermodynamic trap?

a| None at all ☐
b| The trap can air bind at start-up unless fitted with a special disc ☐
c| The trap discharges condensate at a lower temperature ☐
d| The trap discharges at a higher temperature ☐

4. On what principle does a thermodynamic trap operate?

a| It senses the difference in density between water and steam ☐
b| It senses the difference in temperature between water and steam ☐
c| It operates on the difference in velocity between water and steam ☐
d| It operates with a fixed orifice position ☐

5. What is a disadvantage of any fixed orifice device?

a| They cannot shut off and can therefore waste steam ☐
b| They can waterlog and corrode the application steam space ☐
c| They regularly block up due to the size of the orifice ☐
d| All of the above ☐

6. What effect does a high backpressure have on a thermodynamic trap?

a| It will reduce the trap capacity ☐
b| It will increase the trap capacity ☐
c| It will cause the trap to air bind ☐
d| It will have no effect at all ☐

Answers

1: d, 2: b, 3: b, 4: c, 5: d, 6: a

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Module 11.5

Considerations for Selecting Steam Traps

Considerations for Selecting Steam Traps

Considerations

By definition, a steam trap must trap or hold back steam whilst at the same time not restricting the passage of condensate, air, and other incondensable gases. The basic requirements of good steam trapping have already been outlined but it is worth repeating that the performance of the plant is paramount. The trap selection follows on the basis that the requirements of pressure, condensate load and air venting have been met, in the provisional selection. However, system design and maintenance needs will also influence performance and selection. Please refer to the following sub-sections in this Module for further advice on this matter.

Waterhammer

Waterhammer is a symptom of a problem in the steam system. This could be due to poor design of the steam and condensate pipework, the use of the wrong type of trap or traps or a leaking steam trap, or a combination of these factors. It is often futile to install the correct trap for an application if the system layout will not allow the trap to operate correctly. It is equally pointless to install the correct layout and not pay proper attention to steam trapping. The Modules 11.6 to 11.11 inclusive 'Selecting steam traps' will deal with the correct matching of steam traps to applications and layouts. The proper layout of steam pipework is also dealt with in Block 10 - 'Steam Distribution'. Symptoms of waterhammer are often attributed to malfunction of the steam trap. A more likely explanation is that a faulty steam trap has been damaged by waterhammer. Waterhammer can be caused in a number of ways, including:-

- Failure to remove condensate from the path of high velocity steam in the pipework.
- From an application which is temperature controlled and where condensate has to lift to a return line, or return to a pressurised system.
- The inability of condensate to properly enter or travel along an undersized return line, due to either (a) flooding, or (b) overpressurisation with the throttling effects of flash steam.

Modern design and manufacturing techniques have produced steam traps which are more robust than those of their predecessors. This allows the steam trap to last longer under normal conditions, and will also be better able to withstand the effects of poorly designed systems. Basically, however well a steam trap is made, if it is installed in a poorly designed system it will be less effective and have a shorter working life.

If a steam trap persistently fails on an established system due to waterhammer, it is probably the fault of the system layout, rather than the trap. The solution is to investigate and eradicate the true cause of the problem by correcting the system inadequacies.

Two important applications are the drainage of steam mains, and of temperature controlled heat exchangers.

As a general rule, steam mains should be drained at regular intervals of 30 to 50 metres with adequately sized drain pockets. The bottom of any riser must also be drained.

Temperature controlled heat exchangers can only work effectively if condensate is allowed to drain freely from them. If there is a lift after the trap, there will always be a tendency for waterhammer, whichever trap is fitted. In this situation, the trap should either be complemented with a pump, or changed for a punp-trap . This subject will be dealt with in further detail in Block 13 - 'Condensate Removal'

It is important that the pipework is designed and installed correctly. This will help to maintain thermal performance of the system throughout its service life.

Dirt

Dirt is another major factor which must be considered when selecting traps. Although steam condenses to distilled water, it can sometimes contain trace products of boiler feed treatment compound and natural minerals found in water. Pipe dirt created during installation and the products of corrosion also need to be considered.

An intermittent blast action trap is the least likely to be affected by dirt. In thermostatic traps this means that the balanced pressure thermostatic trap is preferable, although the larger flat valve associated with some diaphragm traps can cause difficulties.

The dribbling action of bimetallic traps, coupled with the arrangement of the valve stem passing through the seat, means that these are most prone to malfunction (due to added friction) or even to blockage. It is sometimes claimed that the sensor element can be readily cleaned and is not subject to fouling. However, fouling of the element is rarely a problem: the relevant parts are the 'dynamic clack' valve mechanism, which tends to be self-cleaning due to its positive opening action.

Float-thermostatic steam traps are quite resistant to dirt. As an extreme example, when draining concrete curing autoclaves, the residual sand which precipitates into the condensate can be carried through large float-thermostatic steam traps quite successfully, due to the low velocity flow through a relatively large orifice.

The inverted bucket trap has an air vent hole in the bucket. If this blocks, it can cause the trap to air-bind and be slow to react. If this happens, the scale or dirt blocking the air vent must be dislodged, which requires the trap to be removed from service.

The impulse trap is intolerant of dirty conditions. The fine clearance between plug and tapered sleeve is susceptible to high velocity flow and the plug will frequently stick in an intermediate position. The trap seizes in a fixed position and will either pass steam or condensate depending on the rate of condensation.

The fixed orifice device is least suited to dirty conditions. The hole is inherently small and frequently blocks. Enlarging the hole (as is sometimes done in desperation) destroys the concept of sizing on a fixed orifice. It is wasteful and in some cases merely delays the time until blockage re-occurs. A strainer is often supplied and fitted but this has to be extremely fine to be effective.

This simply transfers the blockage from the orifice trap to the strainer, which, in turn, requires regular downtime for cleaning.

Strainers

These devices (Figure 11.5.1) are frequently forgotten about in steam systems, often, it seems, in an effort to reduce installation costs. Pipe scale and dirt can affect control valves and steam traps, and reduce heat transfer rates. It is extremely easy and inexpensive to fit a strainer in a pipe, and the low cost of doing so will pay dividends throughout the life of the installation. Scale and dirt are arrested, and maintenance is usually reduced as a result.

Selection is simple. The strainer material is selected to match the type of installation and the system pressure up to which it is expected to operate. Different filter screen sizes may be considered for differing degrees of protection. The finer the filter, the more often it may need cleaning. One thing is certain, strainers are far easier and cheaper to buy and maintain than control valves or steam traps.

Further information on strainers is given in Block 12 - 'Pipeline Ancillaries'

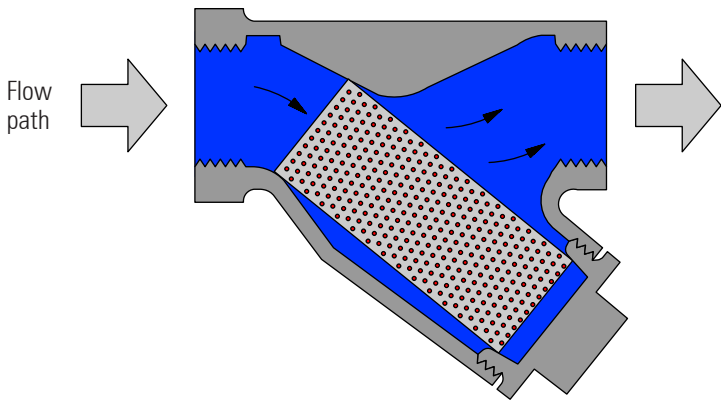


Fig. 11.5.1 Typical Y-type strainer (cut section)

Steam locking

The possibility of steam locking can sometimes be a deciding factor in the selection of steam traps. It can occur whenever a steam trap is fitted remotely from the plant being drained. It can become acute when condensate is removed through a syphon or dip pipe. Figure 11.5.2 illustrates the problem of steam locking in a rotating drying cylinder by using a syphon pipe.

In Figure 11.5.2 (i) the steam pressure is sufficient to lift condensate up the syphon pipe, through the steam trap and away. Figure 11.5.2 (ii) shows what happens when the level of the condensate at the bottom of the cylinder falls below the end of the syphon pipe. Steam enters the syphon pipe and causes the steam trap (in this case a float type) to close.

The trap is temporarily 'steam locked'. Heat loss from the cylinder will result in the formation of more condensate which, as a result, is unable to reach the trap. Figure 11.5.2 (iii) shows the cylinder becoming increasingly waterlogged which will result in a reduced drying rate from the cylinder and an increase in the power required to turn the cylinder. In extreme cases the cylinder may fill to the centre line and damage may then result from mechanical overload.

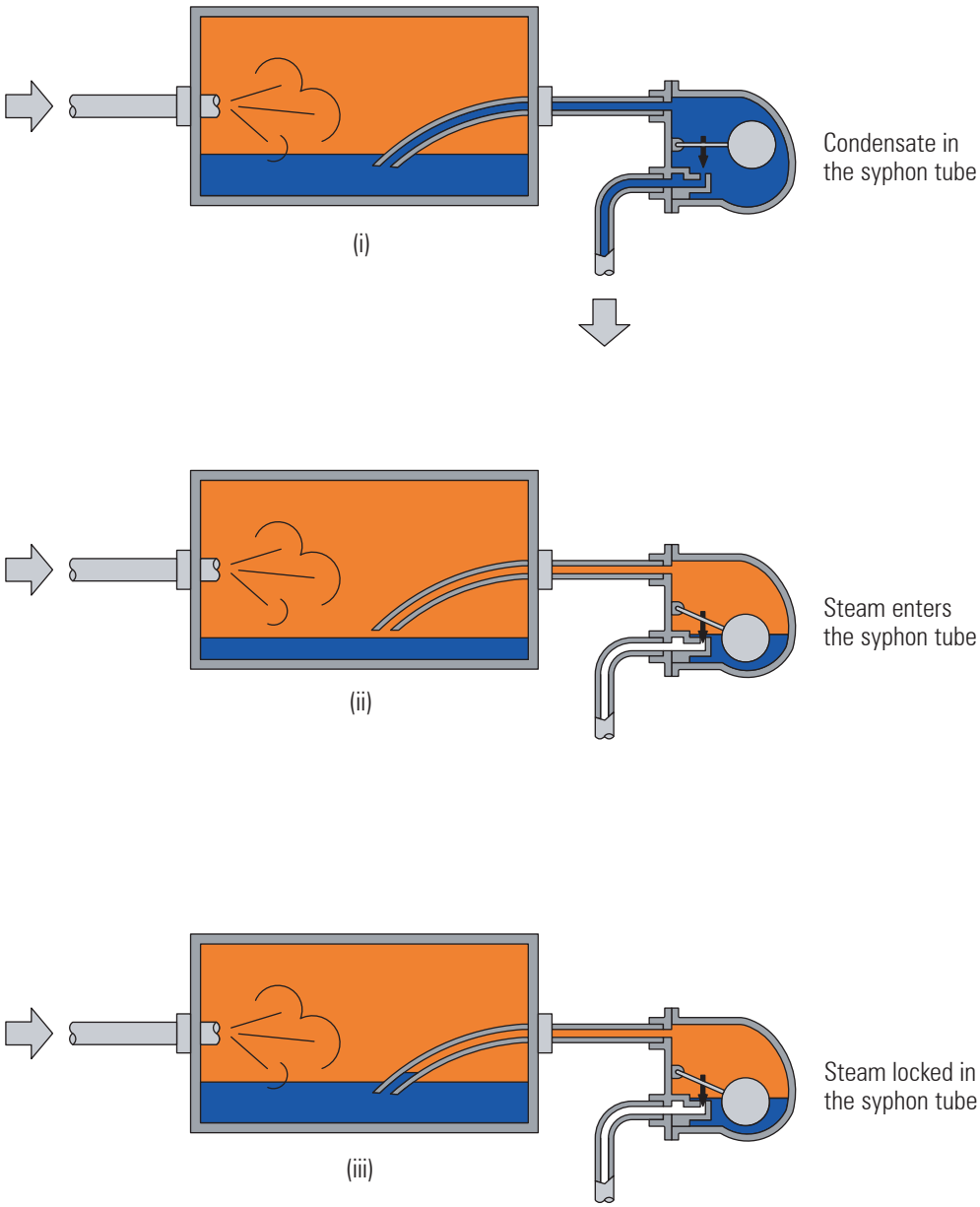


Fig. 11.5.2 Steam locking

To relieve this problem a trap is needed with a 'steam lock release' valve. This is an internal needle valve which allows the steam locked in the syphon pipe to be bled away past the main valve. The float trap is the only type of trap with this facility and is the correct choice on rotating machinery such as drying cylinders. Because the needle valve is just open enough to avoid steam wastage it has a limited capacity to vent air. Traps of this type are often provided with combined air vents and steam lock release (Figure 11.5.3). The manually operated steam lock release mechanism works independently of the automatic air vent action. A standard float-thermostatic steam trap is shown in Figure 11.5.4.

Other types of traps will open and eventually cope with a steam lock, however, the drainage and plant performance will be erratic. This is clearly unacceptable to users of process plant where batch times, quality and efficiency are of high importance.

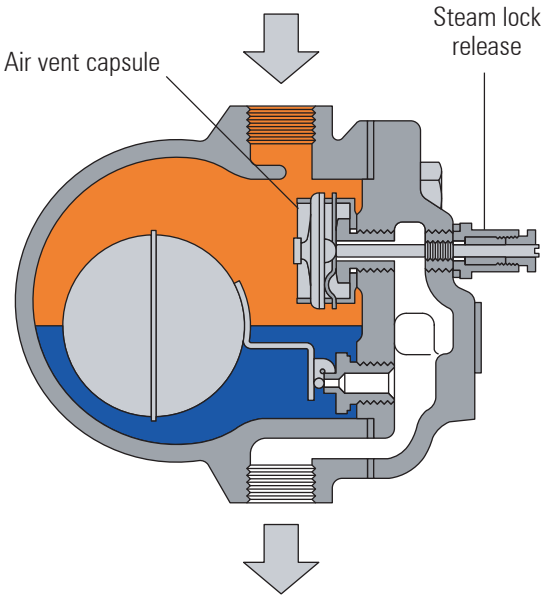


Fig. 11.5.3 Float-thermostatic trap with combined steam lock release valve

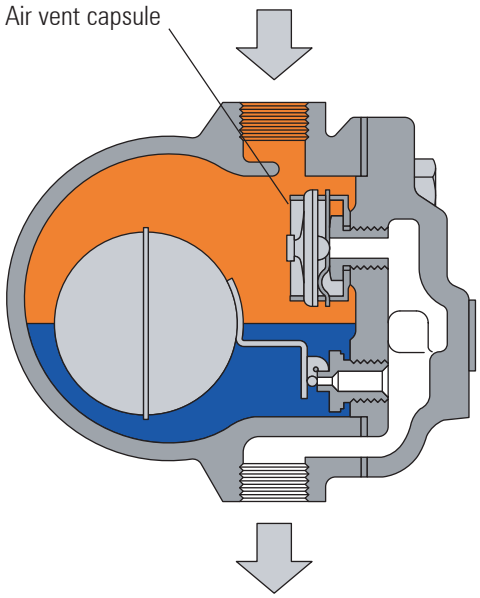


Fig. 11.5.4 Standard float-thermostatic trap

Group trapping

Group trapping describes the use of one trap serving more than one application. Figure 11.5.5 shows two batch processes (jacketed pans) operating at two different steam pressures with the drain line from each connected to one steam trap. The higher pressure in plant **B** will allow condensate from this vessel to drain but will stop condensate being discharged from plant **A** as check valve **C** will be held closed. Plant **A** will waterlog and will suffer a severe drop in performance.

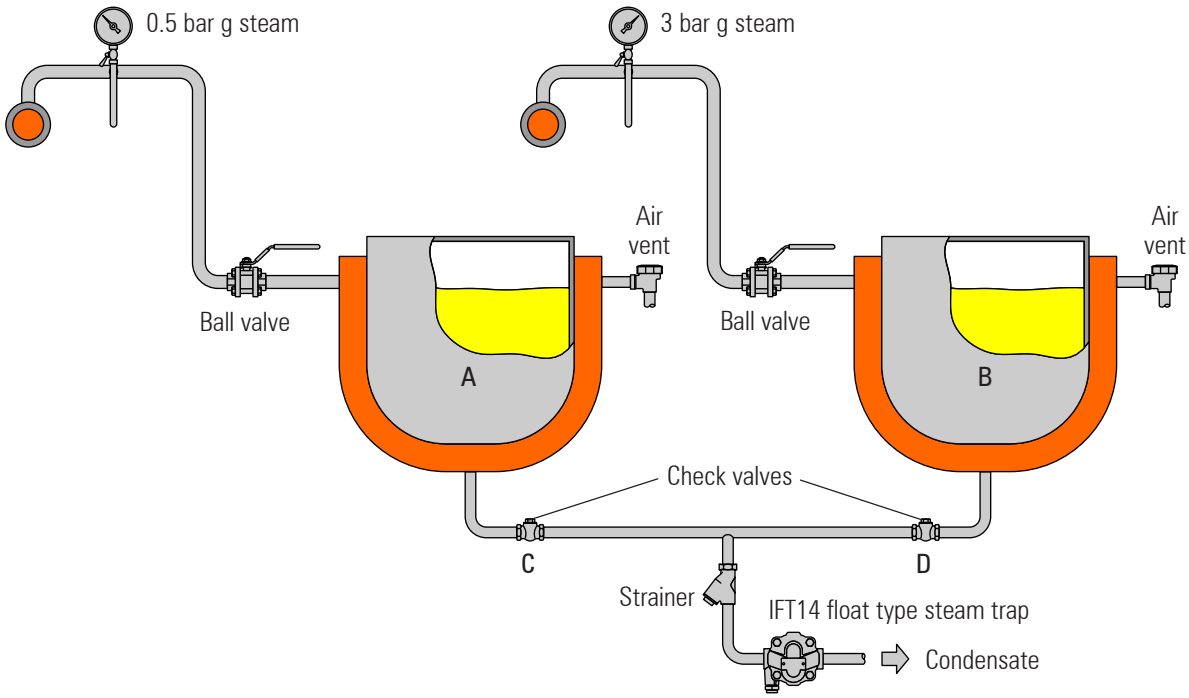


Fig. 11.5.5 Group trapping with different process pressures

For this reason, group trapping of equipment operating at different pressures is not good practice. But what if equipment operates at the same pressure? Consider the following installation shown in Figure 11.5.6.

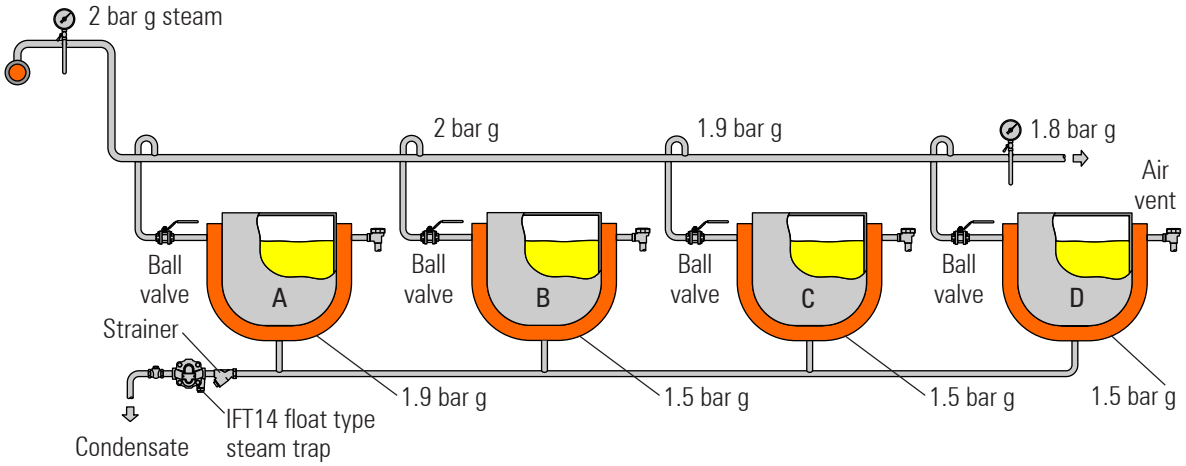


Fig. 11.5.6 Group trapping with same process pressures

In Figure 11.5.6, the content of pan **A** is almost up to temperature and is condensing relatively little steam. Pans **B**, **C** and **D** have just been filled with cold product and, as the steam is turned on, their condensation rates are much higher than pan **A**. Consequently, the steam velocity along these supply pipes is much higher, resulting in a higher pressure drop along each of the branch lines. Lower steam pressures will exist at the pan inlets of **B**, **C** and **D** and in their steam jackets, (due to **B**, **C** and **D** having a higher condensing rate than pan **A**) reducing their heating ability and increasing their production times.

Because of this, the pressures at the drain outlets of pans **B**, **C** and **D** are also lower than that at pan **A**. Steam will flow from pan **A** via the condensate drain line to the other pans to equalise the pressures, and the condensate from the other pans will have to flow against this steam flow. When the drainage points of different vessels at different pressures are connected to one trap, the vessel with the highest pressure (in this instance pan **A**) will impede the flow of condensate from the others. Those vessels with the greatest need to discharge condensate (at this instance pans **B**, **C** and **D**) will tend to waterlog. Hence, the condensate arrangement shown in Figure 11.5.6 is unlikely to be satisfactory. The situation can be aggravated when group-trapped processes have separate temperature control.

One possible application suitable for group trapping is an air handling unit with multiple heater sections in series (Figure 11.5.7).

This 'flow' type application differs from the batch (or non-flow) process in Figure 11.5.6. The heater sections will always share any load change as they are served by the same control valve. It is important that the condensate drain connections and common pipework are generously sized to allow adequate condensate flow in one direction against steam flow in the other. It will only work where all sections are fed by one control valve and the same secondary fluid is being heated by all sections.

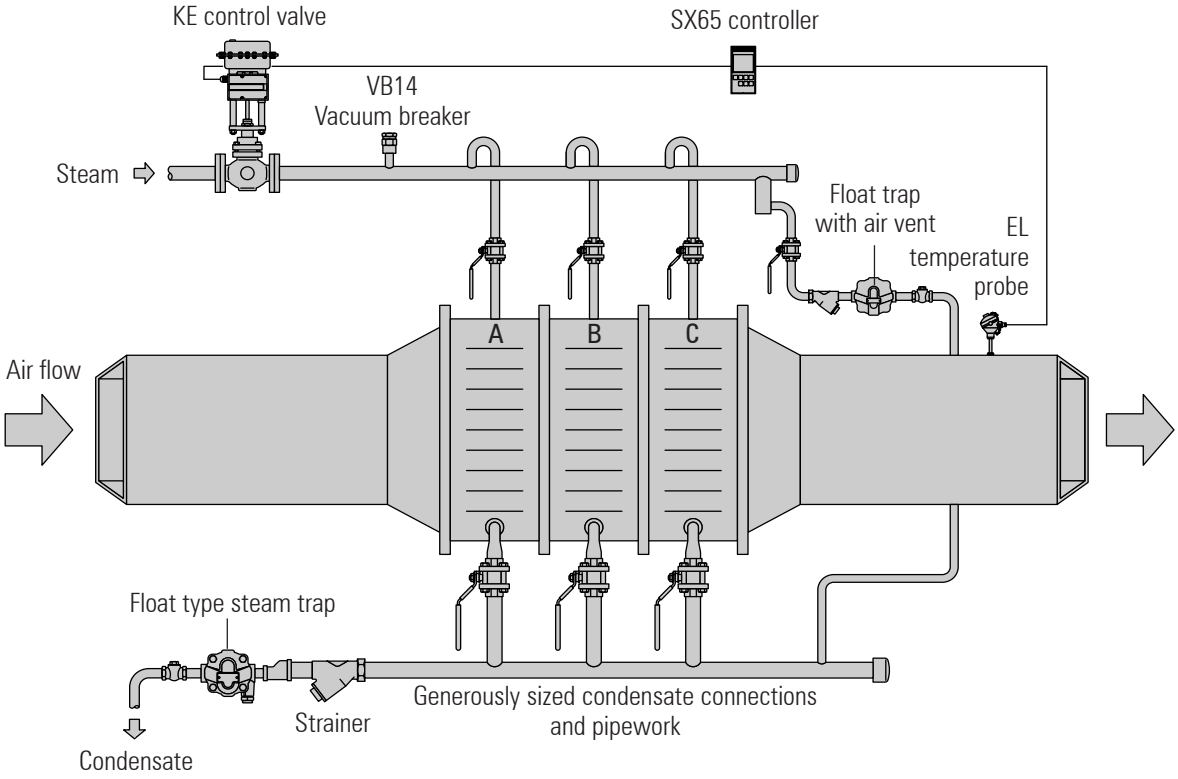


Fig. 11.5.7 Three section air handling unit with one control valve

The original reason for group trapping was that there used to be only one kind of steam trap. It was the forerunner of the present day bucket trap, and was very large and expensive. Steam traps today are considerably smaller and cost effective, allowing individual heat exchangers to be properly drained. It is always better for steam using equipment to be trapped on an individual basis rather than on a group basis.

In many instances it may be necessary to use a pump-trap on temperature controlled equipment, to remove condensate properly.

Diffusers

With steam traps draining to atmosphere from open ended pipes, it is possible to see the discharge of hot condensate. A certain amount of flash steam will also be present relative to the condensate pressure before the trap. This can present a hazard to passers by, but the risks can be minimised by reducing the severity of the discharge. This may be achieved by fitting a simple diffuser (Figure 11.5.8) to the end of the pipe (Figure 11.5.9) which reduces the ferocity of discharge and sound. Typically, sound levels can be reduced by up to 80%.

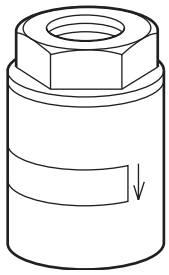


Fig. 11.5.8 Diffuser

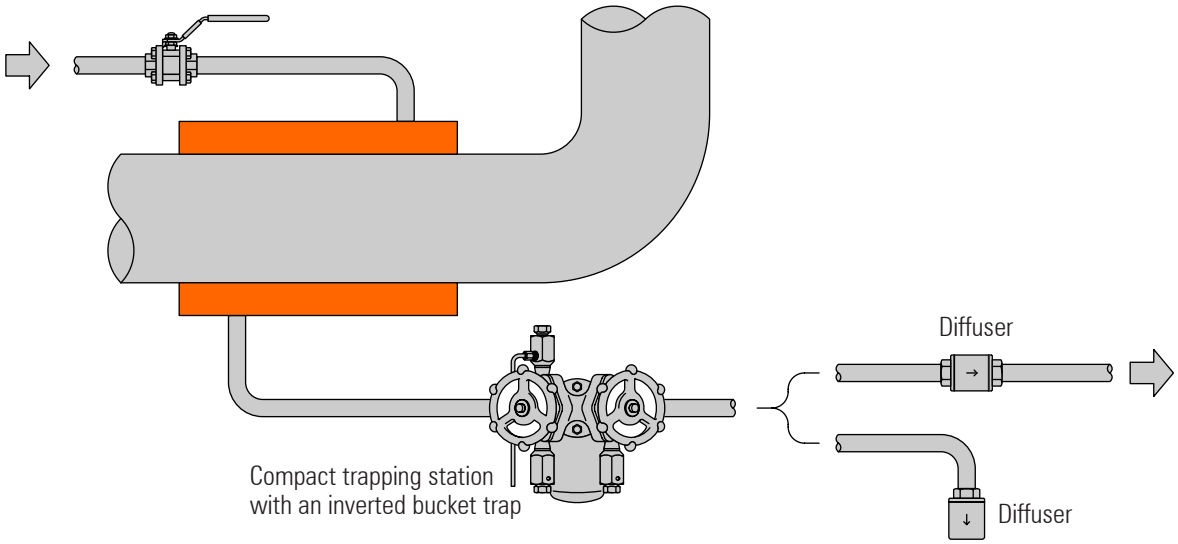


Fig. 11.5.9 Steam tracer line

Special requirements

Vacuum drainage

Condensate removal from a steam space working under vacuum can be a problem. If a steam trap is used, its outlet must be connected to a source of greater vacuum than that in the steam space to ensure a constant differential pressure across the orifice to discharge the condensate. Where this is not possible, a pressure powered pump can be used to drain condensate from the plant (Figures 11.5.10 and 11.5.11).

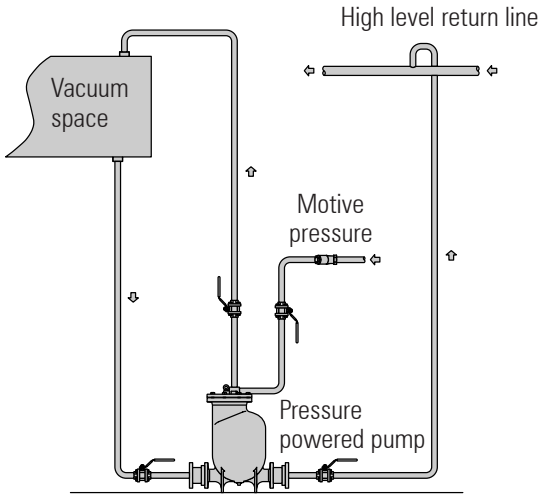


Fig. 11.5.10 Pump draining vacuum system to a high level return line

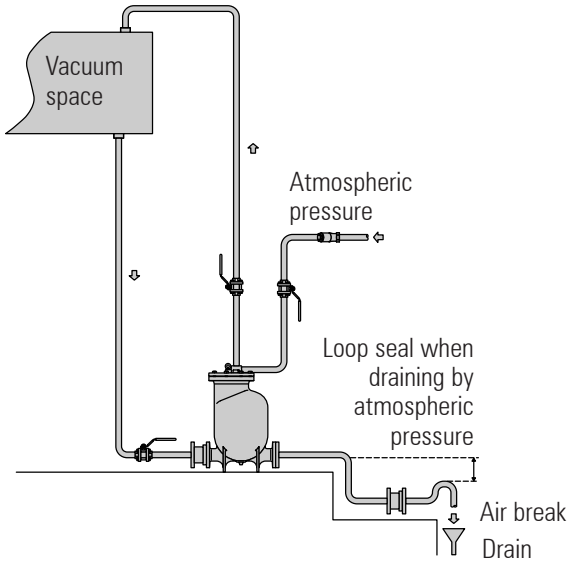


Fig. 11.5.11 Pump draining vacuum system to a low level drain

A soft seated check valve is recommended on the pump outlet where little or no lift is present, and an air break will act as an anti-syphoning device when draining to a point below the pump.

Atmospheric pressure can be used as the motive force when draining below the pump (Figure 11.5.11), but the outlet check valve should be positioned in a loop seal below the pump to induce a minimum opening head (dependant on the type of check valve) and water seal.

Should the pump be draining condensate from a vacuum gas system then compressed air or inert gas can be used as the motive force to drive the pump.

Steam trap drainage of temperature controlled processes

The steam trap is an automatic valve that relies on the system dynamics to provide flow. It has to rely on and react to external factors, such as steam pressure or static head pressure on the inlet side of the trap. The outlet pressure must be lower than the inlet pressure to provide flow in the correct direction. The rate of flow through any steam trap is therefore related to the differential pressure across it.

It is also possible to have negative differential pressures across the trap, which would promote reverse flow through it. When traps are installed to pass condensate into common return lines, it is advisable to fit non-return valves after each trap to prevent reverse flow under negative pressure conditions.

The occurrence of zero and negative differential pressure across steam traps is commonplace. The effects are commonly seen with temperature controlled processes i.e. heater batteries, calorifiers, jacketed pans, plate heat exchangers, in fact any process that has a control valve on the steam supply. It can occur irrespective of steam supply pressure, and depends wholly on the condensate system pressure and the steam pressure in the heat exchanger.

The term 'stall' describes this condition. Whenever it is predicted or diagnosed, another solution, such as a pump-trap is required to remove the condensate from the heat exchanger.

The phenomenon is discussed in greater detail in Block 13 - 'Condensate removal'.

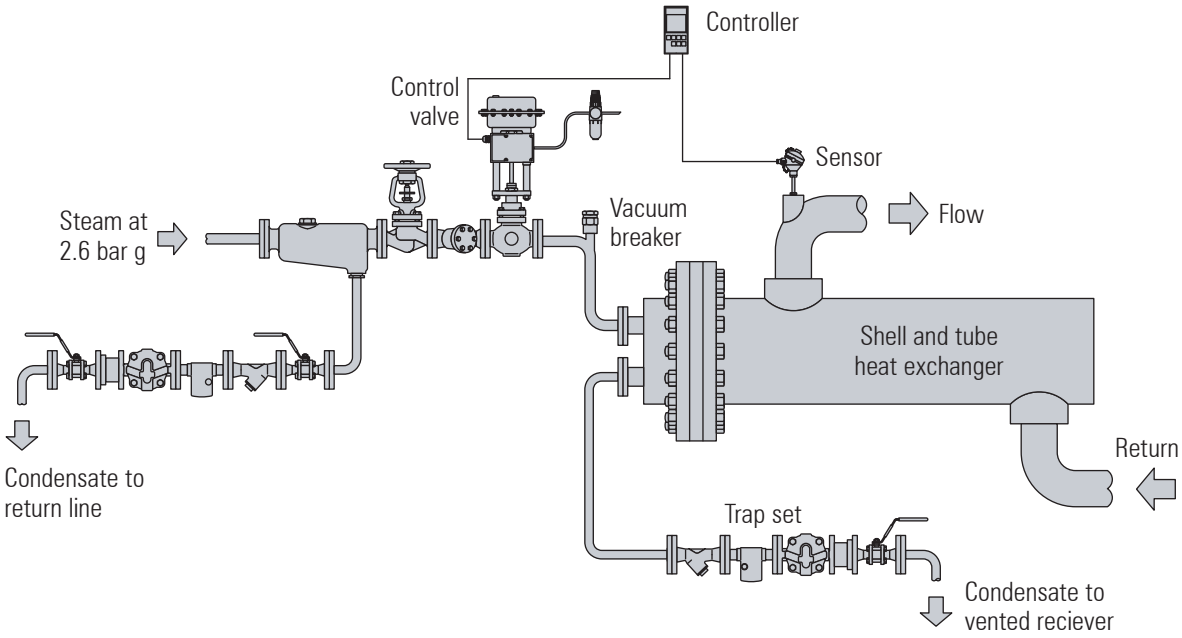


Fig. 11.5.12 Typical temperature controlled process

Questions

1. Name the principle cause of waterhammer:

a | Water particles suspended in steam ☐
b | Water allowed to build up in pipes ☐
c | Water droplets carried along the insides of pipes ☐
d | Wet steam passing through steam traps ☐

2. What effect does dirt have on steam systems?

a | It clogs up control valves ☐
b | It clogs up steam traps ☐
c | It reduces heat transfer performance ☐
d | All of the above ☐

3. What effect does steam locking have on rotating machinery?

a | None at all ☐
b | It reduces the drying rate of drying cylinders ☐
c | It increases the drying rate of drying cylinders ☐
d | It causes the steam trap to air bind ☐

4. When can group trapping be used with success?

a | For multiple batch processes fed by the same steam pressure ☐
b | For multiple batch processes fed by different steam pressures ☐
c | For multiple air heater batteries fed by the same control valve ☐
d | For heater batteries generally ☐

5. What is the best method of draining a vacuum main?

a | A thermodynamic steam trap ☐
b | A check valve fitted in reverse ☐
c | A float trap fitted in conjunction with a check valve ☐
d | A pressure powered pump ☐

6. Name one method of reducing the effect of stall in a temperature controlled application:

a | Increase the size of the steam trap ☐
b | Remove the steam trap altogether ☐
c | Install a pump-trap ☐
d | Increase the steam pressure onto the control valve ☐

Answers

1: b, 2: d, 3: b, 4: c, 5: d, 6: c

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Module 11.6

Selecting Steam Traps - Canteen Equipment; Oil Transfer/Storage; Hospital Equipment

Selecting Steam Traps

Key:

A - Best choice.
B - Acceptable alternative.

1 - With air vent in parallel.
2 - At end of unlagged cooling leg. Minimum length 1 m.
3 - Use special tracing traps which offer fixed temperature discharge option.
4 - If the equipment is temperature controlled, a condensate pump and trap combination may be required.
5 - With close to steam temperature capsule.
6 - Fitted with anti-air-binding disc.

Application

Steam trap:	FT range (float-thermostatic)	FT-C (float-thermostatic with steam release)	TD range (Thermodynamic)	BPT (Balanced pressure thermostatic)	SM (Bimetallic)	No.8 (Liquid expansion)	IB range (Inverted bucket)
Canteen equipment							
Boiling pans - tilting		B		**A**[2, 5]			
Boiling pans - fixed	**A**	B	B[1]	B			
Boiling pans - pedestal	B			**A**[2, 5]			
Steaming ovens				**A**[2, 5]			
Hot plates	B			**A**[2, 5]			
Oil transfer/storage							
Bulk oil storage tanks	**A**						B[1]
Line heaters	**A**						B[1]
Outflow heaters	**A**						B[1]
Tracer lines			B	**A**	B[2] (non-critical only)		B
Jacketed pipes			B[1, 6]	**A**[5]			B[1]
Hospital equipment							
Autoclaves and sterilisers	B	B		**A**[5]			
Industrial dryers							
Hot air dryers	**A**		B[1]	B			
Drying coils			B[1]	**A**			B[1]
Multi-bank pipe dryers	**A**		B[1]	B			B[1]
Drying cylinders	B	**A**					B[1]
Multi-cylinder sizing machines	B	**A**					B[1]
Laundry equipment							
Garment presses	B	B	**A**[6]				
Ironers and calenders	B	**A**	B[1]	B[5]			
Tumbler dryers	**A**	B					
Dry cleaning machines	**A**		B[1]				

Steam trap:	FT range (float-thermostatic)	FT-C (float-thermostatic with steam release)	TD range (Thermodynamic)	BPT (Balanced pressure thermostatic)	SM (Bimetallic)	No.8 (Liquid expansion)	IB range (Inverted bucket)
Presses							
Multi-platen presses (parallel connections)	B		**A**[6]				
Multi-platen presses (series connections)			**A**[1, 6]				
Tyre presses	B		B[1]	**A**			B[1]
Process equipment							
Boiling pans - fixed	**A**	B	B[1]	B			
Boiling pans - tilting	B	**A**					
Retorts	**A**						
Industrial autoclaves	**A**						B[1]
Digesters	**A**[1]		B[1]				
Hot tables	B		B[6]	**A**[2]			
Brewing coppers	**A**[1]	B					
Evaporators, calandrias	**A**[1]	B					B[1]
Vulcanisers	**A**		B[1] (jacket only)				B[1]
Space heating equipment							
Calorifiers	**A**[4]						
Heater batteries	**A**[4]						
Radiant panels and strips	**A**	B[1]	B[1]				B[1]
Radiators and convection cabinets	B			**A**	B		
Unit heaters and air batteries	**A**[4]						
Overhead pipe coils	B			**A**			B[1]
Steam mains							
Pressure reducing valve station	**A**			B[5]			
Horizontal runs	B		**A**				B
Shutdown drain (frost protection)				B[3]	B	**A**	
Separators	**A**		B				B
Steam header drainage	**A**		B[6]				B
Terminal ends	B		**A**[1]				B[1]
Tanks and vats							
Process vats (rising discharge pipe)	B	B	**A**	B[5]			B
Process vats (discharge pipe at base)	**A**		B[6]	B[5]			
Small coil heated tanks (quick boiling)	**A**			B[5]			
Small coil heated tanks (slow boiling)					B	**A**	

Canteen Equipment

A - Best choice, B - Acceptable alternative, [1] (parallel air vent), [2] (with 1 m cooling leg), [5] ('near-to-steam' capsule).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Boiling pans - tilting		B		**A** [2, 5]			
Boiling pans - fixed	**A**	B	B[1]	B			
Boiling pans - pedestal	B			**A** [2, 5]			
Steaming ovens				**A** [2, 5]			
Hotplates	B			**A** [2, 5]			

Canteen boiling pans

Although similar in construction to process jacketed pans, canteen boiling pans do not normally have the same need for rapid heating, consequently the steam pressure is normally lower. Condensate loads will normally be much lower. Whilst air and condensate removal are not so critical, air vents can still be useful in reducing heat-up times.

Tilting boiling pans

Figure 11.6.1 shows a balanced pressure thermostatic trap, draining a slow boiling tilting pan. A balanced pressure air vent (fitted as shown) will speed up the boil of, for example, 140 litres of soup by about 20 minutes. If faster cooking would be an advantage, an air vent should be fitted. A good alternative to the balanced pressure steam trap is a float trap with steam lock release.

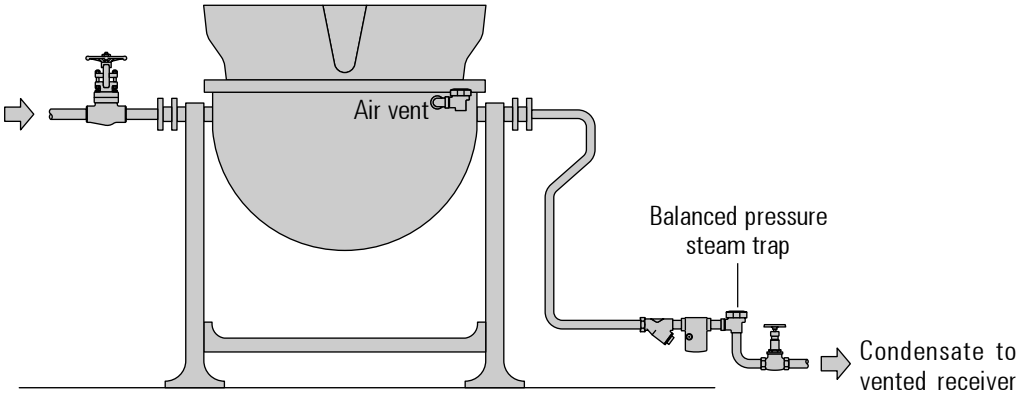


Fig. 11.6.1 Slow boiling tilting pan

Pedestal boiling pans

The correct way to drain pedestal boiling pans is to use a balanced pressure thermostatic trap and strainer. For efficient operation this should be fitted about 1 m away from the outlet at the end of the cooling leg (Figure 11.6.2). There is usually no need to fit an air vent on this type of pan.

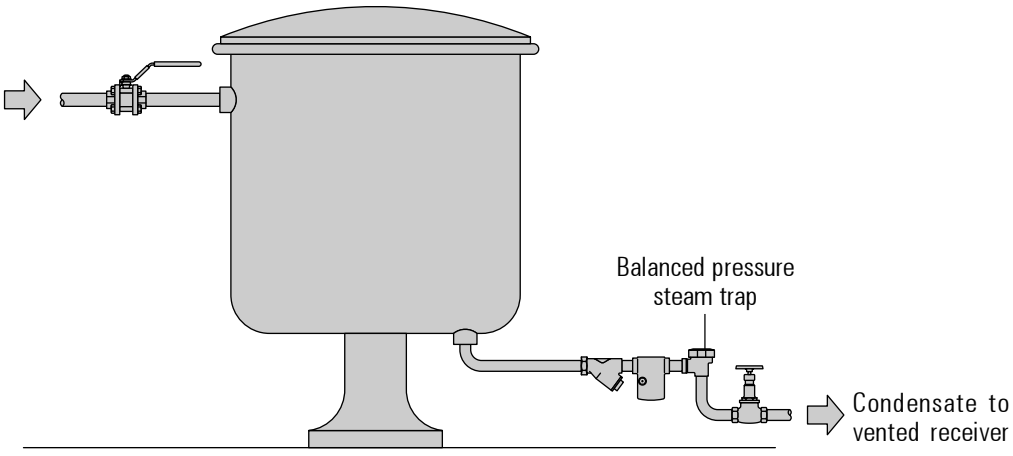


Fig. 11.6.2 Pedestal pan

Steaming ovens and hotplates

Figure 11.6.3 shows an ideal layout for draining and air venting steaming ovens. There are three vital features:

- The steam inlet must be drained just before the inlet valve by a balanced pressure thermostatic trap.
- Each compartment outlet must have a similar trap direct on to the outlet, but without a strainer (to let the greasy condensate pass through before the grease cools).
- The traps draining the compartments, and the air vents, should be fitted with near-to-steam elements. The ovens should be blown through with steam after cooking has finished.

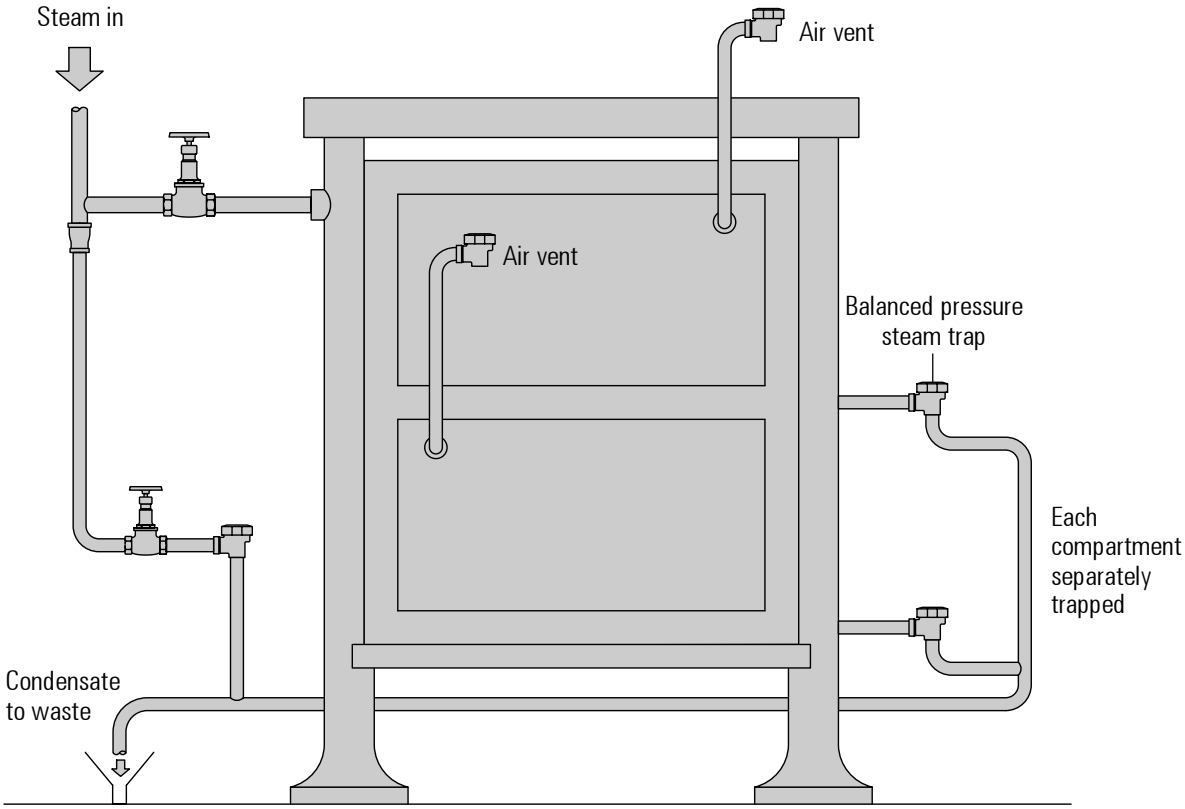


Fig. 11.6.3 Direct steaming oven

Figure 11.6.4 shows a kitchen hotplate fitted with a Fig 5 type strainer, close coupled to a balanced pressure thermostatic steam trap, an ideal combination for this application.

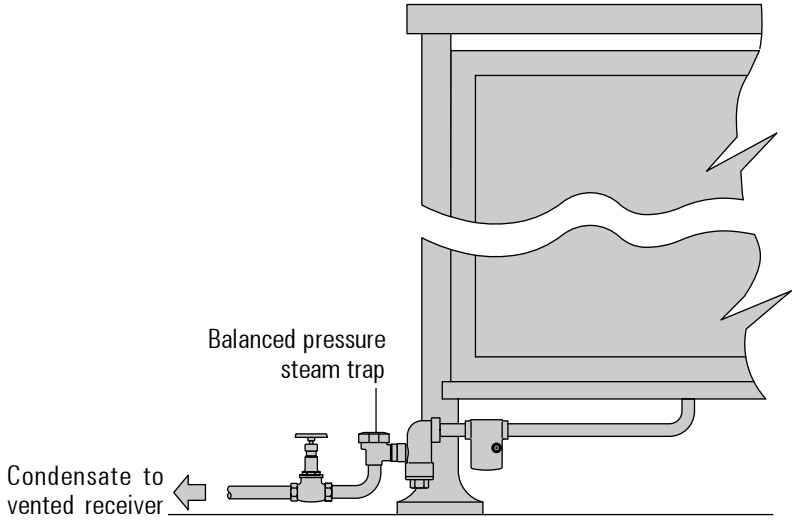


Fig. 11.6.4 Kitchen hotplate

Oil Transfer/Storage

A - Best choice, B - Acceptable alternative,
[1] (parallel air vent), [2] (with 1 m cooling leg), [5] ('near-to-steam' capsule), [6] (anti-air-binding disc).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Bulk storage tanks	**A**						B[1]
Line heaters	**A**						B[1]
Outflow heaters	**A**						B[1]
Tracer lines			B	**A**	B[2] (non-critical only)		B
Jacketed pipes			B[1, 6]	**A**[5]			B[1]

Bulk storage tanks

Oil and other fluids are stored in tanks that are heated by pipe coils or other forms of heating, either alone, or in combination with outflow heaters, to provide the correct temperature for pumping. Line heaters raise the temperature of fuel oil to that required for burning or for process use.

There are several ways to heat small to medium sized storage tanks, such as with pipe coils (Figure 11.6.5) spread across the bottom of the tank, or with 'bayonet' or 'field' heaters (Figure 11.6.6). In these situations a large pipe, sealed at both ends, is fixed through the side of the tank. Steam is fed to the remote end by an internal pipe and condensate is removed from the nearest end. However, on larger tanks, one of the more widely used methods is the fitting of a number of special heaters served from an internal ring main as in Figures 11.6.7 and 11.6.8. With all coil configurations it is essential that each pipe section or each heater is separately trapped.

Long coils are susceptible to waterhammer, as they will collect condensate along their length. Because of this, it makes sense that coils are designed with a constant fall in the direction of steam flow. The modern float-thermostatic trap is equipped to withstand high levels of waterhammer, but if the symptoms are extreme, an inverted bucket trap or balanced pressure trap is a good choice. It may be necessary to lag float-thermostatic traps to protect them against damage by freezing. The inverted bucket trap may require a separate air vent to be fitted in parallel to remove air from the coil on start-up.

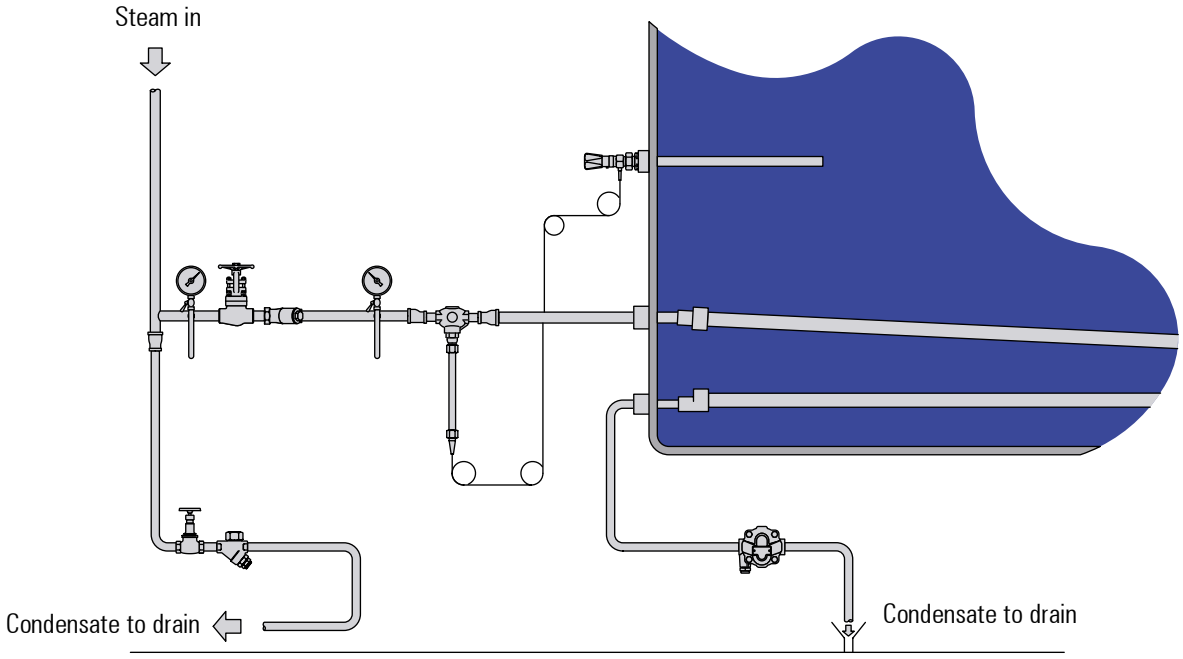


Fig. 11.6.5 Oil storage tank - pipe coil

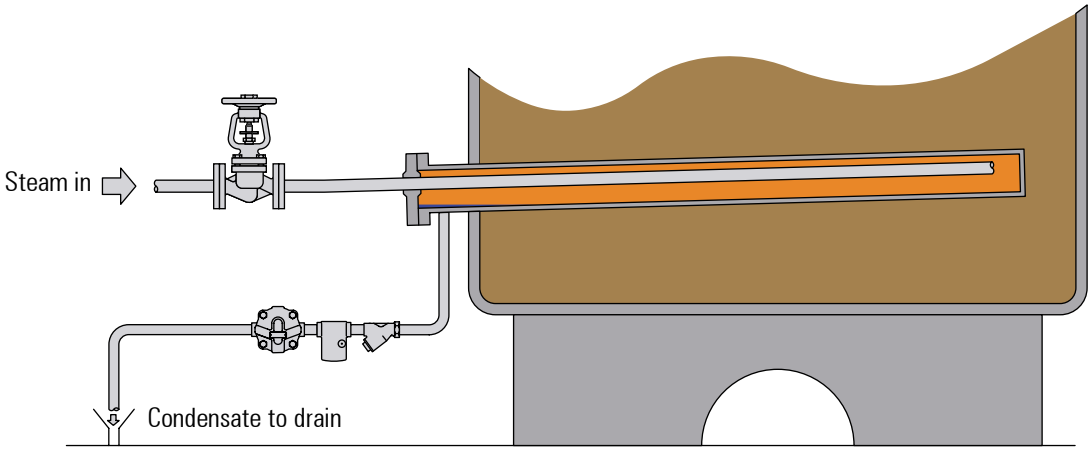


Fig. 11.6.6 Oil tank - bayonet heater

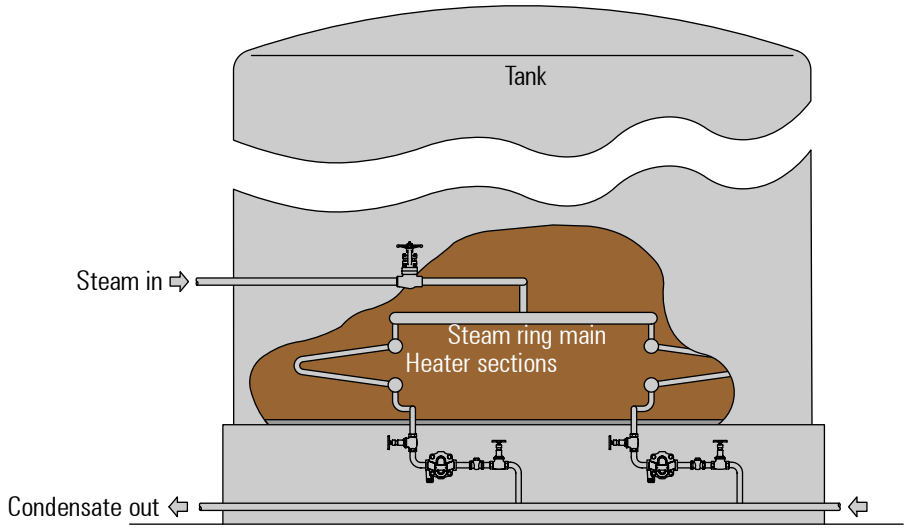


Fig. 11.6.7 Large oil tank with multi-heaters

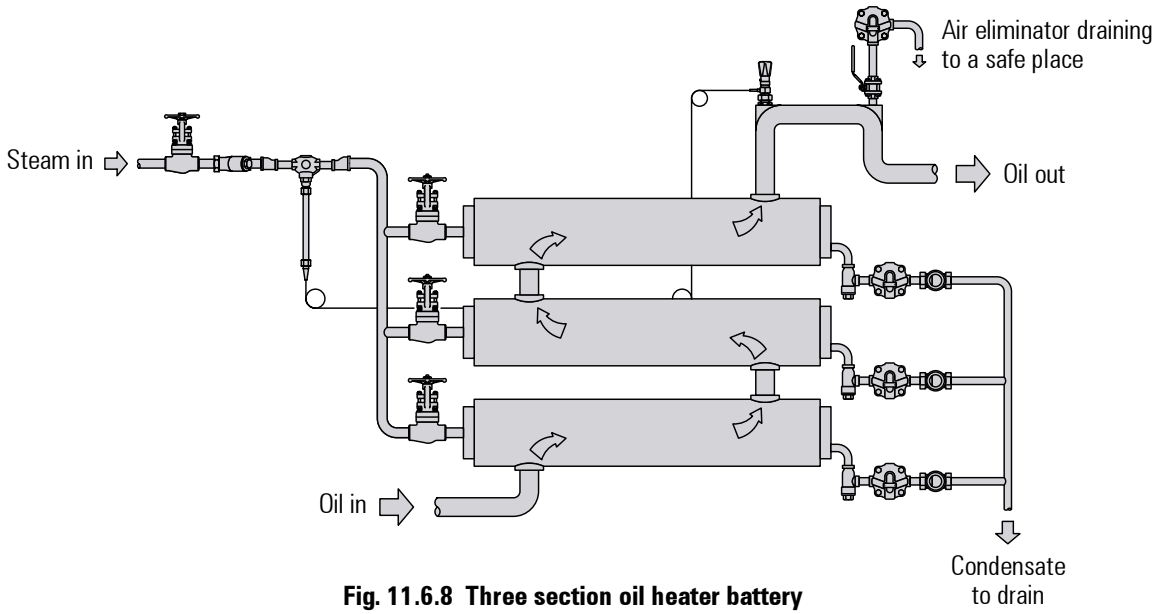


Fig. 11.6.8 Three section oil heater battery

Oil heater batteries

These are single or multi-stage heat exchangers and should be treated in a similar manner to outflow heaters. Each stage should be individually trapped and since they are often fitted indoors where the traps are not likely to freeze, float-thermostatic traps are the best choice.

Outflow heaters

An outflow heater is a shell and tube heat exchanger installed in the side of a storage tank, which heats the oil locally as it is pumped out of the tank. Automatic temperature control is usual and Figure 11.6.9 shows a Spirax Sarco self-acting control with the sensor in the oil outlet, actuating a valve in the steam supply.

The first choice is to use a float-thermostatic trap. If exposed to the elements, it should be insulated. It is normal for condensate to be wasted due to the risk of contamination by the oil, but if condensate is being returned and lifted up to a return main it is not recommended that it is lifted by its own pressure, as flooding and waterhammer are likely at light loads. A pump/trap installation may be used under these conditions.

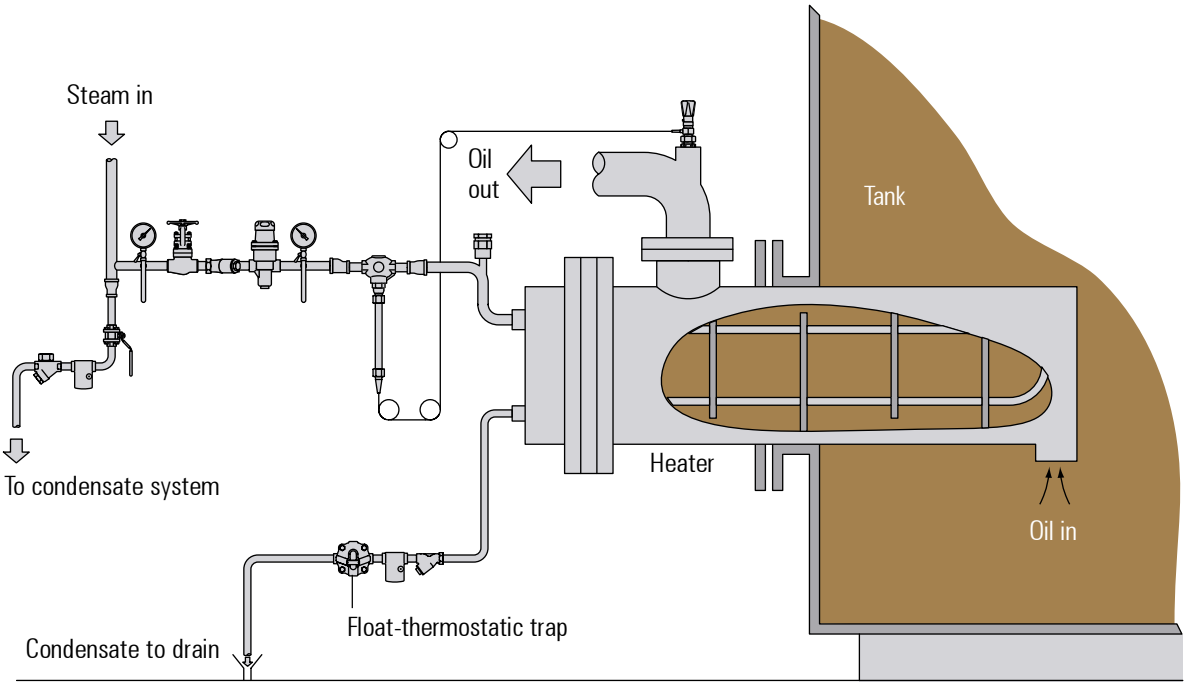


Fig. 11.6.9 Outflow heater

Tracer lines

Tracer lines should be arranged to fall in the direction of steam flow and should not exceed 25 metres in length for 10 mm tracers or 50 metres for all larger sizes, each length being drained by a balanced pressure thermostatic tracing trap or a thermodynamic trap. It is preferable to run single tracers near the bottom of the product line, and where it is necessary to pass flanges, this should be done with a horizontal loop to help maintain a continuous fall towards the trap.

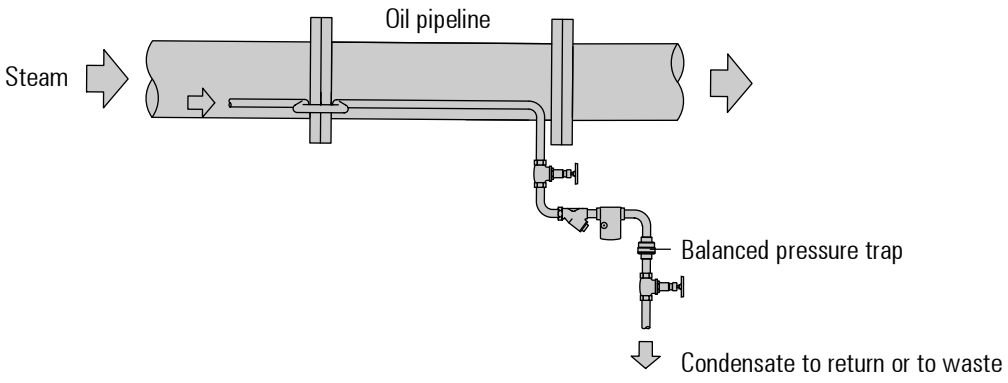


Fig. 11.6.10 Steam tracer line

Oil pipe tracing is not normally considered 'critical', and where condensate is discharged to waste, a bimetallic trap or a balanced pressure thermostatic tracing trap (in the constant temperature discharge mode) can be used. This will conserve energy and prevent unsightly flash steam. However, if critical tracing is considered essential, a thermodynamic or balanced pressure trap, discharging close to steam saturation, should be used.

A convenient method of supplying steam to large numbers of tracers on process lines, and for draining condensate from them, is to use distribution and collection manifolds. These are shown in Figure 11.6.11, along with universal steam traps, and pipeline connectors with integral isolation valves. These allow traps to be changed quickly and without any downtime.

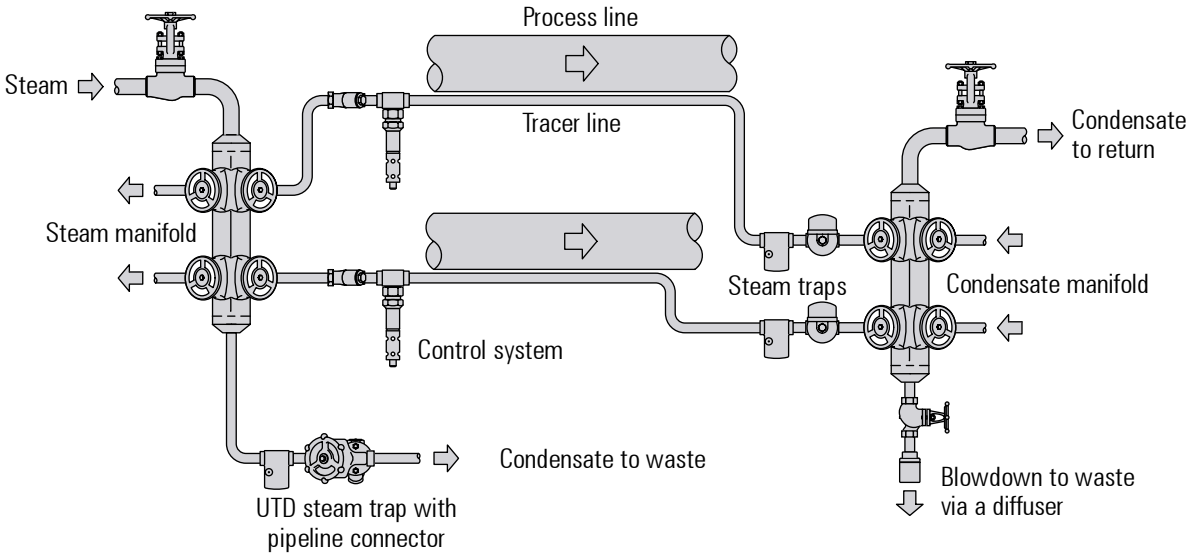


Fig. 11.6.11 Typical tracing application with steam and condensate manifolds

Jacketed pipes

When the temperature of the product is critical (because of the danger of solidification, burning or vaporisation) the complete product pipeline is 'traced' with a steam jacket. This application is often seen in 'sulfur' plants.

Jacketed pipes are generally constructed in not more than 6 m lengths and ideally, each length should be separately trapped using a balanced pressure thermostatic tracing trap, (Figure 11.6.12), or a TD trap.

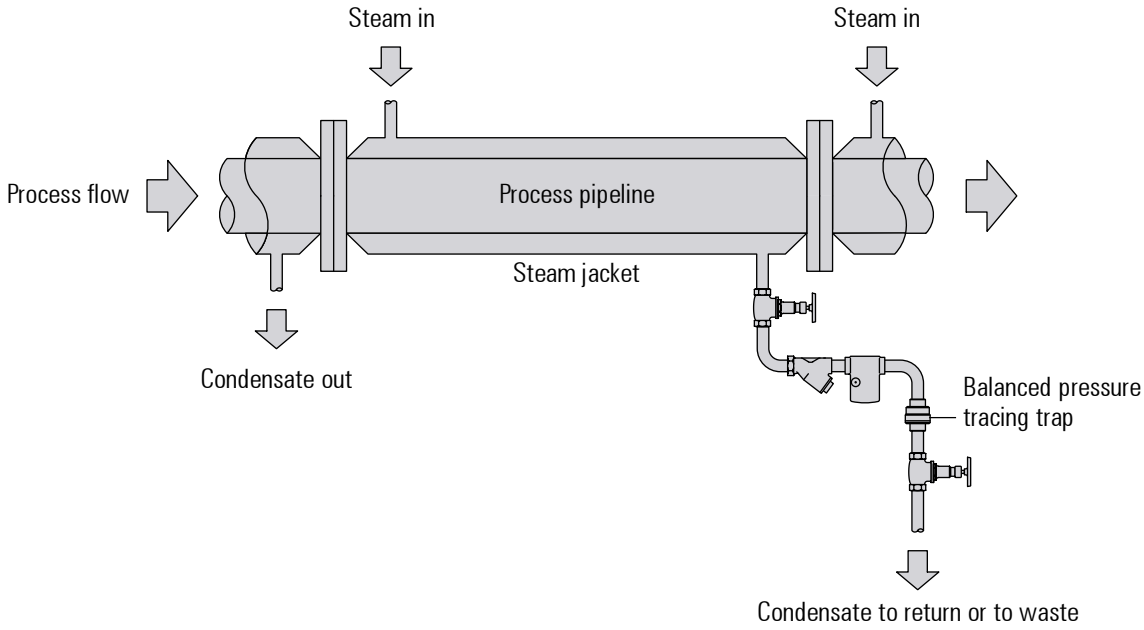


Fig. 11.6.12 Typical steam jacket with balanced pressure trap

It is, however, quite practical to join up to 4 lengths together, but it is important to join the jackets both at the top and bottom (Figure 11.6.13) so that the steam and condensate can flow freely and independently. It is worth noting, since many jacketed pipes are exposed to the elements, that the steel bodies of the thermodynamic and balanced pressure traps are not damaged by freezing.

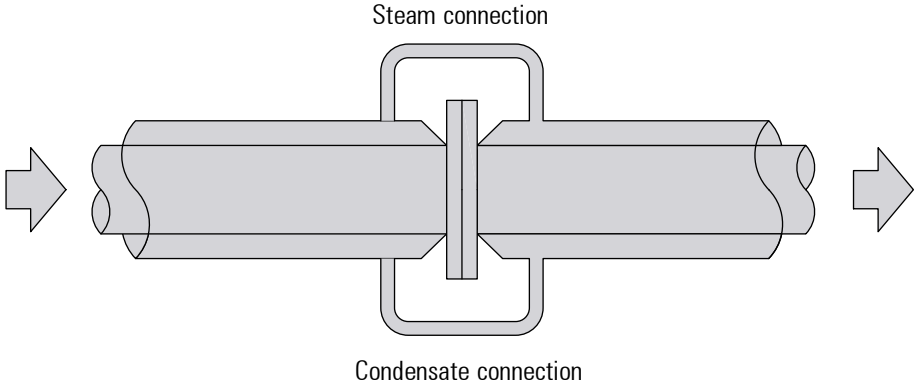


Fig. 11.6.13 Steam and condensate lines between connecting jackets

Hospital Equipment

A - Best choice, B - Acceptable alternative, [5] ('near-to-steam' capsule).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Autoclaves and sterilisers	B	B		**A**[5]			

Autoclave and sterilisers

The draining and air venting of modern high vacuum sterilisers is very important and the manufacturer normally supplies the necessary trapping equipment with the machine.

Figure 11.6.14 shows an autoclave supplied with plant steam for the jacket, and filtered steam for the chamber. The steam supplied to the chamber must be dry, so a separator drained by a float-thermostatic trap should be fitted to the steam line. For the chamber a balanced pressure thermostatic trap with near-to-steam capsule can be used successfully. On large units a float-thermostatic trap may be needed. A strainer to protect the trap is important, as it will catch any fibrous material or broken glass. If the steam inlet to the jacket is at the bottom or at one end, an air vent at the top or the far end will give better heating. The jacket may be drained with a balanced pressure thermostatic trap-strainer unit.

On new systems, there is an increasing requirement to use all stainless steel pipework and fittings to comply with European and International standards. In many cases, this will require the use of 316L steam traps.

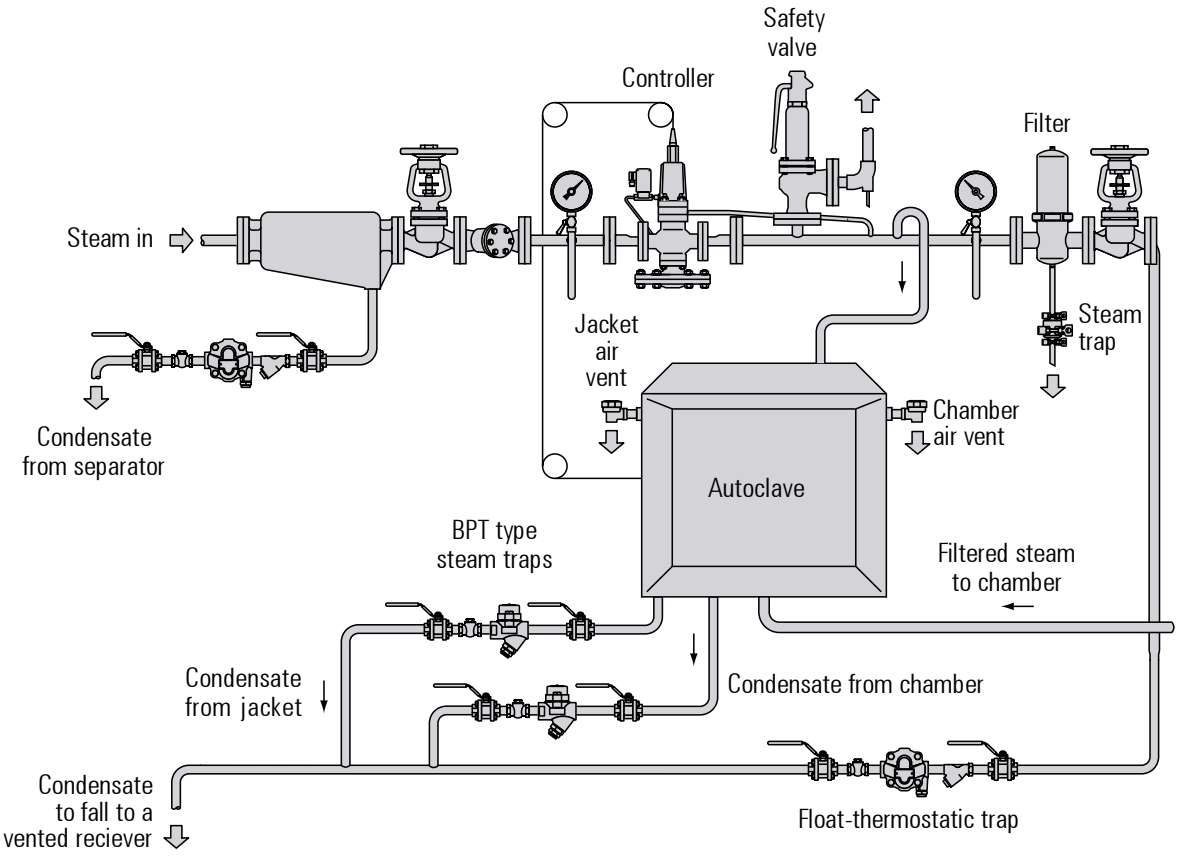


Fig. 11.6.14 Hospital autoclave with filtered steam supply

Questions

1. What steam traps are best suited to draining kitchen boiling pans?

a| Balanced pressure types ☐
b| Thermodynamic types ☐
c| Inverted bucket types ☐
d| Fixed orifice devices ☐

2. Why is it a good idea not to fit strainers on kitchen steaming ovens?

a| They cost too much ☐
b| They block with grease discharged with the condensate ☐
c| There is usually not enough space to fit them ☐
d| They increase radiation losses and effect the trap's operation ☐

3. How should coils be run in large oil tanks to provide good service?

a| Horizontally ☐
b| Vertically ☐
c| Falling with the direction of steam flow ☐
d| Falling against the direction of steam flow ☐

4. Name a convenient method of collecting condensate from multiple tracer lines?

a| Allow the condensate to drain to waste ☐
b| Group trap large numbers of tracers with one steam trap ☐
c| Fit steam traps every 30 m of tracer line ☐
d| Fit manifolds to collect condensate from multiple tracer lines ☐

5. Why is it important to fit a strainer before an autoclave chamber trap?

a| The strainer will help condense any steam in the condensate line ☐
b| To reduce any effect of backpressure that may occur ☐
c| To protect the trap from broken glass or fibres in the condensate ☐
d| It is not particularly important to do this ☐

6. Why are there normally two steam supplies to hospital autoclaves?

a| In case one of them fails during an operating cycle ☐
b| To allow the autoclave to work at two different pressures ☐
c| Because autoclave manufacturers traditionally fit two supplies ☐
d| One to supply the chamber, one to supply the jacket ☐

Answers

1: a, 2: b, 3: c, 4: d, 5: c, 6: d

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Module 11.7

Selecting Steam Traps - Industrial Dryers

Industrial Dryers

A - Best choice, B - Acceptable alternative, [1] (parallel air vent).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Hot air dryers	**A**		B[1]	B			
Drying coils			B[1]	**A**			B[1]
Multi-bank pipe dryers	**A**		B[1]	B			B[1]
Drying cylinders	B	**A**					B[1]
Multi-cylinder sizing machines	B	**A**					B[1]

Hot air dryers

Many industrial substances are dried using hot air. The machines take various forms, but can either consist of heater batteries through which air is forcibly drawn before being blown on to the wet material, or pipes over which air naturally convects (Figure 11.7.1). The need for draining and air venting is the same as for heater batteries used for space heating.

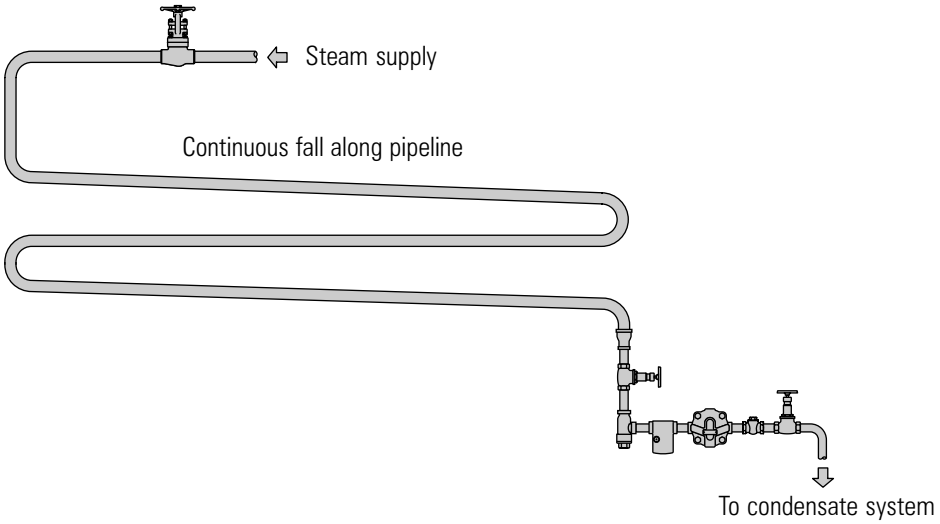


Fig. 11.7.1 Hot air continuous convection coil with a float trap set

Drying coils

These can be continuous or in grid form, horizontal or vertical. Continuous coils should be short with an adequate fall in the direction of steam flow so that condensate can easily reach the drain point. They can then be drained using a float-thermostatic trap or a balanced pressure trap. If the condensate is lifted from the trap using coil pressure only, waterhammer may occur. Waterhammer is likely in grid coils unless all sections fall towards the drain point and the condensate then falls to a lower level. The same recommendations apply as for continuous coils. If thermodynamic or inverted bucket traps are used, an air vent bypassing the trap will shorten 'start-up' time. The inlet header should be drained separately, unless the cross pipes are level with the bottom of it, to allow free flow to the condensate header. Always use an eccentric reducer at the coil outlet (Figure 11.7.2).

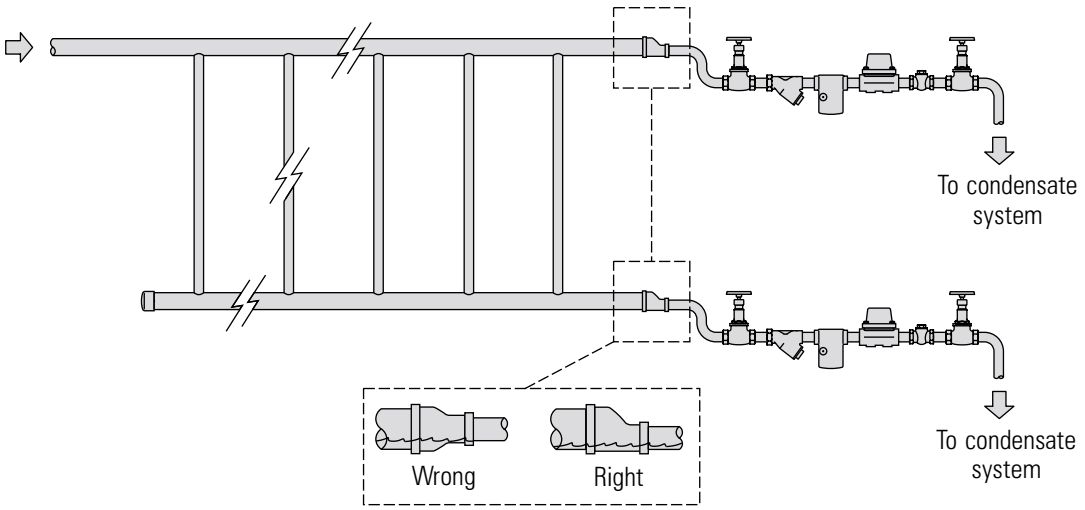


Fig. 11.7.2 Grid type drying coils with balanced pressure traps

Multi-bank pipe dryers

Examples of multi-bank pipe dryers are the older types of tentering and carbonising machines, and kilns used in the textile and wood treatment industries, (but which are now being replaced by steam heated hot air dryers).

At one time, very long, continuous runs of pipe were used, and because it was impossible to arrange for a proper fall and also due to sagging pipes, waterlogging and waterhammer were common. Where this arrangement still exists thermodynamic traps can be used with an air vent in parallel. Later machines of this type were divided into bays, and the improved layout reduced waterhammer. In these cases, float-thermostatic traps or balanced pressure thermostatic traps with stainless steel elements can also be used. They should be fitted outside the machine casing, but as close as possible to the end of the coil.

Where the heating surface consists of horizontal coils running between vertical headers, the top of the vertical condensate outlet header should be air vented separately. This will considerably reduce start-up times. The bottom of the vertical stem inlet header should also be drained (Figure 11.7.3).

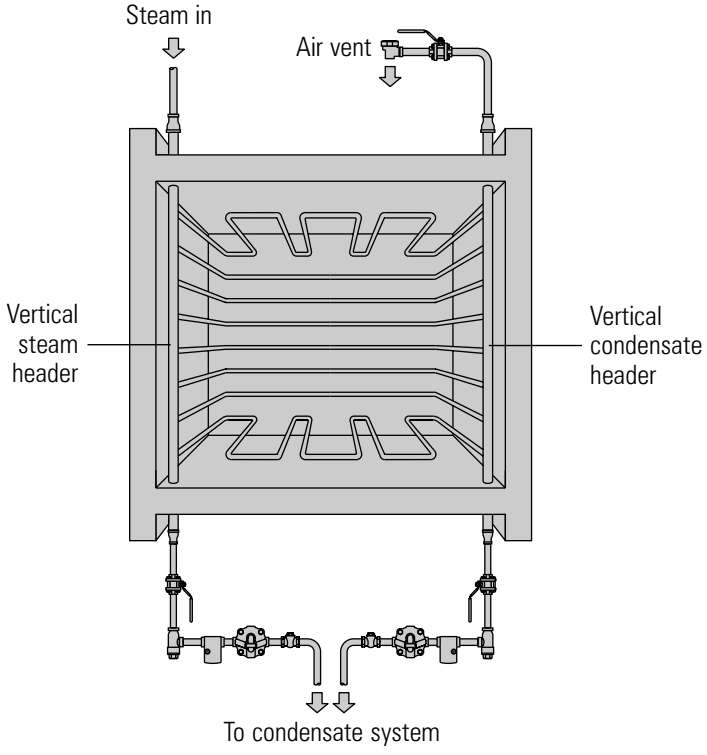


Fig. 11.7.3 Multi-bank pipe dryer with vertical headers and float-thermostatic traps

Heated rotating cylinders

Heated rotating cylinders vary widely in size, speed and condensate handling arrangements, which may be by means of internal scoops or fixed or rotating syphon pipes. The latter are normally associated with high speed machines, and sometimes use a special blow-through system. (Refer to Figures 11.7.4 and 11.7.5).

Slow speed cylinders with scoops and fixed syphons should be trapped and air vented individually, each with an air bottle arrangement. This comprises a float-thermostatic trap with steam lock release, strainer, sight glass, air collector vessel, and air vent, assembled in various forms to fit different outlet nozzle arrangements. This arrangement allows good individual control of cylinder temperature where it is required. The sight glass can be used to set up the steam lock release valve.

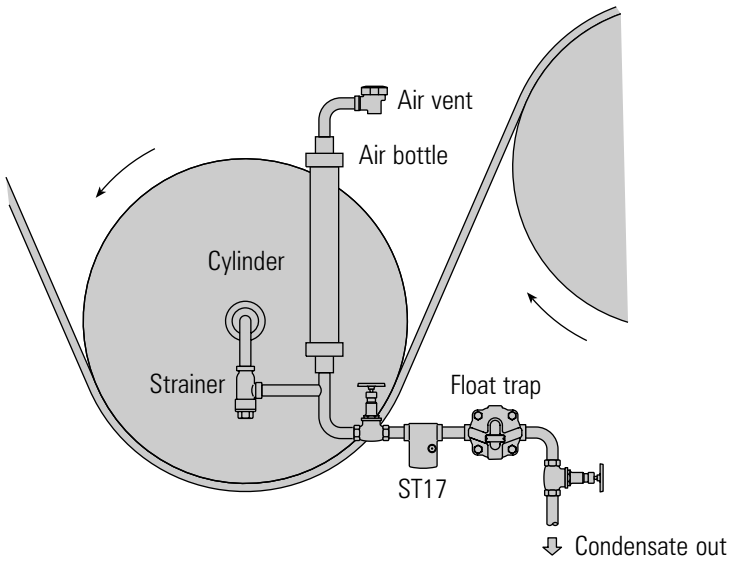


Fig. 11.7.4 Slow speed cylinder drainage with system unit

On faster machines, there is a need for large amounts of blow-through steam to assist the flow of condensate out of the cylinder via the syphon tube. The float trap internal steam lock release cannot handle such large amounts, and an external bypass with needle valve will give better results.

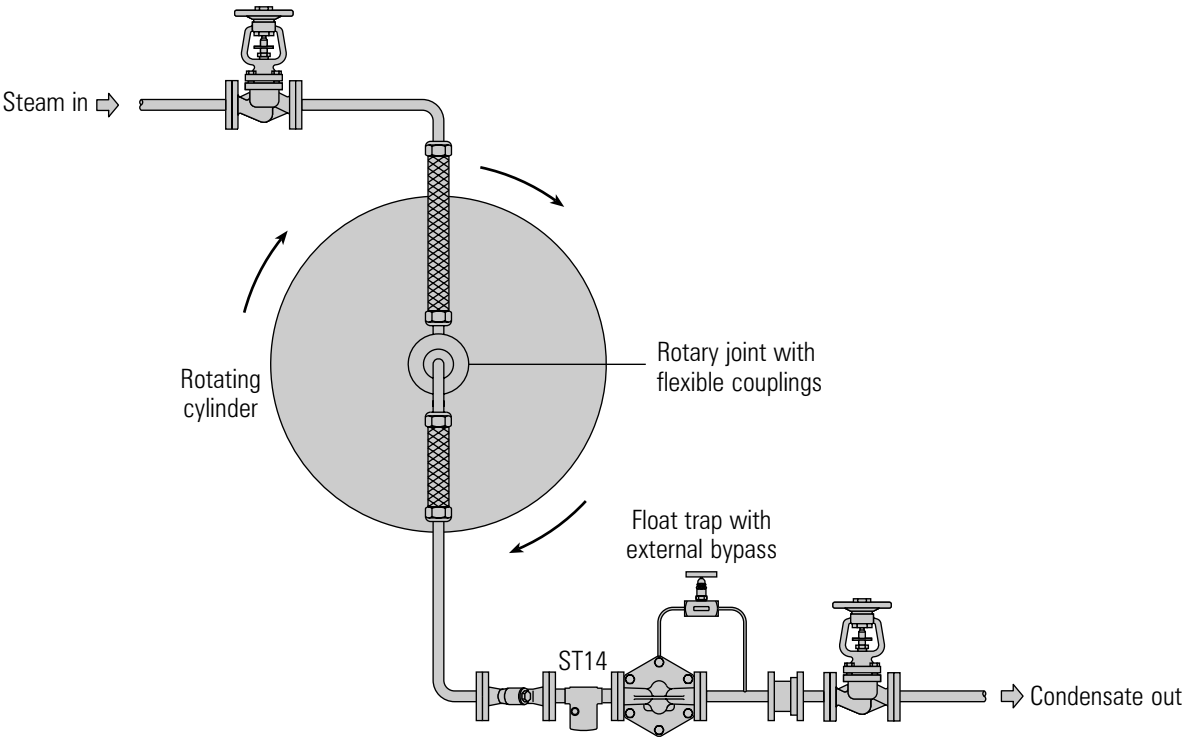


Fig. 11.7.5 High speed cylinder with float trap and parallel blow-through valve

Multi-cylinder sizing machines

Figure 11.7.6 shows how to drain and air vent a typical multi-cylinder textile sizing machine.

The steam manifold supplying the cylinders is drained by a float trap, or thermodynamic trap.

Float traps with steam lock release drain the cylinders. This compact arrangement is particularly suited to small combined inlet and outlet nozzles.

The size bath is usually heated either by direct steam injection or a steam coil and in both cases the supply should be regulated by a suitable temperature control. The coil should be drained by a float-thermostatic trap.

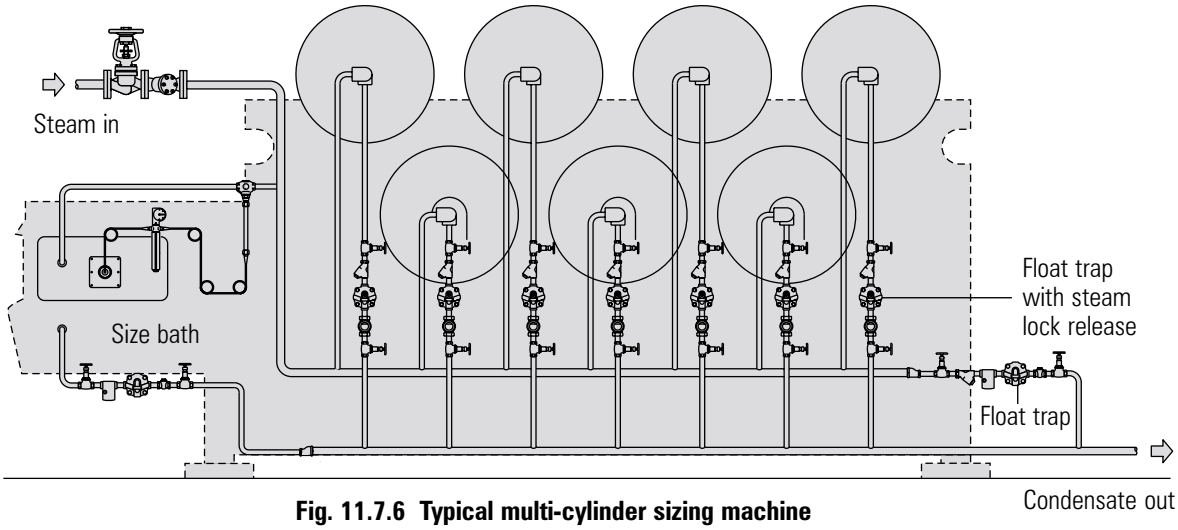


Fig. 11.7.6 Typical multi-cylinder sizing machine

Multi-cylinder dryers

Modern vertical machines should, if possible, have the cylinders drained individually, using float traps with combined steam lock releases and air vent bypasses.

If the cylinders all drain into a vertical condensate manifold, use a float-thermostatic trap at the bottom and an air vent at the top of the manifold.

The steam inlet manifold should be similarly drained and air vented (Figure 11.7.7).

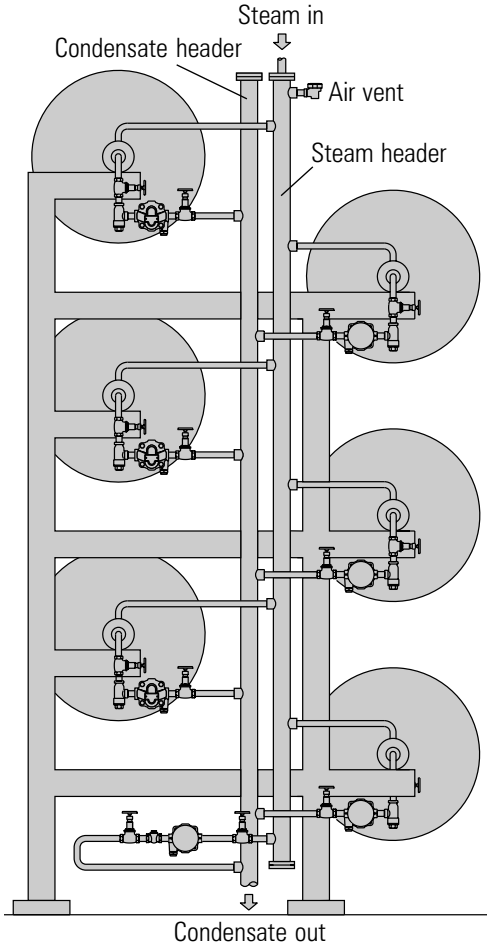


Fig. 11.7.7 Multi-cylinder dryers with float trap and steam lock release draining the cylinders

Questions

1. What is a common cause of waterhammer in drying coils?

a| Wet steam supplied to the coil ☐

b| Too low a steam pressure onto the coil ☐

c| Condensate has to lift after the steam trap ☐

d| The coil falling in the direction of steam flow ☐

2. What will help reduce start-up times in multi-bank dryers?

a| Higher steam pressure in the coil ☐

b| Shorter heating coils ☐

c| Longer heating coils ☐

d| An air vent fitted to the top of the vertical condensate heater ☐

3. Name a trusted method of removing condensate from slow speed rotating cylinders?

a| An air bottle arrangement having a float-thermostatic trap fitted with steam lock release ☐

b| With a pump-trap ☐

c| With a vacuum condensate system ☐

d| With an oversized thermodynamic trap ☐

4. Name a good way of commissioning a float trap fitted with a steam lock release

a| Pass the condensate to drain and adjust the steam lock release for live steam waste ☐

b| Listen to the internal trap noise with a screwdriver and adjust the steam lock release ☐

c| Adjust the steam lock release and observe the process performance ☐

d| Adjust the steam lock release to the fully open position ☐

5. On high-speed rotating cylinders how may the steam trap installation be modified to increase the removal of condensate?

a| Fit the trap close-coupled to the cylinder rotary joint ☐

b| Fit the trap with an external bypass to adjust the blow-through rate ☐

c| Fit a larger float trap to increase the blow-through rate ☐

d| Fit a balanced pressure steam trap with an air vent in parallel ☐

6. Which of the following statements is true?

a| Bimetallic steam traps are an ideal choice for rotating cylinders ☐

b| Rotating cylinders can not suffer from steam locking ☐

c| Strainers cannot be fitted to float traps which have a steam lock release ☐

d| Air vents around thermodynamic and inverted bucket traps can considerably improve start-up times. ☐

Answers

1: c, 2: d, 3: a, 4: c, 5: b, 6: d

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Module 11.8

Selecting Steam Traps - Laundries, Presses

Laundries

A - Best choice, B - Acceptable alternative, [1] (parallel air vent) [5] ('near-to-steam' capsule), [6] (anti-air-binding disc).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Garment press	B	B	**A**[6]				
Ironers and calendars	B	**A**	B[1]	B[1]			
Tumble dryers	**A**	B					
Dry cleaning machines	**A**		B[1]				

Garment presses

Thermodynamic, float-thermostatic traps, and balanced pressure traps can be used. It is important for each press to have a separate trap (Figure 11.8.1). The head and tables of twin presses should also be individually drained for maximum output.

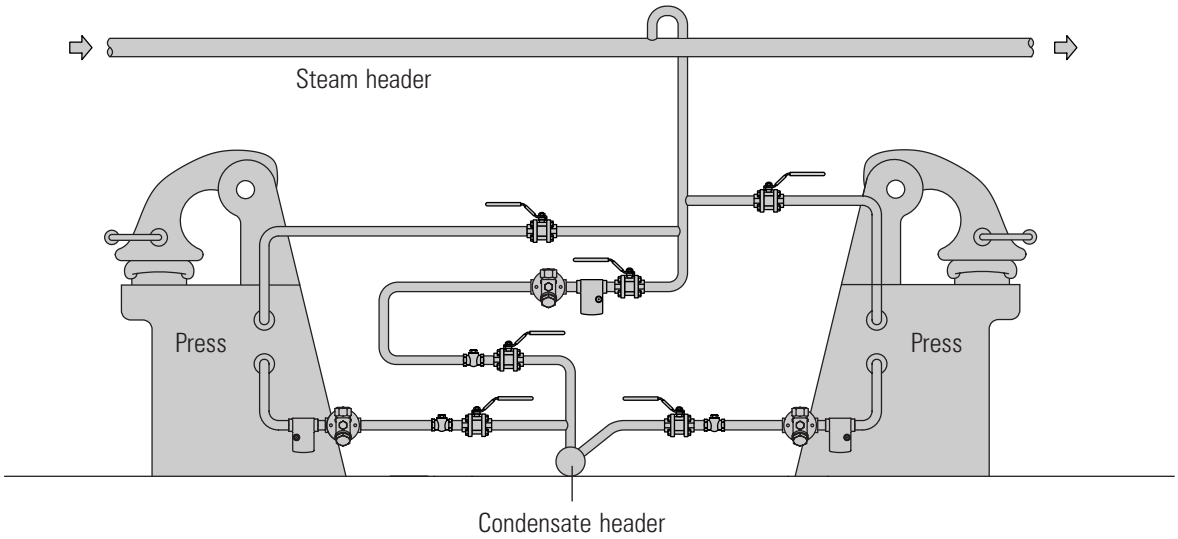


Fig. 11.8.1 Garment presses with thermodynamic traps

Ironers and calenders

Ironers vary greatly in construction, but in all cases proper condensate and air removal are vital for maximum output. In addition, machines with light fabricated beds may distort and tear the work if heating is uneven, due to air pockets or waterlogging. The steam supply should always be drained, preferably using a separator.

Modern, fully enclosed machines often have the traps fitted all at one end for ease of maintenance, with long pipes connecting the traps from the middle of the bottom of the beds to the drain points. Good results are obtained from thermostatic traps if the long connecting pipes are left unlagged (see Figure 11.8.2), otherwise float traps with steam lock release can be used. Thermodynamic traps can sometimes be used, but it is best if air vents are fitted in parallel. Fit vents to the beds at the point furthest from the steam inlet, and also any heated airing gaps. On the roll, if heated, a float trap with steam lock release is the best choice, although a balanced pressure thermostatic trap fitted on an unlagged line at least 1 m down from the outlet has proven to give good results . If preferred, a thermodynamic trap with anti-air-binding disc can also be used.

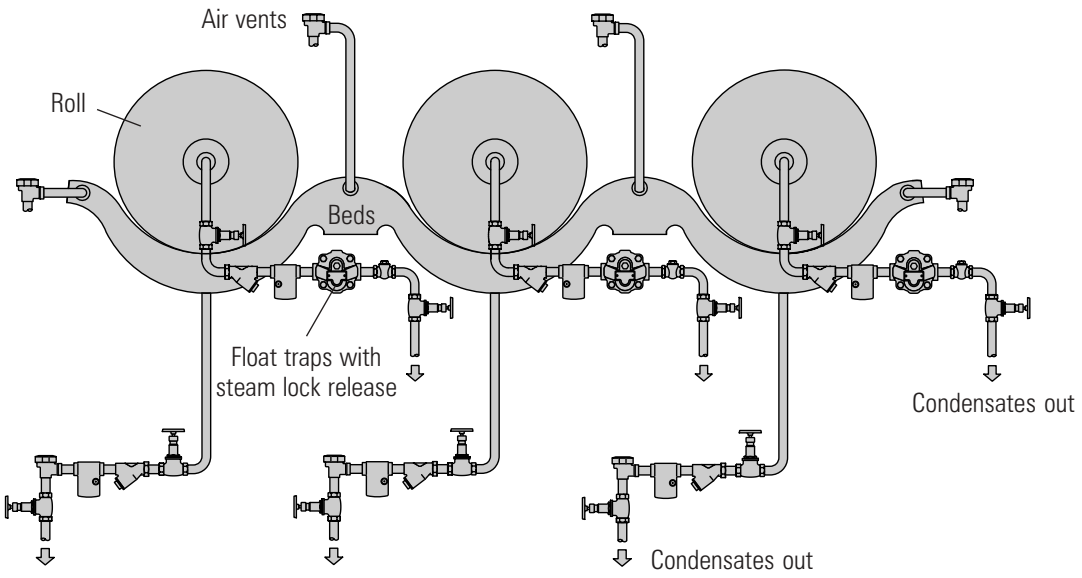


Fig. 11.8.2 Calendar beds drained by float traps with steam lock release

Dry cleaning machines

The air heater battery and the spirit still heating coil should each be fitted with a float-thermostatic trap (Figure 11.8.3). Thermodynamic traps can also be used.

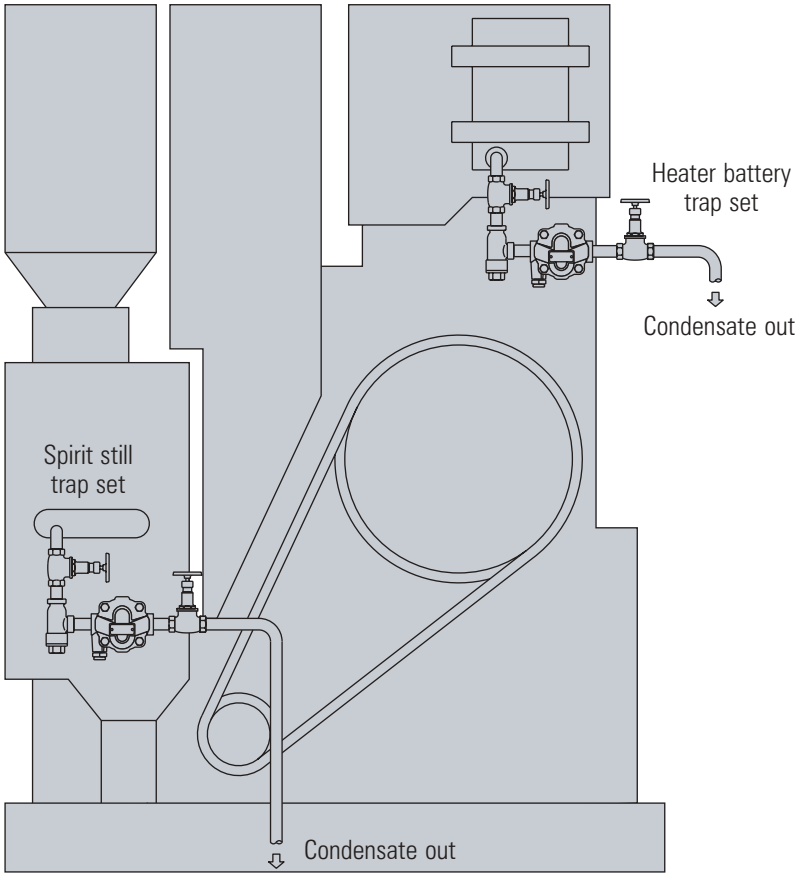


Fig. 11.8.3 Dry cleaning machine with float traps on the heater battery and spirit still

Tumble dryers

The air heater battery should be drained using a float-thermostatic trap but thermodynamic traps can be used with a separate air vent.

Presses

A - Best choice, B - Acceptable alternative, [1] (parallel air vent), [6] (anti-air-binding disc).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Multi-platen presses (parallel connections)	B		**A**[6]				
Multi-platen presses (series connections)			**A**[1, 6]				
Tyre presses	B		B[1]	**A**			B[1]

Multi-platten presses (parallel connections)

To ensure proper platen drainage, the steam supply connection should be above the platen with the condensate outlet below. Where possible each platen should have its own trap (Figure 11.8.4) but where accurate platen temperatures are not required then the 'group trapping' arrangement shown in Figure 11.8.5 can be used. The steam inlet header is drained by a thermodynamic trap. This is also ideal for draining individual platens, as each platen has a relatively small load. Traps should discharge into a generously sized return header through swept connections. This will eliminate back pressure caused by the simultaneous discharge of several traps. If the press is temperature controlled, always use float traps.

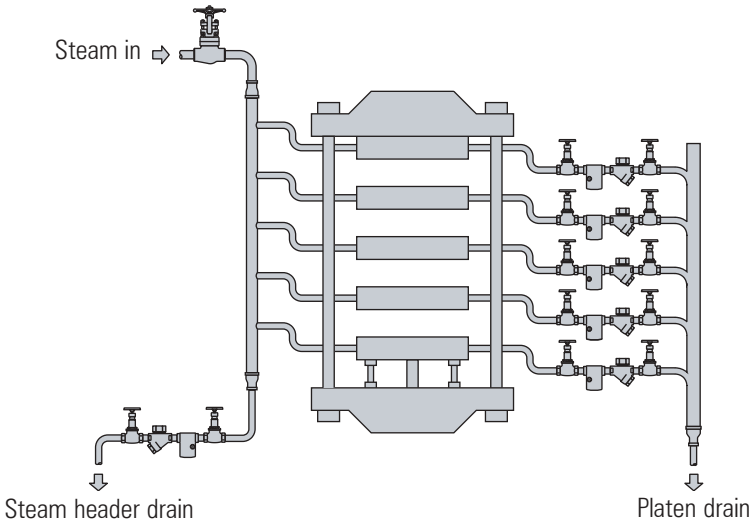


Fig. 11.8.4 Platens trapped individually

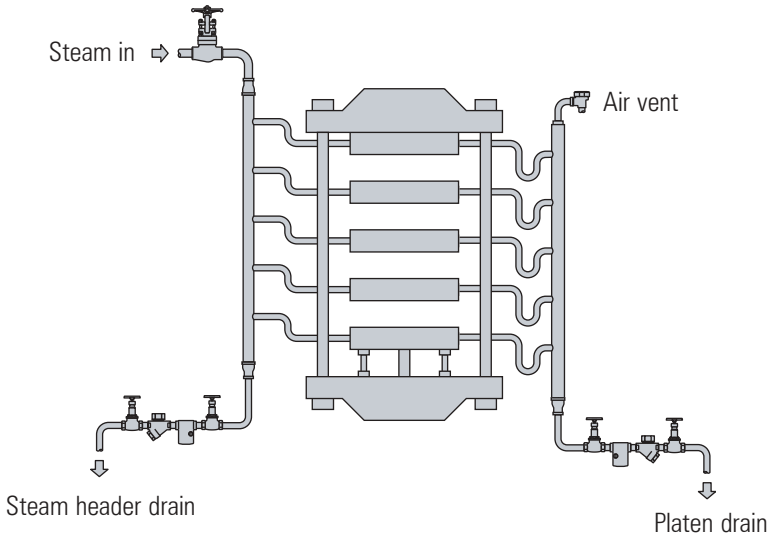


Fig. 11.8.5 Platens group trapped

The thermodynamic trap is able to withstand the extreme waterhammer which usually occurs with this type of press due to the loops often formed in the flexible steam and condensate connections. However, if these are properly fitted to give a continuous fall, then float thermostatic traps can be used. It may be advantageous to fit an air vent in parallel around the trap, as in Figure 11.8.6.

Multi-platen presses (series connections)

This layout is almost certain to have water pockets due to the piping, and the flow of the condensate over the flat platens will be slow. For both reasons use a rugged blast-discharge trap (Figure 11.8.6), which will help purge the condensate out of each platen. This diagram shows the thermodynamic trap with an air vent fitted in a bypass around the trap, but an inverted bucket trap can also be used. The steam supply should be properly drained, and it may be an advantage to fit a separator close to the inlet.

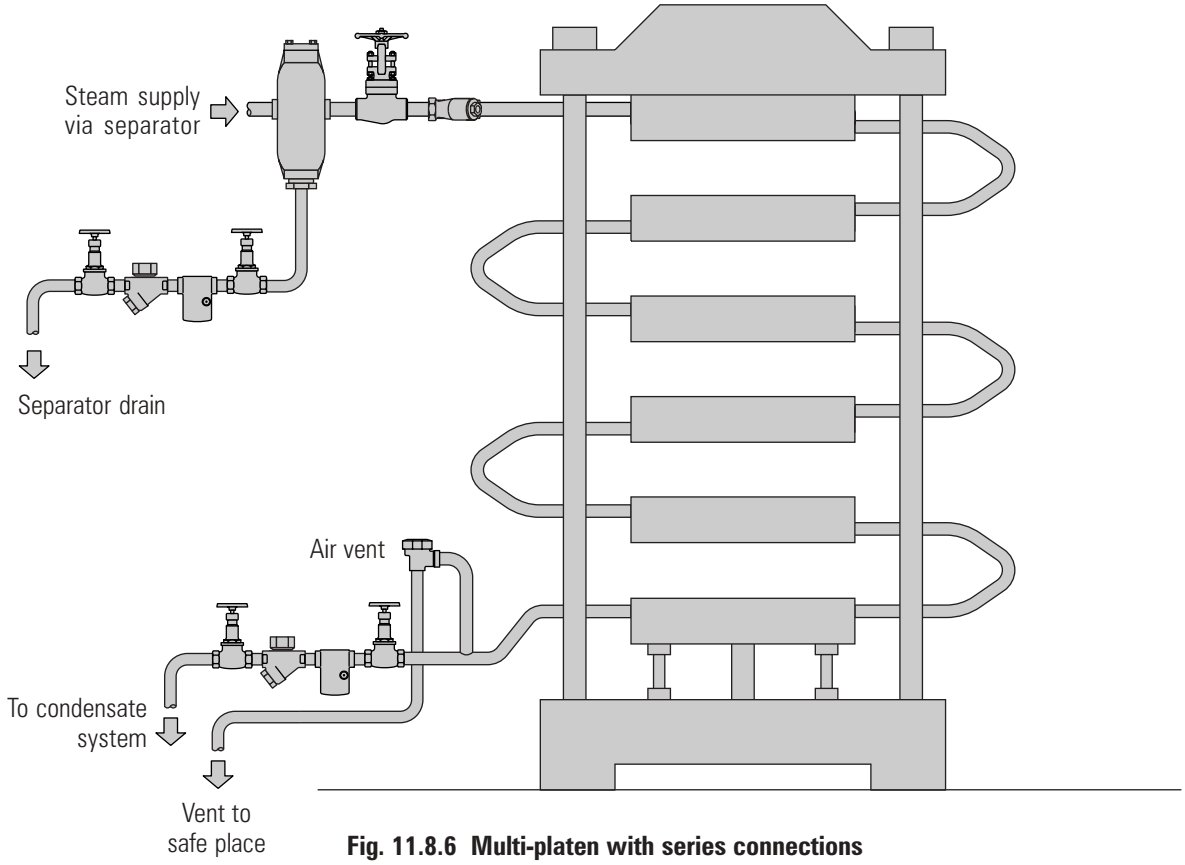


Fig. 11.8.6 Multi-platen with series connections

Tyre presses

Good temperature conditions are vital to avoid 'soft' cures. Condensate must be removed as it forms and there must be free discharge of air. Nitrogen (or other inert gases) are sometimes used to add internal pressure to the 'bladder' during the curing process. The selected trap must therefore be able to remove the gas freely or the process times will be extended. In practice, balanced pressure traps seem to give the best results but float-thermostatic and thermodynamic traps (Figure 11.8.7) can also be used. If solenoid or quick acting valves are used to control the process, then inverted bucket traps may be used successfully, in conjunction with separate air vents.

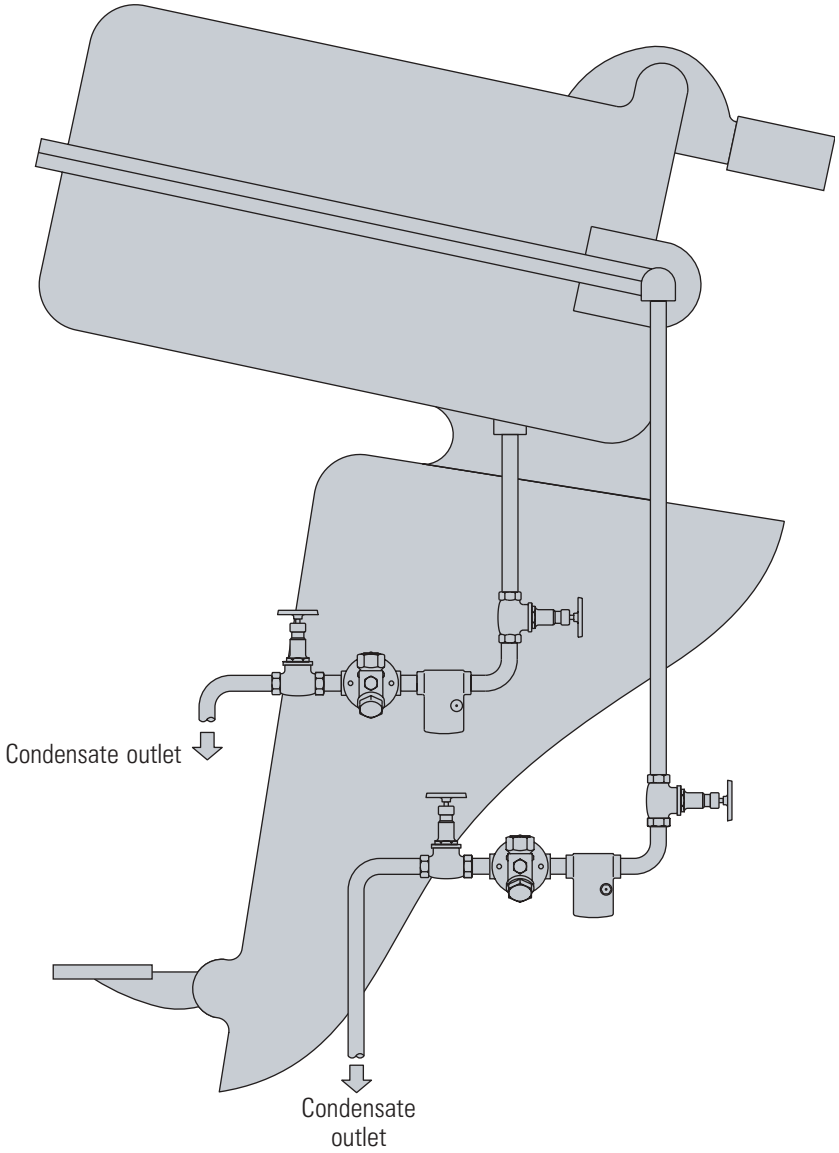


Fig. 11.8.7 Tyre press with thermodynamic traps (steam supply not shown)

Questions

1. Steam traps are often fitted at one end of a laundry calender. Why?

a| To provide easy access for maintenance ☐
b| To reduce steam locking ☐
c| Smaller traps can be used ☐
d| To reduce the number of air vents ☐

2. Where should air vents be fitted on the beds of laundry calenders?

a| On the steam inlet to the bed ☐
b| In the middle of the bed ☐
c| At the point furthest away from the steam inlet ☐
d| It is not necessary to fit air vents on calender beds ☐

3. As calenders can suffer from steam locking, which traps can be used?

a| Float traps fitted with Steam Lock Release ☐
b| Thermodynamic traps with internal air vents ☐
c| Balanced pressure traps ☐
d| Inverted bucket traps ☐

4. When should float traps be used to drain platen presses?

a| When the condensate pipe lifts up after the trapping point ☐
b| When the press is temperature controlled ☐
c| When thermodynamic traps are not available ☐
d| When steam locking is possible ☐

5. On group trapped presses, what consideration must be given to the condensate installation?

a| The condensate manifold should be kept as short as possible ☐
b| The steam trap must be a float trap with steam lock release ☐
c| Use a generously sized condensate manifold with an air vent on top ☐
d| Use a horizontal condensate manifold ☐

6. Which of the following statements is true?

a| Garment presses are ideal for a group trapping arrangement ☐
b| Steam is not used on dry cleaning machines ☐
c| The thermodynamic trap is unable to handle fluctuating condensate loads ☐
d| It is important to vent the air from ironers and calender beds ☐

Answers

1:a, 2: c, 3: a, 4: b, 5: c, 6: d

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Module 11.9

Selecting Steam Traps - Process Equipment

Process Equipment

A - Best choice, B - Acceptable alternative, [1] (parallel air vent), [2] (with 1 m cooling leg), [6] (anti-air binding disc).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Boiling pans - fixed	**A**	B	B[1]	B			
Boiling pans - tilting	B	**A**					
Retorts	**A**						
Industrial autoclaves	**A**						B[1]
Digesters	**A**[1]		B[1]				
Hot tables	B		B[6]	**A**[2]			
Brewing coppers	**A**[1]	B					
Evaporators	**A**[1]	B					B[1]
Vulcanisers	**A**		B[1] (Jacket only)				B[1]

Fixed boiling pans

Process boiling pans are used in many industries for heating a wide range of materials, and nearly always have to heat up their contents as quickly as possible. In this respect they differ from canteen boiling pans. Steam pressures are normally high, and efficient removal of air and condensate is vital. The traps fitted to fixed production pans must discharge condensate and air very quickly and must deal with a condensate load which varies widely between starting and running conditions. The float-thermostatic trap is the ideal choice. The jacket will start up quicker if an air vent is placed opposite the steam inlet position. Provision is usually made for this.

Figure 11.9.1 shows a float-thermostatic trap fitted close up to the drain point. The thermodynamic trap can be a useful alternative particularly where the outlet is close to the ground, but it may be necessary to fit an air vent in a bypass around the thermodynamic trap for maximum production. Balanced pressure thermostatic traps can also be used on small pans but must be fitted on an unlagged cooling leg.

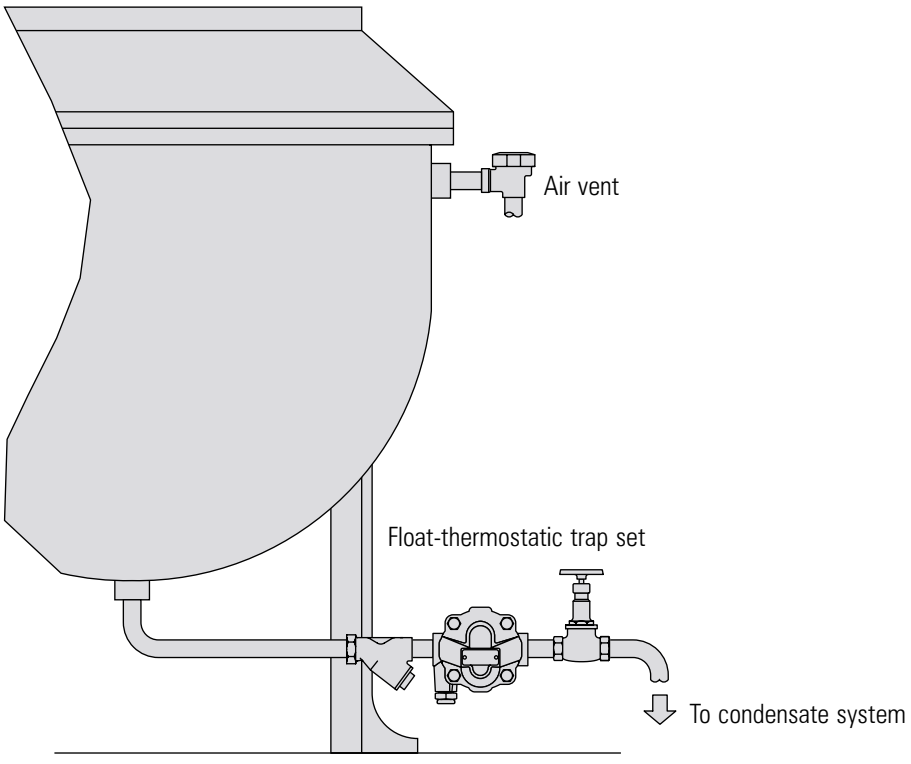


Fig. 11.9.1 Fixed boiling pan with float-thermostatic trap set

Figure 11.9.2, shows the arrangement when the trap cannot be fitted underneath the pan, and the condensate is removed by an internal fixed syphon pipe through a float-thermostatic trap with steam lock release.

Tilting process pans

A feature of all tilting-type jacketed pans (Figure 11.9.2) is that steam locking conditions are always present, however close the trap is fitted to the pan. The reason is that condensate must pass through a rising tube from the bottom of the jacket to the outlet trunnion. This rising passage fills with steam and causes the trap to remain closed, thus holding back the condensate, unless the proper precaution is taken. The trap must have a steam lock release feature.

If steam enters the jacket at the top, an additional air vent on the jacket will improve start-up times.

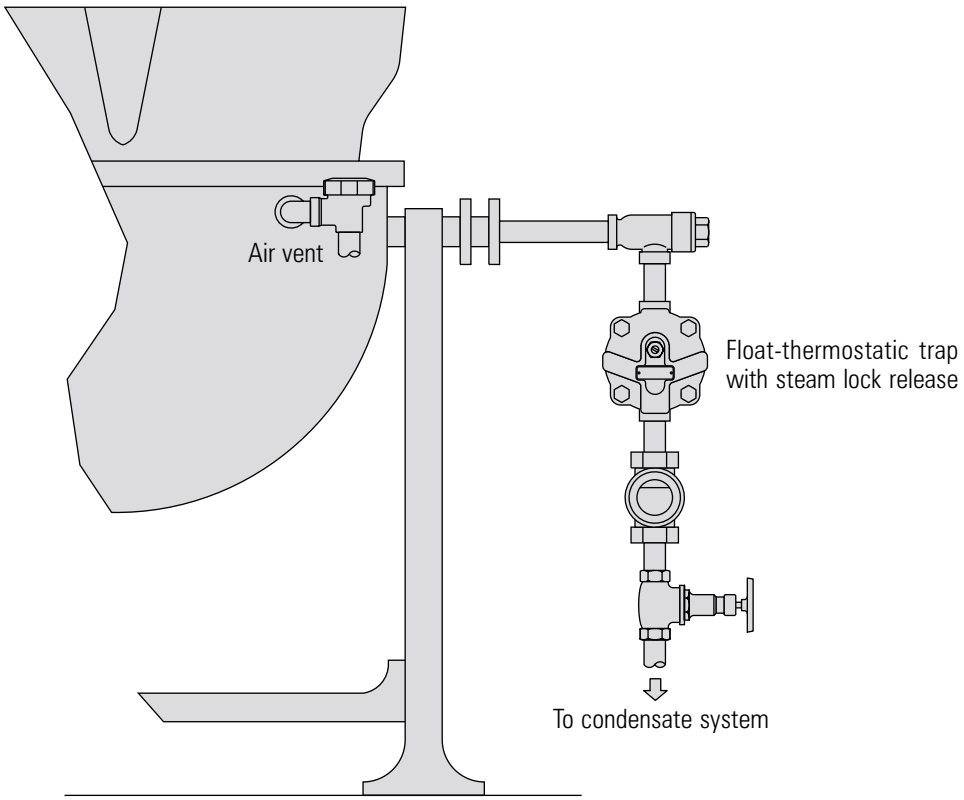


Fig. 11.9.2 Tilting production pan with syphon tube condensate removal

Retorts

Retorts are generally large vessels into which a product is placed for processing or cooking with relatively low pressure steam.

An example would be a canning retort into which sealed tins of food are placed. Steam is then used to heat or cook the contents of the can. Once the door is closed, it is vital to ensure that all the air and condensate is removed and replaced by dry saturated steam. A float-thermostatic trap (with its inbuilt air vent) is ideal, especially due to its ability to pass large volumes of condensate at relatively low pressure.

On such a large steam space, air removal can be a problem. If all the air is not removed, process temperatures will fall, resulting in product spoilage. If the steam inlet is at the bottom, fit balanced pressure air vents at the top. If steam enters at the top, add additional air vents near the bottom (Figure 11.9.3).

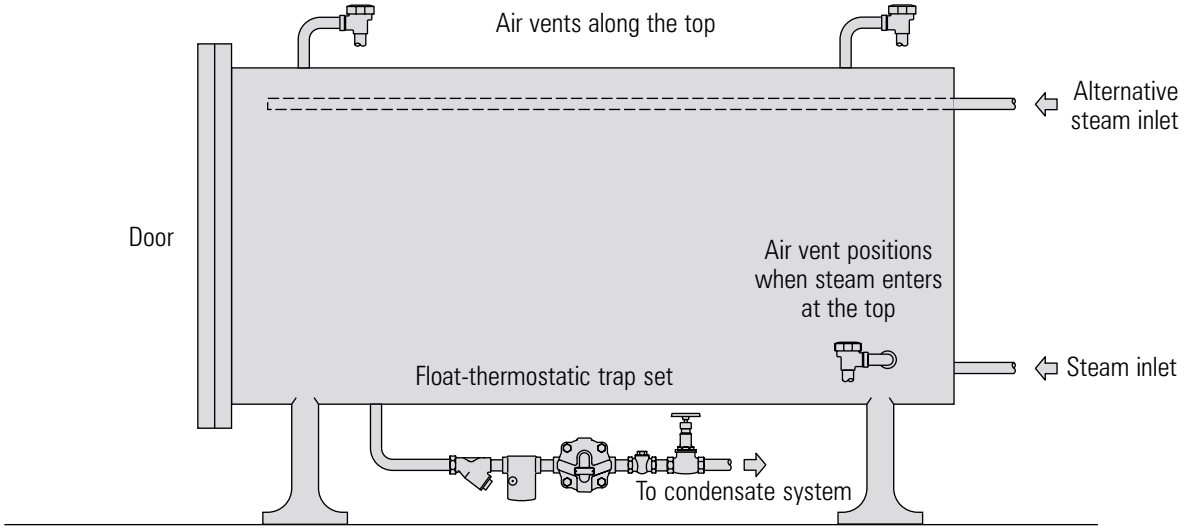


Fig. 11.9.3 Low pressure retort for cooking

Industrial autoclaves

Figure 11.9.4 shows an alternative method of venting a large autoclave using a self-acting temperature control as a large capacity air vent. Where there is a cooling cycle, the traps and air vents must be suitably valved and bypassed.

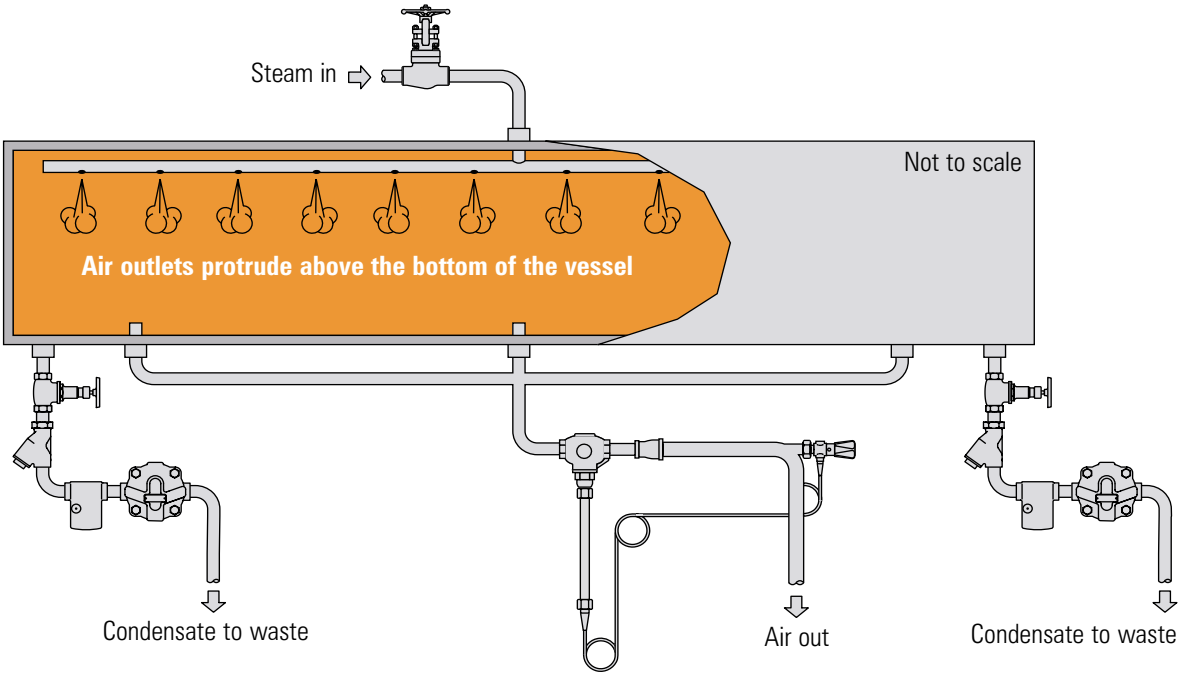


Fig. 11.9.4 Process retort with large air venting capacity (vessel not to scale)

Digesters

Heat is provided by a steam jacket which will be full of air on start-up. The steam inlet position can vary, being at the bottom, in the middle or at the top of the jacket. The first two call for balanced pressure air vents at the top of the jacket (Figure 11.9.5) but for a 'top' inlet, fit the vents near the bottom. In all cases drain the jacket with float-thermostatic traps, as shown. Thermodynamic traps are possible alternatives but additional air venting will usually be required. When the paddle is heated, drain it with a float-thermostatic trap which has a steam lock release.

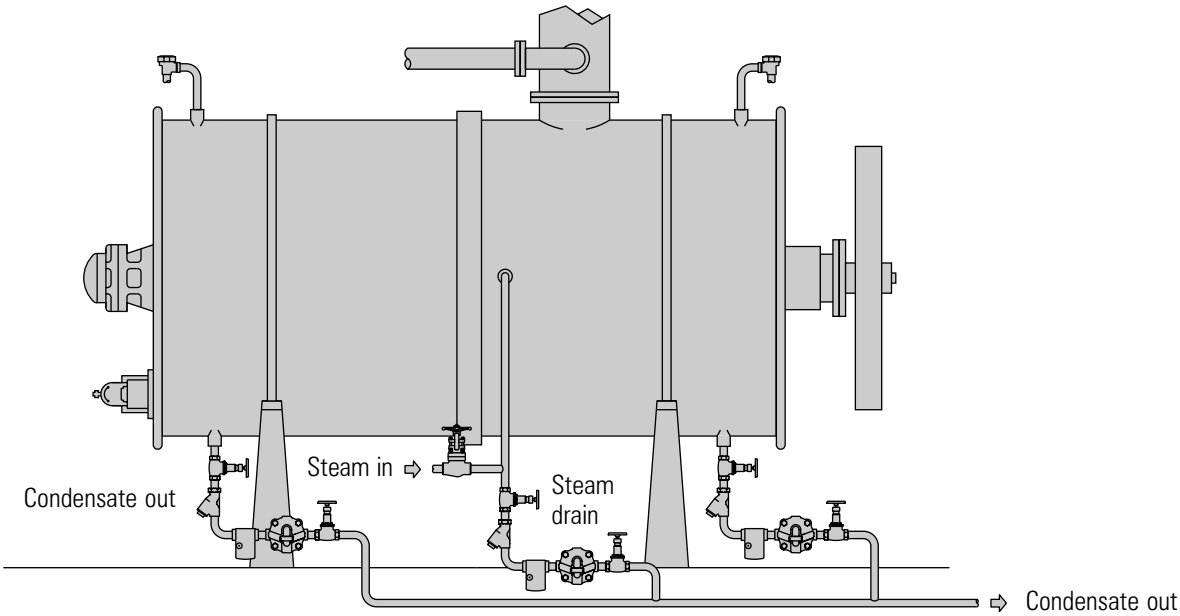


Fig. 11.9.5 An industrial digester

Hot tables and hotplates

These are used in many industries and conditions can be variable, but a typical application would be on the final drying section of a corrugating machine (Figure 11.9.6). Hotplates or steam chests can have varying pressures and condensate loads, due to variations in board thickness. Float-thermostatic or balanced pressure traps are both suited to this application, whilst thermodynamic traps also prove to be a useful alternative.

Generally, steam should not be fed from one end of the table and condensate drained at the other, as the condensate (and air) from any section has to pass through each succeeding section to get to the trap. This will result in longer heat-up times and reduced temperatures on the end sections. An improved method is to feed and drain each section individually. Figure 11.9.6 shows balanced pressure thermostatic traps and strainers which are generally suitable for these tables.

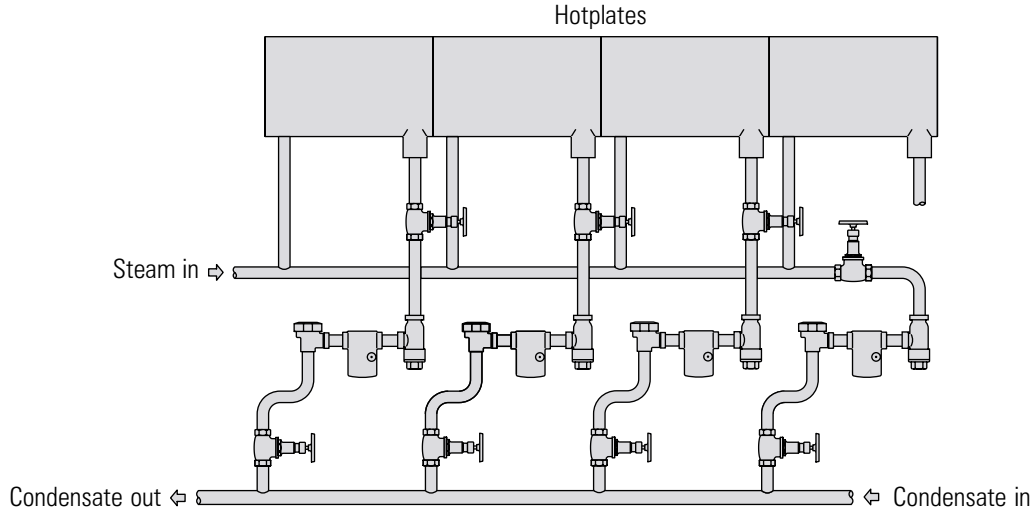


Fig. 11.9.6 Hotplates with balanced pressure trap sets

Brewing 'coppers'

These are specialised types of evaporators requiring special consideration. Steam is usually supplied from below the 'copper', and the high demand of the heater can produce a peak at the boiler plant with the possibility of priming, so a separator in the line close to the 'copper' will ensure that dry steam is available.

The base coil is best drained using a float-thermostatic trap fitted close to the outlet. The heater must be capable of the greatest possible heat transfer with a smooth output to give continuous turbulence in the copper. This calls for a high capacity trap with continuous discharge, capable of handling the heavy starting load as well as the lighter running load. The float-thermostatic trap is ideally suited to this task.

Air venting is extremely important. If the design of the heater means that all the air is discharged through the condensate outlet, additional air venting capacity will be an advantage. Using a balanced pressure air vent fitted around the trap will maximise system purging at start-up (Figure 11.9.7).

Sometimes, the inherent design of the heater will cause air to collect at some other point, in which case separate air venting will be necessary.

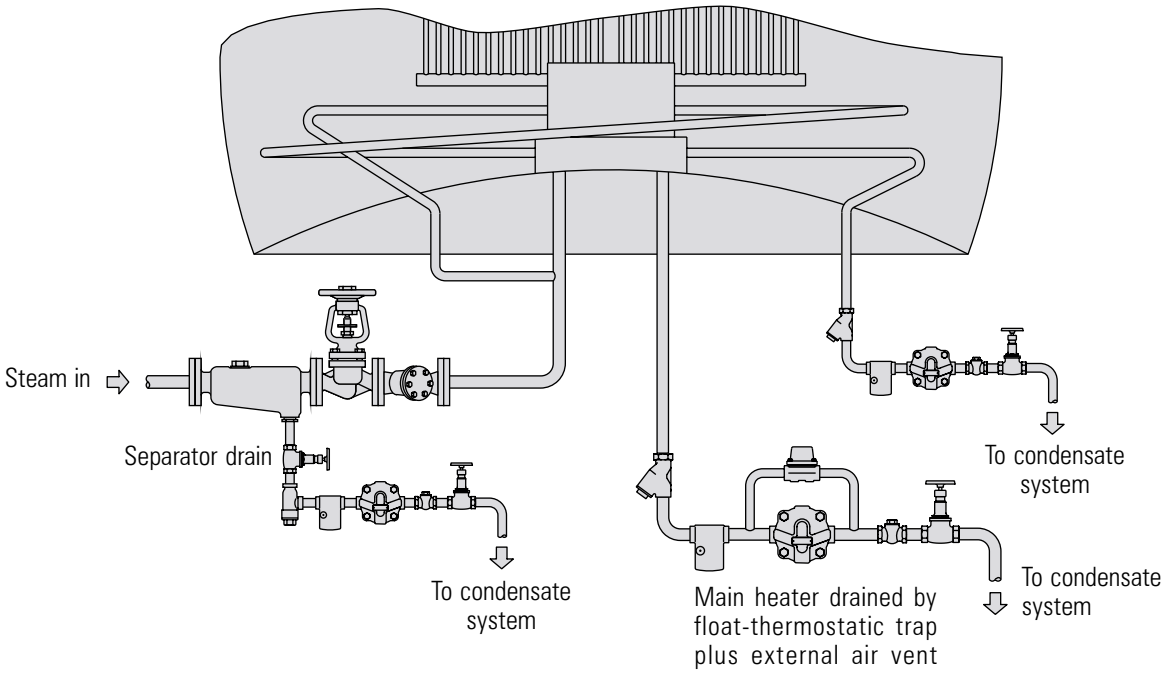


Fig. 11.9.7 Brewing 'copper'

Evaporators, calandrias and reboilers

Evaporators vary widely in design and use, but essentially include some form of heat exchanger to heat a process fluid. The steam heater is usually of the horizontal tube type shown in Figure 11.9.8. Vertical tubes are also used, and these are often arranged in a calandria or a tube basket, with steam outside the tubes. Calandrias may be within the evaporator body, or an external heater or a reboiler may be used. Similar considerations apply in all these cases. The condensing rate may be higher at 'start-up' than when boiling, but a good heat transfer rate is vital at all times. The trap must operate equally well on heavy or light loads and air must be freely discharged.

The float-thermostatic trap is the best choice, and should be fitted close to the condensate drain point. If this is not possible, use the float-thermostatic trap with steam lock release, plus, if necessary, an external air vent in a bypass.

The inverted bucket trap is an alternative when steam pressures are very high, or extreme waterhammer is present. An air vent bypass is always necessary with this arrangement.

With some heaters, output can be improved by additional air venting. The draining and air venting of multi-stage evaporators can be complicated by the fact that one or more stages may operate under vacuum, and special arrangements must be made utilising automatic pump-traps. The condensate may also be corrosive. Always seek expert advice on draining this equipment.

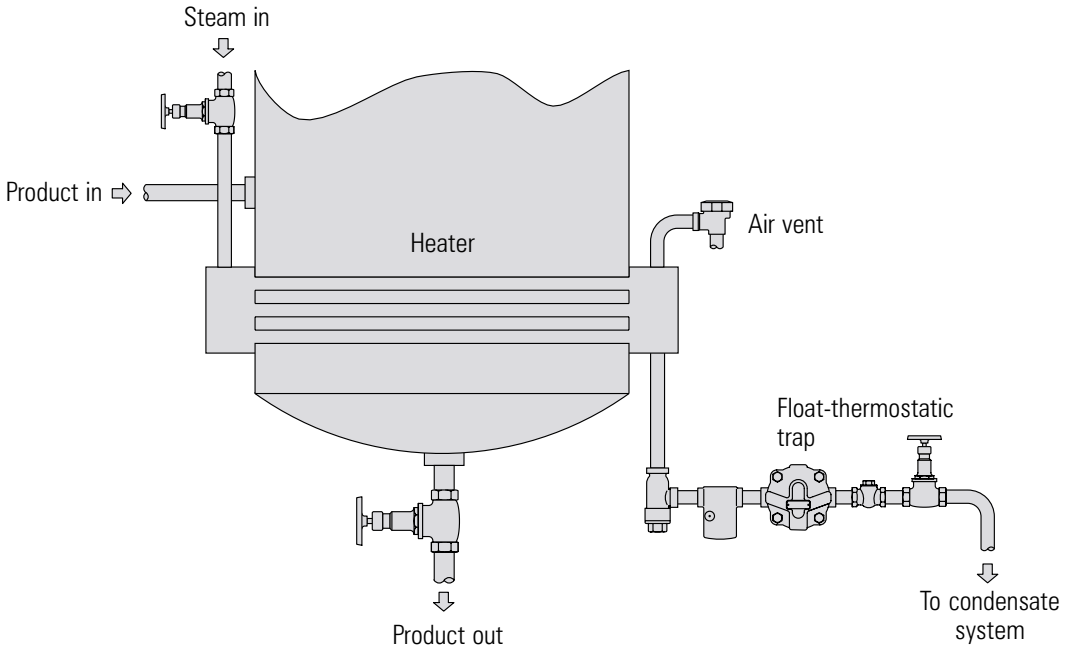


Fig. 11.9.8 Evaporator

Vulcanisers

Condensate from the chamber can become acidic, making it corrosive to some traps. A float-thermostatic trap is still the best choice, or an inverted bucket trap with a separate air vent in parallel. Whichever is chosen, it should be of stainless steel construction to provide resistance to corrosive attack. Condensate must be dumped to waste due to contamination. Trap sets serving the chamber will need to be cleaned regularly.

The entry of steam at one end of the chamber makes it necessary to have air vents at high level at the opposite end of the chamber as well as within (or around) the trap. Draining and venting the jacket is more straightforward. A float-thermostatic trap should be used, together with an additional air vent fitted as far as possible from the steam inlet.

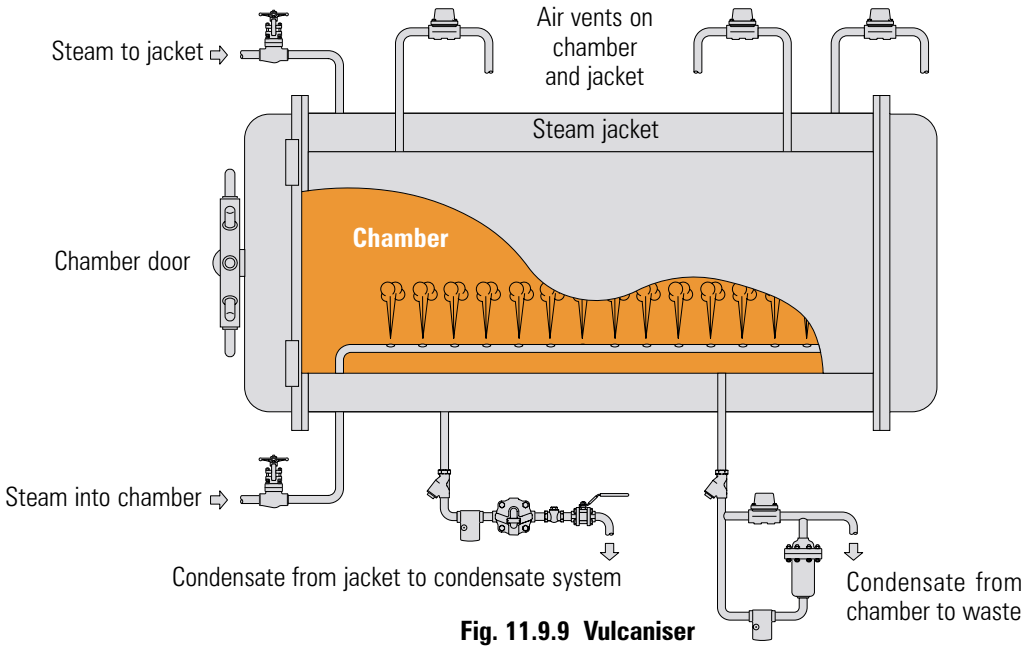


Fig. 11.9.9 Vulcaniser

Questions

1. Which of the following is critical on small jacketed pans fitted with balanced pressure traps?

a| They must be supplied with dry saturated steam ☐

b| The condensate line must rise after the trap to increase backpressure ☐

c| The trap must not be fitted with a 'near-to-steam' thermostatic capsule ☐

d| The trap must be fitted on an unlagged cooling leg ☐

2. Where should air vents be fitted on jacketed process vessels?

a| On the steam inlet to the vessel ☐

b| At the bottom of the vessel ☐

c| At a point furthest away from the steam inlet and horizontal to it ☐

d| It is not necessary to fit air vents on jacketed process vessels ☐

3. How does a tilting pan differ from a fixed pan for condensate removal?

a| Tilting pans produce less condensate ☐

b| Tilting pans can only drain condensate when in the tilting position ☐

c| Tilting pans can suffer from steam locking conditions ☐

d| There is no difference between them ☐

4. What can cause problems on large retorts and industrial autoclaves?

a| Air not venting properly from the vessel ☐

b| Wet steam supplied to the vessel ☐

c| Dry steam supplied to the vessel ☐

d| Steam locking occurring due to long condensate drain lines ☐

5. Where should automatic air vents be placed on large vessels?

a| On the steam supply line to the vessel after the control valve ☐

b| On the top or bottom of the vessel opposite the steam inlet ☐

c| Automatic air vents are not necessary on such vessels ☐

d| On the condensate manifold supplying a group trap arrangement ☐

6. Which of the following statements is true?

a| Tilting boiling pans do not generally suffer from steam locking ☐

b| Retorts should not be air vented under any circumstances ☐

c| Brewing coppers use less steam at start-up than when running ☐

d| Condensate from vulcanisers must be passed to waste ☐

Answers

1: d, 2: c, 3: c, 4: a, 5: b, 6: d

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Module 11.10

Selecting Steam Traps - Space Heating Equipment

Space Heating Equipment

A - Best choice, B - Acceptable alternative, [1] (parallel air vent), [4] (a pump/trap may be required).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Calorifiers	**A**[4]						
Heater batteries	**A**[4]						
Radiant panels and strips	**A**	B[1]	B[1]				B[1]
Radiators and convection cabinets	B			**A**	B		
Unit heaters and air batteries	**A**[4]						
Overhead pipe coils	B			**A**			B[1]

Heat exchangers draining to atmospheric pressure

The trap for this application must be able to handle a very heavy or very light load equally well, and be able to purge air quickly. The float-thermostatic trap is ideal and should always be installed below the outlet of the heat exchanger. Figure 11.10.1 shows a float-thermostatic trap with no backpressure imposed by the condensate system, such as would be found if condensate were draining to a receiver vented to atmosphere, or to a lower, non-flooded condensate return line.

Whenever the output of the heater is controlled, the effect is to reduce the pressure in the steam space, which may then become insufficient to push the condensate through the trap, and the system is said to have 'stalled'. The pressure will reduce to below atmospheric pressure (i.e. vacuum) if the secondary water temperature is controlled to below 100°C. Vacuum retains the condensate which waterlogs the heater tubes. This can cause waterhammer, poor temperature control and, in most cases, eventual corrosion of the heater elements.

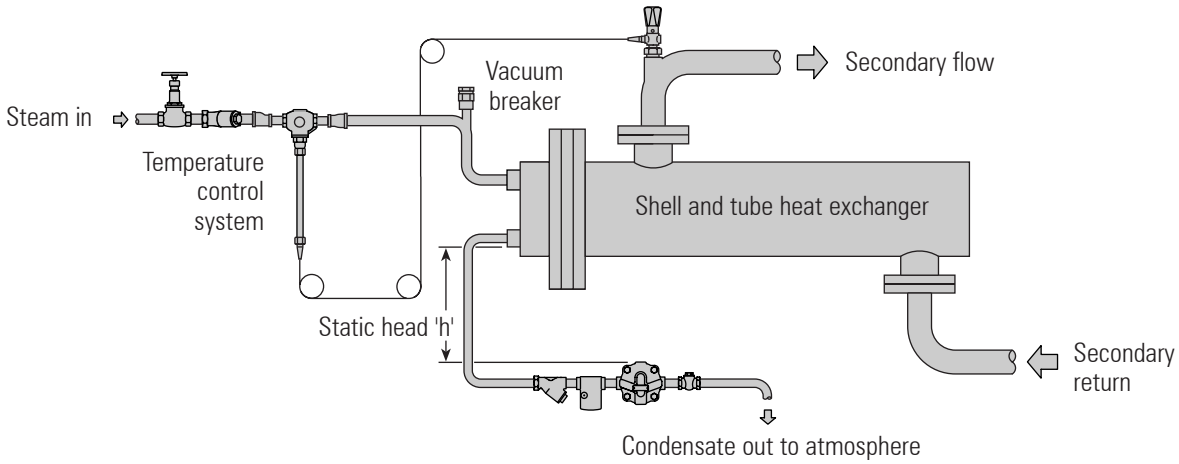


Fig. 11.10.1 Shell and tube heat exchanger with float-thermostatic steam trap

On smaller heat exchangers which drain to atmosphere, a simple remedy is to install a vacuum breaker on the steam inlet to the heat exchanger (see Figure 11.10.1). When vacuum occurs in the steam space, the vacuum breaker opens to allow the condensate to drain down to the steam trap.

The trap itself must be placed below the exchanger outlet, and must be sized to pass the condensate stall load on the static head 'h' (created by the height of the outlet above the trap inlet). The condensate pipe from the trap should slope downwards so that no further backpressure is exerted on the trap.

Heat exchangers draining to a positive pressure

Often, and especially on larger plant, it is usually preferable not to introduce air into the steam space, and the use of a vacuum breaker may not be tolerated. Also, if the condensate lifts after the steam trap up to a higher level, a vacuum breaker cannot assist drainage. In these situations, a pump-trap or pump/trap combination should be used.

If stall is inevitable and a vacuum breaker cannot be used, an active method of condensate removal must be used to give good system performance. A pump-trap (as shown in Figure 11.10.2), will perform as a steam trap if there is sufficient steam pressure in the steam space to overcome the backpressure. If there is not, it will act as a pump. The device is fully self-contained and automatic in its operation.

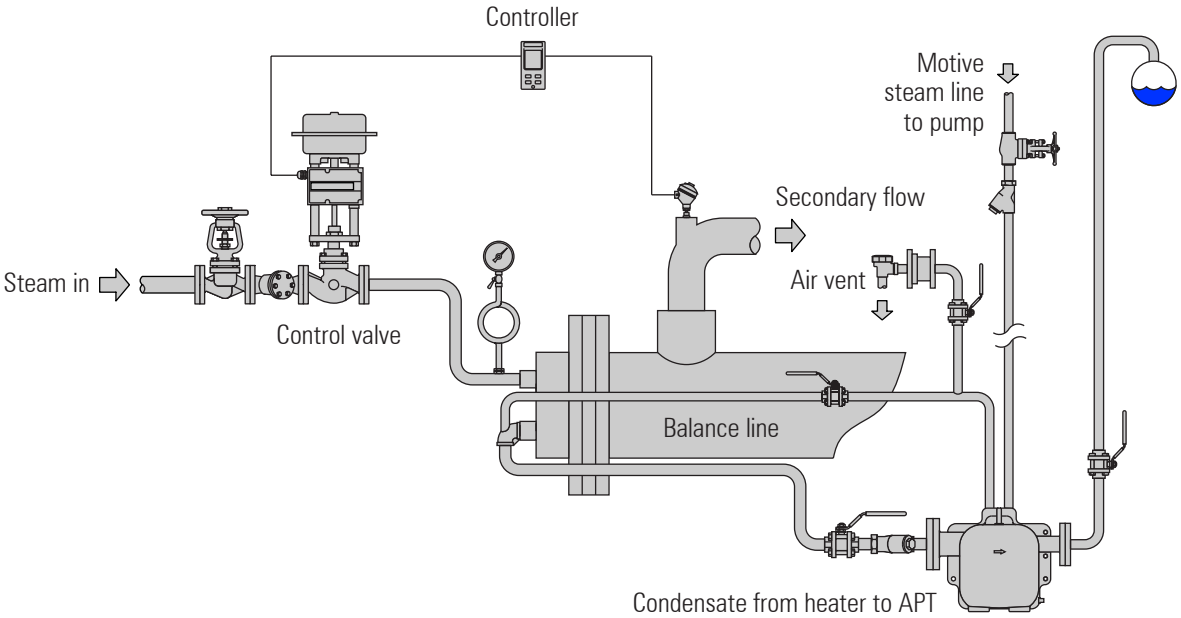


Fig. 11.10.2 Shell and tube heat exchanger with pump-trap arrangement

The pump-trap is also extremely useful where restricted space exists below the heater, for example on air handling units which are often positioned close to the plant room floor. Figure 11.10.3 shows an example draining single and multi-heater batteries to avoid both freezing and corrosion of the coils.

When a pump-trap arrangement is used, condensate will always be removed from the heater under all pressure conditions, ensuring maximum system efficiency at all times, with no escape of flash steam in the plant room.

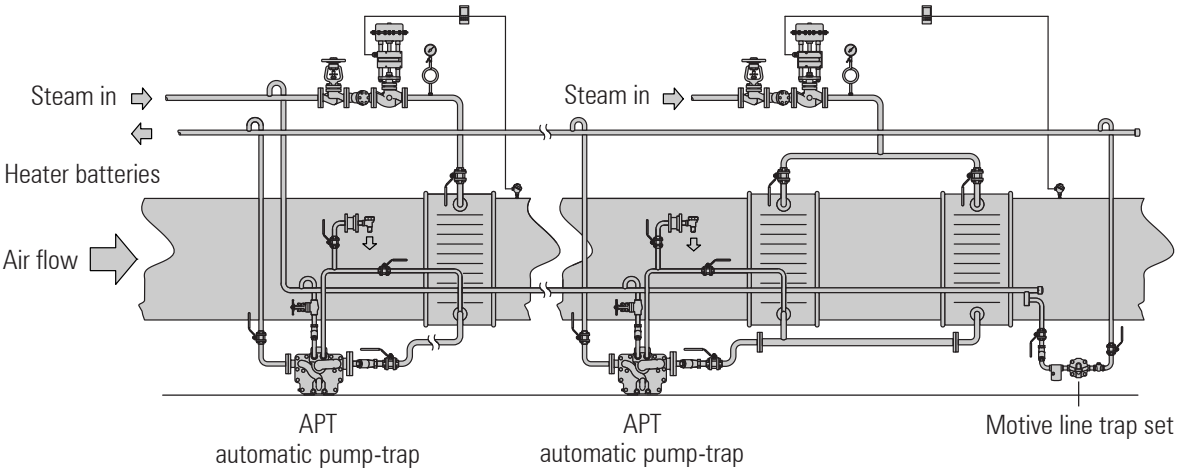


Fig. 11.10.3 Automatic pump traps on heater batteries with low suction heads

Where plant capacity is too large for the pump-trap, it can be replaced by a separate pump and steam trap in combination, such as that shown in Figure 11.10.4. A pressure powered pump is dedicated to a single heater, connected so that the pump chamber, piping, and the steam side of the heater tubes form a common steam space. When the steam pressure is sufficiently high, condensate flows from the steam space and through the pump body and steam trap into the condensate system. When the pressure is lowered as the control valve throttles, condensate fills the pump chamber till full. When the pump chamber is full, a mechanism triggers allowing 'motive' steam to enter the chamber. This pushes condensate out of the chamber and away through the trap.

The pump exhaust line is connected to a reservoir and acts as a balance pipe when the pump is filling. The small amount of exhaust steam is then contained within the system, and pumping occurs with no waste of steam to atmosphere. The system will be energy efficient, and the plant room will be free from flash steam.

If it can be guaranteed that the condensate pressure will always be higher than the steam pressure in the steam space, a trap does not need to be installed with the pump.

Further details on the subject of condensate drainage from temperature controlled heat exchangers can be found in Block 13, 'Condensate Removal'.

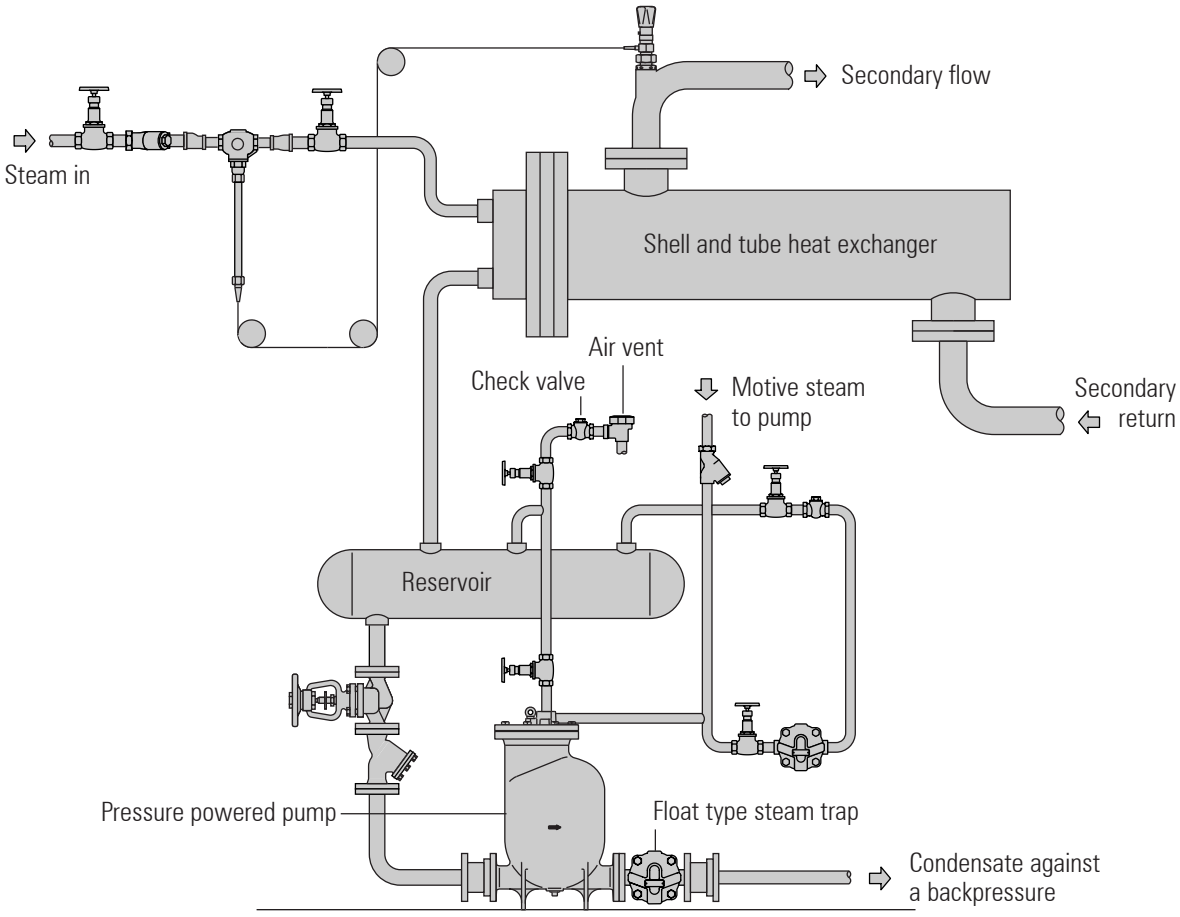


Fig. 11.10.4 Shell and tube heat exchanger with pump and trap arrangement

Radiant panels and strips

Heat output depends on high surface temperature, consequently prompt condensate removal is vital. Best results are achieved by trapping each panel individually with a float trap which handles air and condensate quickly (Figure 11.10.5). Grouping two similar panels to one trap is often satisfactory. Thermodynamic or inverted bucket traps can also be used, but supplementary air vents may be necessary.

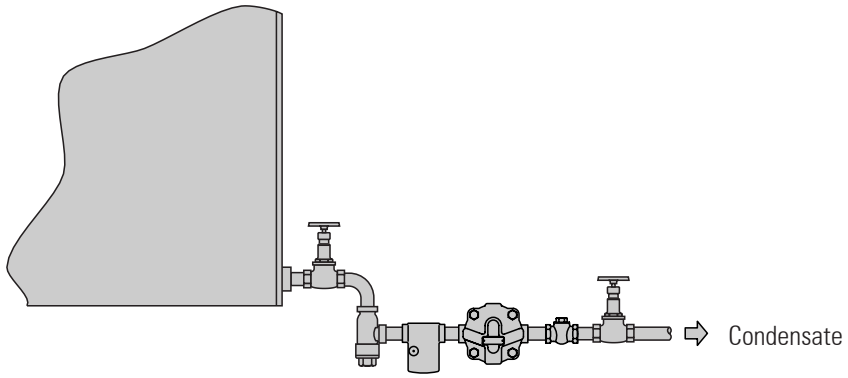


Fig. 11.10.5 Radient panel with float-thermostatic steam trap set

Steam radiators

For the standard type of steam radiator which normally operates at pressures below 2 bar g, a balanced pressure thermostatic steam trap, with union inlet may be used, as shown in Figure 11.10.6. A strainer may not be needed as the radiator collects dirt and can be blown through once a year after temporarily removing the trap capsule. When replacing the capsule, it is useful to ensure the valve and seat faces are clean.

If, however, it is preferred to incorporate a strainer, a balanced pressure trap with strainer is a useful alternative (Figure 11.10.7). In some installations, this type of heater is used in conjunction with a vacuum return system, in which case a special sub-cooled capsule is available.

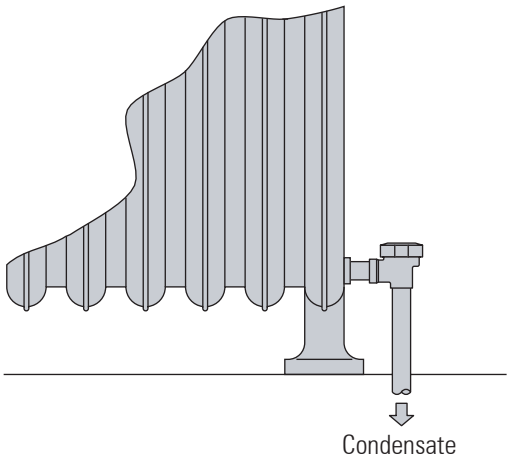


Fig. 11.10.6 Steam radiator

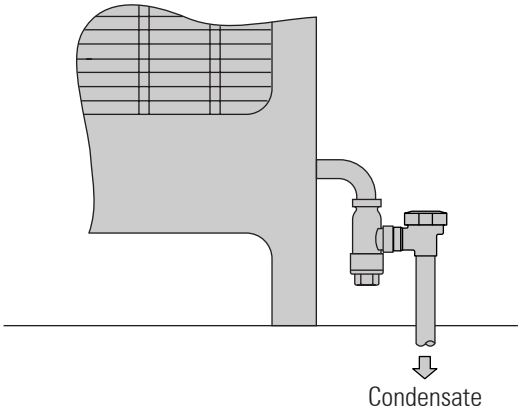


Fig. 11.10.7 Steam convector

Convection cabinet fan heaters

Although these heaters have a small steam space and condensate must not be allowed to build up, design factors call for a neat layout. A balanced pressure trap can provide this, as shown in Figure 11.10.8. If, however, the cabinet is of the forced draught design (with inbuilt fan), the higher duty requires that the steam space should be kept clear of condensate and air. A float trap is ideal but fitting it neatly inside the cabinet may present a problem. A satisfactory alternative is a balanced pressure trap, as Figure 11.10.8 illustrates, to allow a maximum length of cooling leg.

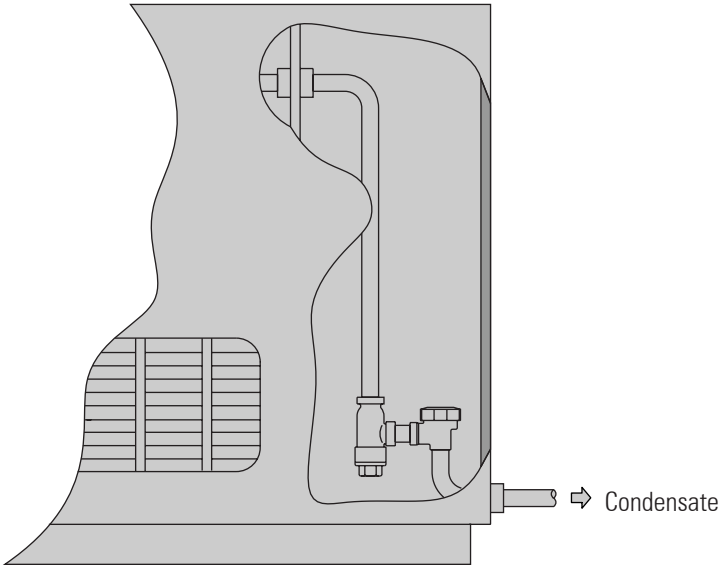


Fig. 11.10.8 Convection cabinet fan heater with balanced pressure trap

Unit heaters and air heater batteries

Unit heaters and air heater batteries produce a lot of condensate from a small steam space. Any accumulation of condensate or air produces uneven temperatures or cold air and may eventually damage the heater battery. Use a small float-thermostatic trap close to the inlet (Figure 11.10.9).

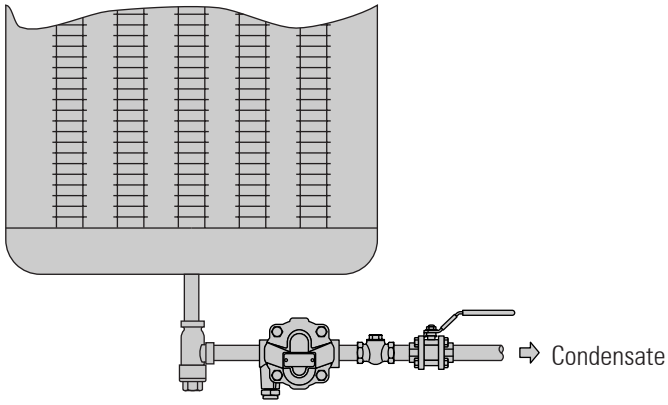


Fig. 11.10.9 Unit heater with float trap

With horizontal batteries such as those used in down-draught heaters, any reduction in the condensate outlet pipe must be made using an eccentric reducer. This will stop condensate backing up in the coils. The trap should be fitted below the outlet as in Figure 11.10.10. Condensate clearance can be improved by fitting the heater battery with a slight fall towards the outlet end.

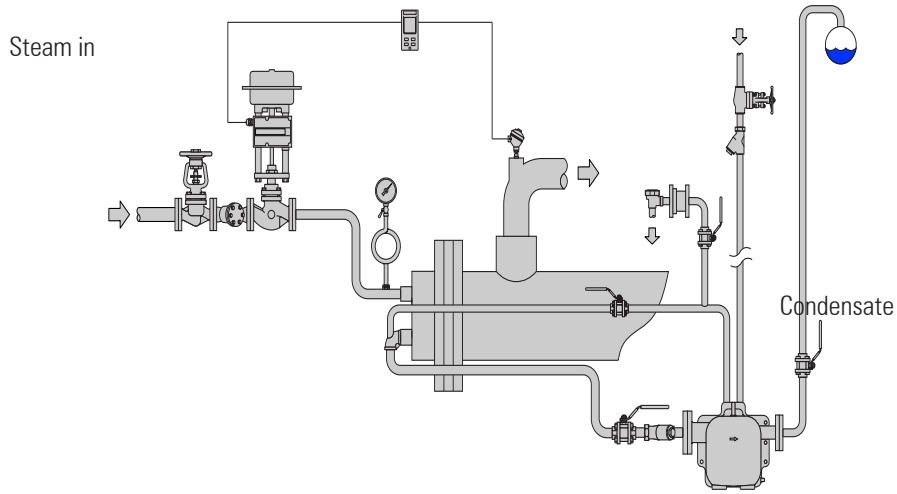


Fig. 11.10.10 Down-draught heater with float trap

Where a number of vertical heater batteries are installed in series with the air flow, successive sections do progressively less work and produce progressively less condensate. Each section should be drained separately with a float trap (Figure 11.10.11). If a float trap is not used, the inverted bucket trap is a possible alternative, but with an air vent fitted in parallel.

When higher pressure steam is used in a multi-heater bank system, savings can be achieved by collecting the condensate, separating the flash steam and using it to heat the first heater section in the bank. When the heater batteries are temperature controlled, stall conditions can occur in the steam spaces preventing efficient condensate removal. A Spirax Sarco vacuum breaker should be fitted to the pipework between the control valve and the heater battery inlet, and the condensate pipework must be allowed to fall to a collecting point i.e. a receiver vented to atmosphere. The float trap must be sized on the stall load. The subject of stall is considered in detail in Block 12.

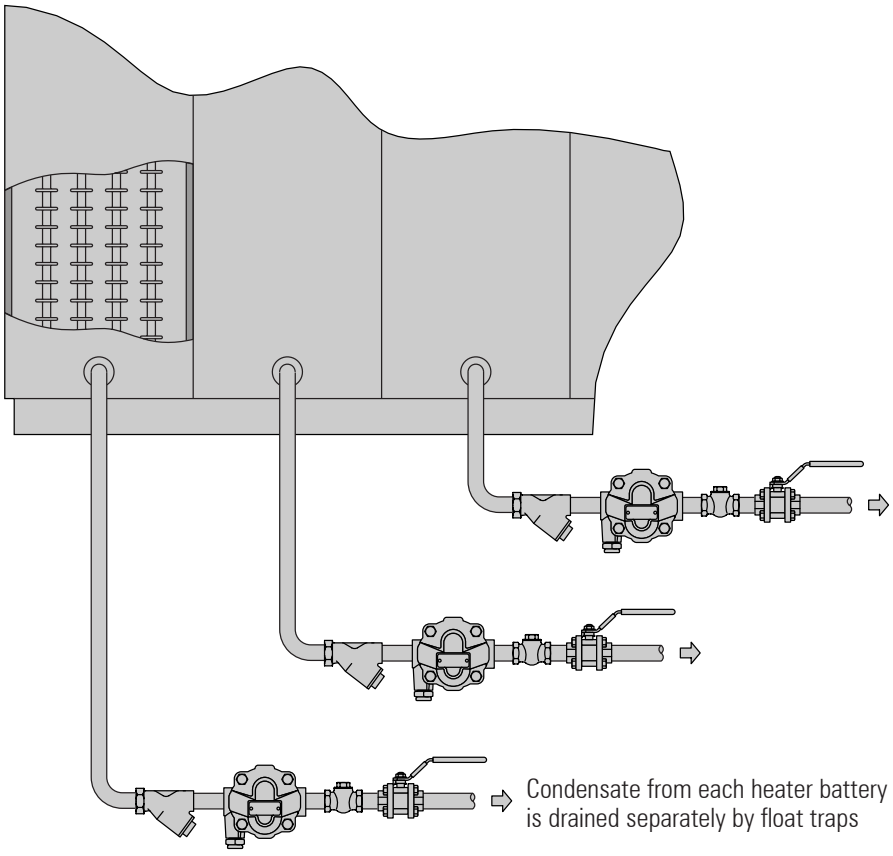


Fig. 11.10.11 Multi-bank heater batteries with float traps

Overhead pipe coils

Long overhead heating pipes, like industrial drying coils, will produce waterhammer if insufficient attention is given to installation. Heat will circulate slowly and temperature control will be difficult. Relaying the pipework as in Figure 11.10.12, using balanced pressure traps with stainless steel capsules, or with float or inverted bucket traps will eliminate these problems. With inverted bucket traps, warm-up speed can be greatly improved by fitting separate air vents, especially on the end of the coil (Figure 11.10.13).

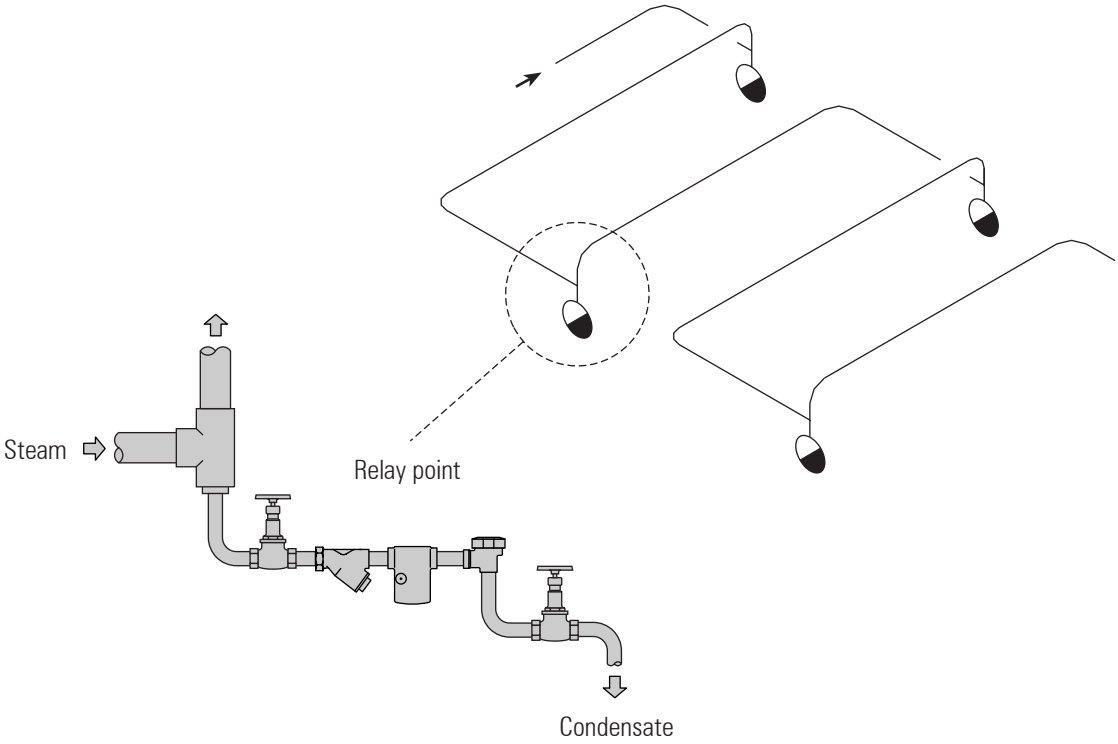


Fig. 11.10.12 Overhead pipe coil

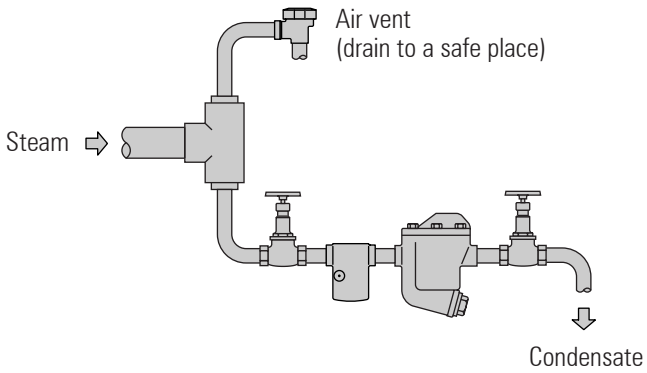


Fig. 11.10.13 Inverted bucket trap with air vent

Questions

1. If vacuum occurs in a temperature controlled plant...

a | The plant must be supplied with higher pressure steam ☐
b | Pump-traps can be fitted to ensure proper condensate drainage ☐
c | A vacuum breaker must always be fitted to the steam trap inlet pipework ☐
d | Vacuum cannot occur in any steam supplied plant ☐

2. If a pump and trap are used in combination to drain a temperature controlled heat exchanger...

a | The trap must be fitted close-coupled to the exchanger outlet ☐
b | The pump and trap must be the same size ☐
c | The trap must be fitted to the pump outlet ☐
d | The trap must be fitted to the pump inlet ☐

3. A heat exchanger has atmospheric backpressure at the trap outlet. If stall conditions occur, which of the following applies?

a | A pressure powered pump need not be fitted ☐
b | A vacuum breaker should be installed on the steam inlet pipe ☐
c | A float trap can be sized on the static head pressure available above it ☐
d | All of the above ☐

4. If a pump-trap is used to drain a heater battery...

a | A vacuum breaker should be fitted to the battery inlet pipe ☐
b | A vacuum breaker should not be fitted to the battery inlet pipe ☐
c | The pump-trap must be close-coupled to the battery outlet ☐
d | A vacuum breaker must be fitted to the battery outlet pipe ☐

5. If backpressure will always be higher than the steam space pressure...

a | A pump-trap must be fitted ☐
b | A pump and trap combination must be fitted ☐
c | A pump only will need to be fitted ☐
d | The steam pressure before the control valve must be increased ☐

6. Which of the following statements is true? Stall cannot occur if...

a | The control valve is oversized ☐
b | Condensate drains down to a vented receiver ☐
c | The set point is higher than 100ºC ☐
d | The steam space pressure is always greater than the backpressure ☐

Answers

1: b, 2: c, 3: d, 4: b, 5: c, 6: d

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Module 11.11

Selecting Steam Traps - Steam Mains; Tanks and Vats; Pressure Reducing Valves

Steam Mains

A - Best choice, B - Acceptable alternative,
[1] (parallel air vent), [3] (with cooling leg), [5] (near-to-steam capsule), [6] (anti-air-binding disc).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Pressure reducing valve station	**A**			B[5]			
Horizontal runs	B		**A**				B
Shutdown drain (frost protection)				B[3]	B	**A**	
Separators	**A**		B				B
Steam header drainage	**A**		B[6]				B
Terminal ends	B		**A**[1]				B[1]

Steam mains

Steam mains carry water droplets in suspension in the steam, as well as a layer of condensate and air on the wall of the pipe. Both the air and water must be removed for maximum plant output. Steam traps should discharge into adequately sized condensate lines, falling towards a vented receiver. Because condensate return lines often run alongside steam mains, there is a temptation to connect into them the discharges from the traps draining the main. If the condensate returns are flooded, as they often are, severe waterhammer will result. This is undesirable if the traps are of the blast discharge type, and the practice of discharging into flooded lines should be avoided to deter waterhammer.

The condensate loads associated with mains drainage are relatively small hence a low capacity thermodynamic trap is more suitable. Thermodynamic traps are very robust and offer long life and efficient operation in exposed conditions.

Horizontal runs

Horizontal runs must not be drained through a small pipe connection in the bottom of the pipe. Use a properly sized pocket into which fast moving condensate can fall - as shown in Figure 11.11.1.

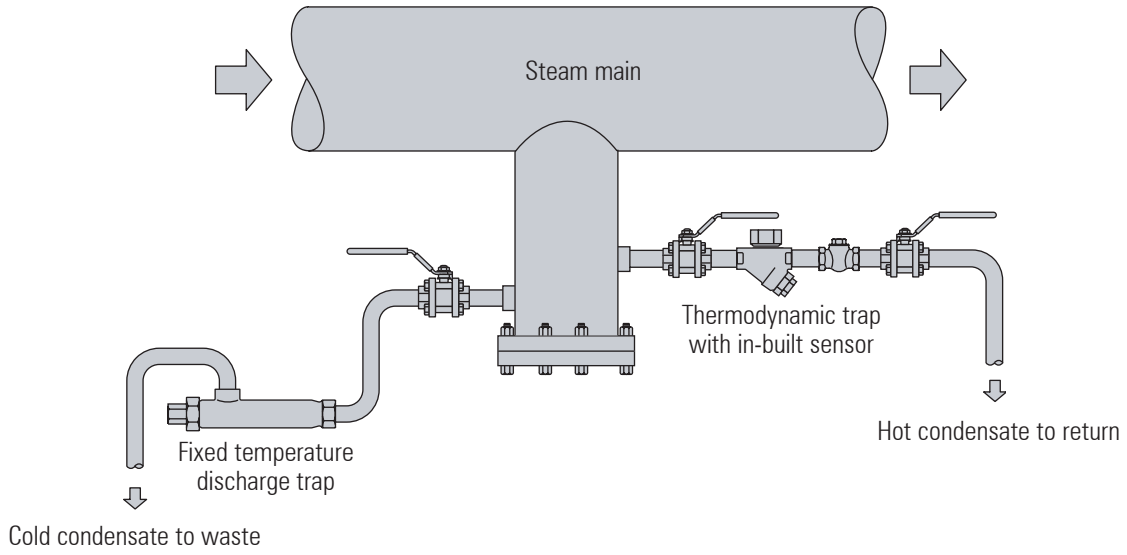


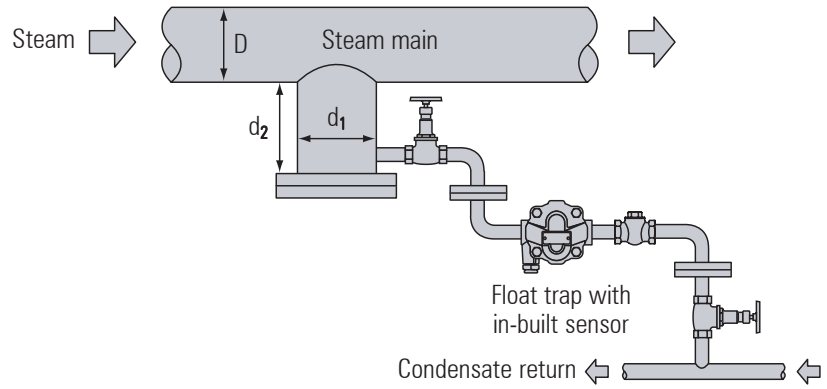
Fig. 11.11.1 Mains drainage with parallel shutdown drain

Drain pocket dimensions

Typical recommended drain pocket dimensions, relative to steam main pipe sizes are given in Table 11.11.1.

Table 11.11.1 Drain pocket dimensions

Mains diameter - D	Pocket diameter - d_1	Pocket depth - d_2
Up to 100 mm nb	$d_1 = D$	Minimum $d_2 = 100$ mm
125 - 200 mm nb	$d_1 = 100$ mm	Minimum $d_2 = 150$ mm
250 mm and above	$d_1 \geq D/2$	Minimum $d_2 = D$



Separators

Separators are normally fitted line size. A separator will remove the suspended droplets as well as the condensate layer and provide drier steam for heating and processes (Figure 11.11.2). As it is essential to clear condensate as it forms, the first choice is a float-thermostatic trap. Alternatively, the inverted bucket trap could be used with a separate air vent as in Figure 11.11.4. The third alternative, the thermodynamic trap, is ideal for outside mains in exposed conditions, as it will not be damaged by freezing.

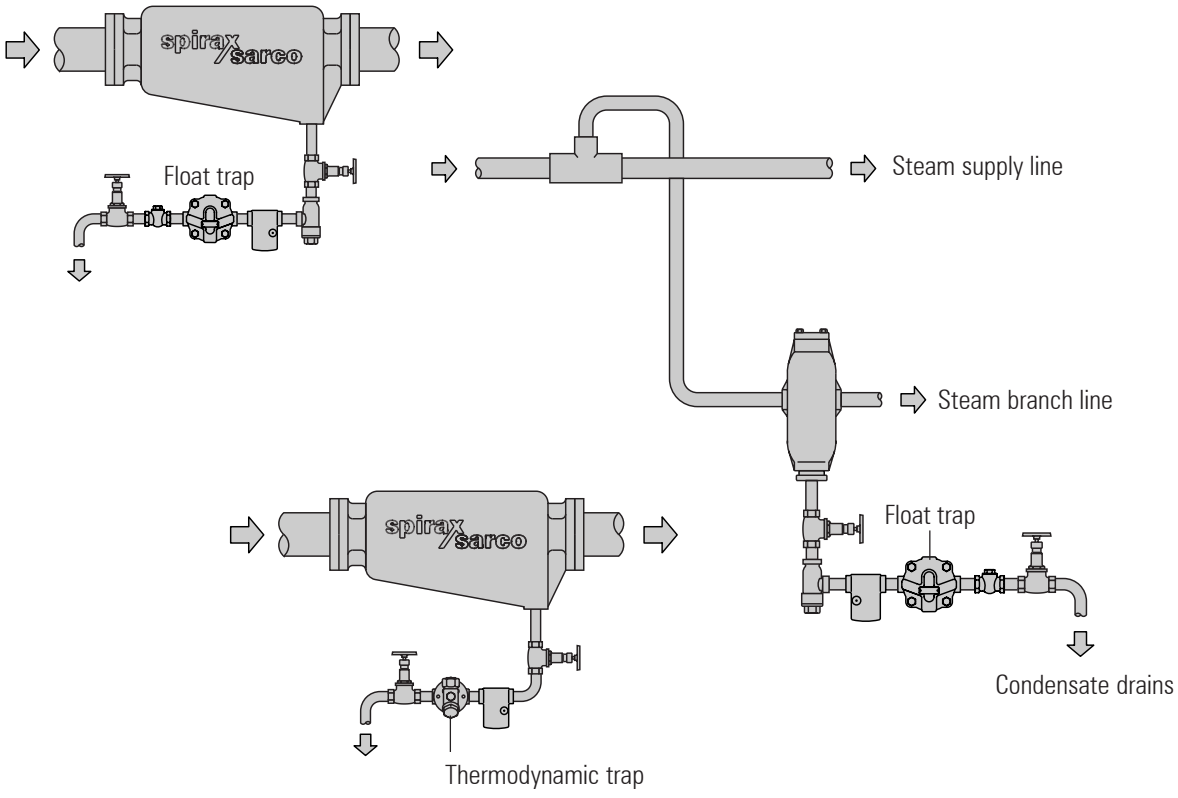


Fig. 11.11.2 Various separator configurations

Steam header drainage

Steam headers should be drained in a similar way to steam mains, with a pocket suitably placed along the bottom of the manifold. A slight fall towards the end which houses the drain pocket assists drainage. Headers longer than 5 m may benefit from a drain pocket at either end. Float traps are best suited to handling fluctuating condensate loads. If headers situated close to boilers are susceptible to carryover, thermodynamic traps with anti-air-binding discs are good alternatives.
Note: The drain pocket should be sized as per Table 11.11.1. The distribution header diameter should be sized on a steam velocity of 10-15 m/s, for the maximum incoming steam load.

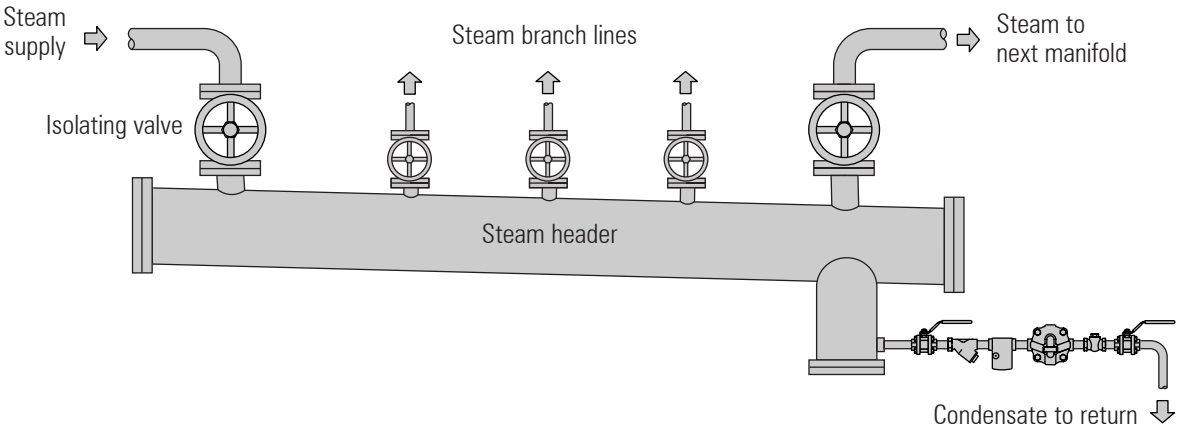


Fig. 11.11.3 Typical steam header with drain pocket and float-thermostatic trap set

Terminal ends

Terminal or 'dead' ends are inherently more susceptible to waterhammer than horizontal runs because of their position in the pipework. Air will also tend to collect at these positions at start-up as steam will push any air in its path to the furthest point in the system. It is sensible therefore to position a steam trap and air vent here.

A 'Tee' piece, shown in Figure 11.11.4, will help to dissipate any mechanical forces caused by waterhammer, thus helping to protect the trap and vent from mechanical damage, whilst offering a simple way to install them.

The best trap for this is the thermodynamic type due to its robust design, but a good alternative is an inverted bucket should this be preferred. Both will require an air vent, for the reasons stated above.

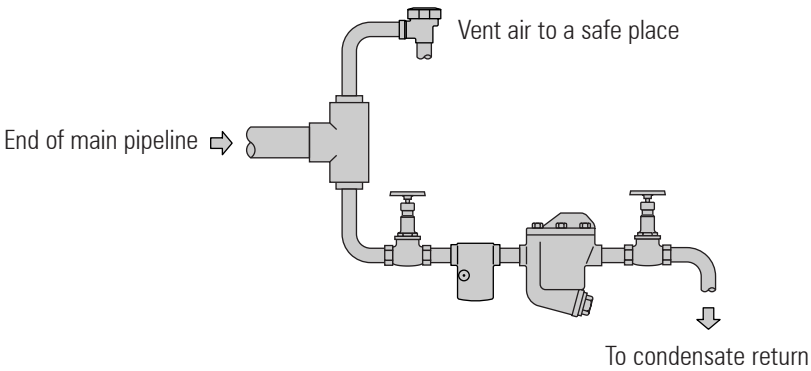


Fig. 11.11.4 Terminal end with inverted bucket trap and air vent

Air venting

Venting the end of the main, as shown in Figure 11.11.4, will provide quicker heating-up and faster production - further details are given in Module 11.12, 'Air Venting Theory'. On a long main, or one which is started up daily, it may also be necessary to fit air vents at certain intermediate drain points. The discharge from an air vent should not be connected into a flooded condensate return line (as waterhammer may result), nor into a line carrying sub-cooled condensate (since this can encourage corrosion of the pipework).

Branch mains to process

Optimum heat transfer will be obtained from any process when it is fed with dry steam. The branch line should be taken from the top of the main, and where it is relatively long or convoluted, the line should be well insulated and fitted with a small separator and trap set before the plant inlet. Figure 11.11.2, shows the arrangement where the separator is drained by a float-thermostatic trap.

Any process having a temperature controlled steam supply would benefit from having a drain trap set situated immediately before the control valve. This will drain the line of condensate when the control valve is shut, preventing waterhammer damage and erosion of the valve seat by wet steam upon opening. The ultimate benefits are to increase the working life and performance of the valve and process. Again, if there is likelihood of wet steam at the end of the branch line, it is better to fit a separator.

Tanks and Vats

A - Best choice, B - Acceptable alternative, [5] (near-to-steam capsule), [6] (anti-air-binding disc).

Application	Ball float-thermostatic	Ball float FT-C	Thermodynamic	Balanced pressure	Bimetallic	Liquid expansion	Inverted bucket
Process vats (rising discharge pipe)	**A**	B	B	B[5]			B
Process vats (discharge pipe at base)	**A**		B[6]	B[5]			
Small coil heated tanks (quick boiling)	**A**			B[5]			
Small coil heated tanks (slow boiling)					B	**A**	

Process vats (rising discharge pipe)

Figure 11.11.5 is most important. A coil in a process liquor vat should have a fall, and finish in a 'U' seal if the outlet rises. The rising pipe must be of small diameter. By placing a small pipe down to the bottom of the seal, and closing the pipe at the top with a convenient coupling, steam locking is prevented. The steam trap can be a float-thermostatic, thermodynamic or a balanced pressure type. A thermodynamic trap can sometimes prove useful in the case of certain corrosive liquors if the coil leaks, because it is less affected by corrosion than the other types. Should there be fear of contamination of the condensate by the tank contents, allow the condensate to drain to waste. Any condensate from corrosive liquors should be carefully disposed of, particularly if there is a fear that the tank contents could contaminate the steam and condensate system. A vacuum breaker should be fitted on the steam inlet side of the coil if the tank content is corrosive, to remove the possibility of corrosive liquor being drawn back into the steam supply.

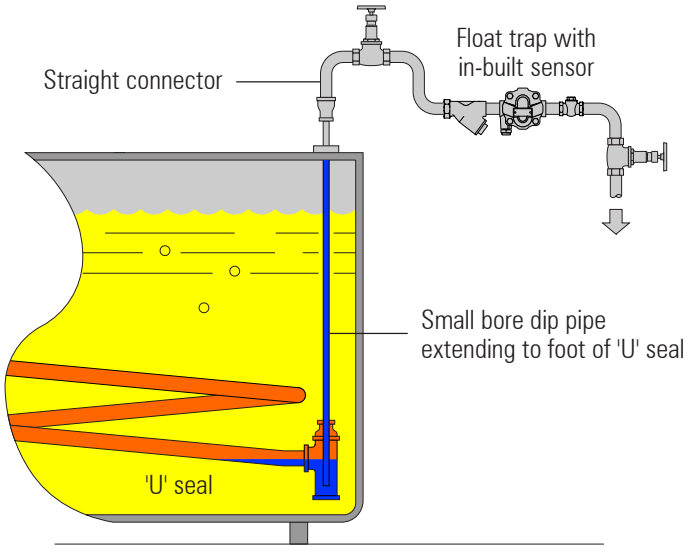


Fig. 11.11.5 Process vat with rising discharge pipe

Process vats (discharge pipe at base)

If the coil has an outlet through the side of the vat, Figure 11.11.6 shows the recommended drain arrangement using a float-thermostatic trap. Thermodynamic and balanced pressure types can also be used. It is important to use an eccentric reducer on the end of a horizontal coil, not a concentric one. A concentric reducer could cause waterlogging of the bottom part of the coil, which would reduce heat transfer, and increase the risk of waterhammer.

The system will operate better if condensate from the trap is allowed to fall to a non-flooded return line or vented receiver for pumping.

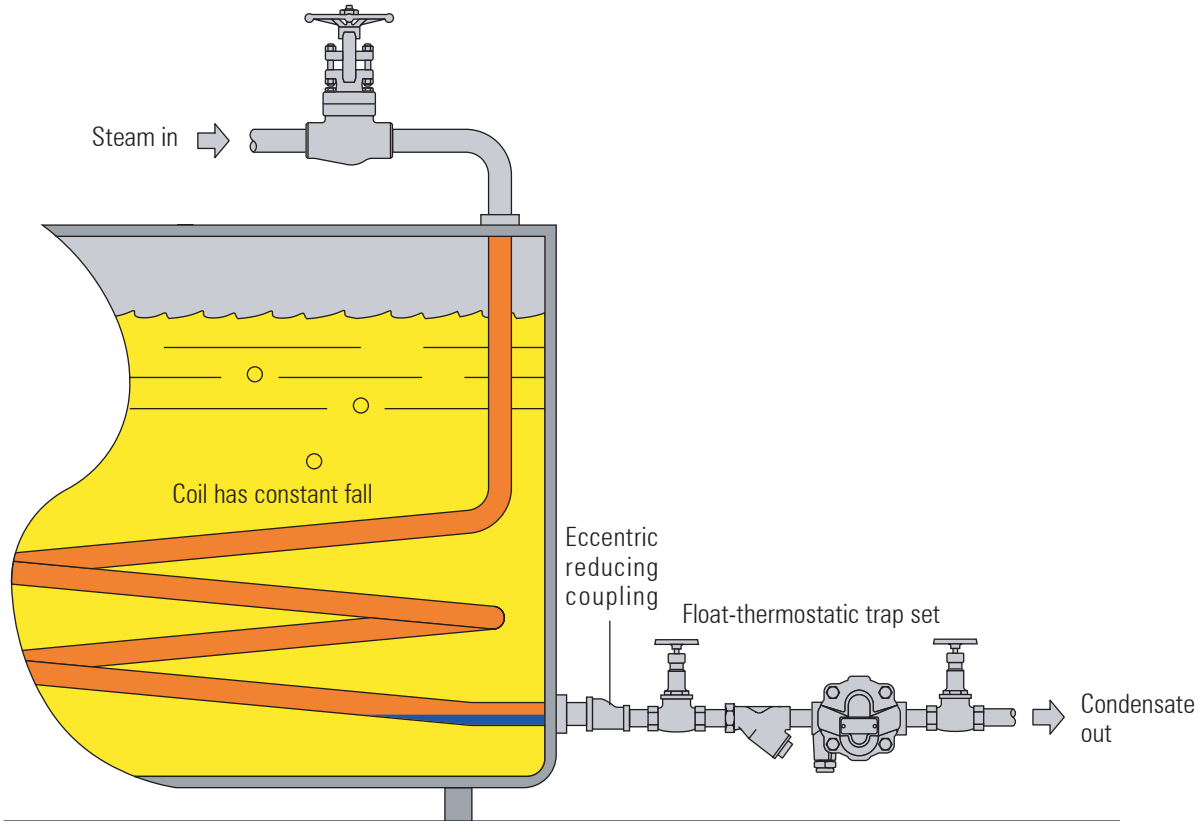


Fig. 11.11.6 Process vat with discharge pipe at the base of the tank

Pressure Reducing Valves

Where there is a possibility that the pipework downstream of reducing valves could be shut off during normal operation, a trapping point should be provided to drain any condensate formed during this period. This keeps the downstream pipework free of water and protects the reducing valve from filling with water and 'locking-up'. Float traps discharge condensate continuously and do not disturb the pressure in the pipe when discharging.

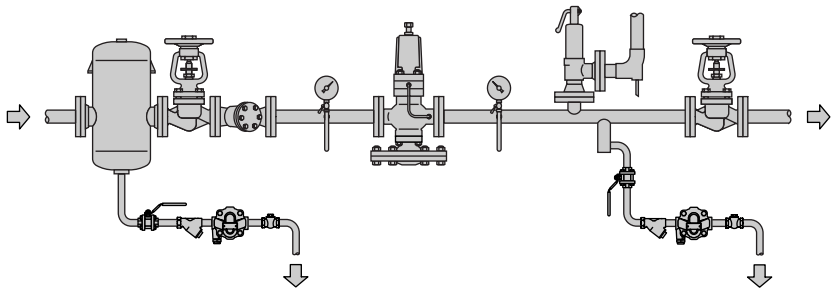


Fig. 11.11.7 Standard pressure reducing valve station

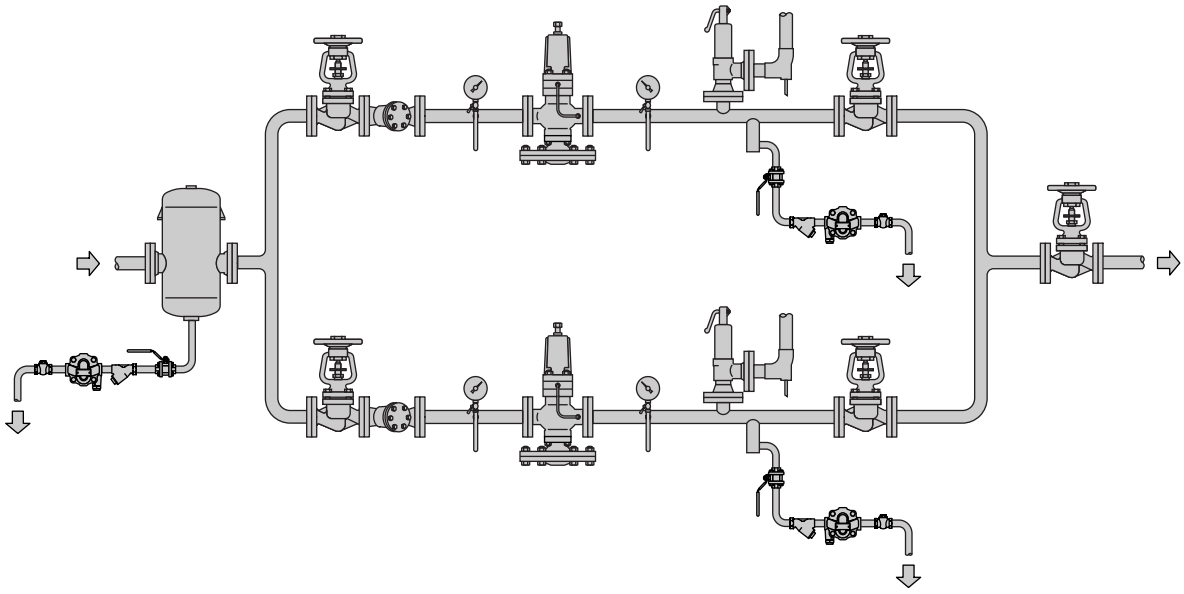


Fig. 11.11.8 Pressure reducing valves in tandem

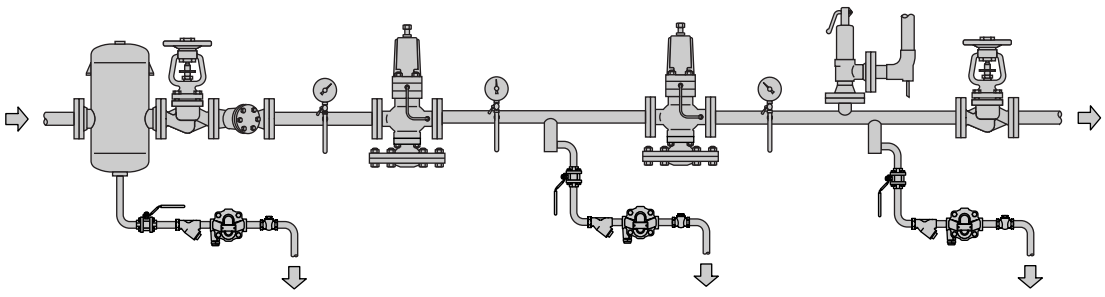


Fig. 11.11.9 Pressure reducing valves in series

Questions

1. On which parameter is a steam distribution header sized?

a | A maximum length of 5 m ☐

b | A minimum diameter of 150 mm ☐

c | An equivalent maximum steam velocity of 15 m/s ☐

d | A maximum number of off-takes ☐

2. What is the recommended diameter and depth of a drain pocket on a DN150 steam main?

a | Pocket diameter DN100: Pocket minimum depth 150 mm ☐

b | Pocket diameter DN150: Pocket minimum depth 100 mm ☐

c | Pocket diameter DN125: Pocket minimum depth 150 mm ☐

d | Pocket diameter DN100: Pocket minimum depth 100 mm ☐

3. Which extra benefit does a separator offer over a drain pocket?

a | It reduces the velocity of steam in the pipe ☐

b | It's cheaper to install than a drain pocket ☐

c | It removes suspended droplets as well as the condensate layer ☐

d | It fits in the pipe rather than under it ☐

4. A steam coil discharge pipe rising out of a tank requires a specific type of installation. What is it?

a | The rising pipe must be the same diameter as the steam coil ☐

b | A vacuum breaker must always be fitted to the steam inlet ☐

c | A pump-trap must be fitted ☐

d | The coil must be fitted with a 'U' seal to prevent steam locking ☐

5. Which steam trapping precautions should be taken with pressure reducing valve stations?

a | A trap should be fitted upstream of the pressure reducing valve station ☐

b | A trap should be fitted somewhere downstream of the pressure reducing valve station ☐

c | Drain pockets should be fitted with float type steam traps ☐

d | All of the above ☐

6. Which of the following statements is true?

a | The purpose of a separator is to prevent waterhammer ☐

b | Ideally, condensate drain lines should not connect into flooded lines ☐

c | Rising condensate lines after traps should be drained with steam traps ☐

d | Steam off-takes are taken from below steam pipes to aid drainage ☐

Answers

1: c, 2: a, 3: c, 4: d, 5: d, 6: b

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Module 11.12

Air Venting Theory

Air Venting

The effect of air

If air is mixed with steam and flows along with it, pockets of air will remain at the heat exchange surfaces where the steam condenses. Gradually, a thin layer builds up to form an insulating blanket, hindering heat transfer as shown in Figure 11.12.1. Air is widely used as an insulator because of its low conductivity (for instance, double glazing used in modern windows is simply two layers of glass with an insulating layer of air sandwiched between them). Similarly, air is used to reduce the heat loss from steam pipes. Most insulating material is made up of millions of microscopic air cells, within a matrix of fibre glass, mineral wool, or polymer-type material. The air is the insulator and the solid material simply holds it in position. Similarly, a film of air on the steam side of a heat transfer surface is resistive to the flow of heat, reducing the rate of heat transfer.

The thermal conductivity of air is 0.025 W/m °C, while the corresponding figure for water is typically 0.6 W/m °C, for iron it is about 75 W/m °C and for copper about 390 W/m °C. A film of air only 1 mm thick offers about the same resistance to heat flow as a wall of copper some 15 metres thick!

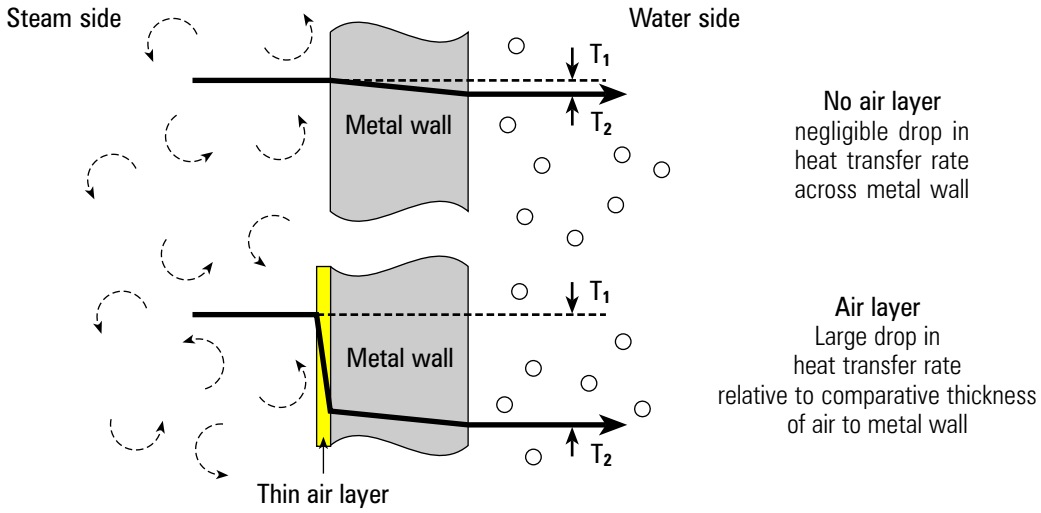


Fig. 11.12.1 Effect of air on heat transfer

It is unlikely that the air exists as an even film inside the heat exchanger. More probably, the concentration of air is higher close to the condensing surface, and lower further away. It is convenient however, to deal with it as an homogenous layer when trying to show its resistance to heat flow.

When air is added to steam, the heat content of a given volume of the mixture is lower than the same volume of pure steam, so the mix temperature is lowered.

Dalton's Law of Partial Pressures states that; 'In a mixture of steam and air, the total pressure is the sum of the partial pressure each gas would exert, when occupying the total volume on its own'.

For example, if the total pressure of a steam/air mixture at 2 bar (absolute) is made up of 3 parts steam to 1 part air by volume, then:

Partial pressure of air = ¼ x 2 bar a = 0.5 bar a

Partial pressure of steam = ¾ x 2 bar a = 1.5 bar a

Total pressure of mixture = 0.5 + 1.5 bar a = 2 bar a (1 bar g)

The pressure gauge would indicate a pressure of 1 bar g, inferring a corresponding temperature of 120°C to the observer. However, the partial pressure due to the amount of steam present in the mixture is only 0.5 bar g (1.5 bar a), contributing a temperature of only 111.6°C. Hence, the presence of air has a double effect:

- It offers a resistance to heat transfer via its layering effect,
- It reduces the temperature of the steam space thus reducing the temperature gradient across the heat transfer surface.

The overall effect is to reduce the heat transfer rate below that which may be required by a critical process, and in worst cases may even prevent a final required process temperature being reached.

In many processes, a minimum temperature is needed to achieve a chemical or physical change in a product, just as a minimum temperature is essential in a steriliser. The presence of air is particularly problematic because it will cause a pressure gauge to mislead. It follows that the temperature cannot be inferred from the pressure.

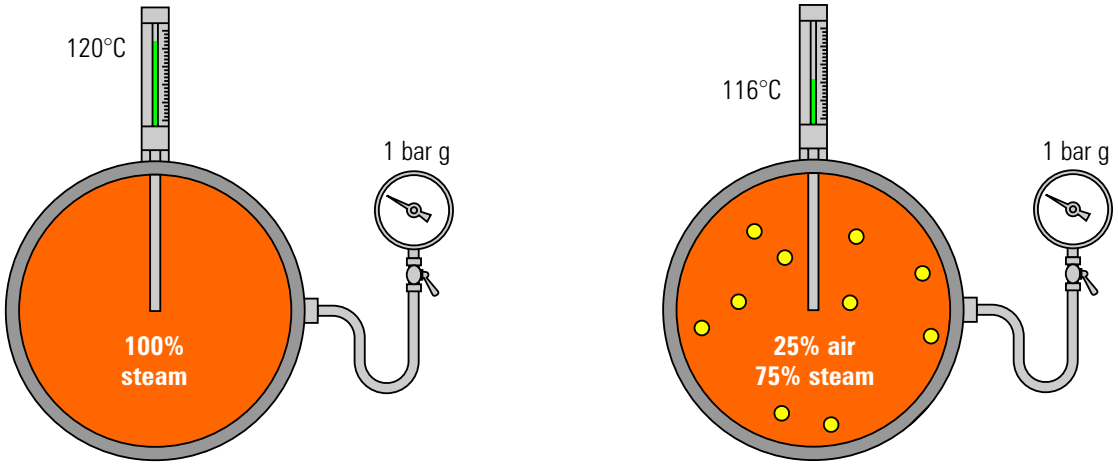


Fig. 11.12.2 Effect of air on steam temperature

Air in the system

Air is present within steam pipes and steam equipment at start-up. Even if the system were filled with pure steam when used, the condensing steam would cause a vacuum and draw air into the pipes at shutdown.

Air can also enter the system in solution in the feedwater. At 80°C, water can dissolve about 0.6% of its volume, of air. The solubility of oxygen is roughly twice that of nitrogen, so the 'air' which dissolves in water contains nearly one part of oxygen to two of nitrogen rather than the one part to four parts in atmospheric air. Carbon dioxide has a higher solubility, roughly 30 times greater than oxygen.

Boiler feedwater, and condensate exposed to the atmosphere, can readily absorb these gases. When the water is heated in the boiler, the gases are released with the steam and carried into the distribution system. Unless boiler 'make-up' water is fully demineralised and degassed, it will often contain soluble sodium carbonate from the chemical exchange of water treatment processes. The sodium carbonate can be released to some extent in the boiler and again carbon dioxide is formed.

With higher pressure boilers, the feedwater is often passed through a deaerator before it is pumped to the boiler. The best deaerators can reduce oxygen levels to 3 parts per million (ppm) in water. This residual oxygen can then be dealt with by chemical treatment. However, such an amount of oxygen will be accompanied by about 6 ppm of nitrogen, which the chemical treatment ignores. If the boiler is of a moderate size producing 10 000 kg per hour of steam, it uses about 10 000 litres per hour of water, in turn producing 60 cm^3 of nitrogen. This will cumulate over time with a significant effect on heat transfer if not removed from the system.

The best of physical and chemical treatments will still allow some untreated incondensable gas to leave the boiler with the steam. Air, frequently unsuspected, is more widespread in steam systems than believed and is the cause of both limitation of output and equipment corrosion.

Signs of air

1. A gradual fall off in the output of any steam heated equipment.

2. Air bubbles in the condensate.

3. Corrosion.

The removal of air from steam systems is paramount. The following pages address the issue by discussing the application of air vents.

Air removal

The most efficient means of air venting is with an automatic device. Air mixed with steam lowers the mix temperature. This enables a thermostatic device (based on either the balanced pressure or bimetallic principle) to vent the steam system. An air vent fitted on the steam space of a vessel (Figure 11.12.3) or at the end of a steam main (Figure 11.12.4) will open when air is present. For maximum removal of air, the discharge should be as free as possible. A pipe is often fitted to carry the discharge to a safe location, preferably not a condensate return line, which could restrict the free release of air and may also encourage corrosion.

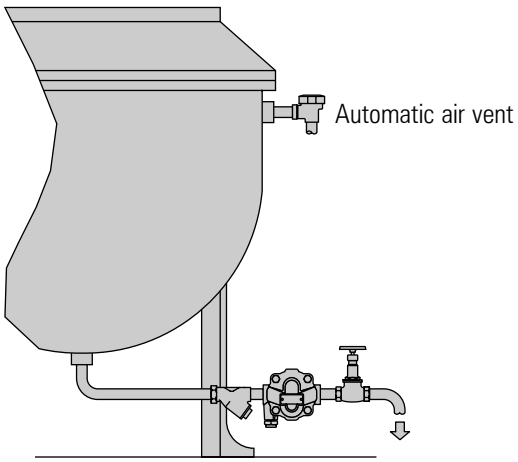


Fig. 11.12.3 Jacketed pan with an automatic air vent

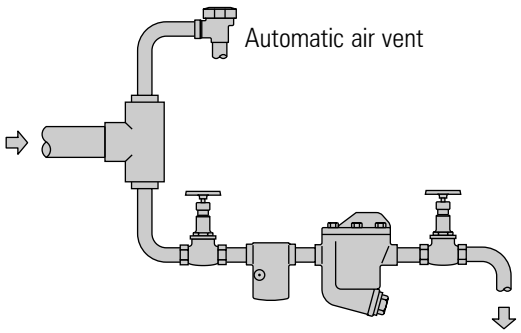


Fig. 11.12.4 End of main automatic air vent

When an air vent is fitted to bypass a steam trap (Figure 11.12.5), it will act as a steam trap after the air is vented, and may from time to time discharge condensate. In such cases it is necessary to reconnect the air vent to the condensate line after the trap.

If the condensate discharge line from a trap rises to high level, the flooded line imposes a backpressure on the trap and the air vent. The ability of the air vent to discharge air is reduced, especially at start-up. This applies equally when the air vent is incorporated within a steam trap. When the shape of the application steam space and the location of the steam inlet mean that most of the air leaves through the condensate outlet, it is preferable if discharge lines from the steam trap and air vent do not rise to high level.

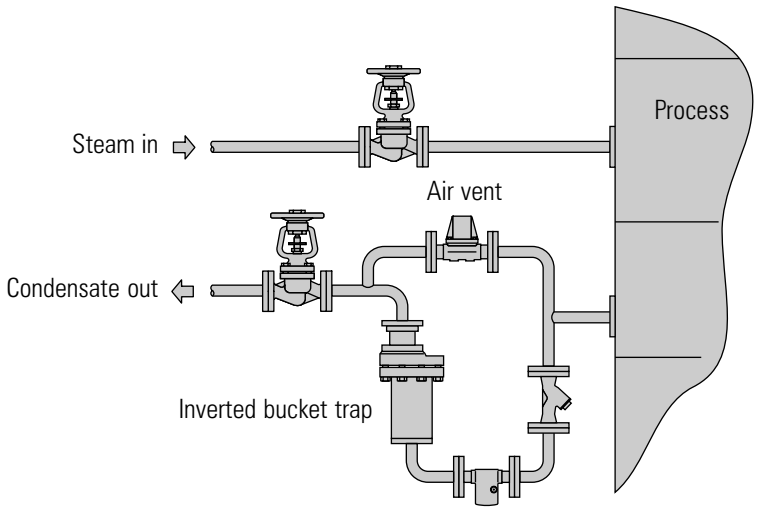


Fig. 11.12.5 Inverted bucket trap with a parallel air vent

The air vent location

When a coil or a vessel has a relatively small cross-section, the steam admitted to it will act like a piston, pushing the air to a point remote from the steam inlet. This 'remote point' is usually the best location for the air vent. In the case of a steam user of the shape shown in Figure 11.12.6, some of the air will pass through the condensate outlet, according to the provision made in the trap, or in a bypass, for handling air. The rest of the air might collect as indicated, forming a cold spot on the heating surface. The unit cannot warm up evenly, and distortion may be caused in some equipment, such as the beds of laundry ironers.

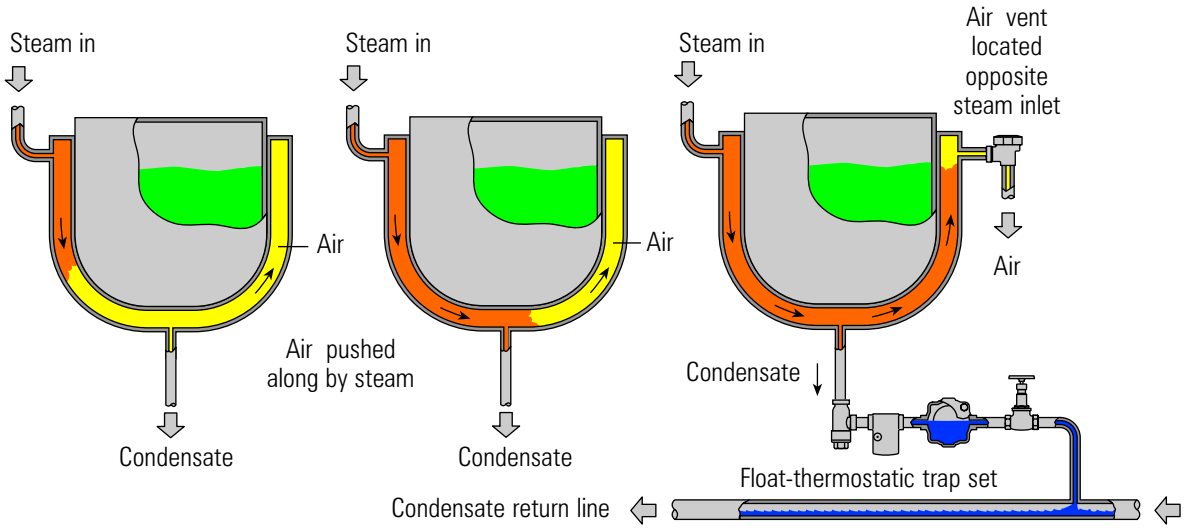


Fig. 11.12.6 Air vent located opposite the steam inlet on the jacketed pan

As an air/steam mixture is denser than pure steam at the same pressure, it is usually sufficient to provide air venting capability within the low-lying steam trap. However, the mode of operation of the trap means that condensate forms a water seal at the trap inlet sometimes preventing air from reaching the trap. There may be the need to consider an automatic air vent connected to the steam space above the level of any condensate. Often it is convenient and sufficiently effective to connect it to the top of the steam space, as in Figure 11.12.6.

However, in the case of two steam spaces of the same size and shape but with different steam inlet positions, the location of the air vent could be different. In Figure 11.12.7 and Figure 11.12.8, condensate drains from the bottom of the vessel but with the bottom steam inlet, at start-up, air would tend to be pushed to the remote point which is at the top. It may be best to locate an air vent at the top whilst a float-thermostatic steam trap will handle any residual air which has collected at the bottom of the vessel.

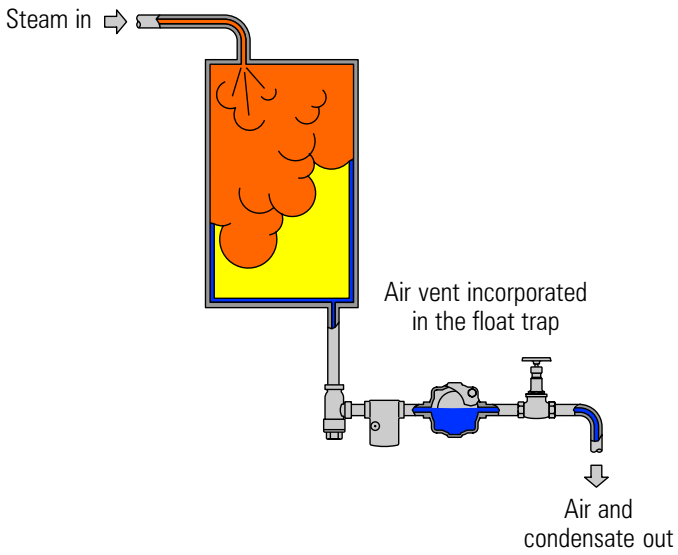


Fig. 11.12.7 Air vent located opposite low level steam inlet

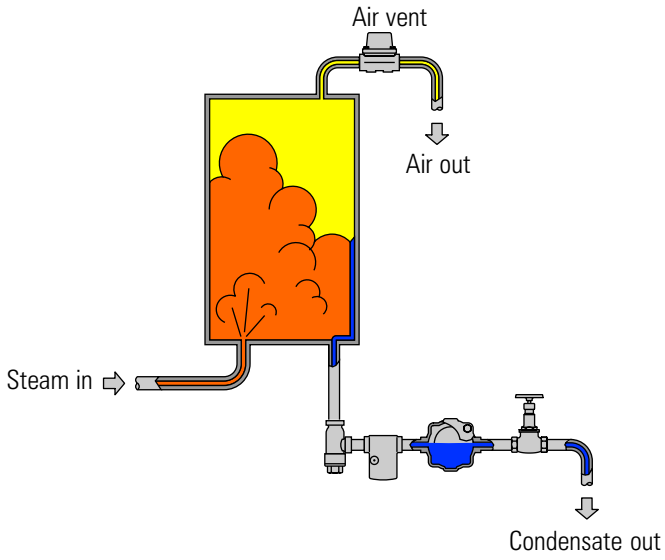


Fig. 11.12.8 Air vent (in steam trap) located opposite high level steam inlet

With top steam entry, the air will tend to be pushed to the bottom at start-up, and provision should be made for venting it at low level. Usually, a trap with a high air venting capability such as a float-thermostatic trap will do the job. However, in practice, to ensure complete removal of air during running conditions, a separate air vent fitted at the top of the vessel (as shown in Figure 11.12.8) may again often prove beneficial, especially on irregularly shaped vessels.

Questions

1. Which of the following will reduce heat transfer performance the most?

a| The layer of air 50 μm thick on a heat transfer surface ☐
b| A layer of water 0.5 mm thick on the same surface ☐
c| A layer of condensate 5 mm thick on the same surface ☐
d| A layer of water and condensate 1.0 mm thick on the same surface ☐

2. What is the effect if air is added to steam?

a| The temperature of the air/steam mixture increases ☐
b| The temperature of the air/steam mixture decreases ☐
c| The enthalpy of the steam/air mixture increases ☐
d| The enthalpy of the steam/air mixture stays the same ☐

3. On which principle does an automatic air vent operate in a steam system?

a| Buoyancy ☐
b| Thermodynamic ☐
c| The fact that air is heavier than steam ☐
d| Thermostatic ☐

4. What is the effect on the air vent if it is discharging into a flooded line?

a| None at all ☐
b| The capacity of the air vent is reduced ☐
c| The capacity of the air vent is increased ☐
d| The air vent totally blocks up ☐

5. What is the effect of air and condensate in a heat exchanger steam space?

a| It promotes noise from the heat exchanger ☐
b| It promotes erosion in the heat exchanger ☐
c| It promotes corrosion in the heat exchanger ☐
d| It promotes waterhammer in heat exchangers ☐

6. Which of the following statements is true regarding an automatic air vent?

a| It is open when cold and will remain open to hot condensate ☐
b| It will not operate on a mixture but only by sensing air separately ☐
c| It should only be used in conjunction with a vacuum breaker ☐
d| Its discharge should be to atmosphere wherever possible ☐

Answers

1: a, 2: b, 3: d, 4: b, 5: c, 6: d

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Module 11.13

Air Venting Applications

Air Venting Applications

Air vent units in general

The automatic air vent is a valve, thermostatically operated, and installed at a location where steam and air, rather than condensate, will reach it. If the air vent is close coupled to a heater of substantial mass, and which is operating at close to steam temperature, then conducted heat may hold the air vent closed, or at least slow down its operation. It is therefore recommended that any air vent and its connecting pipe should be installed unlagged in order for it to operate correctly.

Under these circumstances, the air vent is best installed at the end of a length of about 300 mm of pipe which can act as a collecting bottle, and which permits a temperature gradient from the heater steam space to the vent. The 'bottles' mentioned in 'Rotating cylinders' can be utilised in this way as air collecting units.

When air vents discharge, they invariably do so with an air/steam mixture. This is often perceived as being pure steam, and the logical conclusion is to believe that the air vent is leaking. If operating normally, the degree of discharge should eventually reduce and cease. If the air vent continues to discharge over a long period without any sign of shutting off, it could well be faulty and would benefit from being inspected and repaired.

Steam trap bypasses

It seems natural to fit manual bypasses around steam traps, usually to be opened at start-up. Since condensate loads at start-up are rarely much more than twice the running load, and traps usually have condensate capacities giving safety factors of considerably more than this, it seems that the real function of bypasses is to discharge air. This allows the condensate to reach the trap. Bypasses are often found around bucket traps, which are inherently slow to vent air. The assembly can be made both automatic and efficient by simply replacing the manual bypass valve with an automatic air vent. Manual bypasses are easily forgotten about and left open, and are therefore a potential source of steam wastage.

Vacuum breakers

Vacuum breakers may be used to good effect at times of system shutdown when sub-atmospheric pressures may be experienced within steam pipes and apparatus. Strategically placed, they will allow condensate to gravitate down to the drain trapping point. By allowing the complete removal of condensate from any steam system, fear of waterhammer will be removed at the next system start-up.

Saturated steam mains

The steam main is, in effect, a long steam space with a small cross-section. When steam is turned on at the supply end, it moves along the pipe like a piston, pushing the air in front of it. An air vent fitted at the end of the line as in Figure 11.10.13, Module 10, will clear most of the air. The vent is connected at the top of the pipe, or at least at a point above the expected level of condensate.

Superheated steam mains

Superheated steam mains, generally, only require air venting at start-up. An air vent able to withstand high temperatures is required, consequently a bimetallic type would be the best choice.

Jacketed pans

Selecting the air vent location for these applications can be difficult. Air dissolved in the cold product is forced out of solution as the pan warms up, and bubbles appear on the product side of the jacket. Lack of bubbling on the inside skin of the pan reveals cold spots, indicating where air is collecting inside the jacket.

With the combination of the wrong type of steam trap and no air vent, it is likely that bubbling will occur last at the bottom of the jacket near the condensate outlet, and at the top opposite the steam entry point. The best steam trap will be a float type with air vent, placed below the pan, allowing condensate and air to gravitate to the floor, or to a collecting receiver and pump. The air vent is best placed opposite the steam entry point at high level, and a bonafide manufacturer will place a tapping for this purpose, (Figure 11.9.1, Module 9).

A tilting pan requires a float trap with steam lock release feature as the condensate is removed via a dip pipe passing through a rotary joint. If this does not include an air vent, then a separate air vent bypassing the trap will improve the performance. Likewise, the steam trap should be placed below the outlet, (Figure 11.9.2, Module 9).

Rotating cylinders

One special case of interest is the drying cylinder used in many process industries. A horizontal cylinder is supplied with steam through a rotary joint at one end, and the material being processed is in contact with the outer surface of the cylinder. Condensate is discharged through a dip pipe passing either through the same rotary joint or a similar joint at the opposite end of the cylinder.

With cylinders of appreciable size, the volume of air to be discharged at 'start-up' is large. Air collecting within the cylinder during normal operation leads to cold spots on the outer surface, and improperly processed material is produced. Automatic air venting is paramount, and must be achieved as a matter of course if good results are to be expected.

The best steam trap for this purpose is a float-thermostatic type with steam lock release, but a separate air vent is often still needed due to the large amount of air to be purged. Experience shows an air vent and an air collecting bottle at the condensate outlet, will give an excellent result if fitted as shown in Figure 11.13.1.

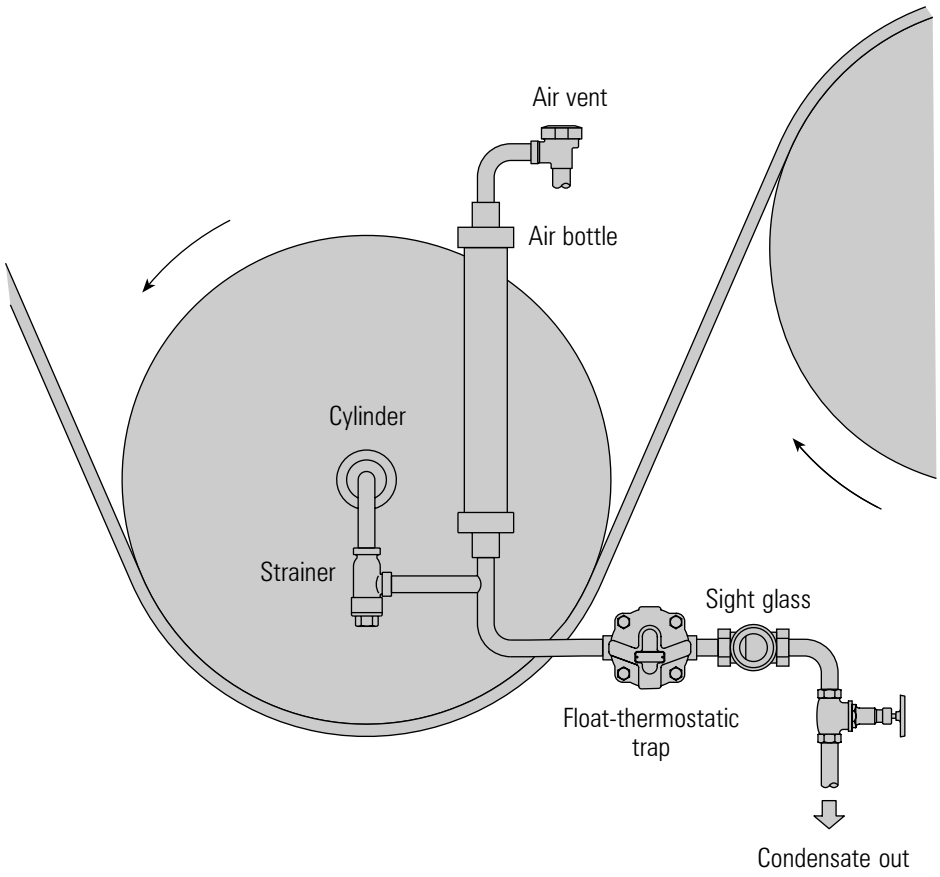


Fig. 11.13.1 Cylinder drainage with system unit

Group air venting

Steam equipment designers sometimes reduce expenditure by connecting the remote points of two or more steam spaces, and fitting a single air vent, rather than using individual air vents for each steam space. Unfortunately such an arrangement is often unsuccessful. A multi-coil air heater can have each of the coils supplied from a common steam header which is fed through a single control valve. Here, the air vent will close when steam from one section reaches it. Air, present in the other sections, would simply not reach the vent to open it. Later, the steam in the air vent body condenses, and is replaced again. Again, when the incoming steam is from the coil containing the least air, the vent tends to quickly close. The air/steam mixtures in the other coils are not induced to flow towards the vent position. Group air venting is not successful, and should be avoided, in the same way as the group steam trapping of condensate drain lines.

Extra large air vents

The capacity of an air vent depends on the size of the orifice, the differential pressure across the seat, and the properties of the gas being discharged. In some instances, the steam spaces being vented are very large, as in large sterilisers and retorts in the food industry, large autoclaves, rubber curing vessels etc. The amount of air to be vented may then be so great as to require large numbers of air vents to be fitted in parallel. An alternative answer is to use a self-acting temperature control, fitted similarly to that in Figure 11.13.2.

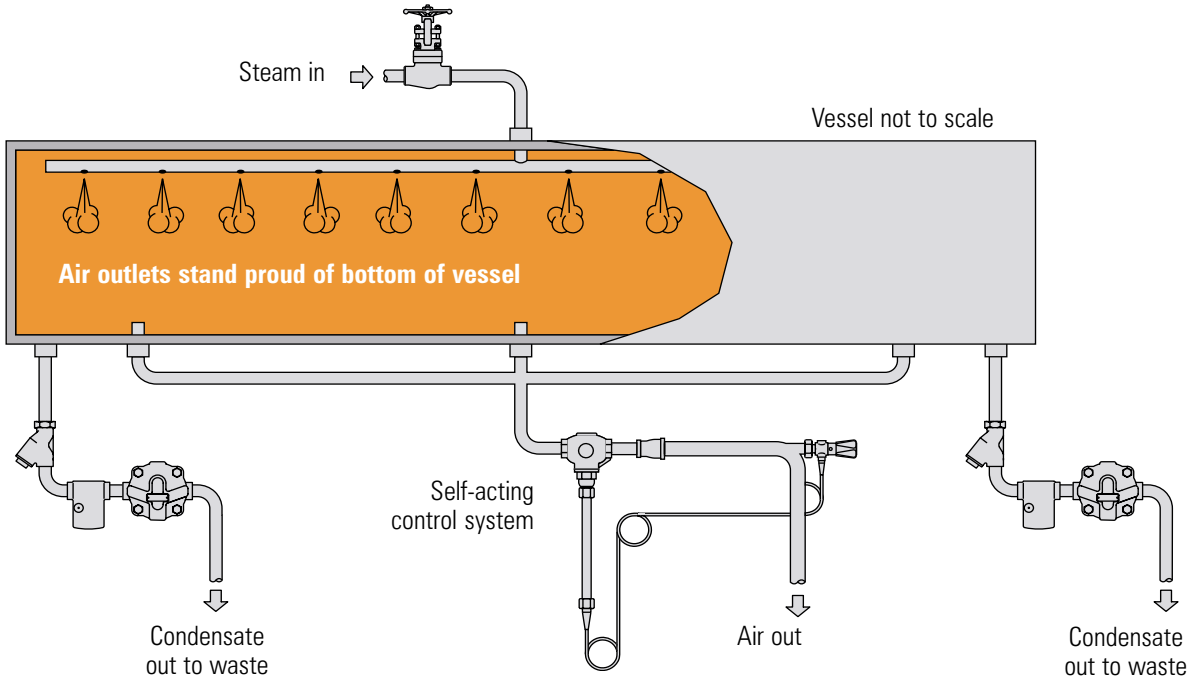


Fig. 11.13.2 Large volume air venting provided by a self-acting control system

The valve must be of a pattern suitable for use on steam service. The valve is positioned by the control system, and the temperature sensor is located on the downstream side of the valve. The temperature setting is adjusted to 100°C, or just below this value. Since the pressure in the tail pipe at the temperature sensor is atmospheric, the temperature at this point would be 100°C if air-free steam had reached the sensor after flowing through the valve. At this temperature, the valve should just be closed. Any lower temperature at the sensor location means that some air is present, and the valve will be slightly opened.

Positioning the temperature sensor in this way, downstream of the valve where the line pressure is atmospheric, nullifies the effect of pressure upstream of the valve. The control system has only to close the valve when the sensor temperature reaches 100°C and open it at lower temperatures. This arrangement makes it quite practical to use air vent valves as large as the DN50, which enables large volumes of air to be discharged.

Venting air through thermostatic steam traps

Any thermostatic steam trap, such as the balanced pressure bellows or capsule, or the bimetallic type, can be used as an air vent. Clearly the operating unit should be one that reacts quickly to temperature changes, and traps incorporating bimetal strips of large dimensions are probably less suitable. But, if a thermostatic steam trap is used primarily to drain condensate, how effectively will it vent air?

Since the trap will be open at start-up when the steam is turned on, it will discharge the air being pushed towards it.

During normal running however, the trap may not be quite as effective as an air vent. As a steam trap, it will close to condensate just below saturation temperature. It follows that with a water seal present at the inlet side of the trap, air and any other non-condensables will be sealed within the process steam space for a little time by the condensate.

When the condensate at the steam trap eventually loses some of its heat, only then will the trap open and discharge both condensate and the cool air/steam mixture.

The most effective way to release air by a steam trap from a steam space is to use a float type steam trap with an inbuilt air vent. As condensate should always get to the trap, the passage of non-condensables to the integral air vent is not held up during normal operation.

It must be made clear that the automatic device which is being used to discharge air/steam mixtures, whether it be described as a steam trap or as an air vent, is best positioned above the water level in the trap. In all other cases, the addition of air vents (at positions where the air/steam mixture can reach them under all conditions) can have beneficial results out of all proportion to the extra costs involved.

Questions

1. Name an ideal position for an automatic air vent on a steam main

a| After a pressure reducing valve ☐
b| On a drain pocket ☐
c| At the end of the main ☐
d| After any steam trap used to drain the main ☐

2. Name the best type of air vent on a superheated steam main

a| A balanced pressure thermostatic type ☐
b| A bimetallic thermostatic type ☐
c| A manually operated valve ☐
d| A superheated steam main does not need air venting ☐

3. How can a failed automatic air vent be detected?

a| By clouds of steam discharging from the outlet ☐
b| By testing the temperature of the discharging air ☐
c| By observing a steam discharge over an extended period of time ☐
d| By hot condensate discharging from the outlet ☐

4. What is the real effect of having bypasses around steam traps?

a| They increase the condensate discharge capacity of the trap ☐
b| They reduce waterhammer ☐
c| They stop the steam trap from air binding ☐
d| They reduce start-up times by quickly venting condensate and air ☐

5. Where is it acceptable for group air venting to be installed?

a| On multi-coil heater batteries fed by the same steam pressure ☐
b| On large vessels such as industrial autoclaves and retorts ☐
c| A vertical condensate discharge manifold on platen presses ☐
d| Where the installation suggests a common discharge line ☐

6. Which of the following statements is true?

a| Large capacity air venting is provided by a self-acting control valve ☐
b| An air vent is not needed on a rotating drying cylinder ☐
c| An air vent is not needed on a vessel fitted with a float/thermostatic trap ☐
d| Vacuum breakers cannot be installed if air vents are fitted ☐

Answers

1: c, 2: b, 3: c, 4: d, 5: c, 6: a

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Module 11.14

Testing and Maintenance of Steam Traps

Testing and Maintenance of Steam Traps

Testing of Steam Traps

Traditional and contemporary methods

Indiscriminate maintenance of steam traps costs money. Steam traps will either be:

- In good working order.
- Leaking steam.
- Blocking flow.

A major problem has always been the accurate identification of faulty traps. Wrong diagnosis can allow faulty traps to remain troublesome, and perfectly sound traps to be replaced unnecessarily. Accurate diagnosis is therefore important to any maintenance programme.

Historically, diagnostic methods have included listening devices, optical sight glasses, temperature monitoring, and ultrasonic techniques. All of these can give an indication of flow, but become inaccurate as system conditions change. Noise level will vary with disturbance from adjacent traps, and condensate load. Interpretation of signals is difficult even for experienced operators.

Sight glasses offer a partial solution, especially the combined sight/check valve that gives a visual indication of flow plus a non-return facility, however, glasses will require changing occasionally. The inadequacies of listening devices have led to temperature monitoring, but it is perfectly feasible (and normal) for condensate and steam to coexist at the same temperature in the same system, making accurate diagnosis difficult on temperature alone.

A modern version of the listening rod is the ultrasonic trap tester which detects ultrasound generated by a leaking trap. It is, unfortunately, unable to differentiate between live steam and flash steam passing through the trap. It is also unable to detect the subtle differences explained above.

The unreliability of the above methods has prompted the development of an integrated steam trap testing device.

This consists of a sensor, fitted inside the steam trap, which is capable of detecting the physical state of the medium at that point by conductivity (Figure 11.14.1). It is not affected by flash steam disturbance. The result is finite and not subject to interpretation. Monitoring can be done locally, remotely, manually or automatically, and can detect immediate failure, thus minimising waste and maximising investment (Figure 11.14.2).

An integral thermocouple in the sensing chamber can detect and help to predict blockages, which is particularly useful, especially to those Hydrocarbon and Processing Industries which require process continuity.

For steam users preferring to use steam traps without integral sensors, or for larger applications requiring larger traps, sensors can be provided in separate sensor chambers (see Figures 11.14.3, 11.14.4 and 11.14.5).

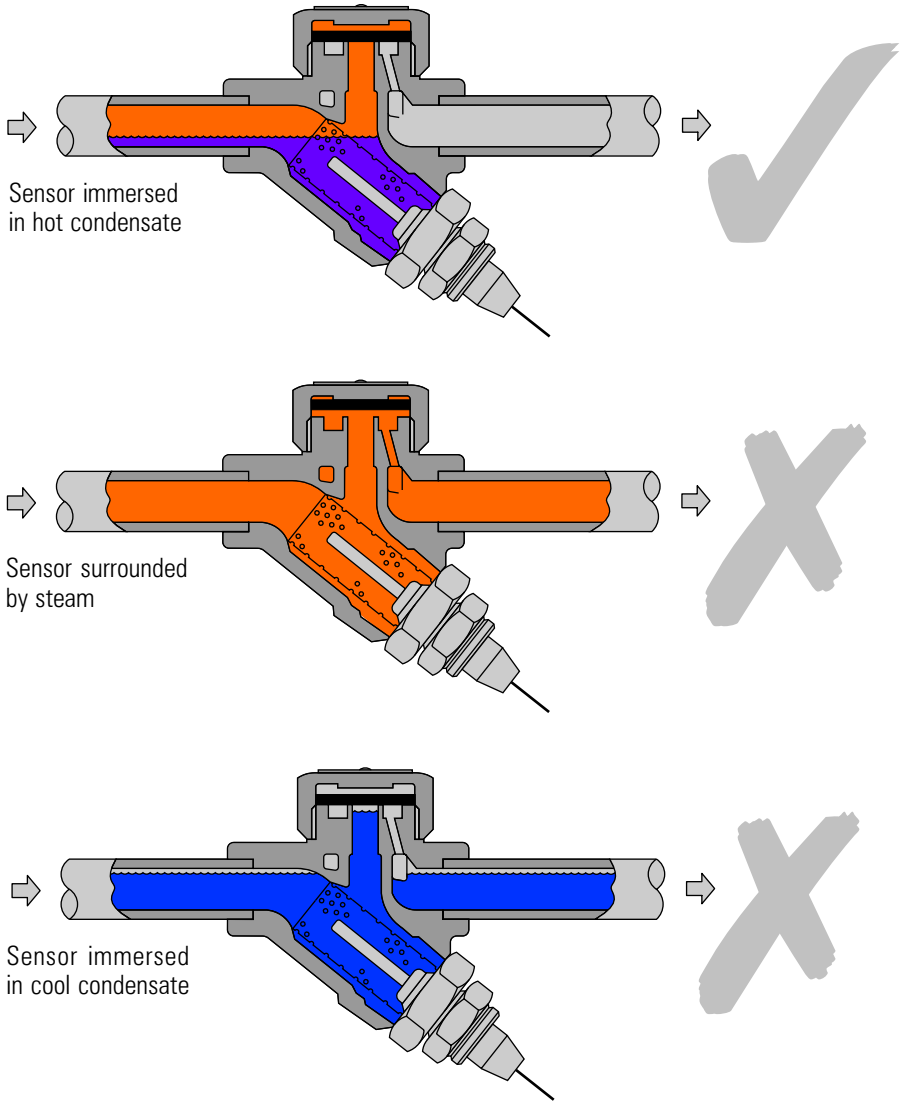


Fig. 11.14.1 How traps with integral sensors work

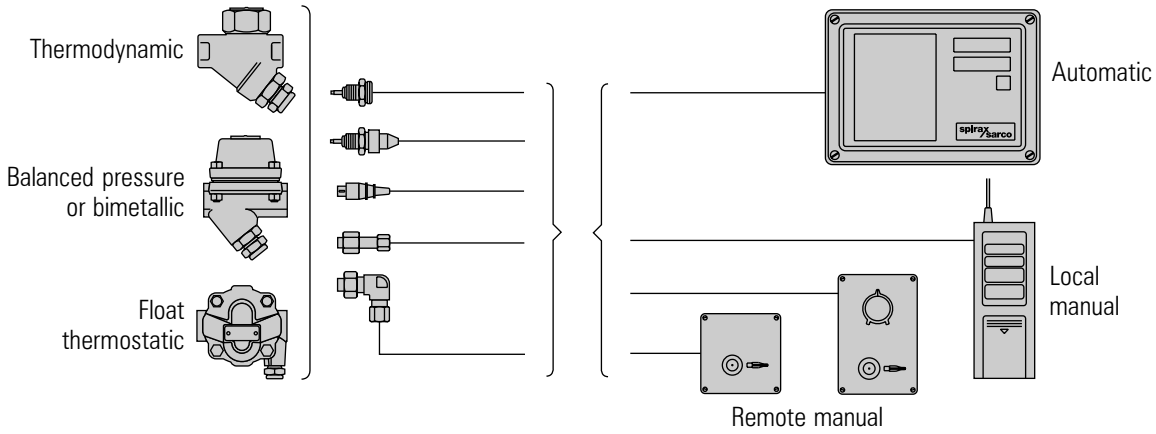


Fig. 11.14.2 Manual, remote, or automatic monitoring with integral traps

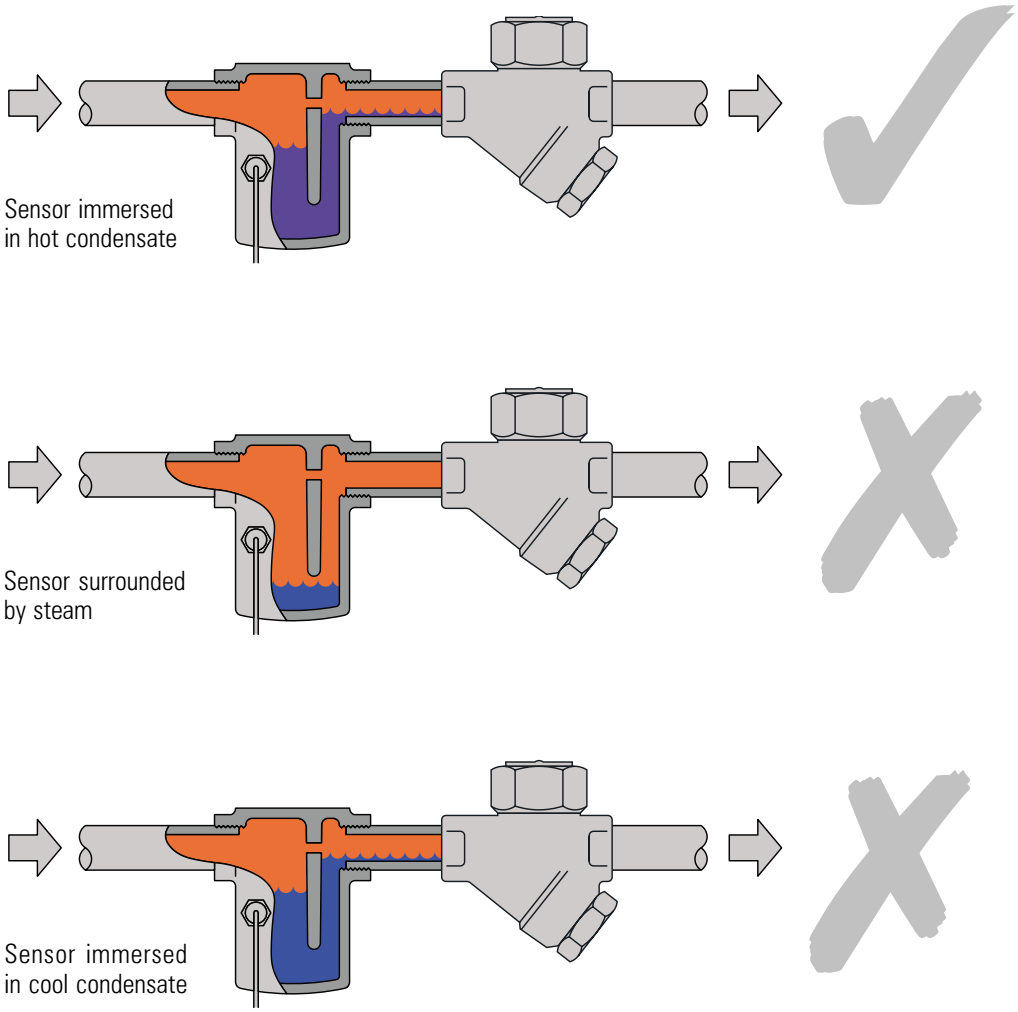


Fig. 11.14.3 How separate chambers work

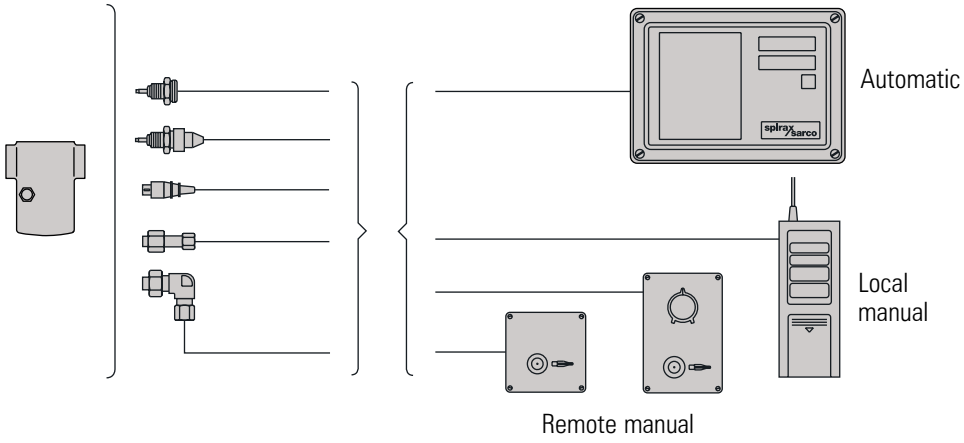


Fig. 11.14.4 Manual, remote, or automatic monitoring with separate chambers

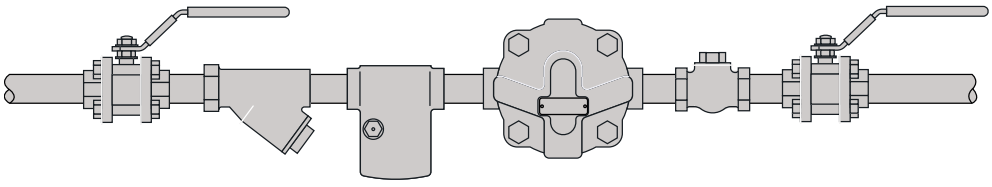


Fig. 11.14.5 Typical steam trap set with separate sensor chamber

Maintenance of steam traps

Routine maintenance

Routine maintenance depends on the type of trap and its application. The balanced pressure steam trap for example, has an element which is designed for easy replacement. Changing these on a regular basis, maybe once every three years or so, might seem wasteful in time and materials. However, this practice reduces the need for trap checking and should ensure a trouble free system with minimal losses through defective traps.

Routine maintenance which involves cleaning and re-using existing internals uses just as much labour but leaves an untrustworthy steam trap. It will have to be checked from time to time and will be prone to fatigue. Any routine maintenance should include the renewal of any suspect parts, if it is to be cost effective.

Replacement of internals

The renewal of internal parts of a steam trap makes good sense. The body will generally have as long a life as the plant to which it is fitted and it is only the internal parts which wear, depending on system conditions. There are obvious advantages in replacing these internals from time to time. It depends on the ease with which the new parts can be fitted and the reliability and availability of the refurbished trap. The elements of thermostatic traps can generally be changed by removing a screwed in seat. Replacement is simple and the remade trap will be reliable assuming the maintenance instructions are correctly carried out.

If the seat or disc faces of a thermodynamic trap become damaged, the disc can simply be replaced (Figure 11.14.6). Damage to seating faces can be rectified by lapping gently. Replacing the seats of some higher pressure thermodynamic traps is more complicated. Two separate gasketed joints may have to be made or a single gasket may have to cope with two or more steam/condensate passages. The weakest point is often the joint between trap body and seat, particularly if this has been allowed to blow steam.

Always check with the manufacturer regarding the correct technique for any maintenance work required on steam traps. A reputable manufacturer will always be able to supply appropriate literature, advice, and spare parts.

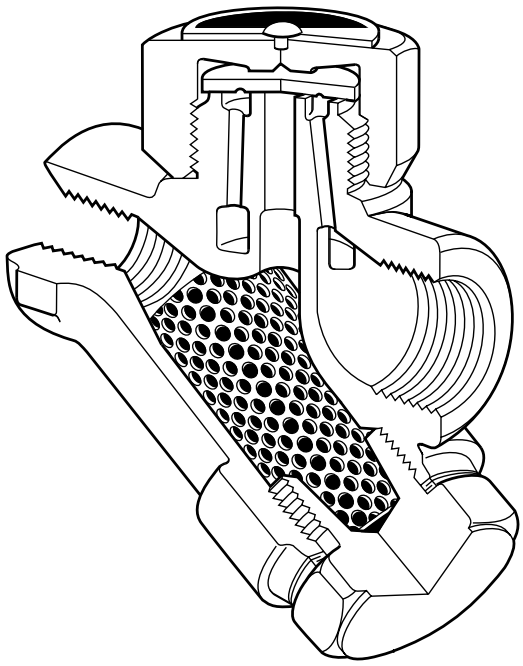


Fig. 11.14.6 Sectional view of a thermodynamic trap with the disc as one moving part

A lot will depend on site conditions. The small float trap, shown in Figure 11.14.7, is designed so that the cover with the internals attached can be taken to the workshop, leaving the main body attached to the pipe. This is often preferable to renewing the seats of inaccessible traps, which have been welded into the pipework under dirty site conditions.

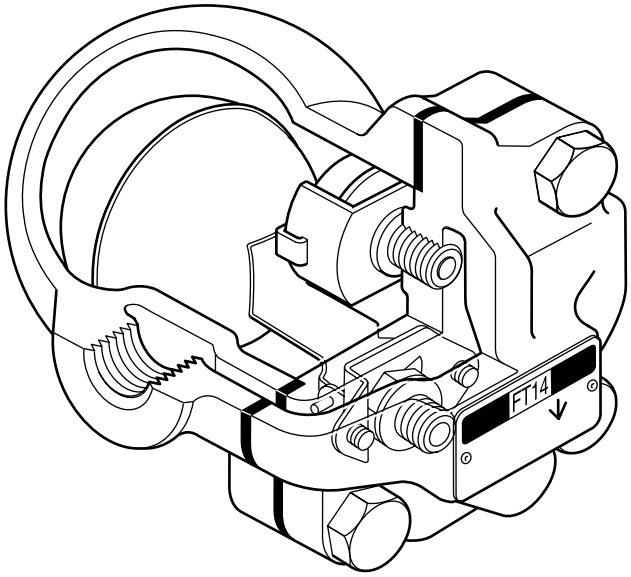


Fig. 11.14.7 Internals of float-thermostatic trap with steam lock release and air vent

Replacement of traps

On occasions, it will be easier and cheaper to replace traps rather than repair them. In these cases it is essential that the traps themselves can be changed easily. Flanged connections provide one solution, although the flanged trap is more expensive than the equivalent screwed trap. Mating flanges are an additional expense.

A swivel connector allows rapid easy removal and replacement of the sealed trap. The trap shown in Figure 11.14.8 is specifically designed for easy replacement for such a system. It comprises a pipeline unit or connector which remains in the pipeline during the maintenance procedure. The trap can be replaced simply by attending to two bolts. This type of trap can be matched to the same connector providing flexibility of choice and rationalisation of spares. Connectors are also available with integral piston isolation valves ensuring downtime is kept to a minimum.

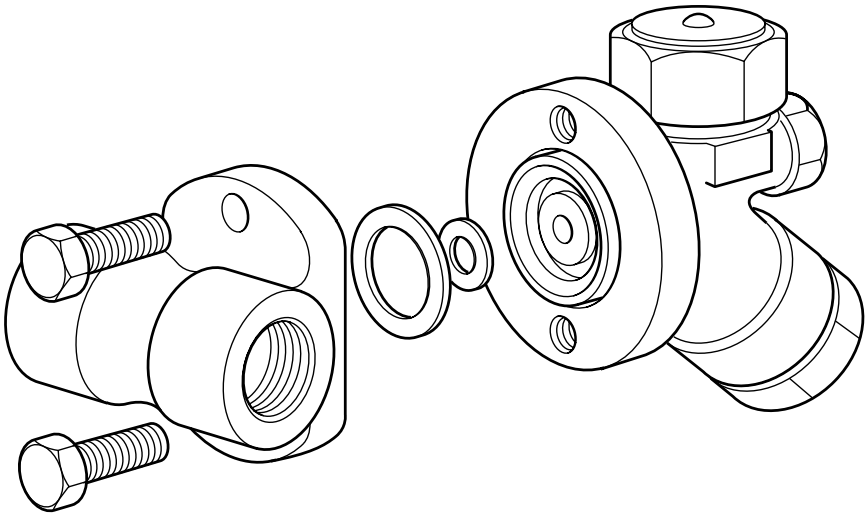


Fig. 11.14.8 Swivel connector trap for quick replacement

Questions

1. What effect does a steam trap have when failed in the open position?

a| It will stop the plant from operating ☐

b| It will loose steam and cost money ☐

c| Plant efficiency is maintained ☐

d| Plant efficiency is increased ☐

2. Which of the following is the most reliable steam trap tester?

a| A sight glass ☐

b| An ultrasonic listening device ☐

c| A temperature monitoring device ☐

d| An integrated steam trap sensor or separate chamber ☐

3. What are the operating principles of an integrated trap sensing system?

a| Ultrasound detection ☐

b| Conductivity and temperature for leaks and blockages respectively ☐

c| Temperature sensing only ☐

d| Pressure sensing ☐

4. Why is it not feasible to rely on temperature sensing for testing steam leaks?

a| Because it is too difficult ☐

b| Temperature sensing devices cost too much ☐

c| Because steam and condensate can co-exist at the same temperature ☐

d| Because saturation temperature varies with steam pressure ☐

5. What are the most convenient steam trap connections to consider for maintenance purposes?

a| Screwed connections ☐

b| Flanged connections ☐

c| Socket weld connections ☐

d| Universal swivel connections ☐

6. Which of the following statements is true?

a| Ultrasonic trap testers cannot differentiate live steam and flash steam ☐

b| A Spiratec hand held unit cannot indicate a failed closed steam trap ☐

c| Sight glasses cannot differentiate between live steam and flash steam ☐

d| All of the above ☐

Answers

1: b, 2: d, 3: b, 4: c, 5: d, 6: d

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Module 11.15

Energy Losses in Steam Traps

Energy Losses in Steam Traps

A large amount has been written about this subject, much of which has been inaccurate or deliberately misleading in order to make the case for using various manufacturers' traps.

An argument is made in favour of replacing one type of trap with another and claiming a steam saving which may be real or imaginary. The truth is that replacing any group of traps with new ones will inevitably reduce steam consumption because any leaking traps are thereby eliminated. This says nothing about the old or new traps.

In other cases, tests have been carried out to establish 'steam wastage'. Some tests are carried out under unrealistic no-load conditions and attempt to overvalue and confuse the amount of energy lost through the trap. Energy loss from the trap due to radiation, which will also increase condensate load, is conveniently ignored. However, these losses will occur at all times and are directly related to the size and shape of the body.

Steam trap users are often confused by subjective information which is intended primarily to create interest in a product. It is therefore worth going back to objective principles and considering the inherent energy requirements of the main types.

Thermostatic steam traps

Under normal operating conditions, the thermostatic trap holds back condensate until it has cooled to a certain temperature. Steam does not reach the main valve so there is no apparent steam wastage.

However, waterlogging of plant can lead to reduced output. Operating times may be extended or additional heaters or heating surfaces may be required. More steam may be required although this will not appear as an energy requirement attributable to the steam trap.

In some cases a cooling leg may be incorporated so that the steam space is kept clear of condensate. Energy is thereby lost due to radiation from the cooling leg and from the trap body. This in itself creates an additional condensate load, but there is no passage of live steam through the trap.

The situation can change under no-load conditions. Heat loss from the trap body cools the condensate surrounding the element which then opens. The minimal amount of condensate involved is discharged and is then replaced by steam. However, hysteresis means that the element has yet to respond and live steam is lost. Laboratory tests indicate typical losses up to 0.5 kg/h. Ironically, under cold outdoor conditions there will be increased heat loss from the trap and steam loss through the trap is less likely. Any attempt to lag a thermostatic trap will result in a serious delay in the opening of the trap. Severe waterlogging will result and hence lagging is not recommend for thermostatic traps.

Mechanical steam traps

The float-thermostatic trap is another example where the valve and seat are normally flooded and there is no steam loss through the trap. Conversely, the float-thermostatic trap is relatively large in size, and there may be a noticeable loss from the trap caused by radiation. Mention should be made of the thermostatic air vent fitted in this type of trap. This will be situated in the steam space above the water level in the trap. Once initial air has been cleared this will normally remain tight shut and there will be no loss from this source.

The float-thermostatic trap can be lagged to reduce heat losses and this will not affect its operation. Lagging is normally recommended on outdoor applications to minimise the danger of damage due to freezing when steam might be turned off.

The inverted bucket trap has surprisingly little in common with the float type trap. The trap closes when steam enters and bubbles through into the bucket to make it buoyant. It will not open until the steam has been dissipated.

This will occur as the steam leaks away through the hole in the bucket which serves as an air vent. The steam will collect in the top of the trap itself and when the main valve opens, this steam is vented.

Laboratory tests again indicate losses of around 0.5 kg/h for ½" traps under these low load conditions. However, there is additional radiation loss from the body, which can be quite large. Lagging is sometimes recommended but the heat loss and its resulting condensate will be much the same as an equivalent float type trap.

Thermodynamic steam traps

This type of trap has attracted most attention under the heading of steam wastage.

The operation depends on condensate approaching steam temperature, producing flash steam at the orifice and causing the trap to close. It does this with condensate on the upstream side and again the flooded valve means that there can be no loss through the trap. However the trap will open periodically as heat is lost from the cap.

Under no-load conditions, i.e. when condensate is being produced only by heat loss from the upstream pipeline, the condensate on the upstream side may exhaust and the trap will then require a small amount of live steam to cause it to close. Much will depend on ambient conditions but the loss will generally be around 0.5 kg/h and this could be doubled in severe weather. Conversely, such losses can be halved by simply fitting an insulating cover over the top cap.

It is important to remember that these losses disappear as the condensate load increases while the radiation losses from the trap are minimal due to its small size. Independent tests have shown that radiation losses are not more than 0.25 kg/h which is at least a quarter of that experienced by equal sized inverted bucket traps.

Mention should be made of misleading figures quoted by some sources. These have their origins in tests carried out simultaneously on a large number of thermodynamic traps. Some tests were carried out at minus 45°C with the cumulative steam loss being measured. The effect of testing at unusually low temperatures and under no-load conditions was to produce an accelerated life test. The loss through a small number of defects averages out to produce a curve showing losses increasing with time. As already indicated, the thermodynamic trap has the great simplicity in that it either works correctly or fails. To suggest a varying loss is totally misleading and fundamentally flawed.

Comparisons

Quantifying the energy requirements of steam traps is not easy. Energy can be lost through the trap but this may depend on load. Energy will be lost from the trap due to radiation but this can be reduced considerably by lagging.

Table 11.15.1 summarises the energy requirements of a variety of ½" traps at 5 bar g. Clearly traps vary in size and performance so the figures must serve as a guide only.

Table 11.15.1 Energy requirement of traps - expressed in kg/h of steam

	No-load			Reasonable load		
	Through trap	From trap	Total	Through trap	From trap	Total
Thermostatic	0.50	0.50	1.00	Nil	0.50	0.50
Float	Nil	1.40	1.40	Nil	1.40	1.40
Inverted bucket	0.50	1.20	1.70	Nil	1.20	1.20
Thermodynamic	0.50	0.25	0.75	Nil	0.25	0.25

The International Standard ISO 7841 (1988) and European Standard CEN 27841 (1991) - Determination of steam loss of automatic steam traps - describe a reliable and accurate test methodology for losses from any type of steam trap. **Any manufacturers' test figures that are not obtained within the parameters of these standards should be treated with caution.**

The purpose of Table 11.15.1 is not to establish the fact that one type of trap is marginally more efficient than another. It is simply to make the point that steam traps use a minimal amount of energy. Losses only become significant when traps are defective. **The important thing therefore is to combine selection, checking and maintenance to achieve reliability. Properly done, costs and steam wastage will be minimised.**

Questions

1. Energy losses from steam traps can consist of which of the following?

a| Energy lost from the trap body by radiation and convection ☐
b| Energy lost through the trap by live steam leakage ☐
c| Energy lost through the trap when under no-load conditions ☐
d| All of the above ☐

2. Which of the following is true of an inverted bucket steam trap?

a| It will always fail in the closed position ☐
b| It will leak steam if the steam pressure exceeds its maximum ☐
c| It has the highest energy loss due to surface area and steam loss via the vent hole ☐
d| It does not lose steam under no-load conditions ☐

3. Which of the following is true of a float trap?

a| Lagging will effect its operation ☐
b| Lagging will reduce its energy losses to virtually nil ☐
c| Air cannot pass through the trap under no-load conditions ☐
d| Air cannot pass through the trap when backpressure exists ☐

4. Which of the following is true of a bimetallic thermostatic trap?

a| On start-up, it is wide open and able to pass large quantities of air ☐
b| Lagging the cooling leg will not effect the trap's operation ☐
c| Lagging the trap will not affect the trap's operation ☐
d| None of the above ☐

5. Which of the following is true of a thermodynamic trap?

a| It loses more steam off-load than any other type of steam trap ☐
b| It loses less steam off-load than any other type of steam trap ☐
c| Its radiation losses are higher than any other type of trap ☐
d| It will always fail in the open position ☐

6. Which of the following statements is true?

a| A leaking steam trap will affect the drainage of plant to which it is fitted ☐
b| A trap failed closed will not affect the plant performance ☐
c| Float traps cannot waste live steam because the orifice is flooded ☐
d| International and European standards exist for the testing of steam trap losses ☐

Answers

1: d, 2: c, 3: b, 4: a, 5: b, 6: d

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Module 12.1

Isolation Valves - Linear Movement

Isolation Valves - Linear Movement

Isolation valves are a key component in any fluid system as they are used to stop the flow of fluid into a particular area of the system. They are also sometimes used to manually control the flow of the fluid. The European standard EN 736-1:1995 distinguishes between isolating, regulating and control valves as follows:

- **Isolating valve -** A valve intended for use only in the closed or fully open position.
- **Regulating valve -** A valve intended for use in any position between closed and fully open.
- **Control valve -** A power-operated device which changes the fluid flowrate in a process control system.

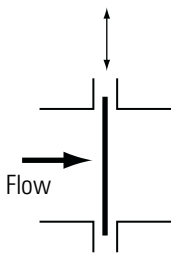
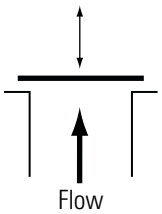
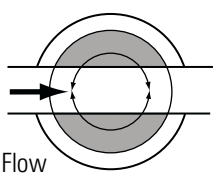
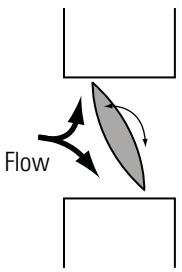
Isolation valves are used in a wide variety of different applications where on/off type control is required, these include:

- Diverting process media.
- Flow isolation to:
  - Facilitate maintenance
  - Allow the removal of equipment
  - Allow the shut down of plant

A multitude of different types and designs of isolation valve have been developed in order to meet this range of applications and the diverse operating conditions in which they are used. Valves are commonly classified into two groups (see Table 12.1.1), according to the operating motion of the closure device (or obturator):

- **Linear movement valves -** The obturator moves in a straight line. Included in this category are gate valves, globe valves, diaphragm valves and pinch valves. These valves are covered in greater depth within this module.
- **Rotary movement valves -** The obturator rotates about an axis at right angles to the direction of flow. Ball valves and butterfly valves are the two most important rotary valves associated with steam applications and are covered in greater depth in Module 12.2, Isolation Valves - Rotary Movement.

Table 12.1.1 Obturator motion in the basic valve types

Valve movement	Linear		Rotary	
Operating motion of the closing device (obturator)	Straight line		Rotating about an axis at right angles to the direction of flow	
Direction of flow in the seating area	At right angles to the operating motion of the obturator	Longitudinal to the operating motion of the obturator	Through the obturator	Around the obturator
Basic types	Gate valve	Globe valve	Ball valves	Butterfly valve
Schematic				

Linear movement valves

Linear movement valves have been developed from the early forms of sluice gates used to control the flow of water in irrigation channels. Since then, a large number of different designs and types have been developed for use in almost every type of flow application. Although linear movement valves are characterised by straight-line obturator movement, the flow of the fluid may be at right angles to this movement (as in the case of gate valves), or in the same direction, as with globe valves. The main feature of the linear movement valve is that tight shut-off may be achieved by tightening down the obturator on a threaded stem.

Gate valves

Gate valves are probably the most common valves in use today due to their widespread use in domestic water systems, but it should be noted that their popularity in industry has declined in recent years. However, they are still used where an uninterrupted flow is required, because the gate fully retracts into the bonnet, creating a minimal pressure drop, when the valve is in an open position. Gate valves are specifically intended for use in isolation applications.

A gate valve consists of four main components, the body, bonnet (or cover), gate and stem. A typical gate valve is shown in Figure 12.1.1.

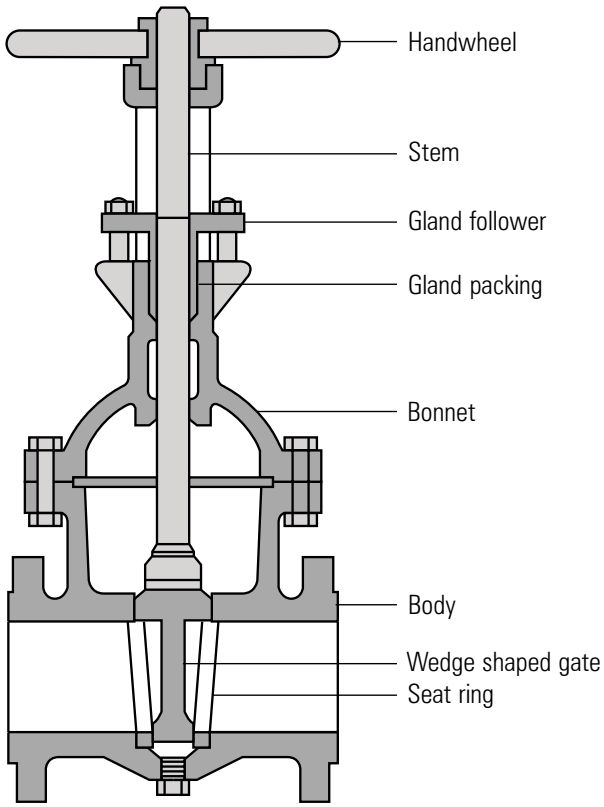


Fig. 12.1.1 Typical wedge gate valve

The gate, which slides between the seats, is lifted in a direction at right angles to the flow until clear of the flow path. The fact that the gate fully retracts into the bonnet ensures that the pressure drop across the valve is low.

Gate valves are divided into a number of different classes, depending on the design of the gate and its seating faces.

Solid wedge gate valve

The gate is wedge shaped and it seats on corresponding faces in the valve body. The mechanical advantage of the activating thread, together with the wedge angle, enables adequate seating forces to be applied against the fluid pressure without excessive handwheel effort. The seat can sometimes be coated with PTFE to assist a high integrity shut-off. A typical solid wedge gate valve is shown in Figure 12.1.1.

Flexible wedge gate valve

Although there are several types of flexible wedge gate valves, they all make use of a flexible two-part disc, which is shaped like two wheels on a very short axle. The flexibility of the disc ensures tight seating over a wide range of temperatures and pressures.

The most common type of flexible wedge gate valve used in steam applications is the parallel slide valve. The two plates that constitute the gate are held against the seat by a spring, encased between them. The fluid pressure moves the upstream disc off its seat, and the force is transferred onto the downstream disc, thereby ensuring a tight shut-off. The high degree of flexibility in the gate allows for expansion and contraction when subjected to temperature variations, making it suitable for use in steam systems.

Globe valves

Globe valves constitute a major class of linear movement valves; they have become more popular than gate valves as there is a wide variety of configurations available to suit most applications. The movement of fluid through the valve seat is longitudinal to the operating motion of the obturator; this means that for a valve in which the inlet and outlet are horizontally opposed, the fluid must follow a changing course. The main advantage of this arrangement is that a globe valve opens more rapidly than a gate valve as the disc only needs to move a small distance from its seat to allow full flow. This is an advantage when there is frequent operation of the valve. The disadvantage is that the fluid has to change course, increasing the resistance to flow and generating turbulence. This results in a higher pressure drop across a globe valve than a gate valve.

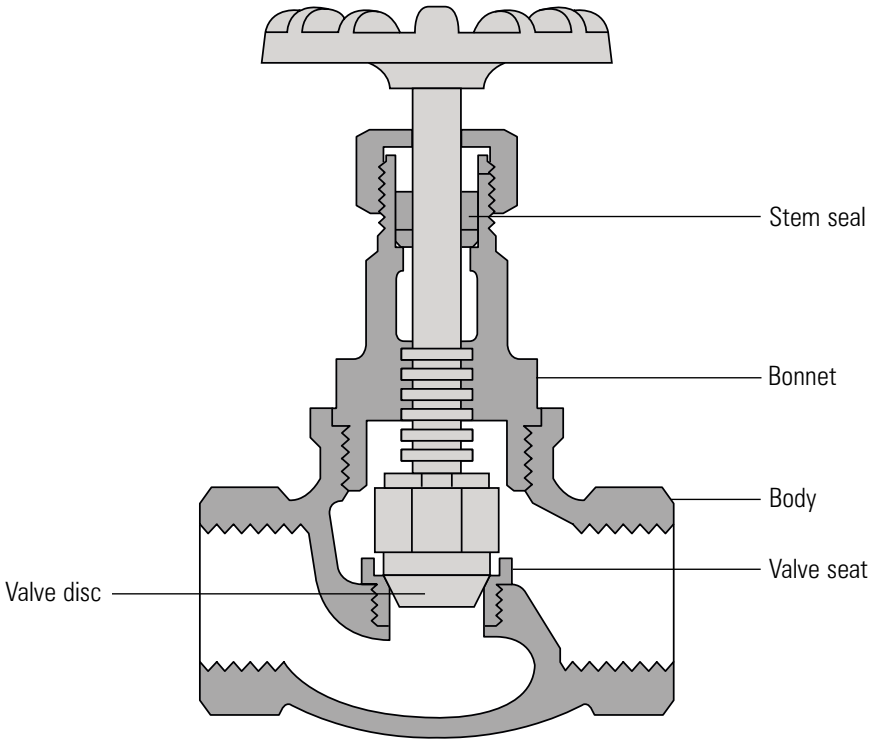
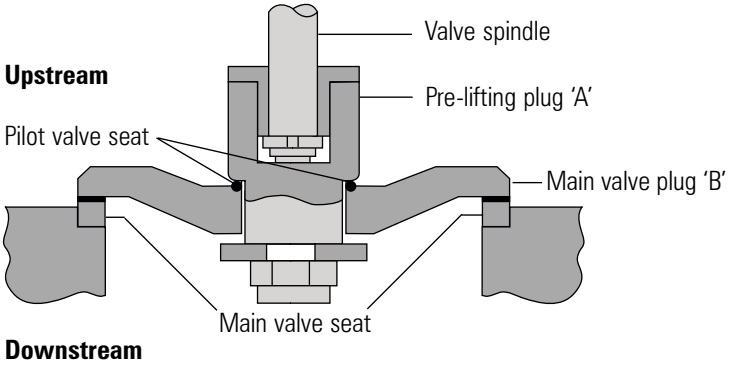


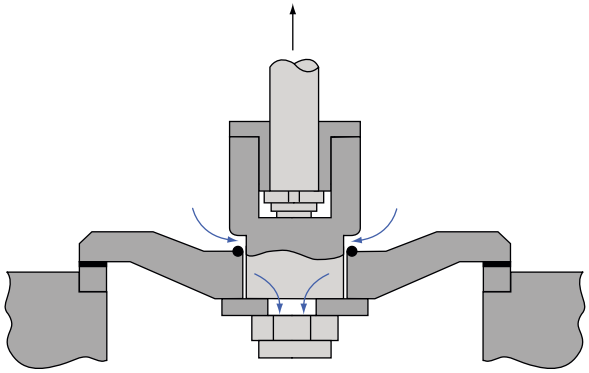
Fig. 12.1.2 A conventional globe valve

Globe valves are less likely to leak than gate valves, which means that they can be used for higher pressure or higher volume applications, for example in steam systems, or where fluid loss can be hazardous or costly. The increased cost of globe valves over gate valves is therefore offset by the additional safety they provide, and a reduced chance of fluid loss.

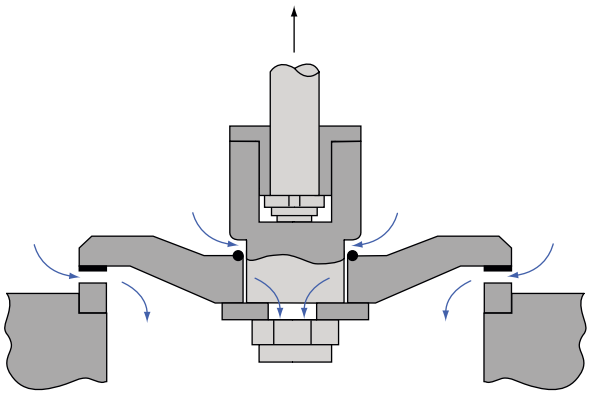
The pressure of the fluid acting over the area of the disc generates an axial load on the stem. This makes closing the valve difficult, so much so, that it limits the size of a standard globe valve to DN250. On high differential pressure closed systems, balancing plugs can be used to overcome this effect, allowing valves with a nominal diameter of up to 500 mm to be used (Figure 12.1.3(a)). The balancing plug contains a pre-lifting plug that acts as a pilot valve. When the valve is opened, the pre-lifting plug opens first, allowing the medium to pass through it at a controlled rate (Figure 12.1.3(b)). This reduces the differential pressure across the valve, enabling the disc to be easily lifted off its seat (Figure 12.1.3(c)). To assist closing of the valve, isolation valves fitted with a balancing plug have to be fitted in reverse so that the top of the plug is acted on by the upstream pressure.



(a) Valve closed



(b) Pilot valve open reducing pressure drop across the valve



(c) Main valve open

Fig. 12.1.3 Schematic of a typical balancing plug valve

Piston valves

One of the main disadvantages of linear movement valves is the fact that their seats are prone to damage from dirt and wiredrawing, and therefore, depending on the application may require regular maintenance. Although these seats are replaceable in theory, it usually involves significant time and cost, and it is often more advantageous to replace the entire valve. To overcome this problem, piston valves have been developed.

The piston valve is a variant of the conventional globe valve, with the traditional seat and cone replaced by a piston and lantern bush. The piston is connected to the valve stem and handwheel, and passes through two sealing rings that are separated by a lantern bush. When assembled, the two sets of sealing rings are compressed around the piston by the load exerted along the stem. The upper set of sealing rings acts as conventional gland packing, and the lower set acts as the seat. Furthermore, the large sealing area between the piston and rings assures a high level of shut-off tightness.

The piston valve is not designed for throttling duties and must be used in the fully open or closed positions. When the valve is fully opened, only the bottom face of the piston is exposed to the fluid as the rest of the body is protected by the upper sealing rings. This means that the sealing surfaces (the sides of the piston) are protected from erosion by the fluid flow.

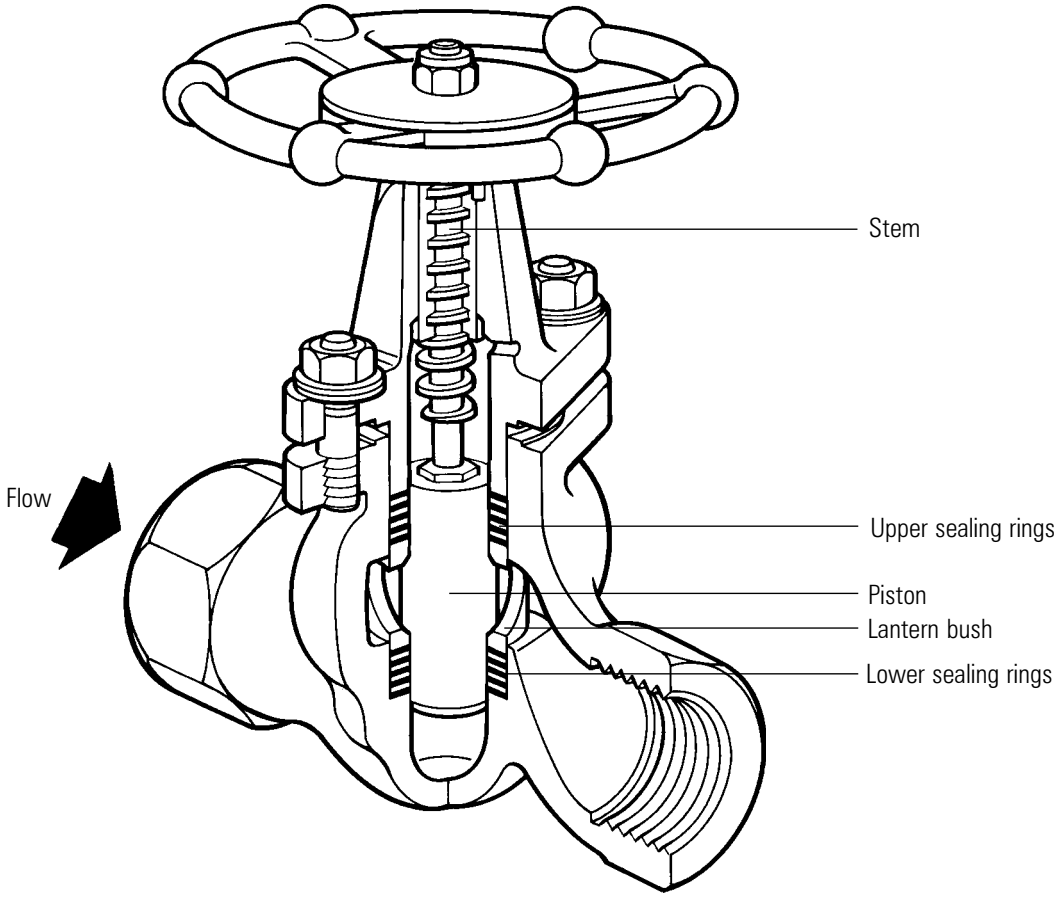


Fig. 12.1.4 A piston valve

If the valve requires maintenance, all the internals can be easily removed by undoing the cover nuts and withdrawing the piston. The rings and the lantern bush can then be removed using an extractor tool. This operation is simple and can be undertaken without having to remove the valve from the pipeline. In general, the piston should never have to be replaced, but the sealing rings may wear over a long period with frequent operation.

Diaphragm valves

Diaphragm valves constitute the third major type of linear movement valves. The stem of the valve is used to push down a flexible diaphragm, which in turn blocks the path of the fluid. There are two different classifications of diaphragm valve based on the geometry of the valve body:

- **Weir type -** A weir is cast into the body, and when closed, the diaphragm rests on the weir, restricting the flow (see Figure 12.1.5 (a)).
- **Straight-through type -** The bore runs laterally through the body and a wedge shaped diaphragm is used to make the closure (see Figure 12.1.5 (b)).

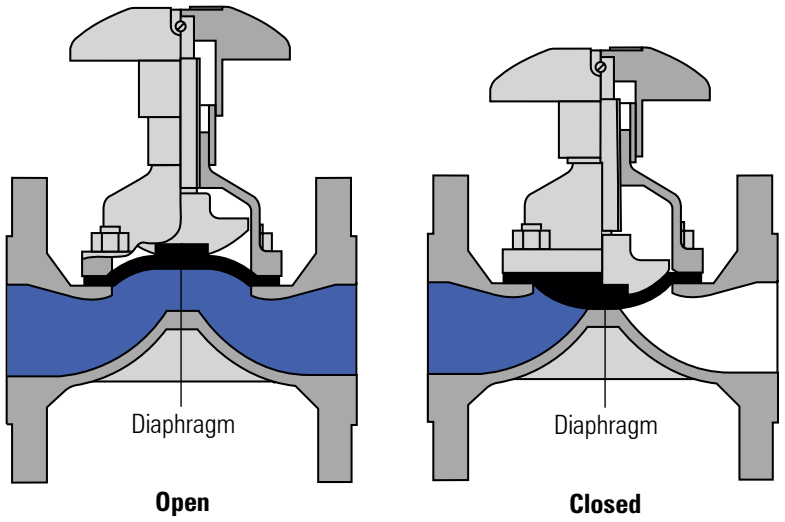


Fig. 12.1.5 The weir type (a) and straight-through type (b) diaphragm valves

The main advantage of a diaphragm valve is the fact that the diaphragm isolates the moving parts of the valve from the process fluid. They are therefore suitable for handling aggressive fluids and for those containing suspended solids. In addition, as the bonnet assembly is not exposed to the fluid, it can be made from inexpensive materials such as cast iron, thereby reducing the overall cost. The development of new diaphragm materials enables diaphragms to be used on most fluids. Their application is however limited by the temperature that the diaphragm can withstand - typically less than 175°C. Diaphragm valves are generally used on process fluid applications.

Linear movement valve stem options

Linear movement valves are available with a number of different stem arrangements:

- **Rising/non-rising stems -** If the stem is rising, it will move vertically upwards when the valve is opened, as opposed to only rotating, as with a non-rising stem. The rising stem indicates the degreee of valve opening, which in turn roughly reflects the amount of flow through the valve. Valves with rising stems do however require more space above the bonnet to accommodate the stem in the fully open position. The use of non-rising stems is recommended on gland packed valves, as they reduce the wear on the packing.

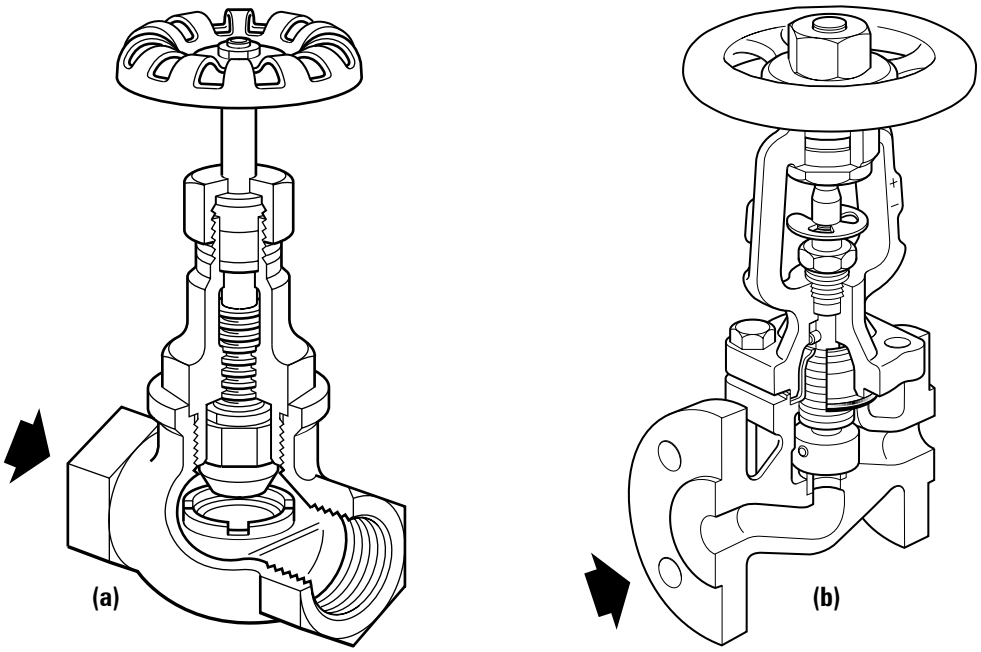


Fig. 12.1.6 Rising (a) and non-rising (b) stem valves

- **Inside/outside stem screws -** On a stem with an outside screw, the actuating threads on the stem are situated outside the valve body and are not exposed to the process fluid. As screw threads are particularly susceptible to corrosion, outside screws should always be used on fluids with corrosive or erosive properties. They are also beneficial where the valve is frequently exposed to large temperature variations, as the expansion and contraction of the stem may cause binding of the threads inside the body.

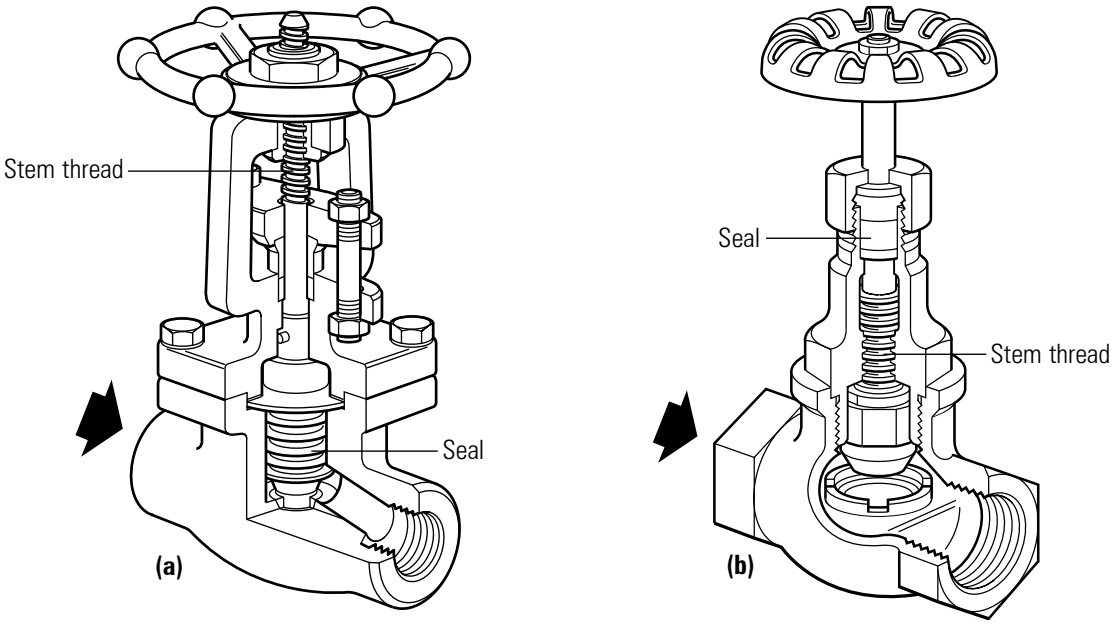


Fig. 12.1.7 Outside (a) and inside (b) stem valves

Stem sealing

In order to prevent leakage of the process media from around the stem of a valve, a barrier must be placed between the fluid and the environment. Stem sealing is usually achieved by one of two methods, namely gland packing and bellows sealing. Gland packing consists of a polymeric material, typically PTFE, packed tightly between the stem and the bonnet of the valve, thereby preventing any process media escaping.

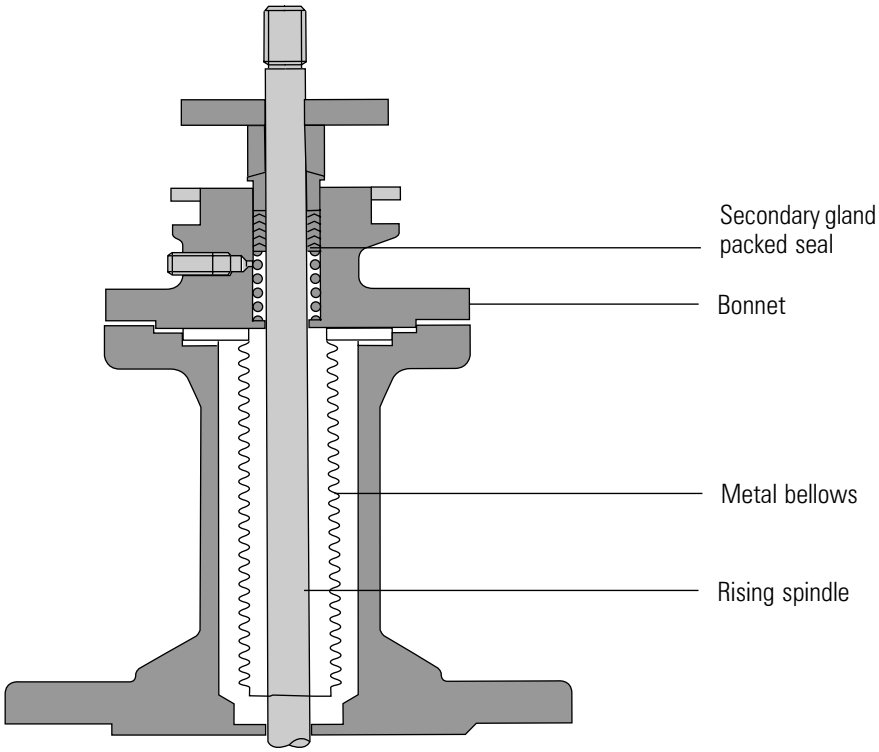


Fig. 12.1.8 Bellows sealed valve

In bellows sealed valves, a flexible metallic bellows is used. It is connected on one end to the stem and the other end is connected to the bonnet, effectively producing a barrier between the fluid and the environment. This bellows extends and contracts as the stem moves up and down. The bellows is so effective, it produces a 'zero emissions' seal. Fitted to the bellows is an anti-torsion device, which prevents the bellows from rotating with the stem. Such a device is essential, otherwise the repeated twisting of the bellows would lead to the failure of the seal.

Although less costly than the bellows sealed valves, the gland packed valve does not produce such a tight seal as the bellows. If a gland packed valve is not used for a significant period, the gland packing can stiffen, and leakage will occur the next time the valve is used. The bellows sealed valve does not suffer from this problem. Furthermore, gland packed valves require regular re-packing of the gland, whereas a typical bellows requires no maintenance for over 10 000 cycles.

Questions

1. What is the main advantage of a gate valve?

a| They are better than all the other linear movement valves for producing a tight shut-off on steam systems ☐

b| They can be used in throttling applications as well as for isolation ☐

c| There is a low pressure drop across the valve ☐

d| They are easily automated ☐

2. In which of the following applications should an outside, non-rising stem be used?

a| Where a gland packed valve is used in a corrosive fluid ☐

b| Where a bellows sealed valve is used in a steam system ☐

c| Where there are no temperature variations of the fluid passing through the valve ☐

d| Where a bellows sealed valve is used in a corrosive fluid ☐

3. Why must balancing plugs be used in globe valves that are larger than DN250?

a| The pre-lifting plug enables more precise control of the fluid ☐

b| It reduces the pressure drop across the valve allowing the valve to open easily ☐

c| It allows the valve to be balanced on water circuits ☐

d| A balancing plug has to be used with a bellows seal ☐

4. What is the main reason for choosing a bellows sealed stem over a gland packed one?

a| A bellows seal will never require maintenance ☐

b| The bellows seal produces a 'zero emissions' seal ☐

c| Gland packed seals on valves above DN250 are prone to leakage ☐

d| All of the above ☐

5. Which of the following valves should be used where the valve is to be welded into a pipeline and rapid seat wear is expected?

a| A globe valve ☐

b| A parallel-side valve ☐

c| A diaphragm valve ☐

d| A piston valve ☐

6. Why is a diaphragm valve not suitable for most steam applications?

a| Condensate collects in the weir, increasing the pressure drop across the valve ☐

b| Diaphragm valves are incapable of producing a tight shut-off above 4.0 bar ☐

c| The diaphragm valve is only suitable for handling fluids containing suspended solids ☐

d| Diaphragm materials are not suitable for temperatures above 175°C ☐

Answers

1: c, 2: a, 3: b, 4: b, 5: d, 6: d

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Module 12.2

Isolation Valves - Rotary Movement

Rotary Movement Valves

Rotary movement valves, often called quarter-turn valves, include ball valves and butterfly valves. Regardless of the type of rotary movement valve, the obturator rotates about an axis perpendicular to the direction of flow. Fluid may flow through the obturator, as is the case with ball valves, or around it, as with butterfly valves. Rotary movement valves tend to have a simple operating mechanism and are therefore easy to automate and maintain.

Ball valves

Ball valves were developed during World War II and were initially intended for use in aircraft fuel systems, where weight and space are at a premium. They consist of a body which houses a rotating ball which has an orifice or bore machined directly through it. The ball is located in the body by two sealing rings.

Rotation of the ball through 90° opens and closes the valve and allows fluid to flow directly through the orifice. In the closed position, the blank sides of the ball block the inlet and the outlet preventing any flow. There are two basic designs of ball valves – the floating ball design, which relies on the valve seats to support the ball, and the trunnion mounted ball, which uses a trunnion to support the ball. Trunnion mounting is used on larger valves, as it can reduce the operating torque to about two-thirds of that provided by a floating ball.

Conventionally, the handle that is attached to the ball is in-line with the axis of the pipe when the valve is open; conversely, if it is at right angles to the pipe axis, this indicates that the valve is closed.

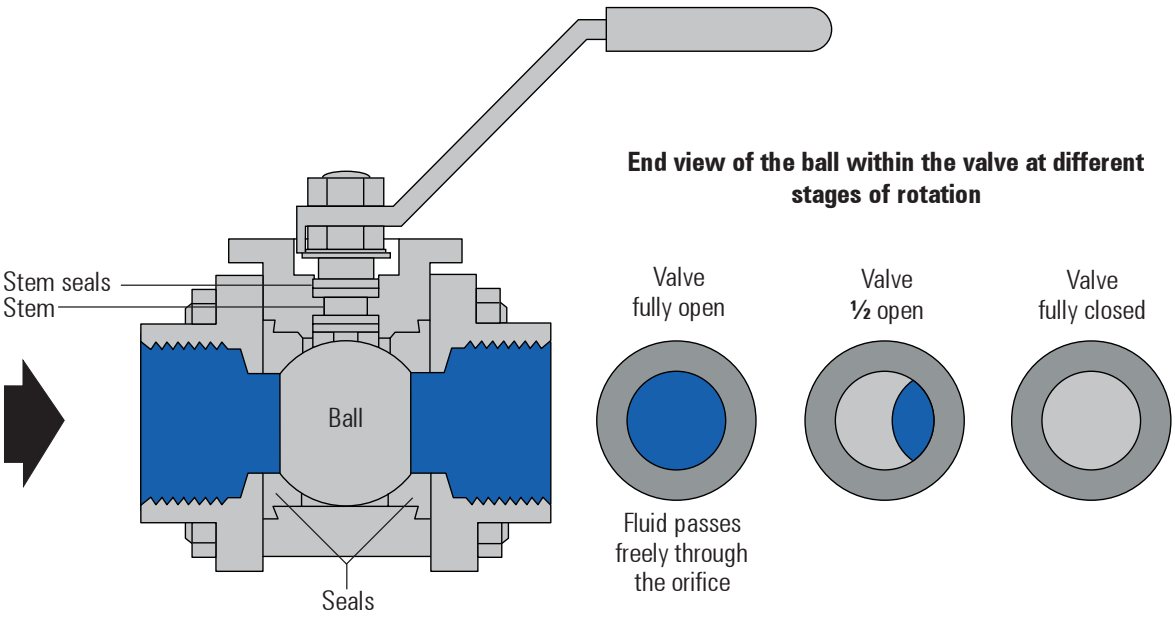


Fig. 12.2.1 Ball valve (shown in its open position)

Ball valves are available as reduced bore or full bore. Full bore valves have an orifice that is the same size as the diameter of the pipe, whereas in reduced bore valves, the orifice diameter is less than that of the pipe. Full bore valves cost more than reduced bore valves, and they should be used where the pressure drop across the valve is critical or where ball valves are used upstream of flowmeters. Full bore valves can be used in flowmeter applications to minimise fluid turbulence upstream of the measuring device.

In order to insert the ball into the body, three different types of assembly exist. Not only does the type affect the ease of assembly, but it also influences the maintainability of the valve.

- **Two and three piece valves -** The body of the valve is split in one or two places in the same plane as the valve flange, and these pieces are bolted together. This has the advantage of simplified, in-line maintenance.
- **Top entry valves -** The ball is inserted through a bonnet in the top of the valve. This facilitates in-line maintenance.
- **Single piece valves -** The ball is enclosed in the body by an insert fitted along the valve's axis. This eliminates the possibility of body joint leakage and any chance of disconnection whilst in service, but when maintenance is required, the whole valve has to be removed from the pipeline.

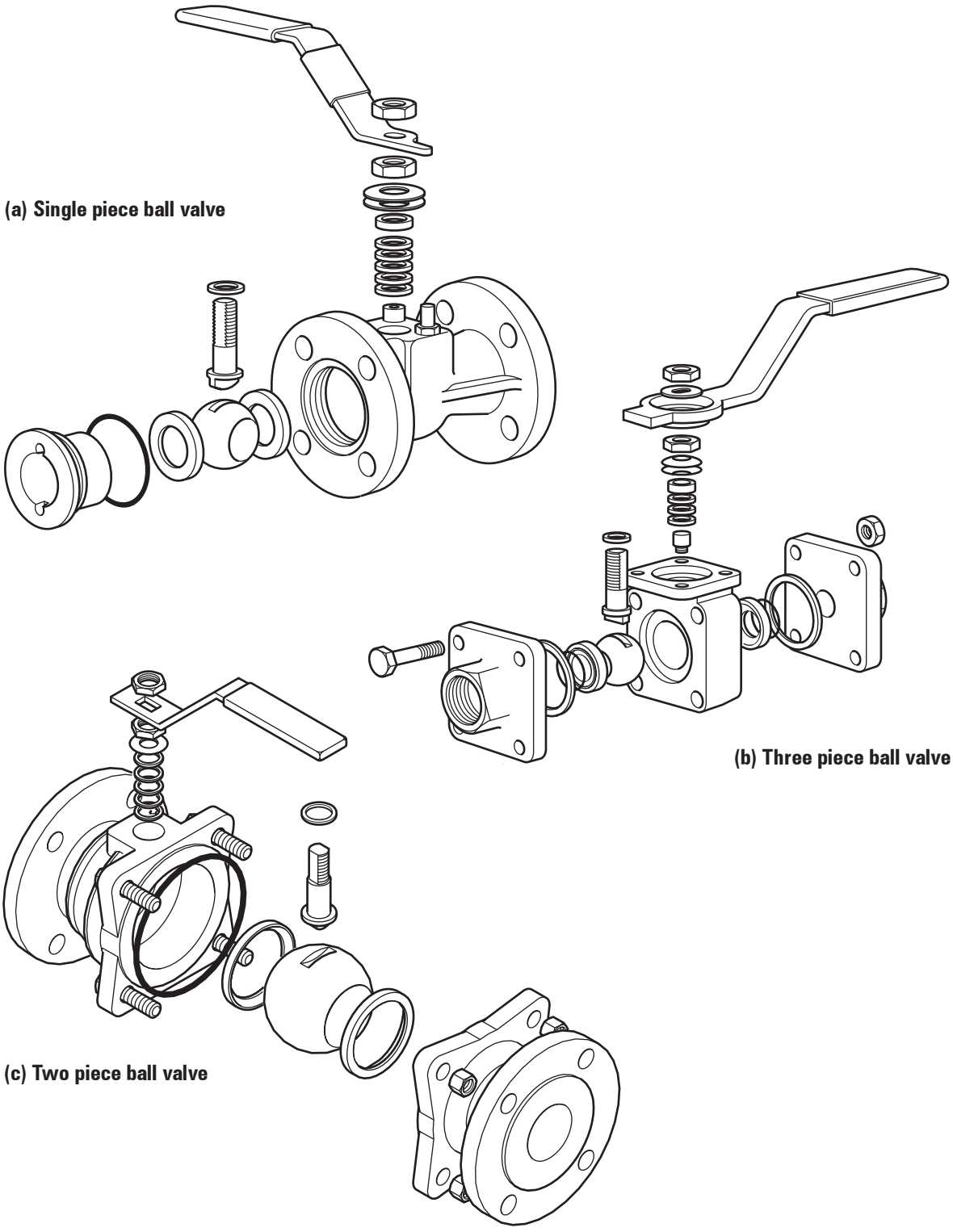


Fig. 12.2.2 Single piece (a) three piece (b) two piece (c) ball valves

The choice of seat material determines the conditions for which a particular ball valve is most suited. Although new seat materials are continually being developed, Table 12.2.1 lists some of the more common materials in use today.

Table 12.2.1 Common ball valve seat materials

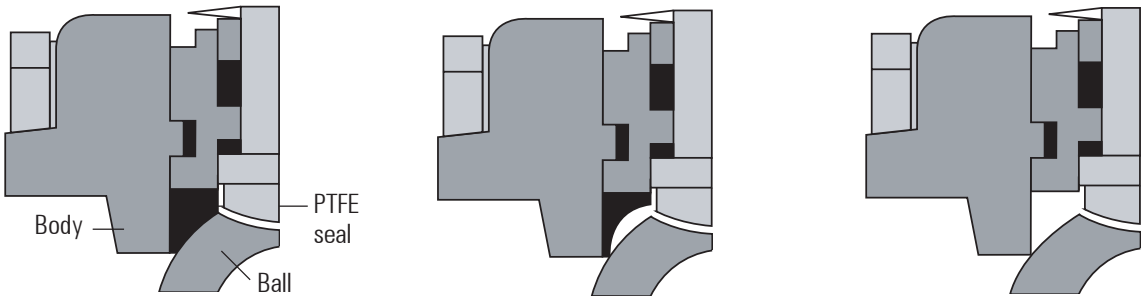
Application	Seat material	Maximum operating temperature
Low temperatures	PTFE	200°C
	Carbon reinforced PTFE	230°C
High pressures	Polyetheretherketone (PEEK)	250°C
High temperatures	Metal	1 000°C

Ball valve options

Ball valves can be produced with a number of options to meet the demands of a wide variety of applications:

- **Actuators -** Ball valves, and indeed all rotary valves, are suitable for automation. This is usually accomplished by using either an electrically or pneumatically operated actuator. The actuator is connected to the valve through a linkage kit. Although not essential, an ISO standard mounting pad enables the linkage kit to be installed without dismantling the valve, which maintains valve integrity. Refer to Module 6.6 for more information on actuators.
- **Firesafe -** As ball valves are commonly used in gas and oil pipelines, it is essential that the valves used in such applications are firesafe. A valve is considered firesafe if, when exposed to fire conditions, it will continue to provide minimal leakage through the seat and stem, and provide effective shut-off during or following a fire or exposure to excessive temperatures. Standards relating to fire-safety are set out in BS 6755 and API RP 6FA.

  The main concern is that burning temperatures will destroy soft seats and seals; a number of methods have been developed to overcome this. One approach is to include secondary metal sealing surfaces behind the polymeric seats as an integral part of the body. When exposed to burning temperatures, the seat begins to deform and the pressure of the process media displaces the ball so that it extrudes the polymeric seat (Figure 12.2.3(b)). When the seat has been completely destroyed, the ball will seat against the body metal sealing surface, providing a tight shut-off (Figure 12.2.3(c)).



(a) PTFE seal intact

(b) PTFE seal melting

(c) PTFE seal destroyed and a metal-to-metal seal is established

Fig. 12.2.3 Operation of a firesafe ball valve

In addition to the inherent safety of the seating mechanism, the stem seal must also be capable of preventing leakage to atmosphere under 'fire' conditions. This can be achieved by using high temperature seals made from flexible graphite or Grafoil®; alternatively, a bellows sealed arrangement can be used (see Figure 12.2.4).

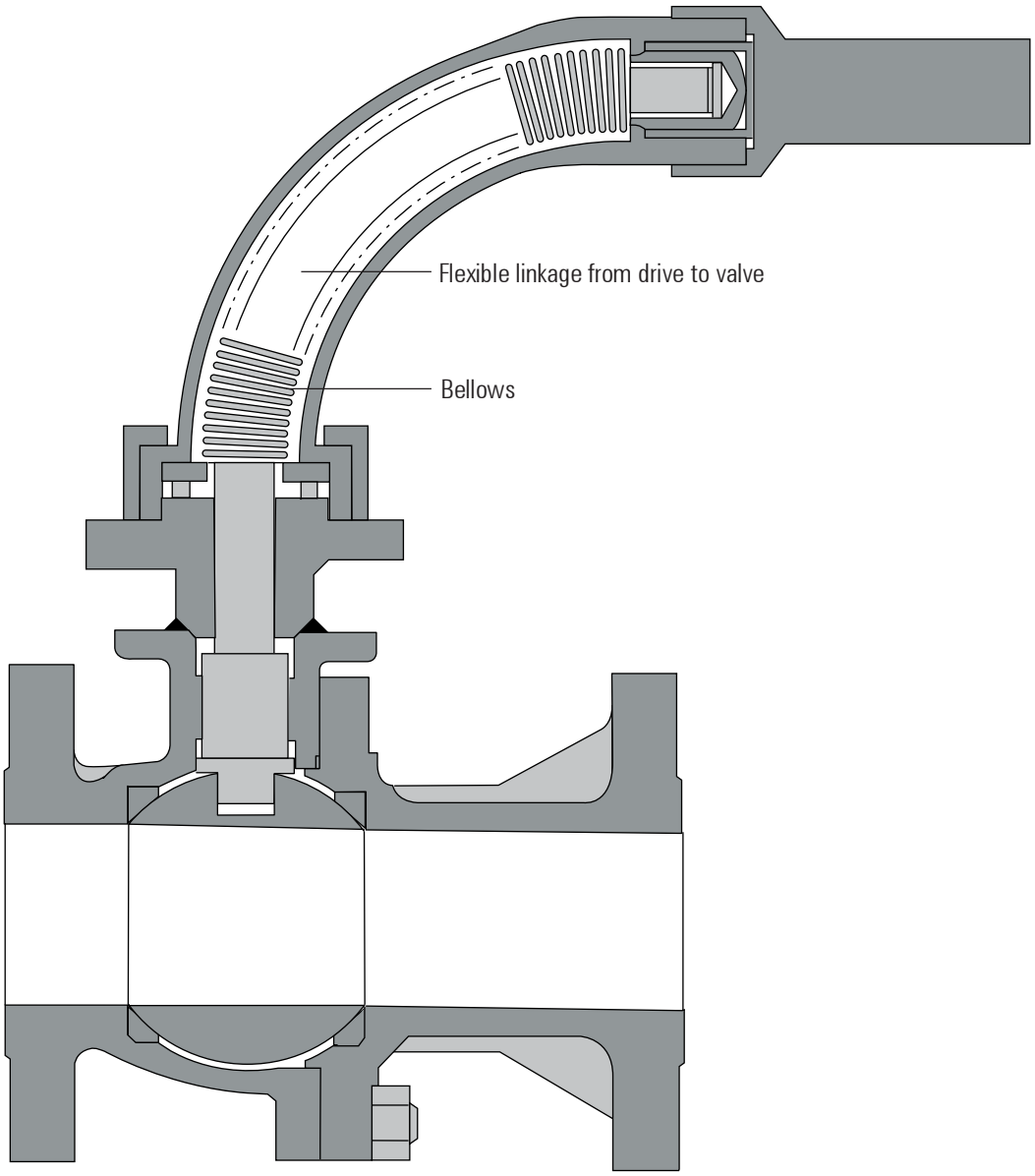


Fig. 12.2.4 A bellows sealed ball valve

- **Clean steam valves -** A number of applications exist that require the valve to be of a 'clean' design; these include steam applications where there is direct injection of steam into the product and process fluid lines in the biotechnology, food and electronics industries. The main area of concern in such applications is the space between the body and the ball; process fluid may accumulate in these spaces leading to contamination and corrosion. This can be overcome by inserting cavity fillers in these spaces. The cavity filler may be an integral part of the seat or a separate component in the valve assembly. Furthermore, ball valves used in clean steam applications should be made from stainless steel with a good surface finish (less than 81 microns Ra is recommended).

- **Throttling applications -** When ball valves are used in throttling applications, high velocity flow can impinge against a localised area of the ball and seals, causing premature deterioration of the seating material. Modifications to the standard design are required for ball valves to be used for throttling; these include the use of metal seats, hard coatings and, sometimes, modifications to the ball, to give a characterised flow pattern.

Butterfly valves

Although there are many different designs of butterfly valve, they all consist of a disc that rotates on a shaft at right angles to the fluid flow. When open, the disc is edge-on to the flow and the fluid passes around it, offering limited resistance. In the closed position, the disc is rotated against a seat in the body of the valve.

Butterfly valves usually take up little more room than a pair of pipe flanges, and are therefore an attractive alternative to the ball valve where space is limited. In fact, some butterfly valves are designed specifically for insertion between pipe flanges, these are known as wafer butterfly valves.



Fig. 12.2.5 Butterfly valves

The main disadvantage of butterfly valves is that the shut-off is not as tight as that achieved by other valve types. This can be alleviated to an extent by offsetting the axis of rotation of the disc and using pressure assisted seats. By using an offset axis of rotation, a 'camming' action is generated, which means that the disc creates a tight seal with the seat during the last few degrees of shut-off. These high performance or eccentric-type butterfly valves have improved shut-off capabilities and their design enables them to be used for throttling.

For steam applications, butterfly valves have largely been superseded by ball valves. Butterfly valves are more commonly used in liquid systems or where space is limited. The compactness of butterfly valves means less material is required and they are therefore ideal where the application specifies the use of costly materials, for example, in seawater applications where nickel is specified.

Selection and sizing of isolation valves

A process fluid must be fully contained in a properly designed piping system to avoid endangering personnel and the environment, and contamination of the fluid itself. The pipeline system can have many potential leak paths, such as pipe joints, seams, equipment connections and, most importantly, valves. Valves can be one of the biggest contributors to plant problems if they are wrongly selected or are poorly designed or manufactured. Furthermore, a valve, when selected correctly for the application should last at least the life of the plant, if maintained properly.

When selecting an isolating valve for a particular application, a number of factors need to be considered; these are shown in Table 12.2.2, along with the valve selection parameter that is affected.

Table 12.2.2 Factors affecting the selection of an isolation valve

Factors affecting the selection of an isolation valve	Areas of concern	Affected parameter
Process medium	Fluid – liquid or gas Pressure Temperature Flowrate Corrosive Abrasion	Type of valve Material of construction Maintainability Valve size
Functional requirements	Speed of operation Fails – safe Frequency of operation Emission loss to atmosphere	Type of valve
Method of operation	Manual Pneumatic Electric Electropneumatic Hydraulic	Type of valve Type of actuator
Pipeline	Pipeline material Pipeline size Pressure loss	Valve size End connections Type of valve Material of construction Availability
Special requirements	Firesafe Free draining Antistatic	Cost Type of valve

Table 12.2.3 summarises the main characteristics of the different types of isolation valve.

Table 12.2.3 Typical sizes and operating ranges of isolation valves

Valve type	Size		Pressure range		Temperature range		Pressure drop[1]
	Min. (mm)	Max. (mm)	Min. (bar)	Max. (bar)	Min. (° C)	Max. (° C)	bar
Gate	3	2 250	>0	700	-196	675	0.007
Globe	3	760	>0	700	-196	650	0.590
Diaphragm	3	610	>0	21	-50	175	0.021
Ball (full bore)	6	1 220	>0	525	-55	300	0.007
Butterfly	50	1 830	>0	102	-30	538	0.120

[1] **Note:** Typical values for a DN150 bore valve passing saturated steam at 24 bar, flowing at 40 m/s.

Table 12.2.4 summarises the applications of the most common isolating valve types in use today.

Table 12.2.4 Applications of isolating valve types

Valve type	General applications	Actuation	Remarks
Globe valve	Shut-off/regulation of liquid / gas flow. Steam and condensate applications.	Usually manual, but may be: - Electric - Manual - Hydraulic - Pneumatic	Usually applied to higher pressure or high volume systems, due to cost. Less suitable for viscous or contaminated fluids.
Piston valve	Used fully open or fully closed for on/off regulation on steam, gas and other fluid services. Typically used on fluids that cause excessive seat wear.	Usually manual, but may be: - Electric - Manual - Hydraulic	Usually used where the valve body is to be permanently installed and maintenance needs to be minimised.
Gate valve	Normally used fully open or fully closed for on/off regulation on water, oil, gas, steam and other fluid services.	Usually manual, but may be: - Electric - Manual - Hydraulic	Not recommended as a throttling valve. Solid wedge gate is free from chatter and jamming. Parallel slide valve used in steam systems.
Butterfly valve	Shut-off and regulation in larger pipelines in waterworks, process industries, HPI, power generation.	Handwheel Electric motor Pneumatic actuator Hydraulic actuator Air motor	Relatively simple construction. Can be produced in very large sizes. Eccentric design essential for steam systems. Typically used on liquid systems.
Ball Valve	Wide range of applications in all sizes, including HPI. Steam and condensate applications.	Handwheel Electric motor Pneumatic actuator Hydraulic actuator	Can handle all fluid types. Limited maximum pressure rating.

Table 12.2.5 is a generalised guide to the selection of isolation valves for particular steam and condensate applications. It should be noted that the choice of isolation valve is subjective and different industries and those in different geographical regions have their own unique preferences.

Table 12.2.5 Selection of valves for steam/condensate isolation purposes

Note: In this table, bellows sealed refers to a bellows sealed globe valve and globe refers to a standard, gland packed globe valve.

Application	Choice	Standard application	Dead tight shut-off	Energy and maintenance savings	Zero emissions
Trap sets up to 100 mm	1st	< DN50 Ball > DN50 Globe	< DN25 Piston > DN25 Ball	< DN25 Piston > DN25 Ball	Bellows sealed Globe
	2nd	< DN50 Ball > DN50 Bellows sealed	Bellows sealed	Bellows sealed	< DN25 Piston > DN25 Ball
Mains and equipment < 50 mm	1st	Globe	Ball	Piston	Bellows sealed
	2nd	Ball	Piston	Bellows sealed	Piston
Mains and equipment 50 mm - 100 mm	1st	Bellows sealed	Bellows sealed	Bellows sealed	Bellows sealed
	2nd	Globe	Ball	Ball	Ball
Mains and equipment > 100 mm	1st	Bellows sealed	Bellows sealed	Bellows sealed	Bellows sealed
	2nd	Globe	Globe	Globe	Globe
Automated mains and equipment	1st	Bellows sealed	Bellows sealed	Bellows sealed	Bellows sealed
	2nd	Globe	Ball	Ball	Ball

Once the most suitable type of valve has been chosen, it is necessary to choose the correct size. Valves are typically sized according to the pipeline size. It is however advisable to check that the pressure drop across the valve (when it is fully open) is within acceptable limits. The pressure drop is a function of the valve flow coefficient (or K_{VS} value), the flowrate and the inlet pressure. Specification sheets usually contain data about the K_{VS} value when the valve is fully opened. With knowledge of the typical operating pressure, and the mass flowrate, it is possible to determine the pressure drop across a chosen valve. Alternatively, if the maximum acceptable pressure drop is known, it is possible to select a suitable valve size. Although there are many formulae and charts available to predict the relationship between flowrate and pressure drop, the following simplified empirical formula (Equation 3.21.1) produces reliable results for steam and is therefore commonly used:

$$\dot{m}_s = 12\ K_v\ P_1 \sqrt{1 - 5.67(0.42 - \chi)^2} \qquad \textbf{Equation 3.21.2}$$

Where:

$\dot{m}_s$ = Mass flowrate in kg/h
K_v = Valve flow coefficient
χ = Pressure drop ratio = $\frac{P_1 - P_2}{P_1}$
P_1 = Upstream pressure in bar absolute
P_2 = Downstream pressure in bar absolute

This formula forms the basis of the chart shown in Figure 12.2.7, which was first introduced in Block 3, Module 21.

If the isolating valve is to be used in a liquid system, the pressure drop across the valve is determined using the following equation:

$$K_V = \dot{V} \sqrt{\frac{G}{\Delta P}} \qquad \textbf{Equation 6.3.1}$$

Where:

K_v = Valve flow coefficient (m^3/h bar)
$\dot{V}$ = Flowrate in m^3/h (m^3/h)
G = Relative density of liquid (non-dimensional)
ΔP = Pressure drop across the valve in bar (bar)

Rearranging the formula gives:

$$\Delta P = G\left(\frac{\dot{V}}{K_v}\right)^2 \qquad \textbf{Equation 12.2.1}$$

This sizing chart is empirical and should not be used for critical applications

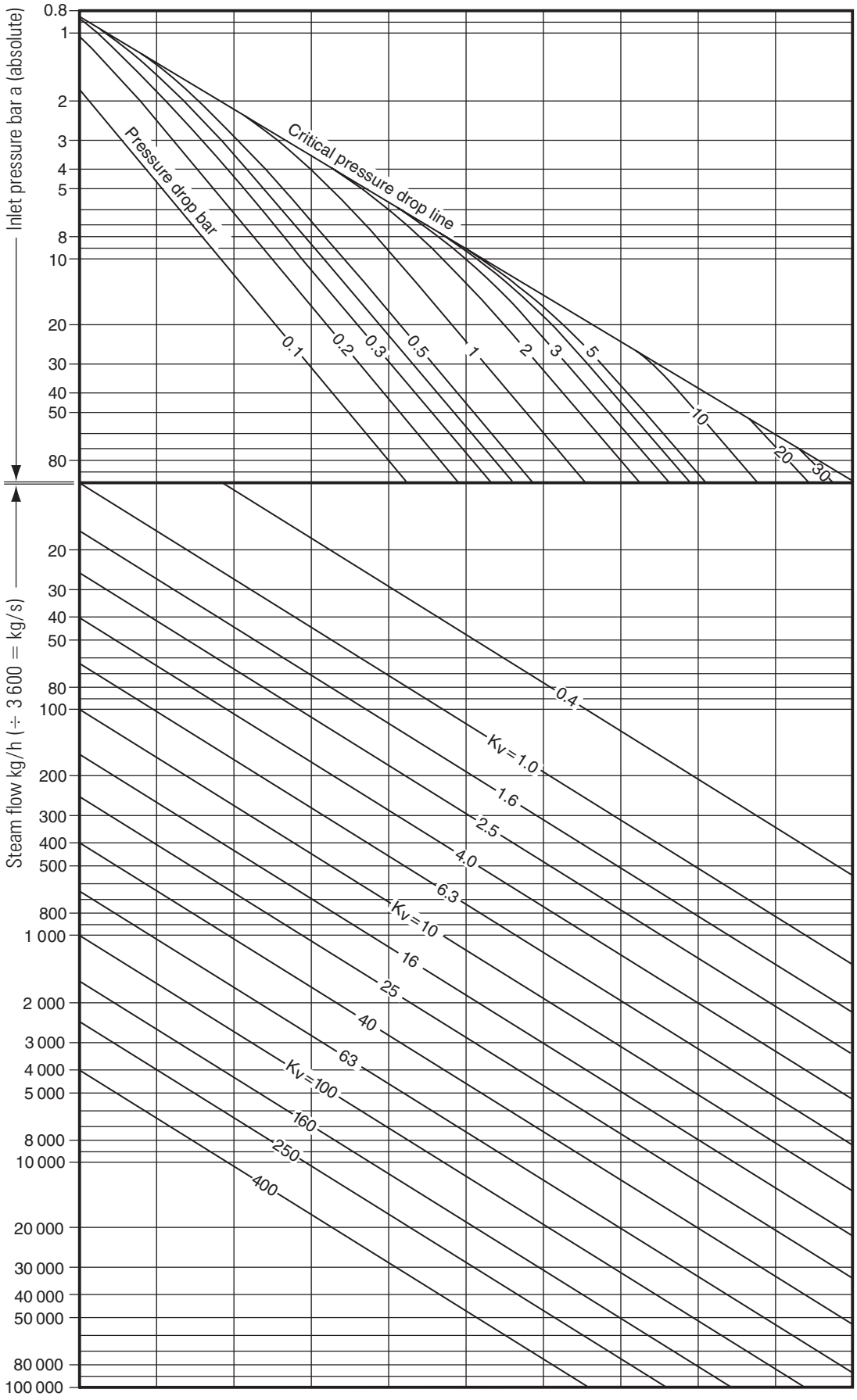


Fig. 12.2.6 Saturated steam sizing chart

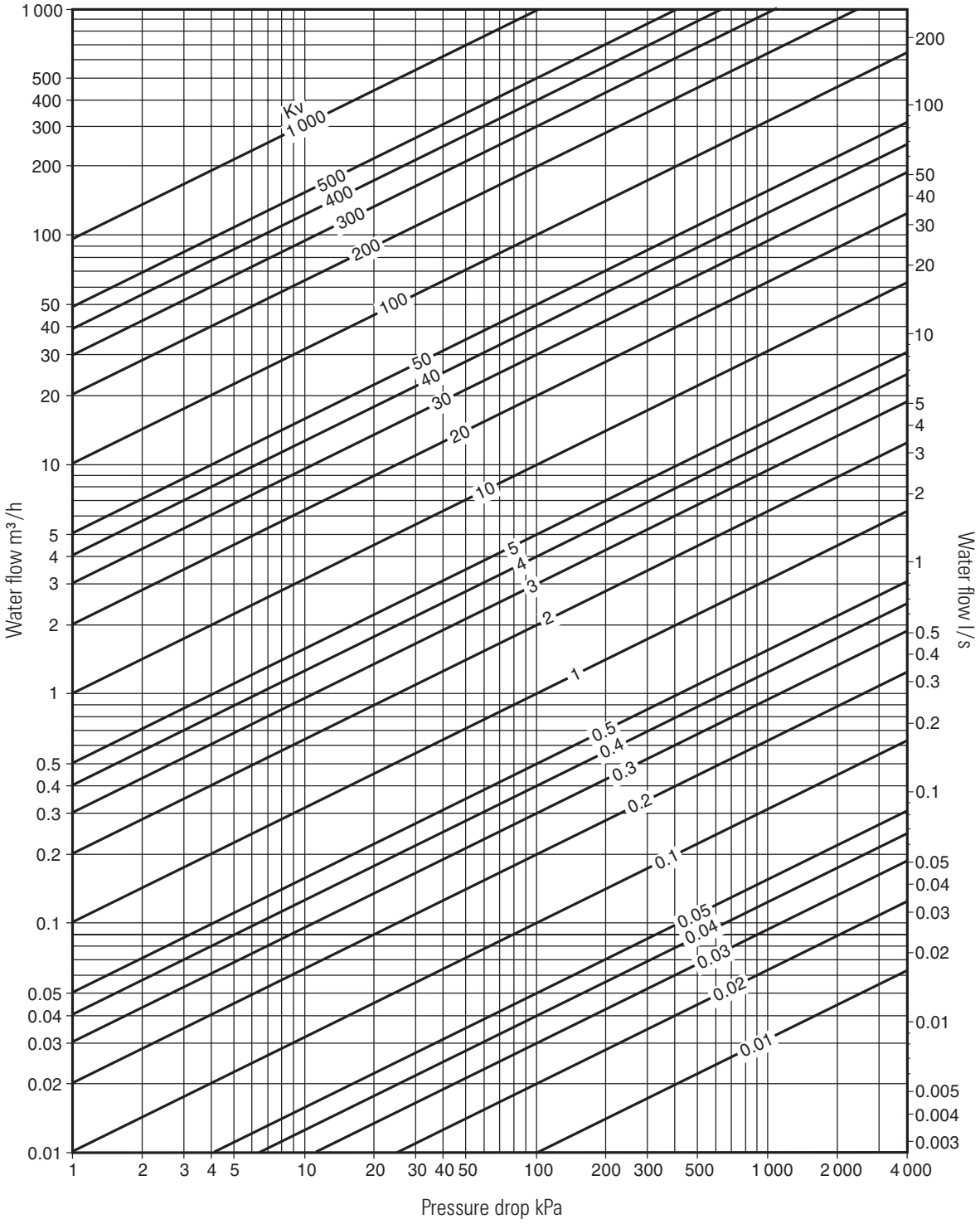


Fig. 12.2.7 Water sizing chart

Questions

1. Which of the following situations would warrant the use of a full bore ball valve?

a| Upstream of a flowmeter ☐

b| For isolation of plant when cost is an important consideration ☐

c| After a steam trap set ☐

d| The end of a steam main ☐

2. What is the main advantage of a three piece ball valve over a one piece ball valve?

a| Eliminates the chance of disconnections whilst in service ☐

b| Each piece can be selected individually to customise the valve to suit a unique application ☐

c| Higher valve integrity ☐

d| Easier in-line maintenance ☐

3. Which application would a standard butterfly valve be most suitable for?

a| Temperature control ☐

b| In small mains applications ☐

c| Automated isolation of a large steam jacket ☐

d| In hazardous gas applications that require a dead tight seal ☐

4. What would be the K_V value of a steam isolation valve with a pressure drop of 0.3 bar? Given that it is to be used upstream of a heat exchanger with a steam demand of 3 000 kg/h and a supply pressure of 5 bar g.

a| 70 ☐

b| 88 ☐

c| 100 ☐

d| 420 ☐

5. A bellows sealed globe valve is available in sizes DN25, DN32, DN40 and DN50 and the table below shows the corresponding K_{VS} values?

Size	DN25	DN32	DN40	DN50
K_{VS} value	12	20	30	47

Choose the correct size globe valve if it is to be used downstream of a pressure reducing station passing 500 kg/h of steam at 10 bar a, given that the pressure drop across the chosen globe valve must be less than 0.1 bar.

a| DN25 ☐

b| DN32 ☐

c| DN40 ☐

d| DN50 ☐

6. Using selection tables determine the most suitable types of valve for use on a 150 mm steam main to give a dead tight shut-off?

a| Bellows sealed globe valve / globe valve ☐

b| Ball valve / bellows sealed valve ☐

c| Ball valve / piston valve ☐

d| Bellows sealed globe valve / eccentric butterfly valve ☐

Answers

1: a, 2: d, 3: c, 4: b, 5: b, 6: a

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Module 12.3

Check Valves

Check Valves

Check valves, or non-return valves, are installed in pipeline systems to allow flow in one direction only. They are operated entirely by reaction to the line fluid and therefore do not require any external actuation. In this text, the expected, or desired direction of flow is termed 'forward flow', flow in the opposite direction is 'reverse flow'.

There are a number of reasons for using check valves, which include:

- Protection of any item of equipment that can be affected by reverse flow, such as flowmeters, strainers and control valves.
- To check the pressure surges associated with hydraulic forces, for example, waterhammer. These hydraulic forces can cause a wave of pressure to run up and down pipework until the energy is dissipated.
- Prevention of flooding.
- Prevention of reverse flow on system shutdown.
- Prevention of flow under gravity.
- Relief of vacuum conditions.

Although check valves can effectively shut off reverse flow, they should never be used in place of an isolation valve to contain live steam, in a section of pipe.

As with isolation valves, there are a number of different check valve designs, each suited to specific applications. The different types of check valve and their applications are discussed in this module, along with the correct sizing method.

Lift check valves

Lift check valves are similar in configuration to globe valves, except that the disc or plug is automatically operated. The inlet and outlet ports are separated by a cone shaped plug that rests on a seat typically metal; in some valves, the plug may be held on its seat using a spring. When the flow into the valve is in the forward direction, the pressure of the fluid lifts the cone off its seat, opening the valve. With reverse flow, the cone returns to its seat and is held in place by the reverse flow pressure.



Fig. 12.3.1 A lift check valve

If a metal seat is used, the lift check valve is only suitable for applications where a small amount of leakage, under reverse flow conditions, is acceptable. Furthermore, the design of a lift check valve generally limits its use to water applications, subsequently, they are commonly used to prevent reverse flow of condensate in steam traps and on the outlets of cyclic condensate pumps.

The main advantage of the lift check valve lies in its simplicity, and as the cone is the only moving part, the valve is robust and requires little maintenance. In addition, the use of a metal seat limits the amount of seat wear. The lift check valve has two major limitations; firstly, it is designed only for installation in horizontal pipelines, and secondly, its size is typically limited to DN80, above which, the valve would become too bulky.

The piston-type lift check valve is a modification of the standard lift check valve. It incorporates a piston shaped plug instead of the cone, and a dashpot is applied to this mechanism. The dashpot produces a damping effect during operation, thereby eliminating the damage caused by the frequent operation of the valve, for example, in pipeline systems, which are subject to surges in pressure, or frequent changes in flow direction (one example would be a boiler outlet).

Swing check valves

A swing check valve consists of a flap or disc of the same diameter as the pipe bore, which hangs down in the flow path. With flow in the forwards direction, the pressure of the fluid forces the disc to hinge upwards, allowing flow through the valve. Reverse flow will cause the disc to shut against the seat and stop the fluid going back down the pipe. In the absence of flow, the weight of the flap is responsible for the closure of the valve; however, in some cases, closure may be assisted by the use of a weighted lever. As can be seen from Figure 12.3.2, the whole mechanism is enclosed within a body, which allows the flap to retract out of the flow path.

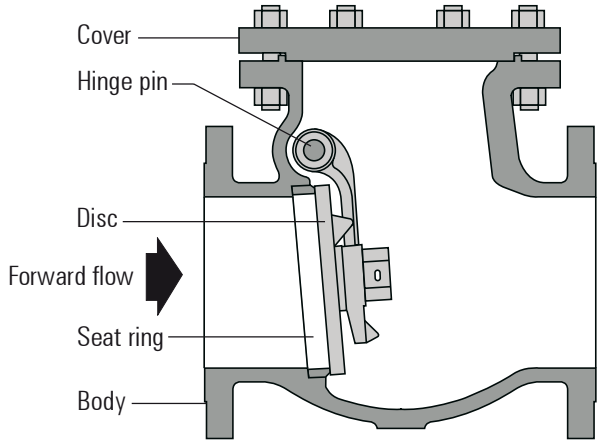


Fig. 12.3.2 A full-bodied, swing check valve

Swing check valves produce relatively high resistance to flow in the open position, due to the weight of the disc. In addition, they create turbulence, because the flap 'floats' on the fluid stream. This means that there is typically a larger pressure drop across a swing check valve than across other types.

With abrupt changes in flow, the disc can slam against the valve seat, which can cause significant wear of the seat, and generate waterhammer along the pipe system. This can be overcome by fitting a damping mechanism to the disc and by using metal seats to limit the amount of seat wear.

Wafer check valves

Both lift and swing check valves tend to be bulky which limits their size and makes them costly. To overcome this, wafer check valves have been developed. By definition wafer check valves are those that are designed to fit between a set of flanges. This broad definition covers a variety of different designs, including disc check valves and wafer versions of swing or split disc check valves.

Disc check valves

The disc check valve consists of four main components: the body, a disc, a spring and a spring retainer. The disc moves in a plane at right angles to the flow of the fluid, resisted by the spring that is held in place by the retainer. The body is designed to act as an integral centring collar that facilitates installation. Where a 'zero leakage' seal is required, a soft seat can be included.

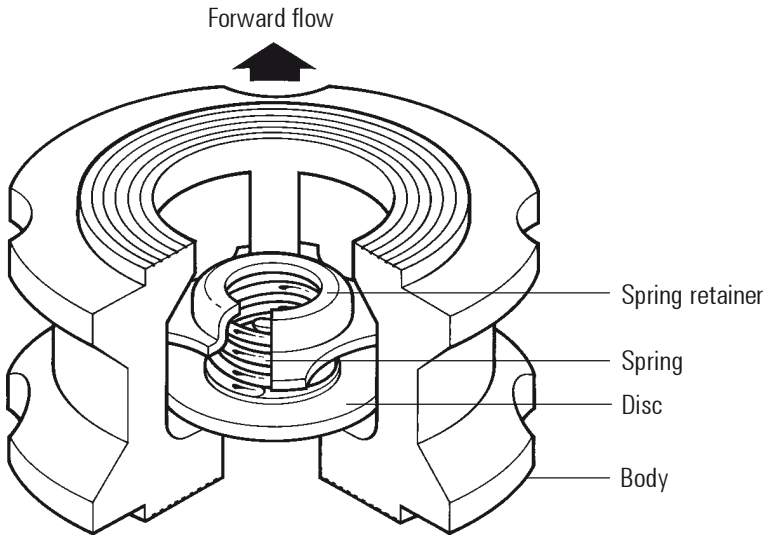


Fig. 12.3.3 A disc check valve

When the force exerted on the disc by the upstream pressure is greater than the force exerted by the spring, the weight of the disc and any downstream pressure, the disc is forced to lift off its seat, allowing flow through the valve. When the differential pressure across the valve is reduced, the spring forces the disc back onto its seat, closing the valve just before reverse flow occurs. This is shown in Figure 12.3.4. The presence of the spring enables the disc check vale to be installed in any direction.

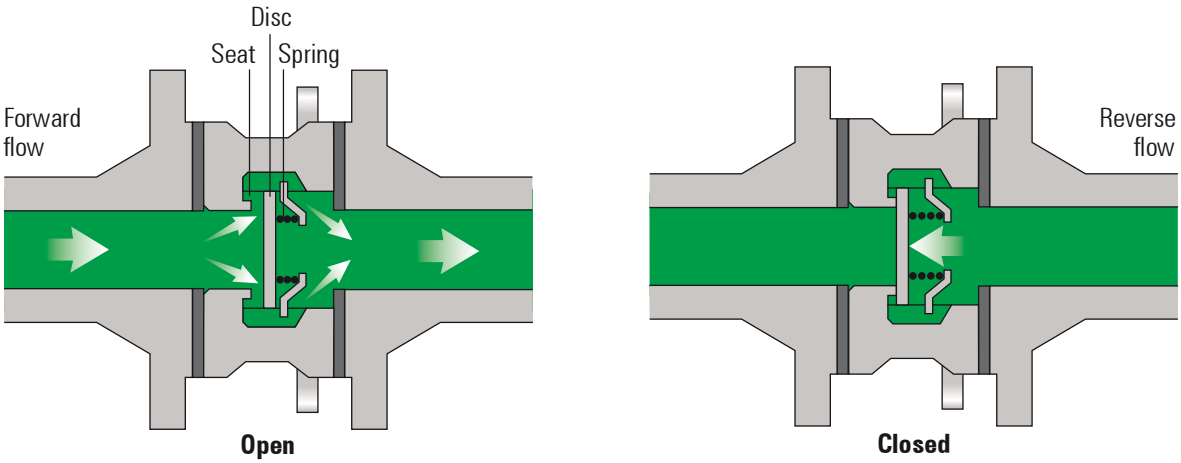


Fig. 12.3.4 Operation of a disc check valve

The differential pressure required to open the check valve is mainly determined by the type of spring used. In addition to the standard spring, there are several spring options available:

- **No spring -** Used where the differential pressure across the valve is small.
- **Nimonic spring -** Used in high temperature applications.
- **Heavy-duty spring -** This increases the required opening pressure. When installed in the boiler feedwater line, it can be used to prevent steam boilers from flooding when they are unpressurised.

As with all wafer check valves, the size of the disc check valve is determined by the size of the associated pipework. This usually ensures that the valve is correctly sized, but there are cases where the valve is over or undersized.

An oversized check valve is often indicated by continuous valve chatter, which is the repeated opening and closing of the valve that occurs when the valve is only partially open. It is caused by the fact that when the valve opens, there is a drop in the upstream pressure; if this pressure drop means that the differential pressure across the valve falls below the required opening pressure, the valve will slam shut. As soon as the valve shuts, the pressure begins to build up again, and so the valve opens and the cycle is repeated.

Oversizing can usually be rectified by selecting a smaller valve, but it should be noted that this will increase the pressure drop across the valve for any one flow. If this is not acceptable, it may be possible to overcome the effects of chatter by reducing the closing force on the disc. This can be done either by using a standard spring instead of a heavy-duty one, or by removing the spring altogether. Another alternative is to use a soft seat; this does not prevent the chatter but rather, reduces the noise. Care must be taken however, as this may cause excessive wear on the seat.

Undersizing results in excessive pressure drop across the valve and, in the extreme, it may even prevent flow. The solution is to replace the undersized valve with a larger one.

Disc check valves are smaller and lighter than lift and standard swing check valves and subsequently cost less. The size of a disc check valve is however limited to DN125; above this, the design becomes complicated. Typically, such a design would include a cone shaped disc and a small diameter spring that is retained and guided along the centre line of the cone, which is more difficult and expensive to manufacture. Even then, such designs are still limited in size to DN250.

Standard disc check valves should not be used on applications where there is heavily pulsating flow, for example, on the outlet of a reciprocating air compressor, as the repeated impact of the disc can lead to failure of the spring retainer and high levels of stress in the spring. Specifically designed retainers are available for such applications. These designs typically reduce the amount of disc travel, which effectively increases the resistance to flow and therefore increases the pressure drop across the valve.

The design of disc check valves allows them to be installed in any position, including vertical pipelines where the fluid flows downwards.

Swing type wafer check valves

These are similar to the standard swing check valves, but do not have the full-bodied arrangement, instead, when the valve opens, the flap is forced into the top of the pipeline. Subsequently, the flap must have a smaller diameter than that of the pipeline, and because of this, the pressure drop across the valve, which is often high for swing type valves, is further increased.

Swing type check valves are used mainly on larger pipeline sizes, typically above DN125, because on smaller pipelines the pressure drop, caused by the disc 'floating' on the fluid stream, becomes significant. Furthermore, there are significant cost savings to be made by using these valves on larger sizes, due to the small amount of material required for the construction of the valve.

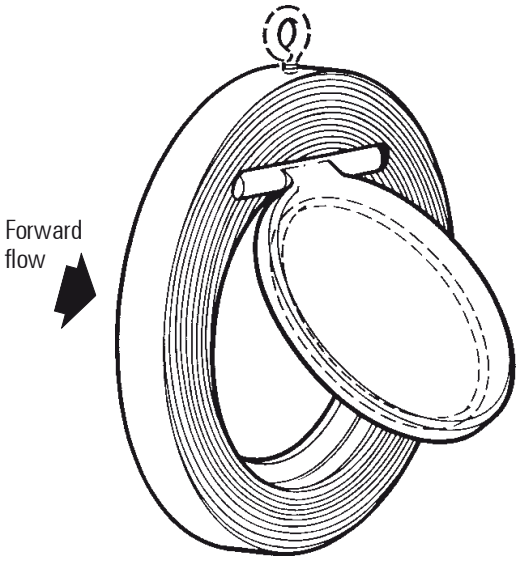


Fig. 12.3.5 Swing type wafer check valve

There is however one problem with using larger size valves; due to their size, the discs are particularly heavy, and therefore possess a large amount of kinetic energy when they close. This energy is transferred to the seat and process fluid when the valve slams shut, which could cause damage to the seat of the valve and generate waterhammer.

Wafer check valve applications

Wafer check valves are becoming the preferred type of check valve for most applications, due to their compact design and relatively low cost. The following is a list of some of their most common applications:

- **Boiler feedlines -** The check valve is used to prevent boiler water being forced back along the feedline into the storage tank when the feedpump stops running. Furthermore, a disc check valve with a heavy-duty spring and a soft seat can be fitted in the boiler feedline to prevent flow under gravity into the boiler when the feedpump is shut off.

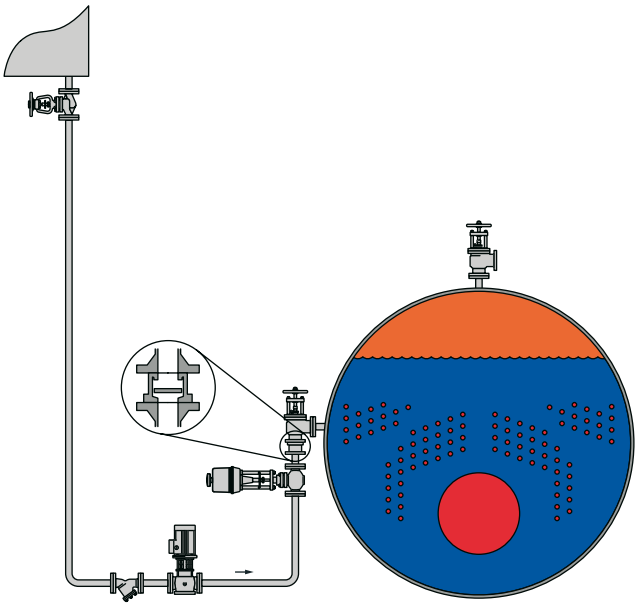


Fig. 12.3.6 Boiler feedline applications

- **Steam traps -** Other than with steam traps discharging to atmosphere, check valves should always be inserted after a steam trap to prevent back flow of condensate flooding the steam space. The check valve will also prevent the steam trap from becoming damaged by any hydraulic shock in the condensate line. It should be noted that when using blast discharge type steam traps, the check valve should be fitted at least 1 m downstream of the trap.

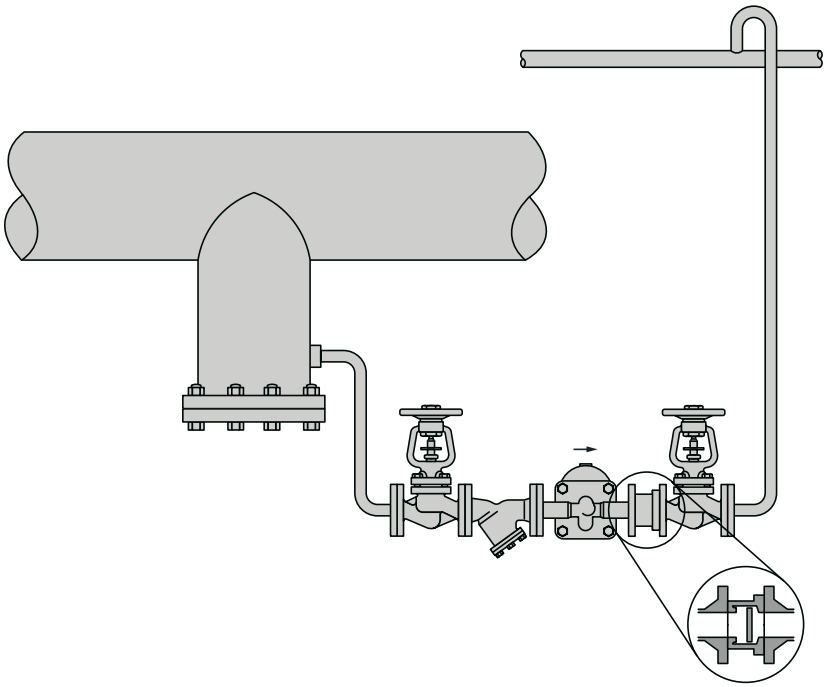


Fig. 12.3.7 Steam trap applications

- **Hot water circuits -** A check valve should be installed after each pump to prevent reverse flow through the pump when it has been shut off (see Figure 12.3.8).

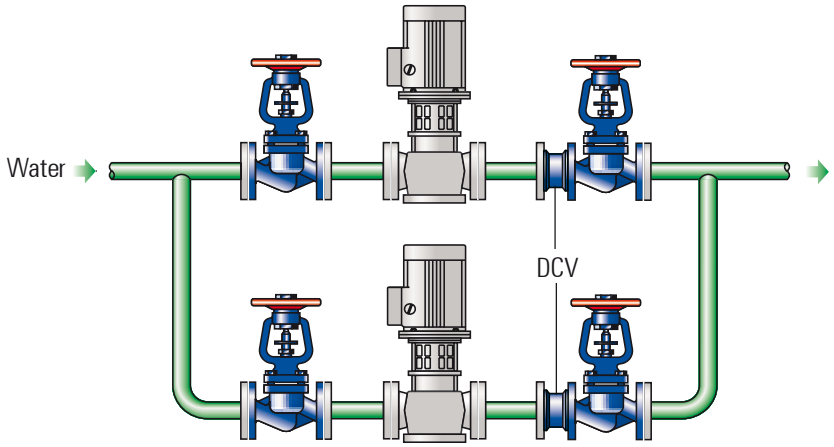


Fig. 12.3.8 Duplex pump set

- **Vacuum breakers -** Check valves can be used as vacuum breakers, by fitting them in reverse. When a vacuum is created, the valve opens, allowing air to be drawn in from the atmosphere (see Figure 12.3.9).

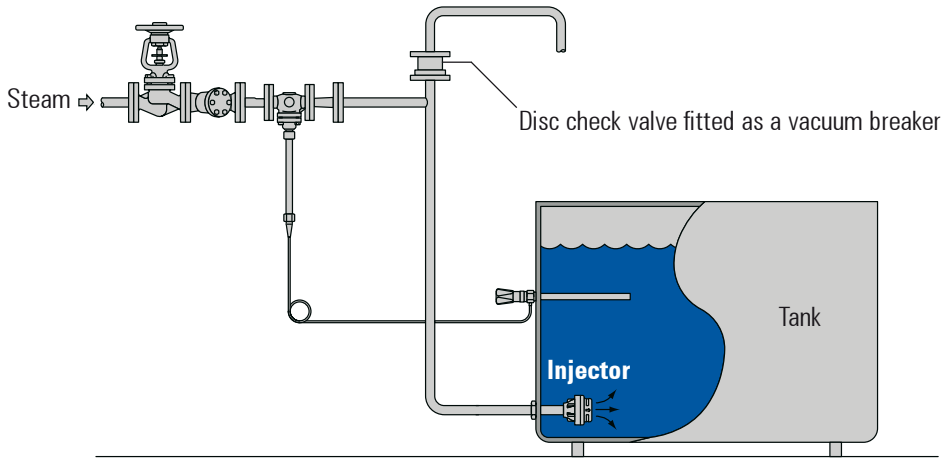


Fig. 12.3.9 Steam injection into a tank

- **Blending -** A check valve should be fitted in each supply line to prevent reverse flow along the different lines which will lead to contamination. A common blending application is the mixing of hot and cold water to provide hot water (see Figure 12.3.10).

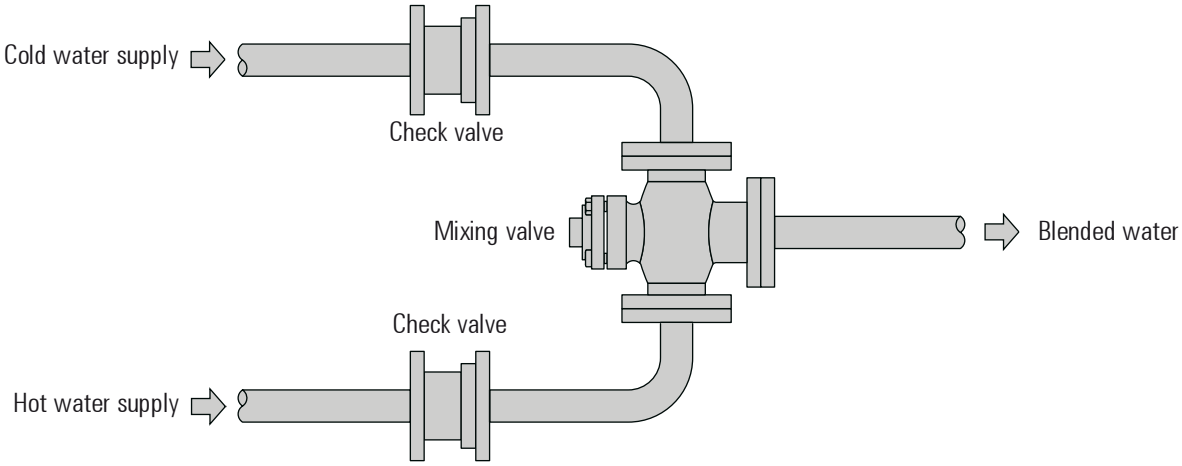


Fig. 12.3.10 Blending applications

- **Pipeline fitting protection -** Check valves are used to prevent damage to equipment such as flowmeters and control valves, all of which can be damaged by reverse flow. Check valves also stop the contents of strainers from being deposited in upstream pipework by back flowing fluid.
- **Multiple boiler applications -** A check valve must be inserted on the outlet of each boiler to prevent any steam flowing into boilers, which may be on hot stand-by (see Figure 12.3.11).

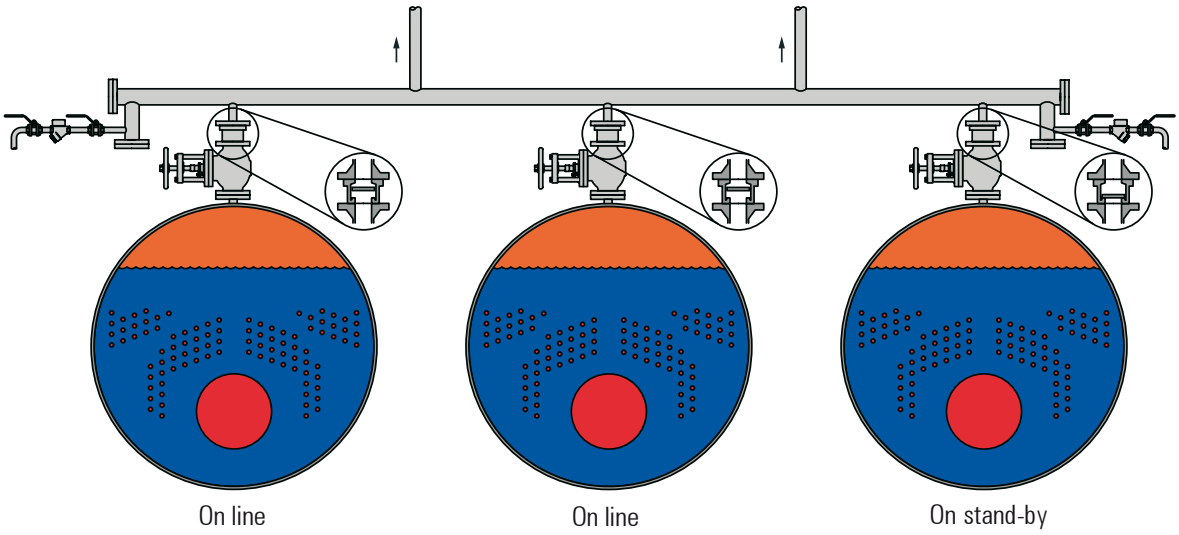


Fig. 12.3.11 Multiple boiler applications

- **Blowdown vessels -** When a blowdown vessel receives blowdown from more than one boiler, a wafer check valve should be installed on each separate blowdown line. This will prevent the blowdown from one boiler flowing back into another boiler. In many countries, this is a statutory requirement.
- **Flash vessels -** A wafer check valve is installed at the flash steam outlet from the flash vessel; this ensures that steam from any make-up valve does not flow back into the flash vessel (see Figure 12.3.12). A check valve is also installed after the steam trap that drains the flash vessel.

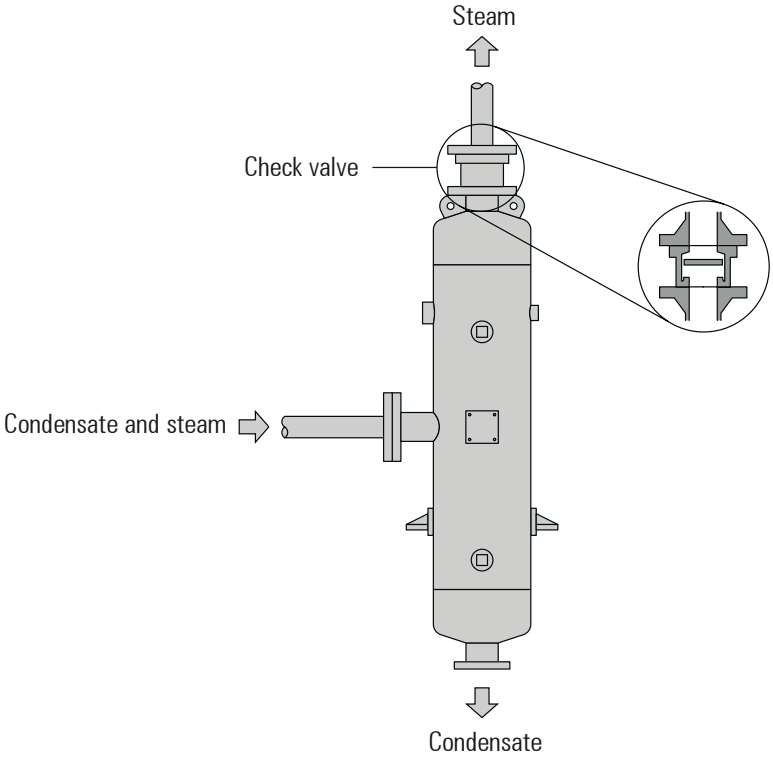


Fig. 12.3.12 Flash vessel applications

Split disc check valves

The split disc check valve or dual plate check valve is designed to overcome the size and pressure drop limitations of the swing and disc type wafer check valves. The flap of the swing check valve is essentially split and hinged down its centre, such that the two disc plates will only swing in one direction. The disc plates are held against the seat by a torsion spring mounted on the hinge.

In order to hold the hinge in the centre of the flow path, externally mounted retainer pins can be used. These retainer pins are a common source of leakage from the valve. An improved design secures the hinge internally, and as the valve mechanism is entirely sealed within the body, leakage to atmosphere is prevented (see Figure 12.3.13).



Fig. 12.3.13 A split disc check valve (retainerless design)

The valve is normally closed, as the disc plates are kept shut by the torsion spring. When fluid flows in the forwards direction, the pressure of the fluid causes the disc plates to hinge open, allowing flow. The check valve is closed by the spring as soon as flow ceases, before any reverse flow can occur.

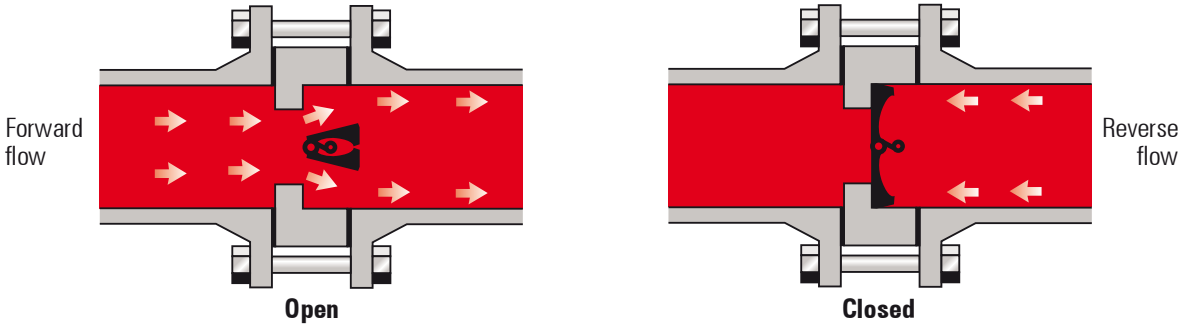


Fig. 12.3.14 Operation of a split disc check valve

The frequent opening and closing of the split disc check valve would soon cause seat damage if the heels of the disc plates were allowed to scuff against the seat during opening. To overcome this, the heel of the disc plates lift during the initial opening of the valve and the plates rotate purely on the hinge as opposed to the seat face.

The split disc type of check valve has several advantages over other types of check valves:

- The split disc design is not limited in size and these valves have been produced in sizes of up to DN5400.
- The pressure drop across the split disc check valve is significantly lower than across other types.
- They are capable of being used with lower opening pressures.
- Split disc check vales can be installed in any position, including vertical pipelines.

Other check valve types

The above mentioned types of check valve are the most commonly encountered types in steam, condensate, and liquid systems. However, several other types are also available. The three types listed below are mainly suited to liquid applications and subsequently may be found in condensate systems:

- **Ball check valve -** This consists of a rubber-coated ball that is normally seated on the inlet to the valve, sealing off the inlet. When pressure is exerted on the ball, it is moved off its seat along a guide rail, allowing fluid to pass through the inlet. When the fluid pressure drops, the ball slides back into its position on the inlet seat. **Note:** Ball check valves are typically only used in liquid systems, as it is difficult to obtain a tight seal using a ball.
- **Diaphragm check valve -** A flexible rubber diaphragm is placed in a mesh or perforated cone with the point in the direction of flow in the pipeline (see Figure 12.3.15). Flow in the forwards direction deflects the diaphragm inwards, allowing the free passage of the fluid. When there is no flow or a backpressure exists, the diaphragm returns to its original position, closing the valve. **Note:** The diaphragm material typically limits the application of the diaphragm check valve to fluids below 180°C and 16 bar.

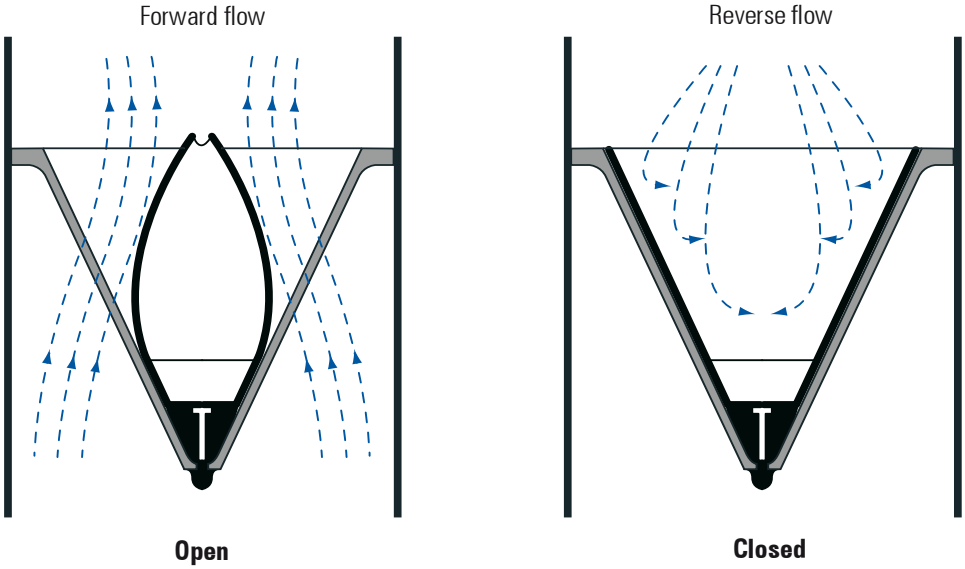


Fig. 12.3.15 A diaphragm check valve

- **Tilting disc check valve -** This is similar to the swing type check valve, but with the flap pivoted in front of its centre of pressure and counterweighted or spring loaded to assume a normally closed position (see Figure 12.3.16). When flow is in the forwards direction, the disc lifts and 'floats' in the stream offering minimum resistance to flow. The disc is balanced so that as flow decreases, it will pivot towards its closed position, closing before reverse flow actually commences. The operation is smooth and silent under most conditions. **Note:** due to the design of the tilting disc check valve, it is limited to use on liquid applications only.

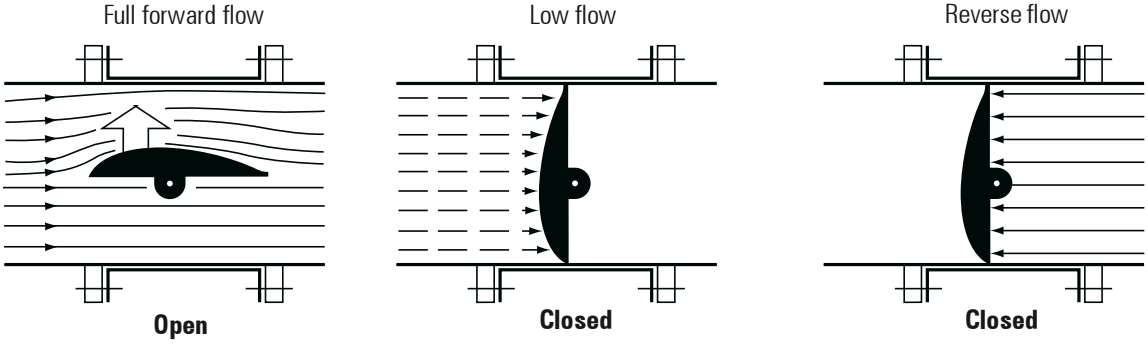


Fig. 12.3.16 Operation of a tilting disc check valve

Pressure loss charts

As most types of check valve are suitable for use on both liquid and gas systems, manufacturers typically show the pressure drop across a valve in the form of a pressure loss chart for water. A typical pressure loss chart is shown in Figure 12.3.17. It shows the pressure drop across a particular check valve for a given valve size and water flowrate in m³/h.

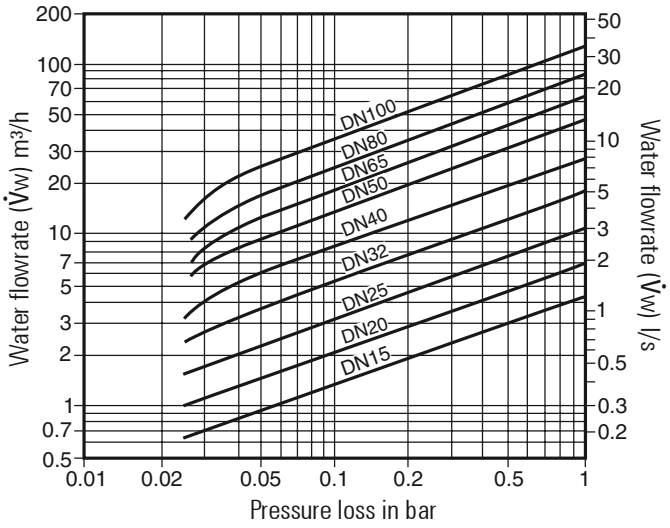


Fig. 12.3.17 A typical manufacturer's pressure loss diagram

In order to determine the pressure drop across the check valve for other liquids, the equivalent water volume flowrate needs to be calculated, this is done using the formula in Equation 12.3.1:

$$\dot{V}w = \sqrt{\frac{\rho}{1000}}\,\dot{V} \qquad \textbf{Equation 12.3.1}$$

Where:

$\dot{V}w$ = Equivalent water volume flowrate (m³/h)
ρ = Density of the liquid (kg/m³)
$\dot{V}$ = Volume flowrate of liquid (m³/h)

Once the equivalent water volume flowrate has been determined, the pressure drop across the valve can be read off the chart using the same method as for water, selecting the equivalent water volume flowrate instead of the actual volume flowrate.

It should be noted that the volumetric flowrate (in m³/h) is typically quoted for liquid applications, whereas, in steam applications, the mass flowrate (in kg/h) is normally used. To convert from kg/h to m³/h, the mass flowrate is multiplied by the specific volume (in kg/m³) for the particular working pressure and temperature (see Equation 12.3.2).

$$\dot{V} = \dot{m}\ \nu_g \qquad \textbf{Equation 12.3.2}$$

Where:

$\dot{V}$ = Volume flowrate (m³/h)
$\dot{m}$ = Mass flowrate (kg/h)
ν_g = Specific volume (m³/kg)

Alternatively, if the K_V value of the valve is specified, the pressure drop across the valve can be determined using the method outlined in Module 12.2.

Example 12.3.1
Determine the pressure drop across a DN65 check valve passing 1 200 kg/h of saturated steam at 8 bar g. Use the pressure drop characteristics shown in Figure 12.3.17.

Solution:
The first step is to calculate the volumetric flowrate:
From steam tables at 8 bar gauge, $\nu_g = 0.214\,9$ m³/kg

Using Equation 12.3.2

$$\dot{V} = \dot{m} \times \nu_g$$

$$\dot{V} = 1\,200 \text{ kg/h} \times 0.214\,9 \text{ m}^3/\text{kg}$$

$$\mathbf{\dot{V} = 257 \text{ m}^3/\text{h}}$$

The next step is to calculate the equivalent water volume flowrate:

Using Equation 12.3.1:

$$\dot{V}w = \dot{V}\sqrt{\frac{\rho}{1000}}$$

Since $\nu = 0.214\,9$ m³/kg, the density, $\rho = \frac{1}{0.2149} = 4.65$ kg/m³

$$\dot{V}w = 257\sqrt{\frac{4.65}{1\,000}} = 17.6 \text{ m}^3/\text{h}$$

$$\mathbf{\dot{V}w = 17.6 \text{ m}^3/\text{h}}$$

Using Figure 12.3.18, the pressure drop across the valve would be approximately 0.085 bar.

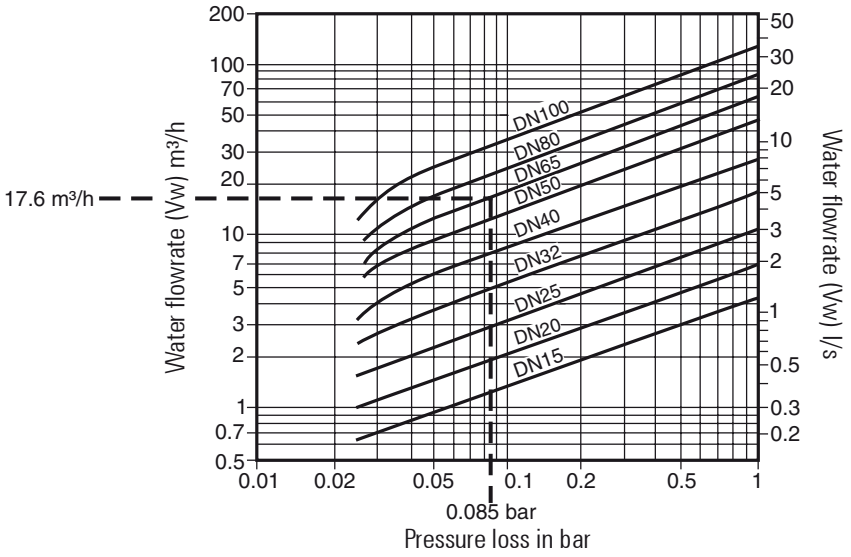


Fig. 12.3.18

Questions

1. Which of the following is not a suitable application of a check valve?

a| To prevent waterhammer ☐
b| To isolate a heat exchanger for upstream maintenance ☐
c| To prevent damage to a flowmeter ☐
d| To divert flow in a blending operation ☐

2. Which of the following can be used to prevent the problems associated with swing check valves, namely waterhammer and seat wear?

a| Limit the velocity of the fluid, by increasing the pipe diameter ☐
b| Replace the metal seat with a soft (PTFE) seat ☐
c| Fit a damping mechanism to the flap ☐
d| Fit a wafer swing check valve ☐

3. A thermodynamic steam trap is used to drain a steam main. How far downstream of the trap should a check valve be fitted?

a| Less than 1 m ☐
b| At least 1 m ☐
c| As close to the outlet as possible ☐
d| It is not necessary to fit a check valve in this situation ☐

4. What advantage does a split disc check valve have over other types of wafer check valves?

a| It is not limited in size ☐
b| The pressure drop across the valve is lower ☐
c| It can be used with lower opening pressures ☐
d| All of the above ☐

5. Which of the following may be used to eliminate the effects of valve chatter caused by oversizing a disc check valve?

a| Use a spring with a lower spring force ☐
b| Use a soft seat ☐
c| Replace the oversized valve with a smaller valve ☐
d| All of the above ☐

6. A disc check valve with the pressure loss diagram shown in Figure 12.3.17 is used downstream of a control valve. The downstream pipeline has a diameter of 32 mm, and passes 200 kg/h of saturated steam at 5 bar g. Determine the pressure drop across the check valve?

a| 0.05 bar ☐
b| 0.25 bar ☐
c| 1.55 bar ☐
d| 5.00 bar ☐

Answers

1: b, 2: c, 3: b, 4: d, 5: d, 6: a

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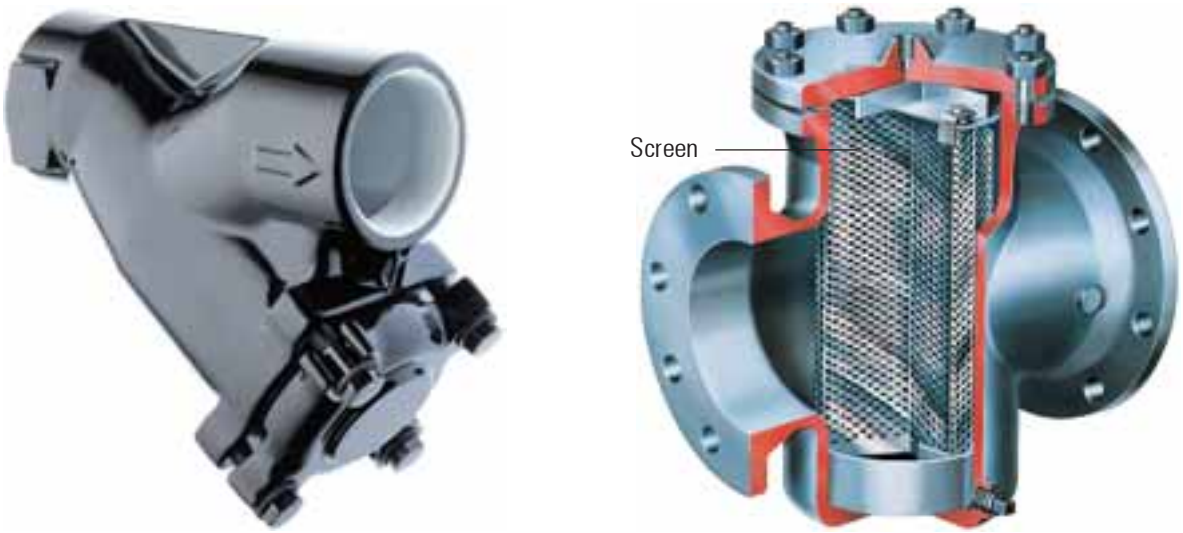
Module 12.4

Strainers

Strainers

As the marketplace becomes increasingly competitive, more emphasis has been placed on reducing plant downtime and maintenance. In steam and condensate systems, damage to plant is frequently caused by pipeline debris such as scale, rust, jointing compound, weld metal and other solids, which may find their way into the pipeline system. Strainers are devices which arrest these solids in flowing liquids or gases, and protect equipment from their harmful effects, thus reducing downtime and maintenance. A strainer should be fitted upstream of every steam trap, flowmeter and control valve.

Strainers can be classified into two main types according to their body configuration; namely the Y-type and the basket type. Typical examples of these types of strainers can be seen in Figure 12.4.1.



Y-type strainer **Basket type strainer**

Fig. 12.4.1 Typical strainers

Y-type Strainers

For steam, a Y-type strainer is the usual standard and is almost universally used. Its body has a compact cylindrical shape that is very strong and can handle high pressures. It is literally a pressure vessel, and it is not uncommon for Y-type strainers to be able to handle pressures of up to 400 bar g. The use of strainers at these pressures is however complicated by the high temperatures associated with steam at this pressure; and subsequently exotic materials such as chrome molybdenum steel have to be used.

Although there are exceptions, size for size, Y-type strainers have a lower dirt holding capacity than basket strainers, which means that they require more frequent cleaning. On steam systems, this is generally not a problem, except where high levels of rust are present, or immediately after commissioning when large amounts of debris can be introduced. On applications where significant amounts of debris are expected, a blowdown valve can usually be fitted in the strainer cap, which enables the strainer to use the pressure of the steam to be cleaned, and without having to shut down the plant.

Y-type strainers in horizontal steam or gas lines should be installed so that the pocket is in the horizontal plane (Figure 12.4.2(a)). This stops water collecting in the pocket, helping to prevent water droplets being carried over, which can cause erosion and affect heat transfer processes.

On liquid systems however, the pocket should point vertically downwards (Figure 12.4.2(b)), this ensures that the removed debris is not drawn back into the upstream pipework during low flow conditions.

Although it is advisable to install strainers in horizontal lines, this is not always possible, and they can be installed in vertical pipelines if the flow is downwards, in which case the debris is naturally directed into the pocket (Figure 12.4.2(c)). Installation is not possible with upward flow, as the strainer would have to be installed with the opening of the pocket pointing downwards and the debris would fall back down the pipe.

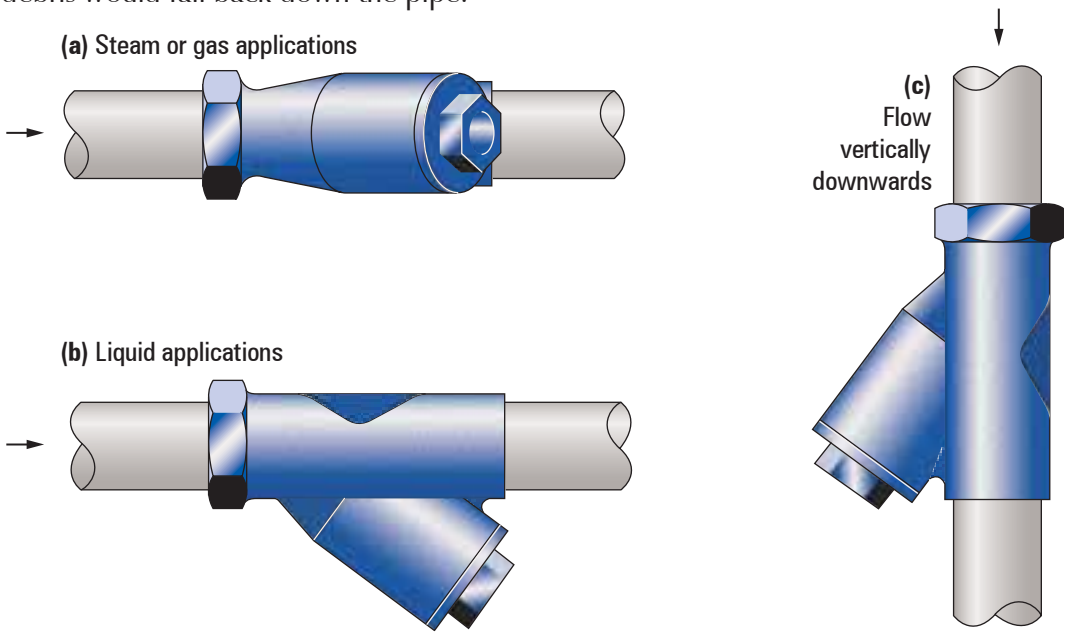


Fig. 12.4.2 Correct orientation of strainers

Straight and angle type strainers

In addition to Y-type strainers, several different body configurations are used in steam systems, namely straight and angle type strainers. These are shown in Figure 12.4.3. These types of strainer function in a similar way to the Y-type strainer and have similar performance. They are used when the geometry of the steam pipework does not suit a Y-type strainer being used.



Straight type strainer **Angle type strainer**

Fig. 12.4.3 Straight type and angle type strainers

Basket type strainer units

The basket type or pot type strainer is characterised by a vertically orientated chamber, typically larger than that of a Y-type strainer. Size for size, the pressure drop across a basket strainer is less than that across the Y-type as it has a greater free straining area, which makes the basket type strainer the preferred type for liquid applications. As the dirt holding capacity is also greater than in Y-type strainers, the basket type strainer is also used on larger diameter steam pipelines.

Basket type strainers can only be installed in horizontal pipelines, and for larger, heavier basket strainers, the base of the strainer needs to be supported.

When basket type strainers are used on steam systems, a significant amount of condensate may be formed. Consequently, strainers designed for use in steam systems usually have a drain plug, which can be fitted with a steam trap to remove the condensate.

Basket type strainers are commonly found in a duplex arrangement. A second strainer is placed in parallel with the primary strainer, and flow can be diverted through either of the two strainers. This facilitates cleaning of the strainer unit whilst the fluid system is still operating, reducing the downtime for maintenance.

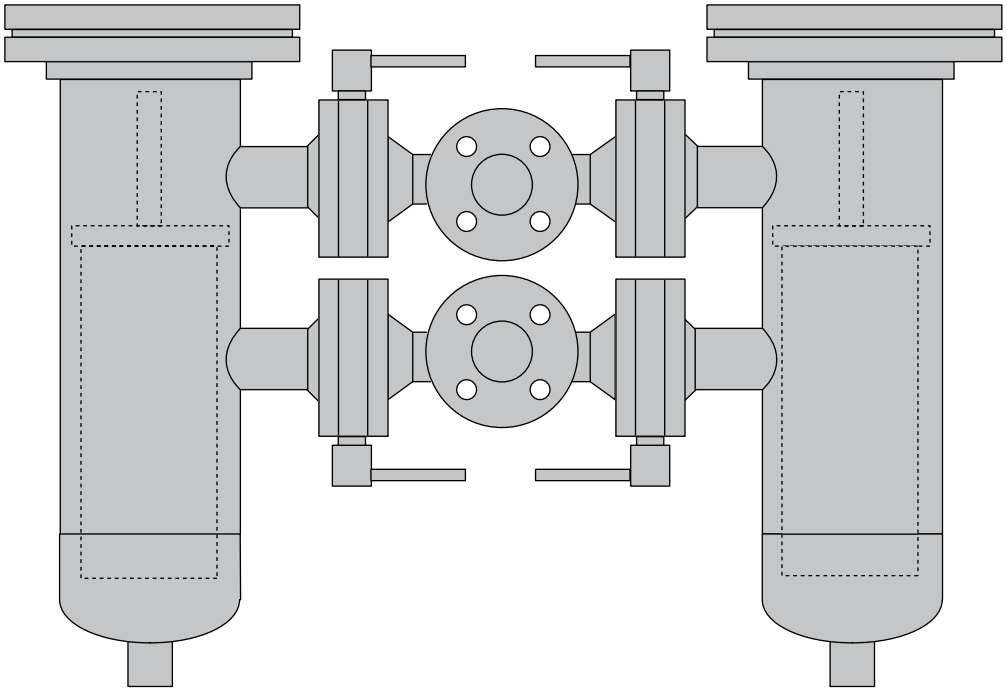


Fig. 12.4.4 A duplex basket strainer

Filters

Whilst strainers remove all visible particles in the steam, it is sometimes necessary to remove smaller particles, for example, in the following applications:

- When there is direct injection of steam into a process, which may cause contamination of the product.

  Example: In the food industry, and for the sterilisation of process equipment in the pharmaceutical industry.

- Where dirty steam may cause rejection of a product or process batch due to staining or visible particle retention.

  Example: Sterilizers and paper/board machines.

- Where minimal particle emission is required from steam humidifiers.

  Example: Humidifiers used in a 'clean' environment.

- For the reduction of the steam water content, ensuring a dry, saturated supply.

In such 'clean steam' applications, strainers are not suitable and filters must be used. A filter used in a steam system typically consists of a sintered stainless steel filter element. The sintering process produces a fine porous structure in the stainless steel, which removes any particles from fluid passing through it. Filters capable of removing particles as small as 1 μm are available, conforming to the good practice needs of culinary steam.

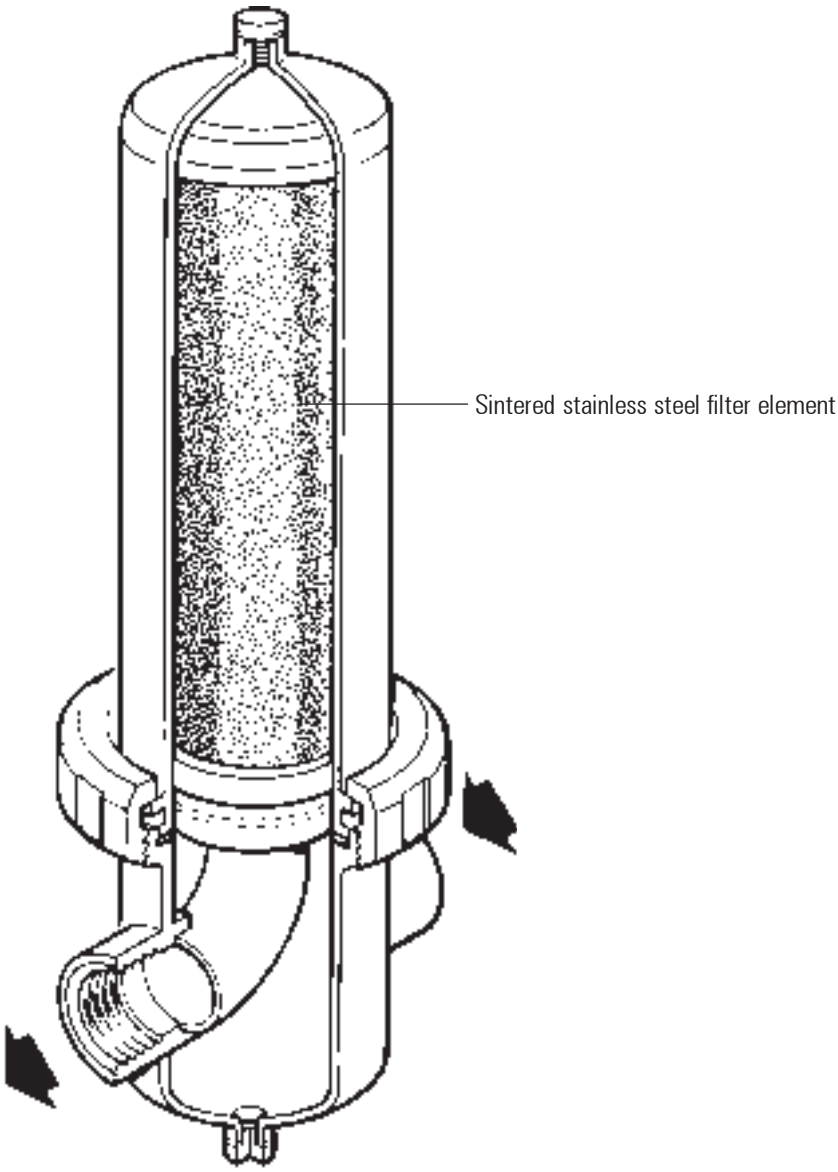


Fig. 12.4.5 A horizontal in-line filter

The fine, porous nature of the filter element will create a larger pressure drop across the filter than that associated with the same size strainer; this must be given careful consideration when sizing such filters. In addition, filters are easily damaged by excessive flowrates, and the manufacturer's specified limits should not be exceeded.

When the filter is used in steam or gas applications, a separator should be fitted upstream of the filter to remove any droplets of condensate held in suspension. In addition to improving the quality of the steam, this will prolong the life of the filter. A Y-type strainer should also be fitted upstream of the filter to remove all larger particles which would otherwise rapidly block the filter, increase the amount of cleaning required and reduce the life of the filter element. By installing pressure gauges either side of the filter, the pressure drop across the filter can be measured, which can then be used to identify when the filter requires cleaning. An alternative to this is to install a pressure switch on the downstream side of the filter. When the downstream pressure decreases below a set level, an alarm light can be switched on in a control room alerting an operator, who can then clean the filter.

Strainer screens

There are two types of screens used in strainers:

- **Perforated screens -** These are formed by punching a large number of holes in a flat sheet of the required material using a multiple punch. The perforated sheet is then rolled into a tube and spot welded together.

  These are relatively coarse screens and hole sizes typically range from 0.8 mm to 3.2 mm. Consequently, perforated screens are only suitable for removing general pipe debris.

- **Mesh screens -** Fine wire is formed into a grid or mesh arrangement. This is then commonly layered over a perforated screen, which acts as a support cage for the mesh.

  By using a mesh screen, it is possible to produce much smaller hole sizes than with perforated screens. Hole sizes as small as 0.07 mm are achievable. Subsequently, they are used to remove smaller particles which would otherwise pass through a perforated screen. Mesh screens are usually specified in terms of 'mesh'; which represents the number of openings per linear inch of screen, measured from the centre line of the wire. Figure 12.4.6 shows a 3 mesh screen.

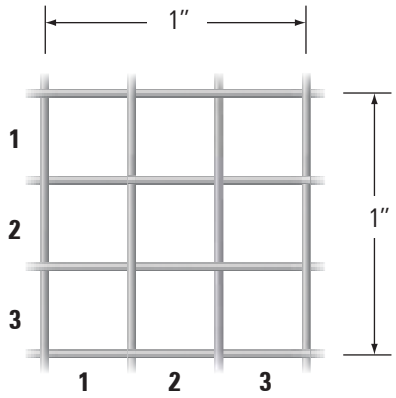


Fig. 12.4.6 Example of a 3 mesh screen

The corresponding hole size in the mesh screen is determined from knowledge of the wire diameter and the mesh size; it is usually specified by the manufacturer. The maximum particle size that will be allowed to pass through the screen can be determined using geometry. If, for example, a 200 mesh screen is specified and the manufacturer's specifications stated that the hole size is 0.076 mm, then the maximum particle size that will pass through the screen can be found using Pythagoras' theorem:

$$c^2 = a^2 + b^2 \qquad \textbf{Equation 12.4.1}$$

Where:

a = 0.076 mm

b = 0.076 mm

c = Particle size

$$c = \sqrt{a^2 + b^2}$$

$$c = \sqrt{0.076^2 + 0.076^2}$$

$$c = 0.1075 \text{ mm}$$

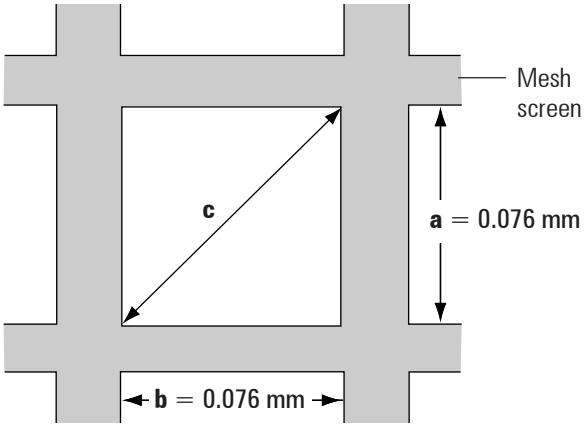


Fig. 12.4.7 Determining the maximum particle size that can pass through the screen

The problem with this dimension is that the screens are two-dimensional and the particle must reach the hole in a certain orientation. Therefore, if a long thin particle reached the strainer 'face on', it may be allowed to pass through the screen. However, if it hit the hole 'side on' it would be stopped. If this is likely to be a problem, a finer mesh should be used.

The screening area is the area available for removing debris. A larger screening area means that the frequency of blowdown for cleaning the screen is considerably reduced.

The free area is the proportion of the total area of the holes to the total screening area, usually expressed as a percentage. This directly affects the flow capacity of the strainer. The greater the free area (and the coarser the screen), the higher the flow capacity and ultimately the lower the pressure drop across the strainer. As most strainer screens have very large straining and free areas, the pressure drop across the strainer is very low when used on steam or gas systems (see Example 12.4.1). However, in pumped water or viscous fluid systems, the pressure drop can be significant. Strainers should have flow capacities quoted in terms of a capacity index or K_{VS} value.

Example 12.4.1
A DN40 strainer with a K_{VS} value of 29, is installed on a 40 mm diameter steam pipe system, which passes 500 kg/h of saturated steam at 8 bar g. What is the pressure drop across the strainer?

Using the empirical formula in Equation 3.21.2:

$$\dot{m}_s = 12\ K_v\ P_1 \sqrt{1 - 5.67(0.42 - \chi)^2} \qquad \textbf{Equation 3.21.2}$$

Where:
$\dot{m}_s$ = Mass flowrate in kg/h
K_v = Valve flow coefficient
χ = Pressure drop ratio = $\frac{P_1 - P_2}{P_1}$
P_1 = Upstream pressure in bar absolute
P_2 = Downstream pressure in bar absolute

This can be rearranged to give Equation 12.4.2:

$$\Delta P = P_1 \left[0.42 - \sqrt{\frac{1}{5.67}\left(1 - \left(\frac{\dot{m}_s}{12\ K_v\ P_1}\right)^2\right)} \right] \qquad \textbf{Equation 12.4.2}$$

Where:
$\dot{m}_s$ = 500 kg/h
K_v = 29
P_1 = 9 bar a

$$\Delta P = 9 \times \left[0.42 - \sqrt{\frac{1}{5.67}\left(1 - \left(\frac{500}{12 \times 29 \times 9}\right)^2\right)} \right]$$

Therefore: **ΔP = 0.05 bar**

This equates to a pressure drop of just over 0.5%.

The pressure drop across a strainer may be determined either from the K_V value or from a pressure loss diagram. The method for doing this for steam flow is shown in Module 12.2, and for water flow in Module 6.3.

Screens are typically available in a number of different materials; most commonly austenitic stainless steels are used in steam applications, due to their strength and resistance to corrosion. Where the strainer is used with specialised chemicals or in offshore applications, a monel screen should be used.

Strainer options

In addition to standard strainers, there are several other options available.

Magnetic inserts

A magnetic insert may be placed in a basket type strainer in order to remove small iron or steel debris. Small particles of iron or steel may be present in a fluid where there is wear of iron or steel parts. These particles will pass through even the finest mesh screens, and it is necessary to use a magnetic insert. The insert is designed so that all the fluid passes over the magnet at relatively low velocity and the magnetic element is powerful enough to catch and hold all the metal particles present. The magnetic material is usually encased in an inert material such as stainless steel to prevent corrosion.

Self-cleaning strainers

There are number of different types of self-cleaning strainer, which enable the build up of debris on the screen to be removed without shutting down the plant. The cleaning process can be initiated either manually or automatically; furthermore, strainers that are automatically cleaned can usually be set to clean either on a periodic basis, or when the pressure drop across the strainer increases.

Mechanical type self-cleaning strainers use some form of mechanical scraper or brush, which is raked over the screen surface. It dislodges any debris that is trapped in the screen, causing it to fall down into a collection area at the bottom of the strainer.

Backwashing type strainers reverse the direction of flow through the screen. A set of valves is changed over so that water is directed across the screen in the reverse direction and out through a flush valve. The fluid dislodges any debris entrained in the screen and carries it out in the backwash fluid to a waste drain.

In addition to the mechanical and backwashing type strainers, there are several types of uniquely designed strainer screens. One of the more common types is the **metallic disc, positive edge type strainer** (see Figure 12.4.8). The straining element is constructed from a pack of circular discs, separated by spacing washers built on a main shaft with tie rods. The thickness of the washers or distance pieces gives the required degree of filtration. The flow direction of the fluid being strained is from the outside of the element to the hollow core, which is formed by the spaces between the main discs. This means that any debris is trapped on the outside surface of the discs.

In order to clean the strainer, the entire strainer pack is rotated by the external handle against a set of stationary cleaning knives interleaved with the main pack. During this rotation, accumulated debris builds up on the leading edge of the cleaning knife, and it is deposited into a solid, vertical groove formed in the outside surface of the strainer element by special packing pieces. As there is no flow through this part of the element there is no force holding the accumulated dirt against the element, and it falls into the sump at the bottom of the strainer.

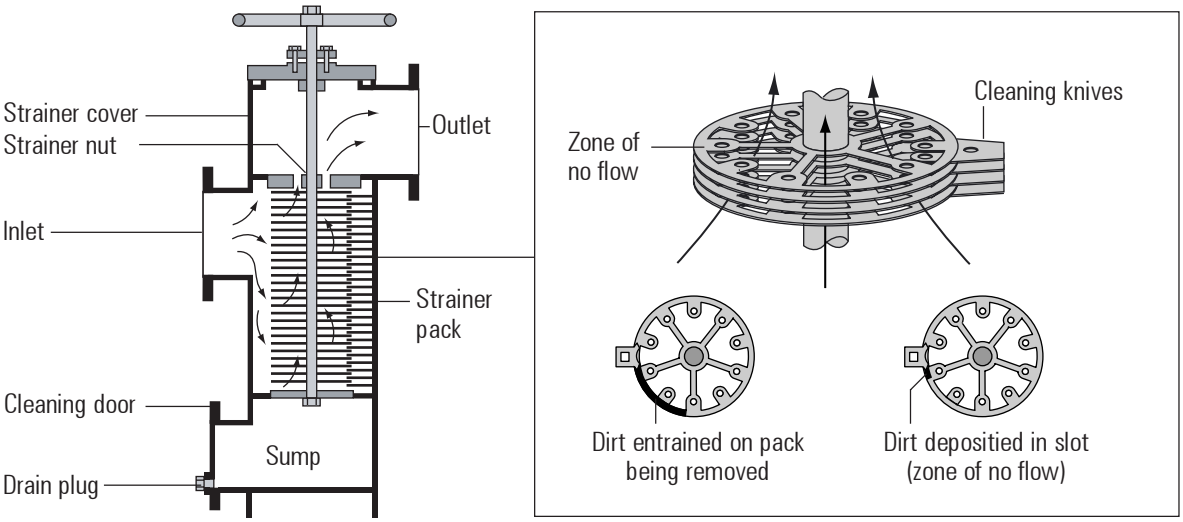


Fig. 12.4.8 The metallic disc, positive edge type strainer

Temporary strainers

Temporary strainers are designed for protection of equipment and instrumentation during start-up periods. The strainer is usually installed between a set of flanges for an initial period after a new plant has been installed. Installation of a spool piece equal or more than the length of the strainer is recommended for ease of installation or removal.

There are three basic configurations of temporary strainers, namely the conical type, the basket type and the plate type. Standard construction is of perforated screen or single ply heavy wire mesh. Wire mesh liners can be added inside or outside of the strainer for finer straining capabilities. If a wire mesh is used, care must be taken to ensure that the direction of flow is against the wire mesh with the perforated metal as a back-up.

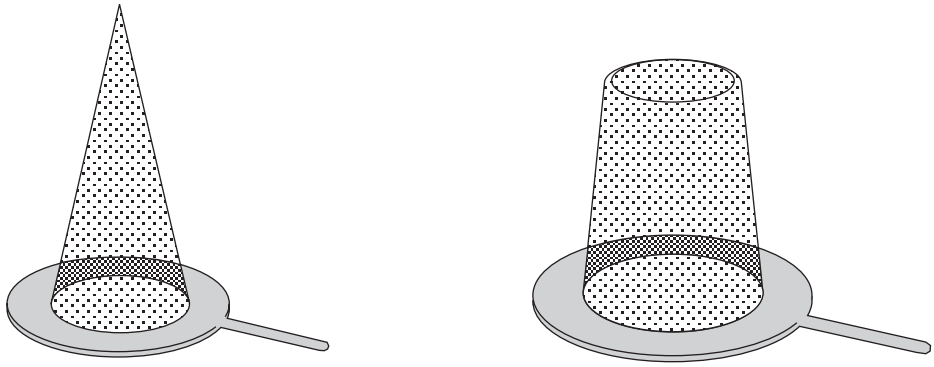


Fig. 12.4.9 Temporary cone (a) and basket (b) type strainers

Questions

1. Why are Y-type stainers commonly used in steam systems?

a| They have a higher dirt holding capacity than basket type strainers ☐
b| The pressure drop across the strainer is neglible ☐
c| The body can withstand high pressures ☐
d| They are available in a duplex arrangement, which reduces downtime ☐

2. What type of strainer would be most suitable to protect a large pressure reducing valve fitted in an old pipeline susceptible to rust?

a| A Y-type strainer ☐
b| A filter ☐
c| A basket type strainer ☐
d| A metallic disc, positive edge type strainer ☐

3. For which of the following steam applications is a clean steam filter not suitable?

a| Where steam is directly injected into a vat of baby food for sterilisation ☐
b| In a pressure reducing station prior to a heater battery ☐
c| In the steam system used to clean new socks prior to final inspection ☐
d| For use in humidifiers in the tobacco industry ☐

4. A manufacturer specifies that its 100 mesh screen is constructed from gauge 37 monel wire and therefore has a hole size of 0.152 mm. What is the maximum size of particle that will be allowed to pass through the screen?

a| 0.046 mm ☐
b| 0.152 mm ☐
c| 0.176 mm ☐
d| 0.215 mm ☐

5. A strainer uses a screen with 3.2 mm diameter perforations. If the total screening area of 73 cm² contains 360 perforations, what is the percentage free area?

(Note that the area of a perforation equals $\frac{\pi \times d^2}{4}$ where d is the diameter of the perforation)

a| 32% ☐
b| 40% ☐
c| 68% ☐
d| 73% ☐

6. When should a temporary strainer be used?

a| When a newly installed steam plant is commissioned for the first time ☐
b| To fit in between flanges where space is limited ☐
c| When the expected amount of debris is small ☐
d| When the plant is only shut down once a year and it is more cost effective to use a disposable, temporary strainer ☐

Answers

1: c, 2: c, 3: b, 4: d, 5: b, 6: a

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Module 12.5

Separators

Separators

Wet steam is steam containing a degree of water, and is one of the main concerns in any steam system. It can reduce plant productivity and product quality, and can cause damage to most items of plant and equipment. Whilst careful drainage and trapping can remove most of the water, it will not deal with the water droplets suspended in the steam. To remove these suspended water droplets, separators are installed in steam pipelines.

The steam produced in a boiler designed to generate saturated steam is inherently wet. Although the dryness fraction will vary according to the type of boiler, most shell type steam boilers will produce steam with a dryness fraction of between 95 and 98%. The water content of the steam produced by the boiler is further increased if priming and carryover occur.

There is always a certain degree of heat loss from the distribution pipe, which causes steam to condense. The condensed water molecules will eventually gravitate towards the bottom of the pipe forming a film of water. Steam flowing over this water can raise ripples that can build up into waves. The tips of the waves tend to break off, throwing droplets of condensate into the steam flow.

The presence of water in steam can cause a number of problems:

- As water is an extremely effective barrier to heat transfer, its presence can reduce plant productivity and product quality. This can be seen in Figure 12.5.1, which shows the temperature profile across a typical heat exchange surface.

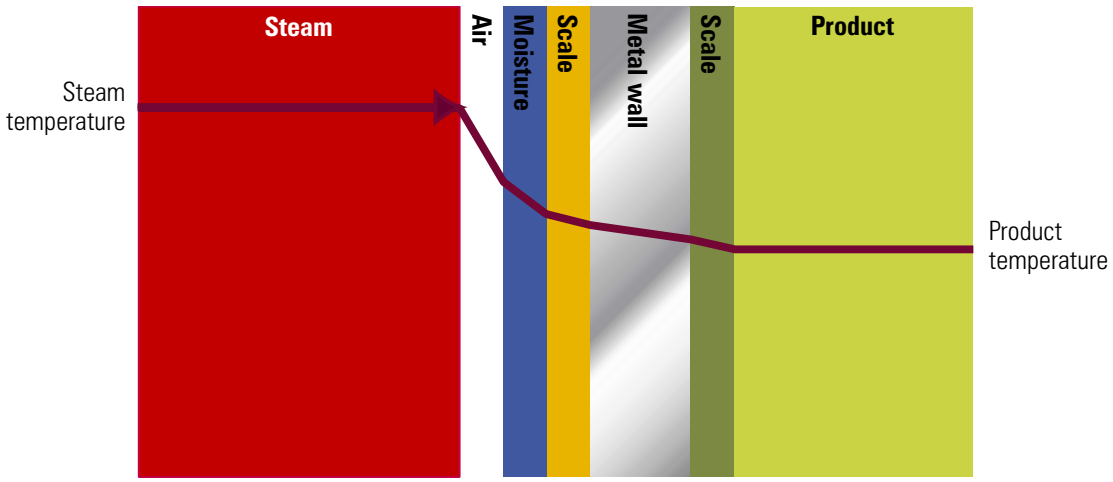


Fig. 12.5.1 Temperature profile across a heat exchange surface

- Water droplets travelling at high steam velocities will erode valve seats and fittings, a condition known as wiredrawing. The water droplets will also increase the amount of corrosion.
- Increased scaling of pipework and heating surfaces from the impurities carried in the water droplets.
- Erratic operation of control valves and flowmeters.
- Failure of valves and flowmeters due to rapid wear or waterhammer.

Although there are a number of different designs of separator, they all attempt to remove the moisture that remains suspended in the steam flow, which cannot be removed by drainage and steam trapping. There are three types of separator in common use in steam systems:

- **Baffle type -** A baffle or vane type separator consists of a number of baffle plates, which cause the flow to change direction a number of times as it passes through the separator body. The suspended water droplets have a greater mass and a greater inertia than the steam; thus, when there is a change in flow direction, the dry steam flows around the baffles and the water droplets collect on the baffles. Furthermore, as the separator has a large cross-sectional area, there is a resulting reduction in the speed of the fluid. This reduces the kinetic energy of the water droplets, and most of them will fall out of suspension. The condensate collects in the bottom of the separator, where it is drained away through a steam trap.

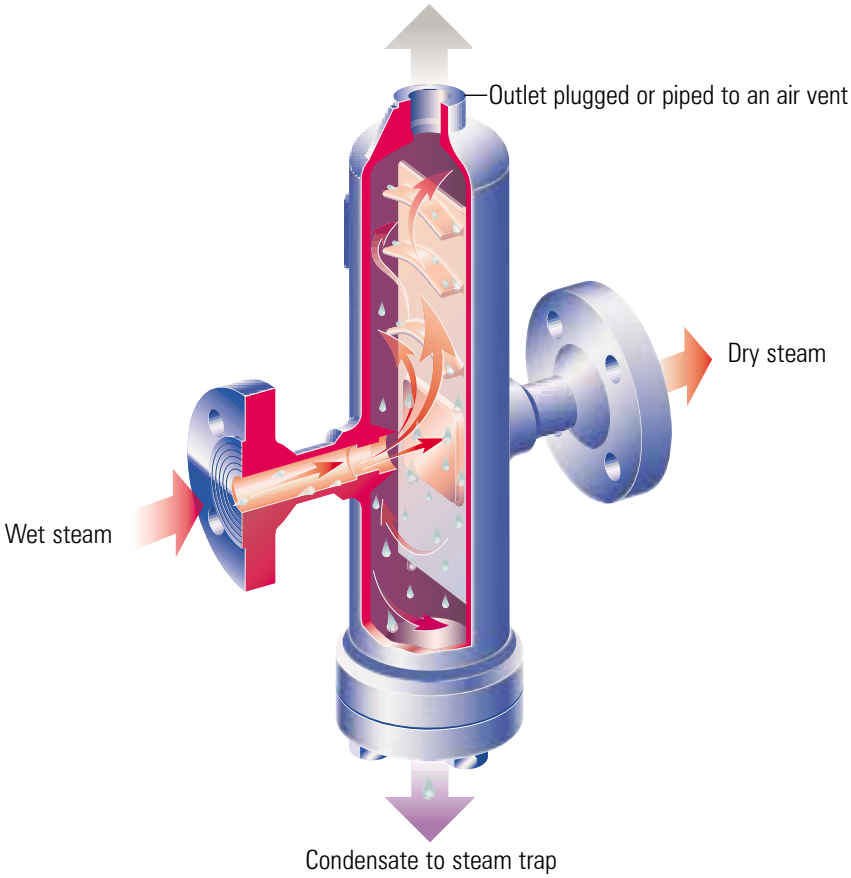


Fig. 12.5.2 A baffle type separator

- **Cyclonic type -** The cyclonic or centrifugal type separator uses a series of fins to generate high-speed cyclonic flow. The velocity of the steam causes it to swirl around the body of the separator, throwing the heavier, suspended water to the wall, where it drains down to a steam trap installed under the unit.

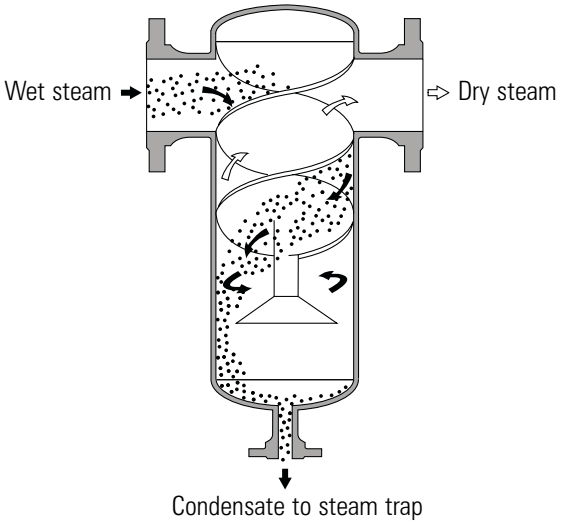


Fig. 12.5.3 A cyclonic type separator

- **Coalescence type -** Coalescence type separators provide an obstruction in the steam path. The obstruction is typically a wire mesh pad (sometimes referred to as a demister pad), upon which water molecules become entrapped. These water molecules tend to coalesce, producing droplets that are too large to be carried further by the gas system. As the size of the droplets increases, they become too heavy and ultimately fall into the bottom of the separator.

  It is common to find separators, which combine both coalescence and cyclonic type operations. By combining the two methods, the overall efficiency of the separator is improved.

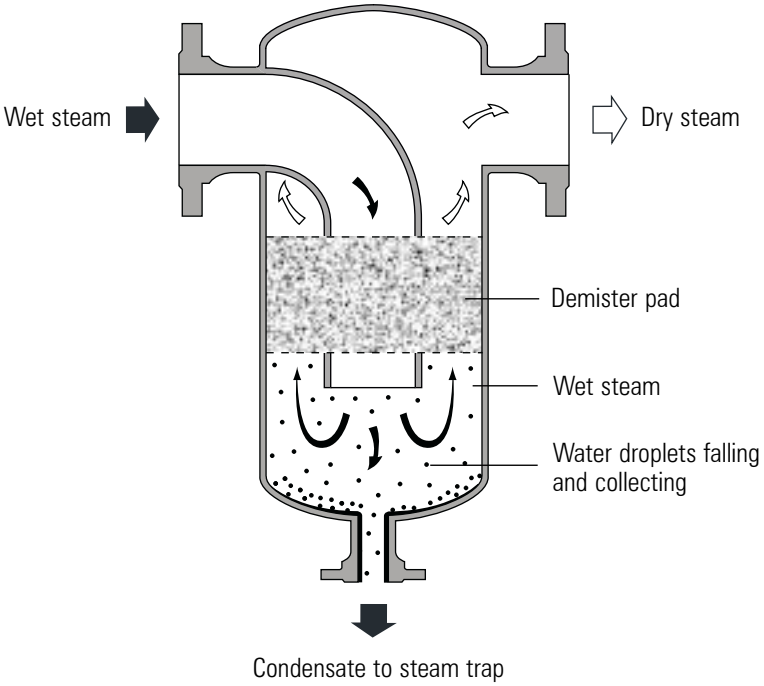


Fig. 12.5.4 A coalescence type separator

Separator efficiency is a measure of the weight of the water separated out in proportion to the total weight of the water carried in by the steam. Outside the laboratory, it is difficult to establish the exact efficiency of a separator, as it depends on the inlet dryness fraction, the fluid velocity and the flow pattern. Erosion of pipe bends, wiredrawing, and waterhammer are, however, indications of the presence of wet steam in steam pipes.

One of the main differences in performance between the baffle type and the cyclonic and coalescence types of separators is that the baffle type is capable of maintaining a high level of efficiency over a wider pipeline velocity range. Cyclone and coalescence type separators typically exhibit efficiencies of 98% at velocities of up to 13 m/s, but this falls off sharply, and at 25 m/s, the efficiency is typically around 50%, according to University research in the UK.

This research has also proven that, for a baffle type separator, the efficiency remains close to 100% over a range of 10 m/s to 30 m/s . The conclusion is that, the baffle type separator is more suited to steam applications, where there is usually some degree of velocity fluctuation. Furthermore, wet steam will be found to run at velocities of over 30 m/s if the pipework is undersized.

One method of overcoming this problem is to use a larger size separator and by increasing the diameter of the pipework immediately upstream of the separator. This will have the effect of reducing the velocity of the steam before it enters the separator.

Example 12.5.1

If a separator with an efficiency of 90% is fitted to a steam main containing steam with a dryness fraction of 0.95, what would the downstream dryness fraction be?

If the initial dryness fraction is 0.95, every kilogram (1 000 g) of steam contains:

(1 - 0.95) x 1000 g = 50 g of water

Since the efficiency of the separator is 90%, only 0.90 x 50 g = 45 g of the water present is removed. This means that the dryness fraction becomes:

$$1 - \left(\frac{50\,g - 45\,g}{1\,000\,g} \right) = 0.995$$

In practical terms, the steam can be considered completely dry.

If however, the separator efficiency is only 50%, only 25 g of the water will be removed. This results in a dryness fraction of:

$$1 - \left(\frac{50\,g - 25\,g}{1\,000\,g} \right) = 0.975$$

Although an improvement on the original dryness of 0.95, the steam will still contain a significant amount of water.

The pressure drop across a baffle type separator is very low due to the reduction in the velocity of the steam, which is created by the large increase in cross-sectional area provided by the separator body. The pressure drop is typically less than the equivalent length of the same nominal diameter pipe. In comparison, the pressure drop across a cyclonic type separator is somewhat higher, as the velocity of the fluid has to be maintained to generate the cyclone effect.

On non-critical applications, baffle type separators are typically sized according to the pipeline size; it is necessary however to check that the chosen size ensures maximum separation efficiency, and that the pressure drop is within acceptable limits. On critical applications, it is more common to select the separator based on operating pressure and flowrate, so as to give a suitable efficiency and pressure drop. Sizing a cyclonic type separator is more complicated, as it is important to ensure that the velocity through the separator is suitable to maintain a high level of efficiency and that the pressure drop across the separator is acceptable.

Example 12.5.2 outlines the selection of a baffle type separator from a typical manufacture's specification chart.

Example 12.5.2

Using the sizing chart in Figure 12.5.5, select a suitably sized separator for a pressure reducing station, with an upstream pressure of 12 bar g and passing 500 kg/h of steam through a 32 mm pipeline, If the flowrate were doubled to 1 000 kg/h, what size should the separator be?

1. Plot point **A** where the steam pressure and the flowrate cross and draw a horizontal line across from this point. Any separator curve that is bisected by this line within the shaded area will operate at near 100% efficiency.
2. Select the line size separator, i.e. 32 mm at point **B**.
3. The line velocity for any size can be determined by dropping a vertical line from this intersection. From point **B**, this line crosses the velocity axis at 18 m/s.
4. To determine the pressure drop across the separator, where the vertical line, extended from point **B**, crosses the line **C**-**C**, plot a horizontal line. Then drop a vertical line from point **A**. The point of intersection, **D**, is the pressure drop across the separator.
5. Repeating this procedure for a 1 000 kg/h flowrate, generates points **X**, **Y** and **Z**. It can be seen that point **Y** falls outside the shaded region and the separator will not operate at maximum efficiency. Here, it would be advisable to use a larger size separator; a DN40 separator would be selected, as depicted by point **Z**, along with a pressure drop of about 0.07 bar at point **W**.

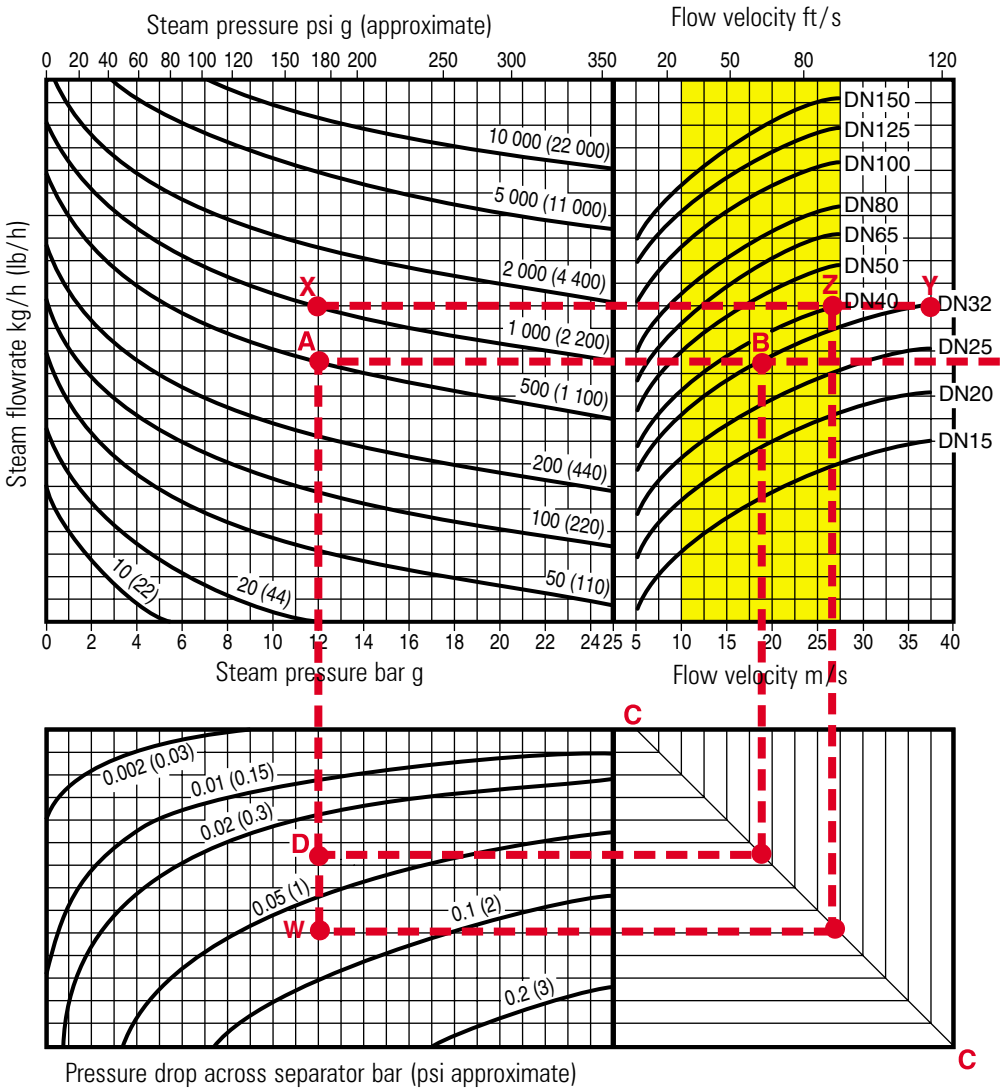


Fig. 12.5.5 Manufacturer's sizing chart for a baffle type separator

Table 12.5.1 summarises the important differences in the performance of baffle and cyclone type separators.

Table 12.5.1 Comparison of baffle and centrifugal type separators

	Baffle type	Cyclonic type
Pressure drop	Relatively low	Relatively high
Efficiency	High over a wide range of velocities	High over a narrower range of velocities
Re-entrainment of water	Little	Significant above a critical velocity
Sizing	Sized according to pipeline size	Sized to ensure maximum efficiency

A suitable steam trap should be fitted to the condensate outlet of the separator to ensure the efficient removal of condensate, without the loss of live steam. The most suitable type of steam trap is the ball float type, which ensures immediate condensate removal. Some separators include the steam trap mechanism inside the separator body.

Most vertical separators have a tapping on the top of the body. This can be used for an air vent, facilitating the removal of air from the steam space during start-up.

Insulation

If a separator is left uninsulated, it can actually induce water droplets to form rather than eliminating them, because of the large surface area exposed to the environment. Furthermore, significant amounts of heat energy can be lost from the surface of the separator. For example, insulating a separator containing steam at 150°C and exposed to ambient temperatures of 15°C, will produce an annual energy saving of 8600 MJ (Based on heat loss due to radiation only, assuming still air conditions and 8760 hours of operation per year). By fitting an insulation jacket, this heat loss can be drastically reduced and the energy savings justify the initial cost of the insulation, within an extremely short time.

Insulation jackets designed to fit over a particular separator should be used, as the shape of the separator, particularly if it is flanged, makes it difficult to insulate. Standard flange covers leave the body exposed, and therefore have a limited effect in the reduction of heat loss.

Even with the best insulation, it is not possible to eliminate all the heat loss from a product. The efficiency of separator insulation is typically above 90%. It is important to use a jacket that is designed for a particular separator; otherwise, the insulation efficiency will decrease. Properly insulated separators also reduce the risk of personal injury from burns.



Fig. 12.5.6 A horizontal separator and insulating jacket

Questions

1. Which of the following causes water entrainment in steam?

a| Priming and carryover of boiler water ☐
b| Heat loss in pipelines ☐
c| Production of saturated steam in a boiler ☐
d| All of the above ☐

2. Although in practice, it is difficult to measure the dryness fraction of steam, which of the following factors provides a good indication that wet steam is present in a steam system?

a| An increase in the steam velocity ☐
b| Valve chatter ☐
c| Erosion of pipe bends, wiredrawing, and waterhammer ☐
d| Increased condensate load ☐

**3. The dryness factor of 10 bar g wet steam is known to be 0.9.
A separator is to be installed to increase this to above 0.98.
What is the minimum efficiency that the separator must have?**

a| 64% ☐
b| 80% ☐
c| 84% ☐
d| 93% ☐

4. What is the main advantage of using a baffle type separator instead of a cyclonic or coalescence type separator?

a| High efficiency over a wider range of flow velocities ☐
b| Very high efficiencies up to a flow velocity of 13 m/s ☐
c| Flanged versions are easy to insulate ☐
d| All of the above ☐

5. Size a baffle type separator using the sizing chart in Figure 12.5.5 for the following conditions:

Operating pressure	**8 bar g**
Steam flowrate	**1000 kg/h**
Pipeline size	**65 mm**

a| DN40 ☐
b| DN50 ☐
c| DN65 ☐
d| DN80 ☐

6. Which of the following is a benefit of using an insulation jacket specifically designed for a particular separator?

a| Reduced heat loss ☐
b| An increase in the efficiency of the separator ☐
c| Protection from the possibility of burns ☐
d| All of the above ☐

Answers

1: d, 2: c, 3: b, 4: a, 5: c, 6: d

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Module 12.6

Gauges, Sight Glasses and Vacuum Breakers

Gauges, Sight Glasses and Vacuum Breakers

Gauges

Pressure gauges

Pressure gauges should be installed in at least the following situations:

- **Upstream of a pressure reducing valve -** To monitor the integrity of the steam supply.
- **Downstream of a pressure reducing valve -** To set and monitor the downstream pressure. Variations in the downstream pressure can lead to reduced plant productivity and product quality. Variations in the downstream pressure may also indicate problems with the pressure reducing valve.
- **On blowdown vessels -** A pressure gauge is used to check the vessel pressure during blowdown. This improves safety, since a higher pressure than normal would give an early indication of pipework blockage.
- **Flash steam vessels -** To monitor the flash steam pressure.

The Bourdon tube pressure gauge is the most commonly used type in steam systems. It consists of a coiled or 'C' – shaped tube that is sealed at one end, and open at the other. The open end of the Bourdon tube is exposed to the process fluid, allowing it to flow into the tube. Any increase in pressure causes elastic distortion of the tube, causing it to unwind. The resulting displacement of the closed end of the tube is translated by a series of gears to an angular displacement of the pointer. The pointer position is therefore proportional to the pressure applied at the gauge's pressure connector. Typically, the maximum deflection of the Bourdon tube corresponds to a pointer angular displacement of 270°.

The tube can be constructed out of a number of different materials, depending on the application; generally, brass or bronze is used for higher pressures, whereas stainless steel is used for lower pressures.

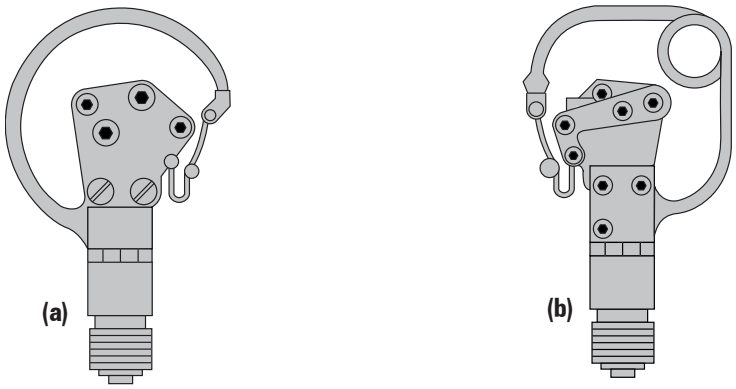


Fig. 12.6.1 'C'-shaped (a) and coiled (b) Bourdon tubes

Bourdon tube pressure gauges often have the option of being liquid filled. The area surrounding the Bourdon tube is filled with a transparent liquid, normally glycerine. This protects the internal mechanisms against damage from severe vibration and to keep out ambient corrosives and condensation. This also damps the movement of the pointer making the gauge less susceptible to small transient pressure fluctuations.

As the Bourdon tube may be damaged by high temperatures, it is common practice on steam systems to install the gauge at the end of a syphon tube. The syphon tube is filled with water which transmits the pressure of the working fluid to the Bourdon tube, enabling the gauge to be located some distance from the actual point where the pressure is being measured. The two most common forms of syphon tube are the 'U' and ring types. The ring tube is used on horizontal pipelines where there is sufficient space above the pipe, and the 'U' type is used when mounting the gauge on a vertical pipeline, or on horizontal pipelines where there is not sufficient space for a ring type siphon.

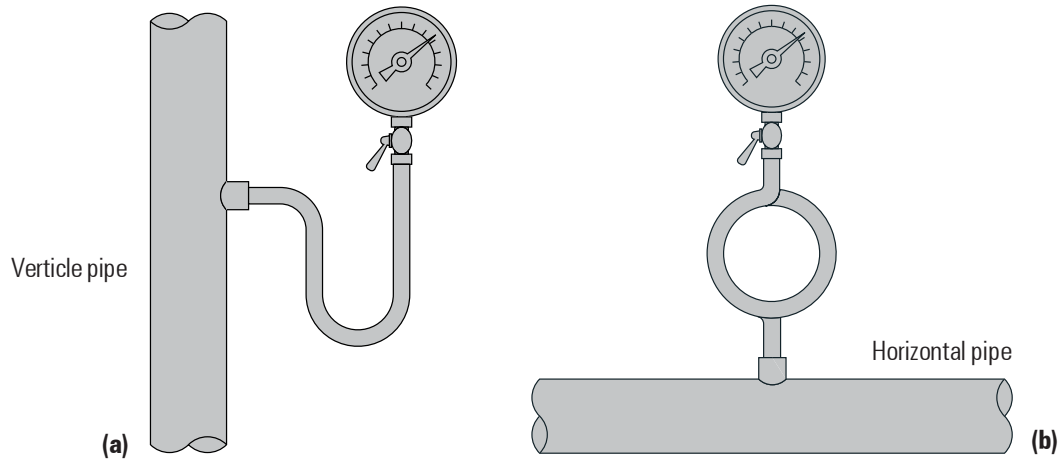


Fig. 12.6.2 'U' (a) and ring type (b) syphon tubes

The Bourdon type pressure gauge is not suitable for use on corrosive liquids or fluids containing suspended solids alone, as these solids may damage the internal elements of the gauge. In such cases, it is necessary to keep the process fluid separate from the Bourdon tube.

This is done by mounting a flexible diaphragm on the inlet to the gauge. The pressure element of the gauge and the space behind the diaphragm form a completely sealed system, which is evacuated and then filled with a suitable filling fluid; in the case of steam this is typically a type of oil. The system pressure causes the diaphragm to deflect, and the pressure is transmitted through the filling fluid to the Bourdon tube.

Diaphragm seals should also be used on 'clean steam' applications where no 'dead space' is allowed.

In addition to the Bourdon tube pressure gauge, several other types of pressure gauge are available which include; Diaphragm type pressure gauges, Piezoresistive pressure gauges and Temperature gauges.

Diaphragm type pressure gauges

A metal diaphragm is clamped between two flanges, and is exposed to the pressure medium on one side. Pressure exerted by the fluid causes elastic deflection of the diaphragm. The amount of deflection is proportional to the pressure applied on the diaphragm and it causes the linear displacement of a linkage rod attached to the internal side of the diaphragm. The movement of the linkage rod is in turn translated to angular movement of the gauge's pointer by a series of gears. Thus, the pointer movement is proportional to the pressure exerted on the diaphragm.

The diaphragm also serves to isolate the fluid from the internals of the gauge; therefore, diaphragm type pressure gauges are suitable for use on most fluid types.

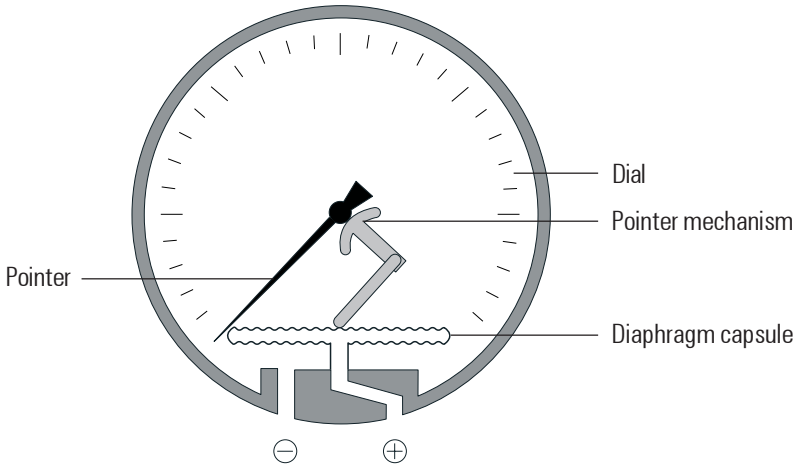


Fig. 12.6.3 Schematic diagram of a diaphragm pressure gauge

Piezoresistive pressure gauges

These pressure gauges consist of a diaphragm made from a ceramic substrate; piezoresistive type strain gauges are bonded to the diaphragm and together with the necessary circuitry, they are integrated on a silicon chip. The diaphragm deflects with changes in pressure, causing a change in the balance of the strain gauge bridge. This is converted by the integrated circuit module to an electronic signal that is proportional to the pressure. The output signal can be fed into a local digital display or further converted into a 4-20 mA signal output for remote transmission.

These gauges are very sensitive and are used where precise measurement of pressure is required. Since they produce an electrical output signal, it is possible to incorporate them into building management systems.

Temperature gauges

Although there are a multitude of different temperature gauges available, five major types are likely to be encountered in steam systems, namely, the bimetallic type, the filled system type, thermistors, thermocouples and resistance temperature devices (RTDs).

- **The bimetallic type temperature gauge** - Consists of a coiled bimetallic element. The gauge is based on the principle of the bimetallic strip, which consists of two metal strips, made from different materials, bonded to each other. The two materials are selected so that they have different thermal coefficients of expansion. The two metals expand by different amounts when heated, and since they cannot move relative to each other, the bimetallic strip bends.

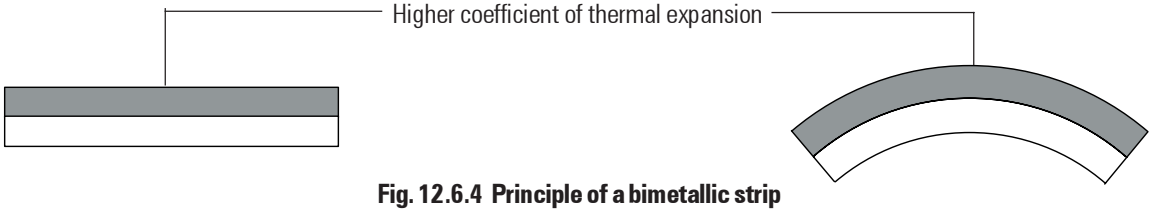


Fig. 12.6.4 Principle of a bimetallic strip

When the temperature of the coiled element rises, it tends to unwind. The degree to which this occurs is indicative of the temperature. A pointer is connected to the coil by a series of linkages, in a similar way to that in the bourdon tube.

Bimetallic gauges tend to be inexpensive, robust and easy to install. They are used where a simple, quick visual indication of temperature is required.

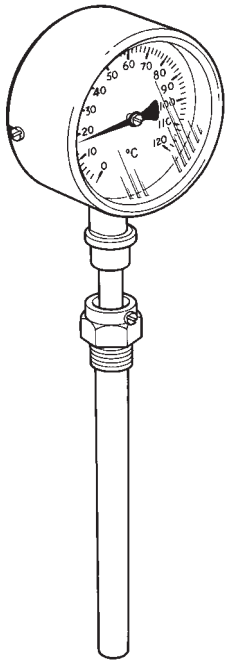


Fig. 12.6.5 A bimetallic temperature gauge

- **Other methods of temperature measurement -** are dealt with in Module 6.7, Controllers and Sensors. These types of temperature sensors are used when a higher level of accuracy is required in measuring temperature, or when this function is to be automated or incorporated into a building management system.

  It is common to place a temperature-measuring probe into a pocket when installed into an item of plant. This enables the sensor to be removed from pipework or equipment without disturbing the integrity of the system. A heat conducting paste is used in the pocket to provide good heat transfer qualities.

  One area of concern when installing a temperature-measuring device is ensuring that it takes a representative reading. It is common, particularly in liquid containing vessels, for there to be some kind of thermal layering of the fluid, and measuring the temperature of the vessels at different levels may produce different results.

  Common applications of temperature-measuring devices include boiler feedtanks, measuring product temperatures and measuring the steam temperature after de-superheating.

Sight glasses

A sight glass, or sight flow indicator, provides a method of observing fluid flow in a pipeline. It has two main functions:

- **Indication -** Sight glasses are used to indicate if fluid is flowing correctly. They are used to detect blocked valves, strainers, steam traps and other pipeline equipment, as well as to detect if a steam trap is leaking steam.
- **Inspection -** Sight glasses can be used to observe the colour of a product at different stages of the production process.

When sight glasses are used to indicate the correct functioning of blast discharge type steam traps, they should be positioned at least 1 m downstream from the trap. For other traps, the sight glass should be positioned immediately after the trap.

Sight glasses do not provide an exact method of monitoring the functioning of steam traps. In practice, a thorough knowledge of the upstream steam system is required and the diagnosis is often subjective, depending on the experience of the observer. For example, depending on the condensate flowrate, pressure and trap discharge pattern, it can be difficult to differentiate if the steam trap is leaking steam or if flash steam is being generated after the steam trap. Sight glasses have generally been replaced by electrical devices such as conductivity sensors, which detect flooding upstream of the steam trap, or leaking traps. These devices do not require steam trap expertise and produce a consistently accurate result.

Sight glasses

The sight glass has a smooth concentric reduction in the inlet connection, which promotes turbulence in the sight glass when fluid is flowing through it. The turbulent flow inside the sight glass permits any fluid to be detected. Sight glasses are available with single, double or multi-viewing windows.

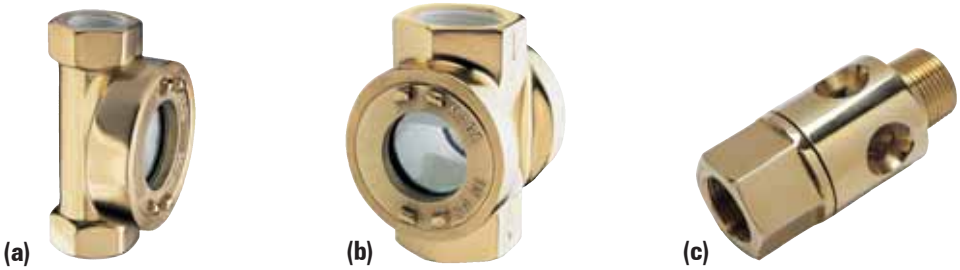


Fig. 12.6.6 Single (a), double (b) and multiple (c) window sight glasses

Some sight glasses may be fitted with a light source, these are useful when the sight glass is fitted in an area of low ambient lighting, or where a single window sight glass has to be used, such as in tanks.

Sight check

The sight check (see Figure 12.6.7) is a combination of a sight glass and a check valve. A ball in the top of the flow tube is lifted off its seat by the fluid as it flows through the cylindrical window to the outlet connection. When there is reverse flow, the ball is forced back onto its seat on the inlet. The ball movement makes the flow easy to see, as well as providing shut-off on reverse flow.

As with sight glasses, the sight check is used to observe the discharge of steam traps. In the sight check, the position of the ball check indicates whether condensate is flowing. Where condensate rises after the trap, the sight check eliminates the need for a separate check valve, thus simplifying installation. The sight check is particularly useful for commissioning steam traps fitted with a steam lock release (SLR).



Fig. 12.6.7 A sight check

Vacuum breakers

Vacuum breakers protect plant and process equipment against vacuum conditions, typically associated with cooling.

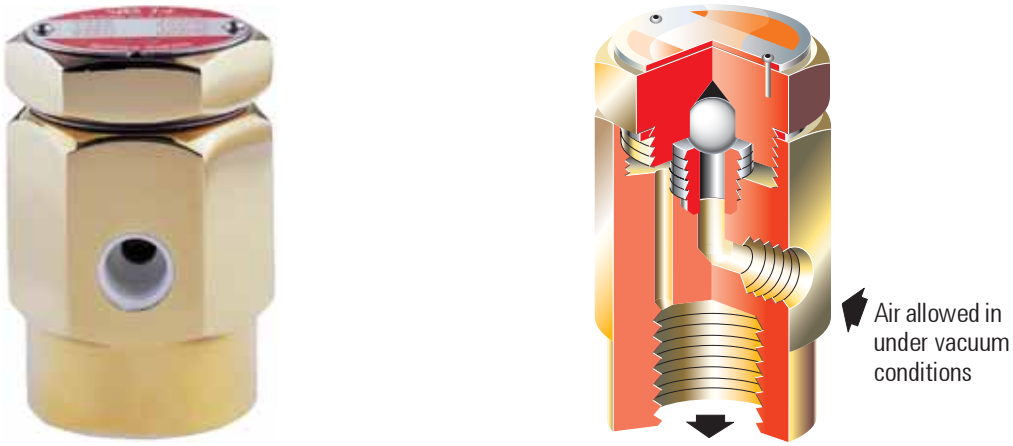


Fig. 12.6.8 Vacuum breaker and a cut section of a vacuum breaker

The vacuum breaker consists of a spherical stainless steel ball that rests on its seat during normal operating conditions. At the point of vacuum, the valve is lifted off its seat and air is drawn into the system.

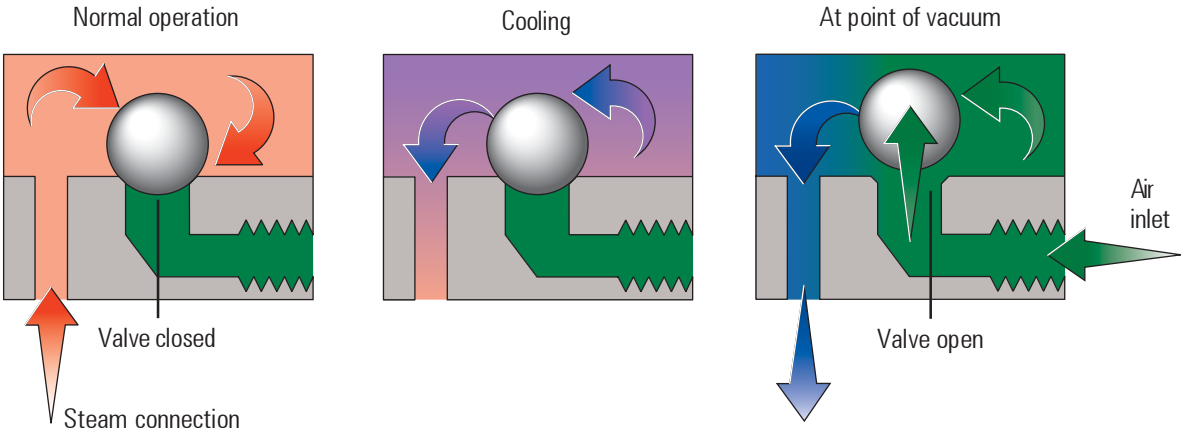


Fig.12.6.9 Operation of a vacuum breaker

In some cases, the valve may be spring loaded, which means that the vacuum is only broken when there is a further pressure decrease. This helps to ensure that the shut-off at near vacuum conditions remains bubble tight.

One of the most common applications of a vacuum breaker is on process equipment such as jacketed pans and heat exchangers. When these items are turned off, they still contain a certain amount of steam. The steam condenses as the vessel cools down, and since condensate occupies a much smaller volume than the steam, vacuum conditions are generated. The vacuum can damage the plant and it is therefore necessary to install a vacuum breaker on the steam inlet to such equipment or onto the plant body. The same situation can occur on steam mains and boilers.

A common application of vacuum breakers is on temperature-controlled heat exchangers that are likely to suffer from stall (see Block 13). On smaller heat exchangers draining to atmosphere, the stall condition can be avoided by installing a vacuum breaker on the steam inlet to the heat exchanger. When the vacuum is reached in the steam space, the vacuum breaker opens to allow condensate to drain down to the steam trap.

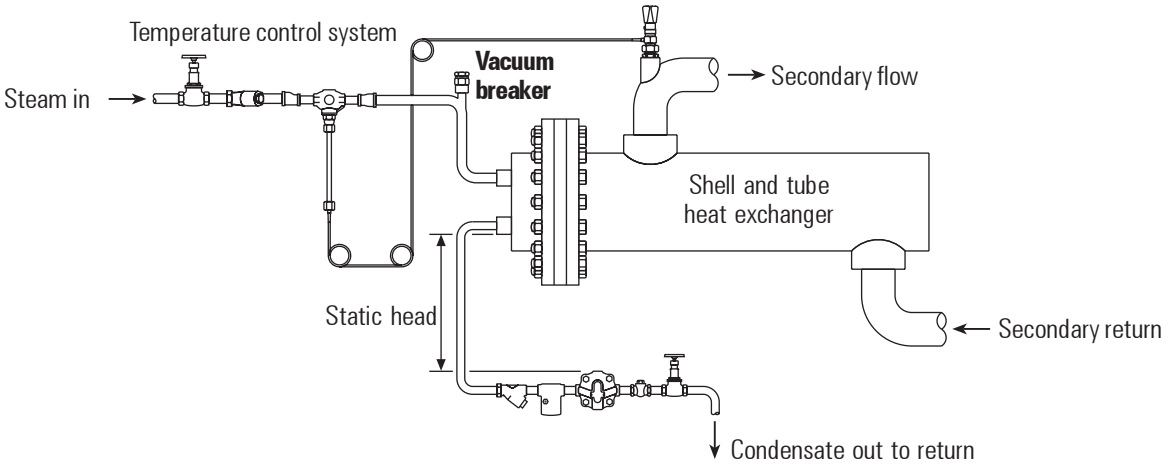


Fig. 12.6.10 The use of a vacuum breaker to prevent stall

In general, it is not desirable to introduce air into the steam space, since it acts as a barrier to heat transfer and reduces the effective steam temperature (refer to Module 2.4). This becomes a problem on larger heat exchangers, where it is not advisable to use a vacuum breaker to overcome stall. Furthermore, if the condensate is lifted after the steam trap, for example, into a raised condensate return main, the vacuum breaker cannot assist drainage. In both these cases, it is necessary to use an active method of condensate removal such as a pump-trap (refer to Module 13.8).

Questions

1. Where is it important to install a pressure gauge?

a| Downstream of a pressure reducing valve station only ☐

b| Upstream of a pressure reducing valve station only ☐

c| Downstream of a steam trap to ensure that live steam is not leaking ☐

d| Both upstream and downstream of a pressure reducing valve station ☐

2. Why should a Bourdon type pressure gauge be fitted to a syphon tube when used on a steam system?

a| To protect the Bourdon tube from erosion in the fast moving steam ☐

b| To protect the pressure gauge from the high temperature associated with steam ☐

c| To ensure that the pressure gauge only measures the static pressure ☐

d| A gauge cock can be fitted to the siphon tube so that the pressure gauge can be isolated when not in use ☐

3. What is the purpose of liquid filled pressure gauges?

a| Keeps out ambient corrosives and condensation ☐

b| Dampens the movement of the pointer ☐

c| Prevent damage to the internal mechanisms from vibrations ☐

d| All of the above ☐

4. What is the main application of a sight check?

a| To monitor steam traps that drain into a raised condensate main ☐

b| To monitor blast action steam traps, in which case it must be installed at least 1 m downstream of the trap ☐

c| To replace check valves on boiler feedlines ☐

d| For inspection of tanks in low light conditions ☐

5. Where should vacuum breakers be installed?

a| On steam mains ☐

b| On process equipment ☐

c| On small heat exchangers that are prone to stall ☐

d| All of the above ☐

6. Why is it disadvantageous to use a vacuum breaker in large heat exchangers to prevent stall?

a| Air acts as a barrier to heat transfer ☐

b| It leads to air locking of the steam trap ☐

c| The amount of air present in a large heat exchanger at start-up is sufficient to prevent any vacuum forming ☐

d| Vacuum breakers can only assist drainage when small quantities of condensate have to be raised to an elevated condensate return main ☐

Answers

1: d, 2: b, 3: d, 4: a, 5: d, 6: a

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Module 13.1

Heat Exchangers and Stall

Heat Exchangers and Stall

Foreword

This Block discusses the removal of condensate from heat exchange equipment supplied by saturated steam and fitted with:

- A temperature control valve on the steam line to the heat exchanger.
- A steam trapping device on the condensate line from the heat exchanger.

The primary side of the heat exchanger will be referred to as the 'steam space', and the steam trapping device will be referred to as the 'trap'. The 'trap' can be a 'steam trap', a 'pump trap', or a 'steam trap and pump' fitted in combination.

On these installations, a control sensor monitors the temperature of the outgoing heated fluid in the secondary circuit. The control valve endeavours to maintain a temperature determined by the controller, regardless of variations in heat load. The valve achieves this by opening or closing to alter the flowrate of steam, thereby varying the steam space pressure.

The discharge from the steam trap may be subject to a lift and/or pressure in the condensate line, or may fall to an open end where it is subjected only to atmospheric pressure. This Block will refer to condensate pressure as 'backpressure'.

The heat exchange equipment can be almost anything that meets the above criteria. Examples include:

- Shell and tube heat exchangers.
- Plate heat exchangers.
- Air heating coils or batteries in ductwork.
- Pipe runs or pipe coils in process equipment, tanks, vats etc.

For brevity, this Block will refer to all such devices as 'heat exchangers' or 'heaters', and the passage of fluid being heated by the heat exchanger will be referred to as passing through the 'secondary' side of the heat exchanger.

The performance of steam heat exchangers is often reduced due to condensate flooding the steam space and waterlogging. The two main causes of waterlogging are:

- Fitting the wrong type of trap.
- Stall.

Important note

Some systems aim to achieve control of temperature by positively encouraging partial flooding of the steam space of the heat exchanger. In these cases, the modulating action of the control valve at the condensate outlet varies the condensate level in the steam space. This changes the area of heating surface exposed to steam, and the effect is to change the heat transfer rate so as to control the secondary outlet temperature.

With systems of this type, it is important that the heat exchangers be designed and manufactured specifically to withstand the effects of flooding. Where this is not done, the presence of condensate in the heat exchanger will have an adverse effect on operating performance and will reduce service life.

This method of control can have certain benefits if the system is designed correctly. One is that the condensate sub-cools in the heat exchanger before it is discharged. This can considerably reduce the amount of flash steam in the condensate pipework, which may improve the performance of the condensate system and also reduce heat losses.

The main operational disadvantage is that systems of this type are slow to respond to variations in heat load.

What is meant by stall?

Stall is the reduction or the cessation of condensate flow from the heat exchanger, and occurs when the pressure in the heat exchanger is equal to, or less than, the total backpressure imposed on the steam trap.

Lower than expected pressure in a heat exchanger may occur as a result of any of the following circumstances:

- The secondary fluid inlet temperature rising as a result of a falling heat load.
- The secondary fluid flowrate falling as a result of a falling heat load.
- The secondary fluid outlet temperature falling due to a lowering of the set point.

As the control valve reduces the steam pressure to meet a falling heat load, the lack of differential pressure across the steam trap causes condensate to waterlog the steam space, as shown in Figure 13.1.1.

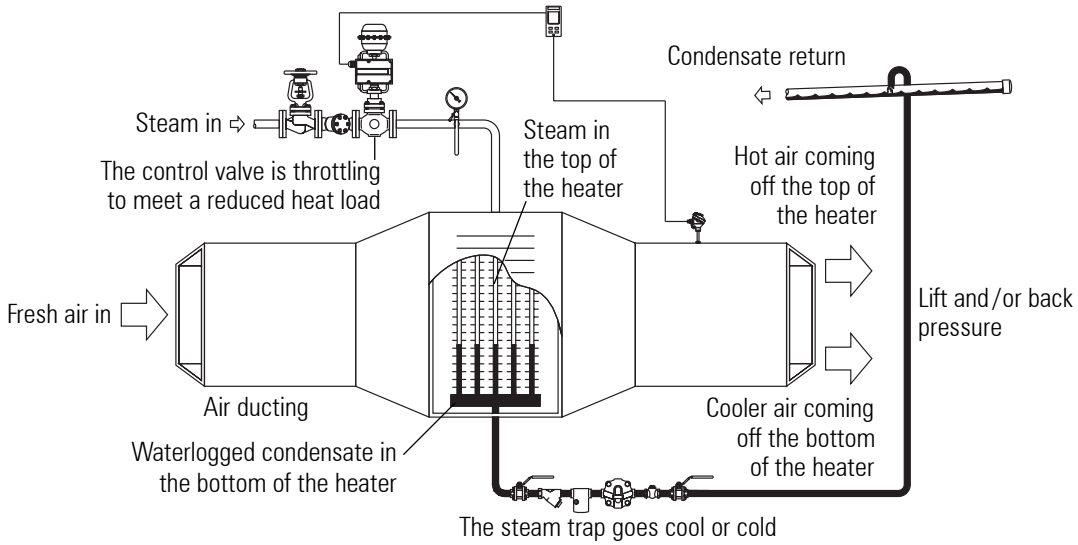


Fig. 13.1.1 An air heater battery suffering the effects of stall

Due to applied safety factors and because heat exchangers are sold in pre-determined sizes, they often have more heating area than required. This has the effect of increasing the heat transfer capability of the exchanger above that required. It also means that the operating steam pressure will be lower than in a comparable heat exchanger perfectly sized for the same duty. The result is that less steam pressure is available to push out the condensate than may be expected. The steam pressure in the heat exchanger is important because it influences the stall condition, which in turn affects trap selection.

Before any trap selection and sizing can take place, it is necessary to determine whether or not stall will occur, and if it does, to what degree. If this is not done, it is likely that the heat exchanger will suffer from waterlogging for some or all of its operating life. This, when it occurs, may not be immediately recognised by the observer or operator, as operating performance might not be reduced in an oversized heat exchanger. However, waterlogging can have severe financial consequences, short and long term, unless the heat exchanger is designed to operate this way.

Short-term problems

Consider an oversized heater battery operating as a frost coil and fitted with the wrong type (or size) of trap, as in Figure 13.1.1.

In this example, the frost coil is preheating chilled air before it passes on to the main heater battery. Though the frost coil is fulfilling its thermal expectations (because it is oversized for the duty), it will do so with the bottom half of its coils waterlogged. Incoming cold air approaching 0°C (typically flowing at 3 m/s) passing over the coils can easily cause the water in them to freeze. This results in having to repair or replace the heater battery, either causing inconvenience or unexpected outlay.

Waterlogging and freezing will not arise if the application is correctly designed.

Long-term problems

Traps that are undersized will sometimes show no immediate adverse effects on heater performance if the heater is oversized.

Ironically, the wrong type of trap fitted to a heat exchanger can often exaggerate a superficial improvement elsewhere in the condensate system. For instance, a thermostatic or fixed orifice fitted to any heat exchanger will hold back condensate so that it sub-cools below the steam saturation temperature. This will have the effect of reducing flash steam from any natural outlet such as a condensate receiver vent. The casual observer can interpret this as a way to save energy and can easily be tempted to fit these devices. Unfortunately, the situation is not as straightforward as it seems. The reality is that holding back condensate until it sub-cools implies waterlogging to some degree. Condensate that continually floods the steam space will cause corrosion with costly results. The service life of the heat exchanger is reduced, and the overall lifetime costs of the installation will increase.

The effects suffered by a waterlogged heat exchanger depend upon the circumstances of the particular installation.

The symptoms and effects of stall are itemised later in this Module.

How does stall occur?

To understand stall it is necessary to appreciate that saturated steam is a condensing vapour, which gives up its heat as it condenses to water. This condensation always occurs at a constant temperature when the pressure in the steam space remains constant.

For example, saturated steam at atmospheric pressure has a temperature of 100°C and will also condense back into water at 100°C, whereas at a gauge pressure of 1 bar, saturated steam has a temperature of 120°C and will condense back into water at 120°C. Steam can also exist inside heat exchangers at below atmospheric pressure i.e. steam at 0.5 bar below atmospheric pressure has a temperature of about 82°C, and will also condense back to water at 82°C. The pressure and temperature relationship of saturated steam is entirely predictable and is documented in steam tables.

Basic heat exchanger theory states that the higher the steam temperature above that of the secondary fluid being heated, the greater the potential heat transfer rate. To vary the transfer of heat from condensing steam, the temperature (and thus the pressure) of the steam in the steam space is varied.

For example, if a heat exchanger uses steam at 160°C at maximum load, and the load is reduced by 50%, steam at a lower temperature is required. To achieve this, the steam pressure must be reduced, and, in many cases, becomes less than the backpressure.

Example:
A heat exchanger running at full-load uses saturated steam at 1 bar g (120°C) to heat water from 40°C to 60°C. Full-load therefore occurs when the water temperature rises by 20°C, and the mean water temperature is:

$$\text{Mean water temperature at full-load} = \frac{40°C + 60°C}{2} = 50°C$$

The difference between the steam temperature and the mean water temperature is termed the Arithmetic Mean Temperature Difference or AMTD, and the heat transfer rate is proportional to this. The full-load AMTD in this example is 120°C - 50°C = 70°C.

Consider the situation where the process load falls to $^2/_3$ load.

At full-load, the water temperature rise is 20°C.

If the load falls to $^2/_3$ full-load, and the outlet water temperature remains constant at 60°C, this means that the temperature rise must be $^2/_3$ of 20°C

Therefore:
At $^2/_3$ load, temperature rise = $^2/_3$ of 20°C = 13.3°C
and the inlet temperature = 60°C - 13.3°C = 46.7°C

Consequently at $^2/_3$ load, the return water temperature will have risen to 46.7°C, and so the mean water temperature is now:

$$\text{Mean water temperature at } \frac{2}{3} \text{ load} = \frac{46.7°C + 60°C}{2} = 53.3°C$$

At $^2/_3$ load, the heat transfer needed will be $^2/_3$ of that at full-load, and equally the AMTD will be $^2/_3$ of that at full-load, i.e.

$$\text{AMTD at } \frac{2}{3} \text{ load} = \frac{2}{3} \times 70°C = 46.7°C$$

It follows that the steam temperature at $^2/_3$ load has to be the mean water temperature at $^2/_3$ load plus the AMTD at $^2/_3$ load, i.e.

$$\text{Steam temperature at } \frac{2}{3} \text{ load} = 53.3°C + 46.7°C = 100°C$$

As the temperature of saturated steam at atmospheric pressure is 100°C, this means that the pressure in the steam space is now atmospheric. Consequently, there is no steam pressure available in the steam space to push the condensate through a steam trap. Even if the condensate line fell to an open-ended steam trap, the condensate might not drain out of the exchanger. The condensate will 'back-up' the drain line and waterlog the heat exchanger unless proper precautions are taken.

If condensate backs up into the exchanger, the surface area available to condense steam is reduced, the heat flow drops and the temperature of the outgoing heated water begins to fall. When the temperature sensor detects this, the controller opens the control valve a little more and the inflow of steam increases. This raises the pressure in the steam space above atmospheric (in this case) and soon becomes high enough to push condensate through the trap. The condensate level falls, but now the steam space pressure is higher than the atmospheric pressure needed to just heat the water to 60°C. The water temperature then climbs. When the sensor detects this, the controller closes down the control valve. The steam space pressure falls to atmospheric - and the flooding begins again.

The result is a continual cycling of the water temperature above and below 60°C. If the secondary medium were other than water this could, in many cases, affect its quality.

What are the symptoms and effects of stall?

One or more of the following symptoms may be evident:

In summary:

1. Cold or cool steam trap.
2. Hunting control valve.
3. Fluctuating outlet temperature.
4. Stratified heater temperatures.
5. Waterhammer.
6. Reduced heat output.
7. Reduced product quality.
8. Corroding heat exchangers.
9. Leaking heat exchangers.
10. Failing heat exchangers.

In detail:

- The steam trap goes cold, or is noticeably cooler than the temperature of the steam pipe inlet to the heat exchanger.
- The control valve is prone to 'hunting', i.e. it cycles regularly somewhere between its open and closed positions.

- The temperature of the secondary fluid flowing from the heat exchanger is less accurate than is expected or required.
- There is stratification of temperature on the output side of the heat exchanger. This will be more apparent on heater batteries and unit heaters.

  For example, it is almost certain to be detectable on the air heater battery depicted in Figure 13.1.1. The design is such that the face of the heat exchanger surface is usually accessible, often via an access panel or door in the side of the ducting. If stall is happening, the top of the battery closest to the steam inlet will be very hot, whereas lower down, it will be much cooler or even cold, and the trap will be cool or cold. The temperature of the air flowing through the top of the battery will be noticeably higher than that flowing through the bottom.
- The heat exchanger makes crackling, banging or thumping noises either continuously or intermittently. Sometimes these noises are associated with severe waterhammer that can cause physical damage to the heat exchanger and any equipment fitted to it. The hot steam condensing into the waterlogged condensate causes the waterhammer and resulting noises, especially when the waterlogging level varies with changes in load.
- In process applications, the result of one or more of the above symptoms may be poor or unreliable product quality.
- Increased corrosion. The waterlogged condensate cools to temperatures much lower than the steam temperature at the inlet to the steam space. Carbon dioxide and oxygen dissolve much more readily into cooler water.
- Carbon dioxide is a common by-product of incorrect boiler water treatment and is carried over into the heat exchanger with the steam. When it dissolves into water it forms carbonic acid, which causes corrosion.

  Oxygen is present in raw water, and if not completely removed by the water treatment process, it too will get carried over with the steam. Its presence in water, especially cool water in which it will readily dissolve, also aggravates corrosion.

  Corrosion rates are greatly accelerated when both gases are present.

  The degree of corrosion will depend upon the heat exchanger material. Copper, carbon steel, and stainless steel will each be affected differently.
- Mechanical stress.

  The hot steam in the top of the steam space will cause the heat exchanger to expand there, while the cool water in the bottom of the steam space has the reverse effect. This uneven expansion / contraction can cause mechanical stress to the heat exchanger structure, notably to the soldered, brazed, welded or expanded joints in 'plate' and 'shell and tube' heat exchangers, and air heater batteries. The most common result is leakage of steam to the surroundings in the former, or into the secondary airflow in the latter. The stress tends to be worse if the waterlogging level continually varies, especially if it varies quickly. The level of waterlogging will vary as the load changes, and as a result; the control valve and steam trap will struggle to achieve stable control.

  It should be said that a properly engineered plate heat exchanger with gasket joints suitably designed for steam will be very resilient to such stress.

The ultimate effect of stall is increased maintenance and shorter service life of the heat exchanger and associated equipment. This increases overall running costs.

Do all heat exchangers suffer from stall?

No. The conditions may be such that there will **always** be sufficient positive pressure upstream of the steam trap to clear the condensate so stall cannot occur.

As a general rule, the higher the secondary temperature above 100°C, and the more stable the running load, (especially if near to the maximum output of the heat exchanger), the less likely for stall to occur. However, each application is unique and will require individual consideration. The only ways to determine the dynamics of the installation are to either plot the application temperatures on a chart or to perform a mathematical calculation. This is explained in Module 13.2, 'Condensate Removal from Heat Exchangers'.

Some applications can appear to operate with partial waterlogging, and show little effect of waterhammer. These tend to be steady load applications, or where the load changes only slightly and very slowly, and/or applications that employ very robust heat exchange equipment.

One such example would be large bore corrosion resistant heating coils inside tanks correctly arranged to have a positive fall towards the trapping points.

Even in applications of this type, if the installation is designed or corrected to eliminate stall, improved operation, improved reliability, and reduced lifetime costs are virtually guaranteed.

Questions

1. What is the prime cause of stall in heat exchangers?

a| Not enough steam pressure upstream of the control valve ☐

b| The heat exchanger pressure is equal to or less than the backpressure ☐

c| The heat exchanger is undersized ☐

d| The condensate discharges to atmosphere ☐

2. What effects will waterlogging cause in some heat exchangers?

a| None at all ☐

b| It increases the steam pressure in the heat exchanger ☐

c| It can cause swings in the temperature of the heated fluid and corrode the heat exchanger? ☐

d| It increases the thermal performance ☐

3. In a heat exchanger at full-load, steam temperature is 140°C, inlet temperature is 20°C, and the outlet temperature is 80°C, what is the AMTD?

a| 40°C ☐

b| 120°C ☐

c| 60°C ☐

d| 90°C ☐

4. In the same heat exchanger at half load, what is the secondary mean temperature?

a| 65°C ☐

b| 50°C ☐

c| 80°C ☐

d| 45°C ☐

5. In the same heat exchanger at half load, what is the steam temperature?

a| 70°C ☐

b| 100°C ☐

c| 110°C ☐

d| 95°C ☐

6. If, for the same duty, a larger heat exchanger were used, what would be expected of the steam temperature at full-load?

a| It would be higher than 140°C ☐

b| It would be lower than 140°C ☐

c| It would also be at 140°C ☐

d| It would be at 100°C, i.e. atmospheric pressure ☐

Answers

1:b, 2: c, 3:d, 4: a, 5: c, 6: b

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Module 13.2

The Heat Load, Heat Exchanger and Steam Load Relationship

The Heat Load, Heat Exchanger and Steam Load Relationship

Saturated steam is used to provide primary heat to a process fluid in a heat exchanger. The term heat exchanger is used to describe all types of equipment where heat transfer is promoted from one fluid to another. For convenience, this broad definition will be applied to the term heat exchanger. While shell and tube heat exchangers and plate heat exchangers will be principally referred to, stall may also be relevant to applications including air heater batteries, submerged tank coils, jacketed vessels and storage calorifiers.

Temperature controlled applications

In a temperature control application, the inlet temperature of the secondary fluid to the heat exchanger may change with time. This means that in order to maintain a consistent secondary fluid outlet temperature, the heat supplied to the heat exchanger must also vary. This can be achieved by using a control valve on the inlet to the primary side of the heat exchanger, as shown in Figure 13.2.1.

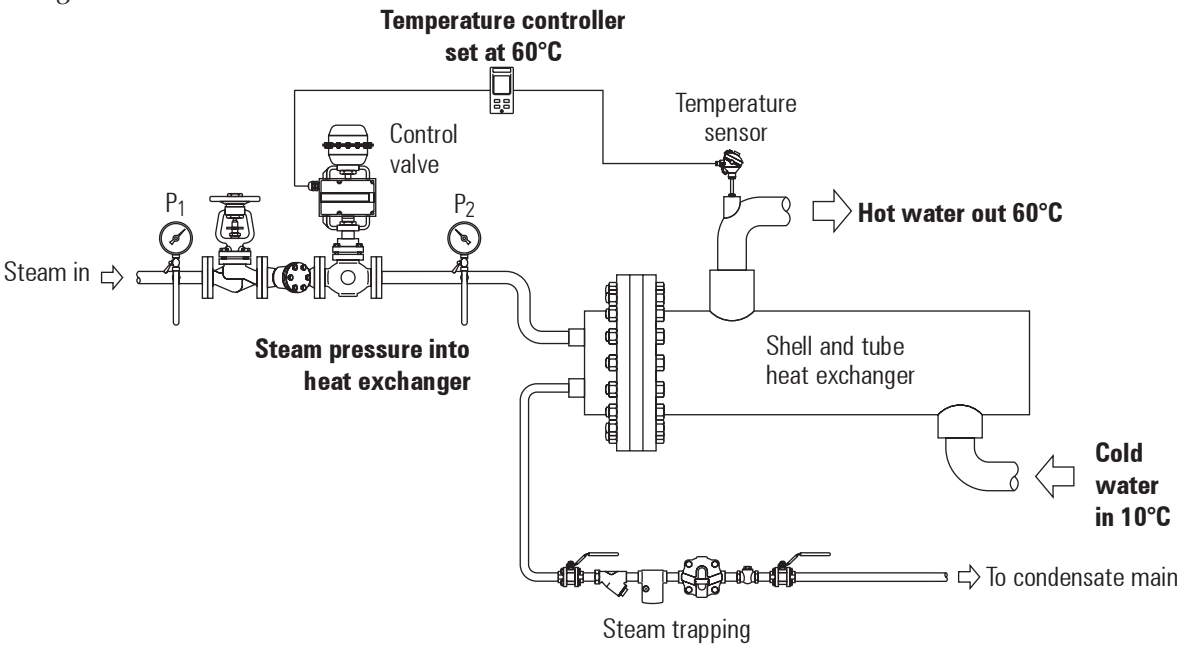


Fig. 13.2.1 Typical temperature control of a steam/water shell and tube heat exchanger

A control valve is used to vary the flowrate and pressure of the steam so that the heat input to the heat exchanger can be controlled. Modulating the position of the control valve then controls the outlet temperature of the secondary fluid. A sensor on the secondary fluid outlet monitors its temperature, and provides a signal for the controller. The controller compares the actual temperature with the set temperature and, as a result, signals the actuator to adjust the position of the control valve.

For a constant heating area and heat transfer coefficient, the rate at which heat is transferred from the steam to the secondary fluid for a particular heat exchanger is determined by the mean temperature difference between the two fluids. A larger difference in mean temperatures will create a large heat transfer rate and vice versa. On partially closing the control valve, the steam pressure and the temperature difference fall. Conversely, if the control valve is opened so that the steam mass flow and hence pressure in the heat exchanger rise, the mean temperature difference between the two fluids increases.

Altering the steam pressure will also slightly affect the amount of heat energy available in the condensing steam as the enthalpy of evaporation actually falls with increasing pressure. This means that the latent heat available per kg of steam reduces as the steam pressure increases. If steam flow accuracy is required, this must be accounted for.

Example 13.2.1

A manufacturer is to design a heat exchanger in which the specification calls for steam at 4 bar g to heat secondary water from 10°C to 60°C. The water flow is to be constant at all loads at 1.5 L/s. It is assumed that 1 litre of water has a mass of 1 kg, so the mass flowrate = 1.5 L/s x 1 kg/L = 1.5 kg/s. The manufacturer uses a heat transfer coefficient 'U' for the heat exchanger of 2 500 W/m²°C. Take the specific heat of water as 4.19 kJ/kg °C.

Determine:

(**A**) The design heat load.

(**B**) The corresponding steam flowrate.

(**C**) The minimum heating area required.

Also, if the customer's minimum heat load occurs when the inlet water temperature rises to 30°C, determine:

(**D**) The minimum heat load.

(**E**) The corresponding steam pressure in the heat exchanger.

(**F**) The corresponding steam flowrate.

Calculations:

(**A**) **Find the design heat load** using the heat transfer flowrate equation (Equation 2.6.5):

$$\dot{Q} = \dot{m}\, c_p\, \Delta T \qquad \textbf{Equation 2.6.5}$$

Where:

$\dot{Q}$ = Mean heat transfer rate (kW)

$\dot{m}$ = Mean secondary fluid flowrate (kg)

c_p = Specific heat capacity of the secondary fluid (kJ/kg K) or (kJ/kg °C)

ΔT = Temperature rise of the secondary fluid (K or °C)

$$\dot{Q} = 1.5 \text{ kg/s} \times 4.19 \text{ kJ/kg °C} \times (60 - 10)\text{°C}$$

$$\mathbf{\dot{Q} = 314.25 \text{ kW}}$$

(**B**) **Find the corresponding steam flowrate** at 4 bar g, saturation temperature (T_S) is 152°C, and h_{fg} = 2 108.1 kJ/kg (from steam tables). Calculate the required steam flow at the design condition using Equation 2.8.1:

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW} \times 3\,600}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\text{Steam flowrate } (\dot{m}_S) = \frac{314.25 \times 3\,600}{2\,108.1} \text{ kg/h}$$

$$\mathbf{\text{Steam flowrate } (\dot{m}_S) = 536.6 \text{ kg/h}}$$

(C) Find the minimum heating area to meet the requirement using Equation 2.5.5. Note; the manufacturer uses the Logarithmic Mean Temperature Difference (ΔT_{LM}) to calculate the minimum amount of heating area to satisfy the design rating:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)}$$ **Equation 2.5.5**

Where:
ΔT_{LM} = Logarithmic Mean Temperature Difference (LMTD)
T_s = Steam temperature (°C)
T_1 = Secondary fluid in temperature (°C)
T_2 = Secondary fluid out temperature (°C)
ln = The mathematical function known as 'natural logarithm'

$$\Delta T_{LM} = \frac{60 - 10}{\ln\left(\frac{152 - 10}{152 - 60}\right)}$$

$$\Delta T_{LM} = \frac{50}{\ln\left(\frac{142}{92}\right)}$$

$$\Delta T_{LM} = \frac{50}{0.434}$$

$$\mathbf{\Delta T_{LM} = 115.2°C}$$

By re-arranging the general heat transfer equation (Equation 2.5.3: $\dot{Q} = U \times A \times \Delta T$) Equation 13.2.1 can be formulated, where ΔT can be represented by the mean value ΔT_M.

$$A = \frac{\dot{Q}}{U\ \Delta T_M}$$ **Equation 13.2.1**

Where:
A = Heating area (m^2)
$\dot{Q}$ = Mean heat transfer rate (W)
U = Heat transfer coefficient ($W/m^2\,°C$)
ΔT_M = Mean Temperature Difference. Note: ΔT_M may be either ΔT_{LM} (LMTD) or ΔT_{AM} (AMTD).

$$A = \frac{314\,250\ W}{2\,500\ W/m^2\,°C \times 115.2°C}$$

$$\mathbf{A = 1.09\ m^2}$$

For the purpose of this example it will be assumed that the heat exchanger is designed to have exactly this area of 1.09 m^2.

(D) Find the minimum heat load, when the inlet water temperature is 30°C, using the heat transfer flowrate equation (Equation 2.6.5) as used in Part 'A' of these calculations:

$$\dot{Q} = \dot{m}\ c_p\ \Delta T$$ **Equation 2.6.5**

$$\dot{Q}_{min} = 1.5\ kg/s \times 4.19\ kJ/kg\,°C \times (60 - 30)°C$$

$$\mathbf{\dot{Q}_{min} = 188.5\ kW}$$

To calculate the corresponding steam flowrate, it is first necessary to determine the steam temperature at the minimum load condition.

It is possible to use the ΔT_{LM} design figures to accurately predict the steam temperature for any load condition, but this requires the use of logarithmic calculations. However, once the exchanger size is fixed and the design temperatures are known, it is much easier to predict operating temperatures using what could be termed a heat exchanger Temperature Design Constant (TDC). The TDC method does not require logarithmic calculations. **Please note:** TDC cannot be used on those applications where the secondary flowrate varies or where control is achieved by varying the condensate level in the steam space.

Note: When sizing a heat exchanger it is normal for heat exchanger manufacturers to use the ΔT_{LM} method. Once sized, by knowing the heating area and the full-load operating temperatures, TDC can be used to accurately predict all operating temperatures resulting from changes in load, as can be seen in the following text.

Operating temperatures can also be predicted graphically by using what is termed a 'Stall Chart'. This method is discussed in Modules 13.5, 13.6, and 13.7.

Temperature Design Constant (TDC)

For any type of steam-heated exchanger with the secondary liquid flowing at a constant rate, TDC can be calculated from the test figures quoted by the manufacturer for full-load. If these data sets are not available and the heat exchanger is already installed in service, TDC can be calculated by observing the steam pressure (and finding the steam temperature from steam tables) and the corresponding secondary inlet and outlet temperatures at any load.

TDC is the ratio of the steam to water temperatures at the inlet and outlet; and is shown in Equation 13.2.2.

$$TDC = \frac{T_s - T_1}{T_s - T_2} \qquad \textbf{Equation 13.2.2}$$

Where:

TDC = Temperature Design Constant

T_s = Steam temperature

T_1 = Secondary fluid inlet temperature

T_2 = Secondary fluid outlet temperature

In Example 13.2.1 at full-load conditions:

The steam pressure = 4 bar g

The inlet water temperature (T_1) = 10°C

The outlet water temperature (T_2) = 60°C

Steam temperature at 4 bar g (T_s) = 152°C

$$TDC = \frac{T_s - T_1}{T_s - T_2}$$

$$TDC = \frac{152 - 10}{152 - 60}$$

$$TDC = \frac{142}{92}$$

TDC = 1.543 5 for this particular heat exchanger

The TDC equation can be transposed to find any one variable as long as the other three variables are known. The following equations are derived from the TDC equation (Equation 13.2.2).

To find the steam temperature at any load use Equation 13.2.3:

$$T_s = \frac{(T_2 \times TDC) - T_1}{TDC - 1} \qquad \textbf{Equation 13.2.3}$$

To find the secondary fluid inlet temperature at any load use Equation 13.2.4:

$$T_1 = T_s - [\, TDC\,(T_s - T_2)\,] \qquad \textbf{Equation 13.2.4}$$

To find the secondary fluid outlet temperature at any load use Equation 13.2.5:

$$T_2 = T_s - \left[\frac{T_s - T_1}{TDC}\right] \qquad \textbf{Equation 13.2.5}$$

For any heat exchanger with a constant secondary flowrate, the operating steam temperature can be calculated for any combination of inlet temperature and outlet temperature.

In Example 13.2.1 the secondary outlet temperature remains at 60°C, and minimum load occurs when the inlet temperature is 30°C. What is the steam temperature at minimum load?

Inlet temperature = 30°C

Outlet temperature = 60°C

Using Equation 13.2.3:

$$T_s = \frac{(T_2 \times TDC) - T_1}{TDC - 1}$$

$$T_s = \frac{(60 \times 1.543\,5) - 30}{0.543\,5}$$

$$T_s = \frac{62.61}{0.543\,5}$$

Steam temperature (T_s) = 115.2°C

(E) Find the corresponding heat exchanger steam pressure and enthalpy at minimum load

From steam tables:

A steam temperature of 115.2°C corresponds with a **steam pressure of 0.7 bar g.**

The specific enthalpy of evaporation at 0.7 bar g (h_{fg}) = 2 215 kJ/kg

(F) Find the steam flowrate at minimum load:

From **(D)** the minimum heat load is 188.5 kW.

From **(E)** the h_{fg} is 2 215 kJ/kg.

Using Equation 2.8.1:

$$\text{Steam flowrate (kg/h)} = \frac{\text{kW rating} \times 3\,600}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\text{Steam flowrate } (\dot{m}_S) = \frac{188.5 \times 3\,600}{2\,215} \text{ kg/h}$$

Steam flowrate ($\dot{m}_S$) = 306.4 kg/h at minimum load

Questions

1. What determines the rate of heat transfer for any given heat exchanger?

a| The heat transfer coefficient ☐
b| The mean temperature difference between the two fluids ☐
c| The heating surface area ☐
d| All of the above ☐

2. How is the temperature of steam controlled in a heat exchanger?

a| By a steam trap ☐
b| By changing the steam pressure upstream of the control valve ☐
c| By controlling the steam flow and pressure in the steam space ☐
d| By an adjustable safety valve ☐

3. What is LMTD?

a| Logarithmic Maximum Temperature Difference ☐
b| Latent Mean Temperature Difference ☐
c| Logarithmic Mean Temperature Difference ☐
d| Lowest Minimum Temperature Difference ☐

4. What is the basic difference between LMTD and TDC?

a| None ☐
b| LMTD is used to accurately calculate a required heating surface area while TDC can be used to easily predict operating temperatures ☐
c| LMTD is easier to use ☐
d| Using TDC is more accurate than using LMTD ☐

5. What effect does lowering the steam pressure have?

a| The steam temperature rises ☐
b| It has no effect on the temperature but increases the latent heat ☐
c| The steam temperature falls ☐
d| The total heat in steam increases ☐

6. Knowing the heat load, what other factor has to be known to accurately determine the steam mass flowrate to any piece of equipment?

a| The temperature of steam ☐
b| The total heat in the steam ☐
c| The enthalpy of evaporation of the steam at the evaporating pressure ☐
d| The specific volume of the steam ☐

Answers

1: d 2: c, 3: c, 4: b, 5: c, 6: c

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Module 13.3

Oversized Heat Exchangers

Oversized Heat Exchangers

The effect of oversizing a heat exchanger

The previous calculations (Module13.2) assumed that the heat exchanger had been sized on the perfect heating area to meet the specification. This would mean that the heat exchanger was exactly sized for the duty.

This is highly unlikely in practice as the designer or specifier will usually add other factors, including those for fouling and uncertainty of maximum operating loads. It is also unlikely that manufacturers can supply heat exchangers to match a specification exactly. As undersized heat exchangers are impractical they are usually bought oversized.

The operating conditions laid down in Example 13.2.1, Part 'C', have been reconsidered in Example 13.3.1 by adding 15% to the required heating area to account for contingencies.

Required heating area is calculated to be 1.09 m² (Example 13.2.1, Part 'C') therefore the specified heating area for Example 13.3.1 is to be 1.09 + 15% = 1.254 m².

The minimum size that the manufacturer can supply has a heating area of 1.31 m², representing an actual heating area of some 20% above that required. A larger heating area requires less steam pressure for the same heat transfer rate, and because of this the steam pressure in an oversized heat exchanger will be lower for the same heat load.

As the steam pressure is less, the steam temperature is less, and the heat exchanger LMTD (Logarithmic Mean Temperature Difference) will also be less.

To determine the steam temperature for the design condition, it is first necessary to find the new LMTD (ΔT_{LM}) for the larger heating area (see Example 13.3.1).

Example 13.3.1

The ΔT_{LM} can be found by re-arranging Equation 13.2.1 to give Equation 13.3.1

$$A = \frac{\dot{Q}}{U\ \Delta T_M} \qquad \textbf{Equation 13.2.1}$$

$$\Delta T_M = \frac{\dot{Q}}{U\ A} \qquad \textbf{Equation 13.3.1}$$

Where:

ΔT_M = Mean temperature difference. Note: ΔT_M may be either ΔT_{LM} (LMTD) or ΔT_{AM} (AMTD)

$\dot{Q}$ = Mean heat transfer rate (W)

U = Heat transfer coefficient (W/m² °C)

A = Heating area (m²)

$$\Delta T_M = \frac{314.25\ \text{kW}}{2\ 500\ \text{W/m}^2\ ^\circ\text{C} \times 1.31\ \text{m}^2} = 95.95^\circ\text{C}$$

From Example 13.2.2, at full-load:

The secondary inlet temperature (T_1) = 10°C

The secondary outlet temperature (T_2) = 60°C

The new steam design temperature can now be determined using Equation 2.5.5:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)}$$ **Equation 2.5.5**

Where:

ΔT_{LM} = 95.95°C

T_1 = 10°C

T_2 = 60°C

T_S = Steam temperature °C

$$95.95 = \frac{60 - 10}{\text{Ln}\left(\frac{T_s - 10}{T_s - 60}\right)}$$

$$\text{Ln}\left[\left(\frac{T_S - 10}{T_S - 60}\right)\right] = \frac{50}{95.95}$$

$$\text{Ln}\left[\left(\frac{T_S - 10}{T_S - 60}\right)\right] = 0.5211$$

By taking antilogs of both sides of the equation . . .

$$\frac{T_S - 10}{T_S - 60} = e^{0.5211}$$

$$\frac{T_S - 10}{T_S - 60} = 1.684$$

$$T_S - 10 = 1.684\,(T_S - 60)$$

$$1.684\,T_S - T_S = (1.684 \times 60) - 10$$

$$0.684\,T_S = 101.4 - 10$$

$$\text{Steam temperature } (T_S) = \frac{91.04}{0.684}$$

$$\text{Steam temperature } (T_S) = 133.1°C$$

This temperature corresponds to a steam pressure of 1.95 bar g. When the heat exchanger was perfectly sized in Module 13.2, the steam pressure was 4 bar g. In this example, with a heat exchanger 20% oversized, the steam pressure is 51% less.

Now that the steam pressure has been predicted at the full-load condition, it is possible to calculate the steam flow at full-load.

By using Equation 2.8.1 find the steam flowrate at the full heat load of 314.25 kW. At 1.95 bar g, steam tables state that the enthalpy of evaporation is 2 164.6 kJ/kg.

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW x 3 600}}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\text{Steam flowrate } (\dot{m}_s) = \frac{314.25 \times 3\,600}{2\,164.4} \text{ kg/h}$$

$$\textbf{Steam flowrate } (\dot{m}_s) = \textbf{523 kg/h at full-load}$$

The steam flow was 536.6 kg/h in the perfectly sized heat exchanger (Example 13.2.1), so it can be seen that there is a slight drop (2.5%) in mass flowrate. This is due to the steam having a slightly larger enthalpy of evaporation in the larger heat exchanger due to its lower pressure.

Determine the TDC for the larger heat exchanger

Now that the steam temperature has been determined for the oversized heat exchanger (using the LMTD equation [Equation 2.5.5]), it is now possible to find its TDC, using Equation 13.2.2.

$$TDC = \frac{T_S - T_1}{T_S - T_2} \qquad \textbf{Equation 13.2.2}$$

Where:

TDC = Temperature Design Constant

T_S = 131.1°C

T_1 = 10°C

T_2 = 60°C

$$TDC = \frac{(133.1 - 10)}{(133.1 - 60)}$$

$$TDC = \frac{(123.1)}{73.1}$$

$$\textbf{TDC = 1.684}$$

At the minimum heat load:

When the heat exchanger was perfectly sized in Example 13.2.1 the steam temperature was 115.2°C at the minimum heat load of 188.5 kW.

Because the oversized heat exchanger in this example is about 20% larger, the steam temperature will also be less at the minimum heat load. The minimum heat load remains the same as in Example 13.2.1 and occurs when the secondary inlet temperature rises to 30°C.

From Equation 13.2.3:

$$T_S = \frac{(T_2 \times TDC) - T_1}{(TDC - 1)} \qquad \textbf{Equation 13.2.3}$$

Where:

T_S = Steam temperature °C = 133.1°C

T_1 = Secondary fluid inlet temperature °C = 30°C

T_2 = Secondary fluid outlet temperature °C = 60°C

TDC = Temperature Design Constant = 1.684

$$T_S = \frac{(60 \times 1.684) - 30}{(1.684 - 1)}$$

$$T_S = \frac{71.04}{0.684}$$

$$\mathbf{T_S = 103.8°C}$$

Comparing the two heat exchangers at minimum load, the steam temperature has dropped from 115.2°C in the perfectly sized heat exchanger to 103.8°C in the oversized heat exchanger.

From steam tables, this steam temperature corresponds with a steam pressure of about 0.15 bar g, and h_{fg} = 2 247 kJ/kg. The steam pressure in the perfectly sized exchanger (at 115.2°C) was 0.7 bar g.

By using Equation 2.8.1, it is possible to find the steam flow at the minimum heat load of 188.5 kW.

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW} \times 3\,600}{h_{fg} \text{ at operating pressure}}$$ **Equation 2.8.1**

$$\text{Steam flowrate } (\dot{m}_S) = \frac{188.5 \times 3\,600}{2\,247} \text{ kg/h}$$

Steam flowrate ($\dot{m}_S$) = 302 kg/h at minimum heat load

The minimum steam flow was 306 kg/h in the perfectly sized heat exchanger (Example 3.2.1), so it can be seen that there is a marginal drop in mass flow in the oversized heat exchanger at the minimum heat load. This is due to the steam having a slightly larger enthalpy of evaporation in the larger heat exchanger due to its lower pressure.

The steam pressure, the steam trap, and effective condensate removal

As the steam gives up its heat across the heat transfer surface to the secondary fluid, it condenses in the steam space. Condensate passes out through the outlet of the heat exchanger, and through a steam trap, which traps the steam in the steam space whilst allowing the condensate to be freely discharged.

If the heat exchanger has not been specifically designed to operate with condensate flooding the steam space, the steam pressure needs careful consideration to ensure the heat exchanger is properly drained of condensate. Any waterlogging of the steam space will reduce the effective heating surface area, and the heat transfer requirement may be satisfied only if the exchanger is sufficiently (perhaps accidentally) oversized.

The capacity of the steam trap will depend upon its type, its orifice size and the differential pressure across it. Differential pressure provides the energy to push the condensate through the trap, and is the difference between the steam pressure in the heat exchanger, and the backpressure exerted on the outlet of the trap by the condensate system.

If the steam trap drains by gravity via a properly sized pipe to a vented condensate receiver or an open end, the backpressure should be very near atmospheric. **Under these conditions, the differential pressure on a sizing chart can simply be read as the gauge pressure in the heat exchanger.**

If, however, there is a lift after the trap (a rise in the trap discharge line), or the trap discharge line is undersized, or this line is pressurised for any other reason, the backpressure may, at times, be greater than the pressure in the steam space. When this is so, the differential pressure across the trap is reversed and is deemed to be a 'negative differential pressure'. The trap capacity is now zero.

As can be seen in the above calculations, the steam pressure in any heat exchanger is governed by its size and the secondary conditions. As the capacity of the steam trap depends on the differential pressure, it follows that changes in the steam pressure and backpressure affect the capacity of the steam trap at all times. As the differential pressure reduces, the capacity of the steam trap will fall. Provided the differential pressure is positive and the steam trap is selected and sized with this in mind, waterlogging and its associated problems will not occur.

Sizing the steam trap for the oversized heat exchanger
The conditions that need consideration are:

- Full-load : 523 kg/h at 1.95 bar g in the steam space
- Minimum load: 302 kg/h at 0.15 bar g in the steam space
- Backpressure: Atmospheric pressure (0 bar g)

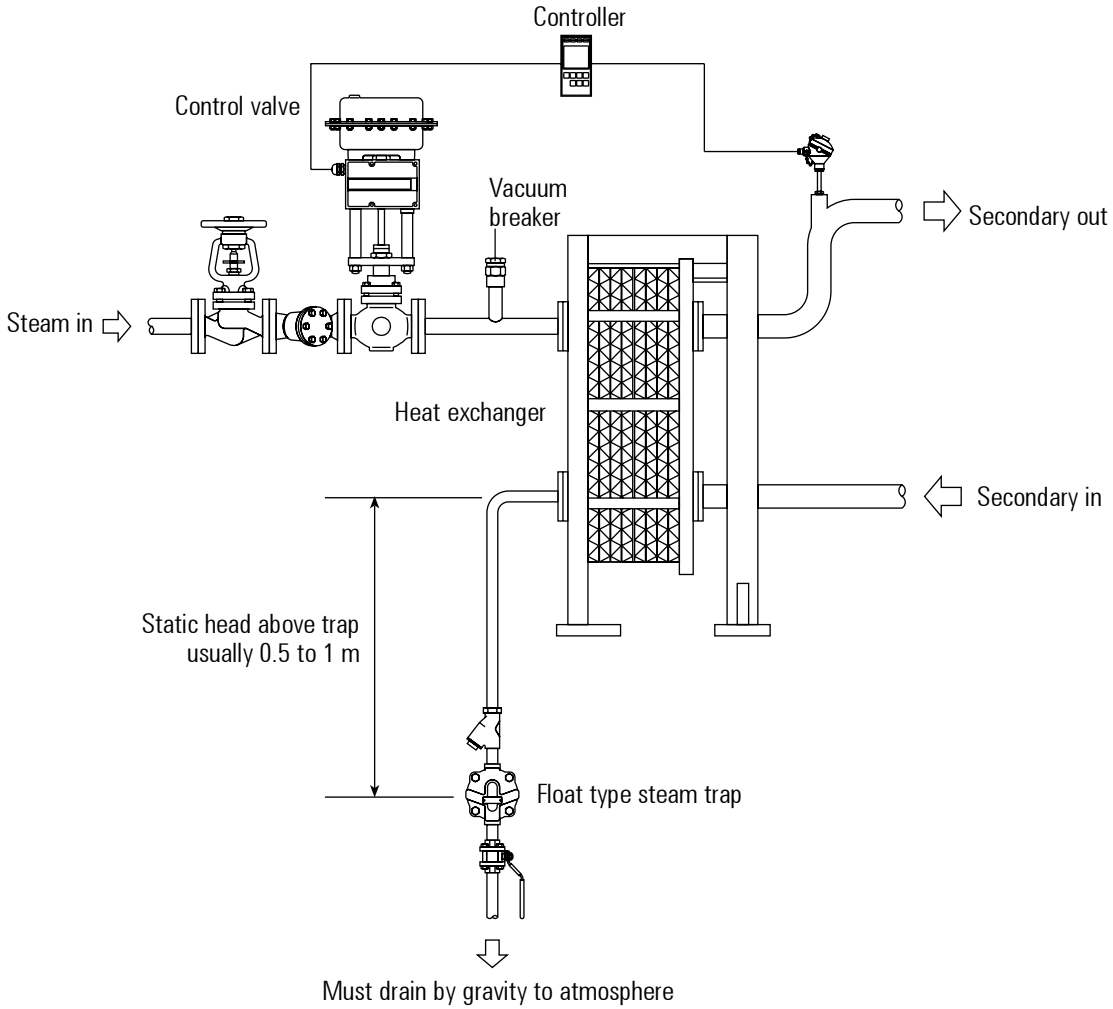


Fig. 13.3.1 Static head and vacuum breaker method of dealing with stall

Consider, on the float trap capacity chart Figure 13.3.2, a DN25 (1") FT14-4.5 ball float steam trap. It can be seen that it will pass 850 kg/h at a differential pressure of 1.95 bar. It may also be seen that at a differential pressure of 0.15 bar it will pass about 370 kg/h. In this example, consider the trap fitted to the oversized heat exchanger and draining by gravity to a vented condensate receiver, as depicted in Figure 13.3.1.

To ensure proper drainage, the steam trap has to be able to cope with all loads between the full-load and minimum load conditions.

As the condensate backpressure is atmospheric in this example, the minimum steam space pressure of 0.15 bar g is always higher than the backpressure. It can be seen from the capacity chart (Figure 13.3.2) that the trap has enough capacity at the minimum and maximum loads, so the DN25 (1") FT14-4.5 ball float steam trap is big enough.

If, however, in this example, the backpressure were higher than the minimum steam pressure of 0.15 bar g, the system would stall somewhere within the normal operating range. (This would only require a lift of just more than 1.5 metres after the trap to cause this). Accordingly, the trap would have to be selected and sized depending upon the amount of backpressure. With larger amounts of backpressure it may be necessary to fit a pump-trap.

Advice on how to select the correct trap for a heat exchanger is given in Module 13.4.

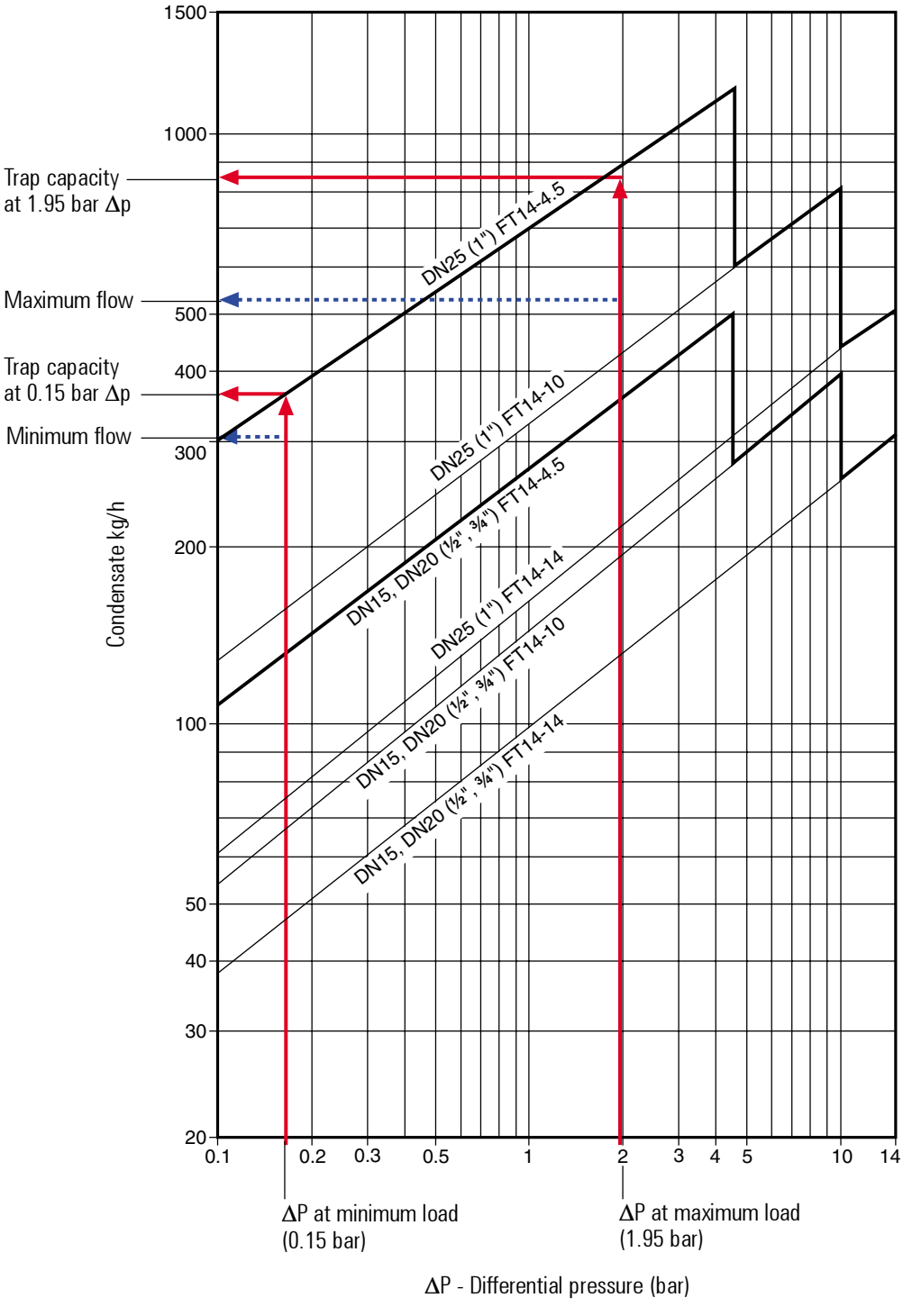


Fig. 13.3.2 FT14 ball float steam trap capacity chart showing data for Example 13.3.1

Questions

1. Why is it usual to fit an oversized heat exchanger?

a| It can cope with contingency loads ☐
b| It can cope with future fouling effects ☐
c| Because this is what the supplier will offer ☐
d| All of the above ☐

2. The oversized exchanger had about 20% more heating area than the one perfectly sized in Example 13.2.1. What was the percentage drop in steam pressure at the same minimum heat load?

a| About 78% drop in pressure ☐
b| About 24% drop in pressure ☐
c| About 10% drop in pressure ☐
d| The steam pressure remained the same in each heat exchanger ☐

3. For the same heat load on two heat exchangers, why is the mass flowrate of steam always less in the exchanger with greater heating surface?

a| Because the control valve is smaller ☐
b| The steam pressure is less and so the enthalpy of evaporation is more ☐
c| Because there is more heating area in the oversized exchanger ☐
d| The steam pressure is less and so the steam will be drier ☐

4. What is the effect of higher backpressure on a steam trap?

a| None whatsoever ☐
b| It reduces the steam pressure in the heat exchanger ☐
c| It reduces the capacity of a steam trap ☐
d| It increases the differential pressure across the steam trap ☐

5. What effect does a lowering steam pressure have?

a| The steam temperature rises ☐
b| It has no effect on the temperature but increases the latent heat ☐
c| The steam temperature falls ☐
d| The total heat in steam increases ☐

6. In Example 13.3.1, if the backpressure were 2 bar g, what size should the trap be?

a| Larger than 1″ ☐
b| 1″ ☐
c| A pump-trap should be used ☐
d| 2″ ☐

Answers

1: d, 2: c, 3: b, 4: c, 5: c, 6:c

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Module 13.4

Example: Selecting the Trap

Example: Selecting the Trap

Example 13.4.1 Selecting the trap

A factory requires a steam/water heat exchanger operating at a nominal 4 bar g to heat process water circulating at 1 L/s (1 kg/s) from 10°C to 80°C, giving a design load of 293 kW. The process is such that a minimum heat load occurs at 60% of the full heat load. This is a permanently running process line with no future load increase. Two suppliers are asked to provide a heat exchanger. The following information is important to selection:

- Supplier 'X' can provide a heat exchanger with a heating area of 2 m², a 'U' value of 2 500 W/m²°C and duty of 350 kW when operating with steam at 4 bar g and with a water flow of 1 L/s.
- Supplier 'Y' is able to provide a heat exchanger with a smaller heating area more suitable for the design heat load of 293 kW, when operating with steam at 4 bar g and with a water flow of 1 L/s. The 'U' value is 2 500 W/m²°C.
- The heat exchanger condensate line will lift 5 metres to a condensate return pipe that falls en route to a vented receiver, and having a total backpressure of 0.5 bar g. **Note:** A one metre column of water under atmospheric pressure will exert a pressure at the bottom of the column of approximately 10 kPa or 0.1 bar g. Any lift in the condensate discharge line will thus exert a static lift due to the column of condensate held in the line, in addition to any pressure in the condensate system.

It is necessary to determine the system operating conditions to select and size the trap for proper condensate removal from both heat exchangers under any operating load condition.

The following questions need to be answered for proper condensate removal:

(A) Will stall occur during normal operation?

(B) At what load will stall occur?

Check the application heat load at the design condition.

From the heat transfer flowrate equation (Equation 2.6.5):

$$\dot{Q} = \dot{m}\, C_p\, \Delta T \qquad \textbf{Equation 2.6.5}$$

$$\dot{Q} = 1.5 \text{ kg/s} \times 4.19 \text{ kJ/kg °C} \times (80 - 10)\text{°C}$$

Heat transfer rate ($\dot{Q}$) = 293 kW (293 000 W)

Consider supplier 'X'

A 350 kW heat exchanger with a 2 m² heating area.

What will be the steam space pressure in this heater at this design heat load?

It is first necessary to determine the LMTD (ΔT_{LM}) for a 2 m² heating area.

From Equation 13.2.1:

$$A = \frac{\dot{Q}}{U\, \Delta T_M} \qquad \textbf{Equation 13.2.1}$$

$$2 \text{ m}^2 = \frac{293\,000 \text{ W}}{2\,500 \text{ W/m}^2\text{°C} \times \Delta T_M}$$

$$\Delta T_M = \frac{293\,000 \text{ W}}{2\,500 \text{ W/m}^2\text{°C} \times 2 \text{ m}^2}$$

$$\Delta T_M = \mathbf{58.6°C}$$

The steam design temperature can now be calculated, by use of Equation 2.5.5:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)}$$ **Equation 2.5.5**

Where:
ΔT_{LM} = 58.6°C
T_1 = 10°C
T_2 = 80°C
T_S = Steam temperature °C

$$58.6 = \frac{80 - 10}{\ln\left(\frac{T_S - 10}{T_S - 80}\right)}$$

$$\ln\left(\frac{T_S - 10}{T_S - 80}\right) = \frac{70}{58.6}$$

$$\ln\left(\frac{T_S - 10}{T_S - 80}\right) = 1.194$$

By taking antilogs of both sides of the equation:

$$\frac{T_S - 10}{T_S - 80} = e^{1.194}$$

$$\frac{T_S - 10}{T_S - 80} = 3.3$$

$$T_S - 10 = 3.3(T_S - 80)$$

$$2.3\,T_S = 264 - 10$$

Steam temperature (T_S) = 110°C

This saturation temperature is equivalent to a steam pressure of 0.45 bar g. This pressure is less than the 0.5 bar g backpressure, and the system will permanently stall.

In this case, if a ball float steam trap were fitted, condensate would permanently flood the heat exchanger, its level modulating relative to changes in load. Operating performance might be unsatisfactory as the secondary outlet temperature will tend to fluctuate, and the heat exchanger might fail prematurely due to corrosion.

If the system is permanently running under stall conditions, a ball float steam trap is the wrong choice for this application, and a pump-trap should be fitted instead.

Consider supplier 'Y'

For the manufacturer to size the heating area that best matches the design condition, it is necessary to find the minimum heating area that will satisfy the operating full-load. It is first necessary to determine the rated LMTD for the heat exchanger with a steam space pressure is 4 bar g (T_S = 152°C).

From Equation 2.5.5:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)}$$ **Equation 2.5.5**

Where:
ΔT_{LM} = LMTD
T_1 = 10°C
T_2 = 80°C
T_S = 152°C

$$\Delta T_{LM} = \frac{80 - 10}{\ln\left(\frac{152 - 10}{152 - 80}\right)}$$

$$\Delta T_{LM} = \frac{70}{\ln\left(\frac{142}{72}\right)}$$

$$\Delta T_{LM} = \frac{70}{\ln\ 1.972}$$

$$\mathbf{\Delta T_{LM} = \frac{70}{0.679} = 103°C}$$

By using Equation 13.2.1 the minimum heating area can now be determined for the rating of 293 kW.

$$A = \frac{\dot{Q}}{U\ \Delta T_M}$$ **Equation 13.2.1**

Where:
A = Heating area (m²)
$\dot{Q}$ = Heat transfer rate (kW)
T_2 = Heat transfer coefficient (W/m² °C)
ΔT_M = ΔT_{LM}

$$A = \frac{293\,000}{2\,500\ x\ 103}$$

Heating area (A) = 1.138 m²

From his standard range, supplier 'Y' can provide a plate heat exchanger that meets the specification with a heating area of 1.198 m². This is oversized (by about 5%) and steam pressure will therefore be less than 4 bar g at the full-load operating condition.

In practice, heat exchangers are likely to be specified at least 10% over capacity. It is for this reason that the operating steam pressure (not the quoted normal working pressure) should always be established before selecting and sizing the steam trapping device. The reputable manufacturer should be willing to supply this information, or, at least, the heating area, the 'U' value, and the heat output. From this data, the rated LMTD can be calculated, from which the operating pressure can be found.

Find the LMTD for the heat exchanger with a heating area of 1.198 m²:

$$A = \frac{\dot{Q}}{U \Delta T_M}$$ **Equation 13.2.1**

$$\Delta T_M = \frac{\dot{Q}}{U A}$$

$$\Delta T_M = \frac{293\,000 \text{ W}}{2\,500 \text{ W/m}^2\,°\text{C} \times 1.198 \text{ m}^2}$$

$$\mathbf{\Delta T_M = 97.8°C}$$

The operating steam temperature at full-load can now be found by use of Equation 2.5.5:

$$\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\frac{T_S - T_1}{T_S - T_2}\right)}$$ **Equation 2.5.5**

Where:
ΔT_{LM} = 97.8°C
T_1 = 10°C
T_2 = 80°C
T_S = Steam temperature°C

$$97.8 = \frac{80 - 10}{\ln\left(\frac{T_S - 10}{T_S - 80}\right)}$$

$$\ln\left(\frac{T_S - 10}{T_S - 80}\right) = \frac{70}{97.8}$$

$$\ln\left(\frac{T_S - 10}{T_S - 80}\right) = 0.715$$

By taking antilogs of both sides of the equation:

$$\frac{T_S - 10}{T_S - 80} = e^{0.715}$$

$$\frac{T_S - 10}{T_S - 80} = 2.045$$

$$T_S - 10 = 2.045\ (T_S - 80)$$

Steam temperature (T_S) = 147°C

This saturation temperature is equivalent to a steam pressure of 3.4 bar g at the design condition. As this pressure is more than the constant 0.5 bar g backpressure, the system will not stall at full-load.

What is the steam flowrate ($\dot{m}_s$) at full-load?
The steam mass flowrate will depend upon the steam space pressure, which is 3.4 bar g at full-load, with an enthalpy of evaporation of 2 122 kJ/kg.

From Equation. 2.8.1:

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW x 3 600}}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\dot{m}_S = \frac{293 \times 3\,600}{2\,122} \text{ kg/h}$$

Steam flowrate ($\dot{m}_S$) = 498 kg/h at full-load

What is the TDC?
It is now necessary to find the heat load at which the system will stall. In order to do so, it is necessary to calculate the TDC for this heat exchanger from the design conditions.

From Equation 13.2.2:

$$TDC = \frac{T_S - T_1}{T_S - T_2} \qquad \textbf{Equation 13.2.2}$$

Where:
TDC = Temperature Design Constant
T_1 = 10°C
T_2 = 80°C
T_S = 147°C

$$TDC = \left(\frac{147 - 10}{147 - 80} \right)$$

$$\textbf{TDC} = \frac{\textbf{137}}{\textbf{67}} = \textbf{2.045}$$

The stall condition
At stall, the pressure in the steam space will equal the 0.5 bar g backpressure.

The saturation temperature of steam at 0.5 bar g is 111.6°C.

From Equation 13.2.4 the inlet temperature can be found:

$$T_1 = T_s - [\ TDC\ (T_s - T_2)\ ] \qquad \textbf{Equation 13.2.4}$$

Where:
T_1 = Inlet temperature °C
T_2 = 80°C
T_S = 111.6°C
TDC = 2.045

$$T_1 = 111.6 - [2.045 \times (111.6 - 80)\ ]$$

$$T_1 = 111.6 - 64.6$$

T_1 = 47°C at stall

What is the heat load at stall?
From the heat transfer flowrate equation (Equation 2.6.5):

$$\dot{Q} = \dot{m}\ C_p\ \Delta T \qquad \textbf{Equation 2.6.5}$$

$$\dot{Q} = 1.0\ kg/s\ \times\ 4.19\ kJ/kg\ °C\ \times\ (80 - 47)°C$$

Heat transfer rate ($\dot{Q}$) = 138 kW (138 000 W)

As full-load is 293 kW, the percentage stall load $= \frac{138}{293} \times 100 = 48\%$ of full-load.

The selection of the trapping device will depend on whether the minimum heat load is higher or lower than the stall load.

The minimum load is quoted as being 60% of the full-load of 293 kW, therefore:

Minimum load $= 0.6 \times 293\ kW = 176\ kW$

Stall load $= 138\ kW$

As the minimum load is greater than the stall load, the system will never stall. It is therefore practical to fit a ball float steam trap, as there will always be a positive differential pressure across it.

However, the ball float steam trap has to be sized to carry both the full-load and the minimum load, and it is therefore necessary to calculate the steam flows and the corresponding steam space pressures at both conditions.

It is first necessary to calculate the secondary inlet temperature at the minimum load. This can be predicted by use of Equation 13.4.1:

$$T_x = [(T_2 - T_1)\ (1 - x)] + T_1 \qquad \textbf{Equation 13.4.1}$$

Where:
T_x = The secondary inlet temperature at any load factor 'x'
T_1 = The secondary inlet temperature at full-load
T_2 = The secondary outlet temperature at full-load
x = The load factor.

For example; the minimum heat load of 60% is equivalent to a load factor of 0.6

$$T_{(0.6)} = [(T_2 - T_1)\ (1 - x)] + T_1$$

$$T_{(0.6)} = [(80 - 10)\ (1 - 0.6)] + 10$$

$$T_{(0.6)} = [70 \times 0.4] + 10$$

$$\mathbf{T_{(0.6)} = 38°C}$$

The minimum load condition

From Equation 13.2.3:

$$T_S = \frac{(T_2 \times TDC) - T_1}{TDC - 1} \qquad \textbf{Equation 13.2.3}$$

Where:

TDC = 2.045

T_2 = 80°C

T_1 = 38°C

T_S = Steam temperature °C

$$T_S = \frac{(80 \times 2.045) - 38}{2.045 - 1}$$

$$T_S = \frac{125.6}{1.045}$$

Steam temperature (T_S) = 120°C at minimum load

This is the steam temperature at the minimum load of 176 kW, and is equivalent to a steam pressure of 1.0 bar g. The condensate pressure is 0.5 bar g. The differential pressure across the ball float steam trap at minimum load therefore equals 1.0 bar g - 0.5 bar g = 0.5 bar.

What is the steam flowrate ($\dot{m}_{S(min)}$) at the minimum heat load of 176 kW?

The minimum steam flowrate will depend upon the steam space pressure, which is 1.0 bar g with an enthalpy of evaporation of 2 201.1 kJ/kg.

From Equation 2.8.1:

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW} \times 3\,600}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\dot{m}_S = \frac{176 \times 3\,600}{2\,201.1} \text{ kg/h}$$

Steam flowrate ($\dot{m}_S$) = 288 kg/h at the minimum heat load of 176 kW

As it has been established that this system will not stall, a ball float steam trap is suitable. It is now necessary to size a ball float steam trap for operation up to the maximum system differential pressure of 3.5 bar and pass . . .

a) the full-load of 498 kg/h with a differential pressure of 3.4 bar g - 0.5 bar g = 2.9 bar g.

b) the minimum load of 288 kg/h with a differential pressure of 1.0 bar g - 0.5 bar g = 0.5 bar g.

It can be seen from the ball float steam trap sizing chart (Figure 13.4.1) that a DN25 (1") FT14-4.5 will satisfy both of these conditions, and could be selected. However, if the minimum heat load were less than the stall load, then a pump-trap would have to be selected.

The methods of selecting trapping devices are further discussed in Module 13.8, 'Practical methods of preventing stall'.

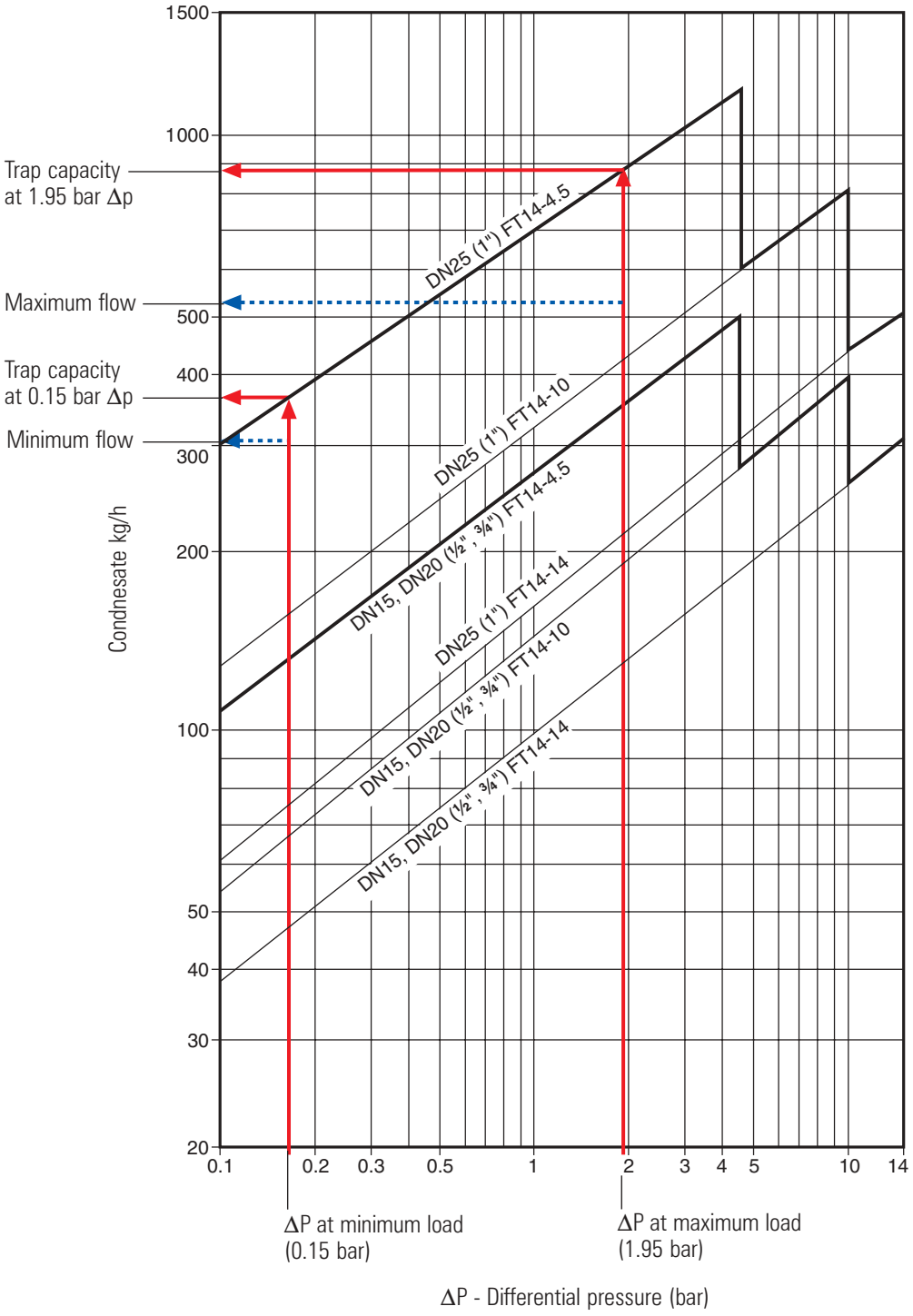


Fig. 13.4.1 FT14 ball float sizing chart showing data for Example 13.4.1

Questions

1. If the heat exchanger from supplier 'X' had a heating area of 1 m² instead of 2 m², what would have been the LMTD for the same secondary full-load conditions?

a| 33.6°C ☐

b| 29.3°C ☐

c| 117.2°C ☐

d| 107°C ☐

2. If the heat exchanger from supplier 'X' had a heating area of 1 m² instead of 2 m², what would have been the steam temperature at the full heat load of 293 kW?

a| 55°C ☐

b| 100°C ☐

c| 117.2°C ☐

d| 165.7°C ☐

3. If the heat exchanger from supplier 'X' had a heating area of 1 m² instead of 2 m², would the system still stall at the full heat load of 293 kW?

a| Yes, because the backpressure is higher than the steam pressure ☐

b| No, because the full-load pressure is 6.1 bar g ☐

c| No, because there is more heating area in the oversized heat exchanger ☐

d| The steam pressure is less, consequently the steam will be drier ☐

4. What is the effect of higher backpressure on a ball float steam trap?

a| None whatever ☐

b| It reduces the steam pressure in the heat exchanger ☐

c| It reduces the ball float steam trap capacity ☐

d| It increases the differential pressure across it ☐

5. What can be done to stop any heat exchanger waterlogging?

a| Increase the pressure upstream of the steam control valve ☐

b| Ensure the condensate discharges to atmospheric pressure ☐

c| Calculate the stall point and fit the correct trapping device ☐

d| Increase the size of the steam trap pipework ☐

6. Steam pressure in a heat exchanger at minimum load is 0.8 bar a. What is the differential pressure across the ball float steam trap if the backpressure is 0.1 bar g?

a| +0.7 bar ☐

b| -0.7 bar ☐

c| +0.9 bar ☐

d| -0.3 bar ☐

Answers

1: d, 2: a, 3: b, 4: c, 5: c, 6: d

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Module 13.5

The Stall Chart
- Constant Flow Secondary
- Varying Inlet Temperature
- Constant Outlet Temperature

The Stall Chart

By definition, stall will occur when the steam pressure in the heat exchanger is less than or equal to the condensate backpressure.

Good results are obtained from heat transfer calculations as shown in Module 13.4. Those not wishing to use a mathematical approach can use a simpler method to arrive at a practical result. This method is graphical and involves the use of a 'stall chart'. It gives slightly less accurate results, but is perfectly adequate for most practical purposes.

A reduction in heat load is usually due to an increasing inlet temperature or a reducing secondary fluid flowrate, and requires a fall in steam pressure for control to be maintained. Sometimes stall may be caused by a combination of these, or perhaps a fall in outlet temperature due to a change in the set point.

Constant secondary flowrate with varying inlet temperature

In this type of heat exchanger, the secondary flowrate and outlet temperature remain constant while the inlet temperature varies with changes in heat load.

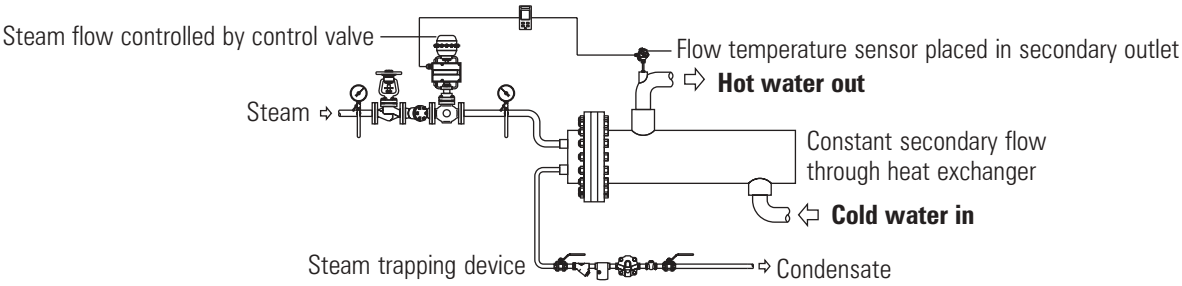


Fig. 13.5.1 Shell and tube heat exchanger with primary control valve

At full-load the inlet temperature will be at its lowest. With a constant secondary flow through the heat exchanger, any reduction in the heat load will cause the inlet temperature to rise. The stall chart can show how the steam temperature and the inlet temperature change as the heat load changes, and predict the inlet temperature at stall and the minimum load condition.

Under full-load conditions, the temperature difference between the steam and secondary fluid will be large. Conversely, under no-load conditions there is no heat exchange so the steam and secondary fluid must be the same temperature, and the temperature difference between them is zero. By proportionality, it follows that at 50% load this temperature difference is 50% of its maximum value.

From this basic principle of proportionality, two straight lines can be drawn onto a chart to represent all these conditions. At full-load the lines are furthest apart, showing that the temperature difference is at a maximum. At no-load the lines converge to a single point, showing that the temperature difference is zero.

A typical stall chart is shown in Figure 13.5.2. It considers a steam temperature of 120°C heating a constant flow of secondary water from 20°C to 80°C. Note that the steam temperature of 120°C is arrived at by one of two means:

- It has been calculated from LMTD design figures, as per the calculations in Module 13.4, which take into consideration the heat exchange surface area.
- The steam space pressure has been observed during operation and the temperature calculated.

Firstly, the steam temperature in the heat exchanger under full-load conditions (Point **A**) is marked upon the left vertical axis in the stall chart in Figure 13.5.2. Secondly, the desired secondary fluid outlet temperature is marked on the right vertical axis (Point **B**). The secondary fluid inlet temperature (Point **C**) at full-load is then marked on the left vertical axis.

If a straight line then joins the points **A** and **B**, the line **AB** will represent how the steam temperature alters relative to changes in heat load. Similarly, if a straight line joins the points **B** and **C**, the line **BC** will represent the changing inlet temperature of the secondary fluid as the heat load varies.

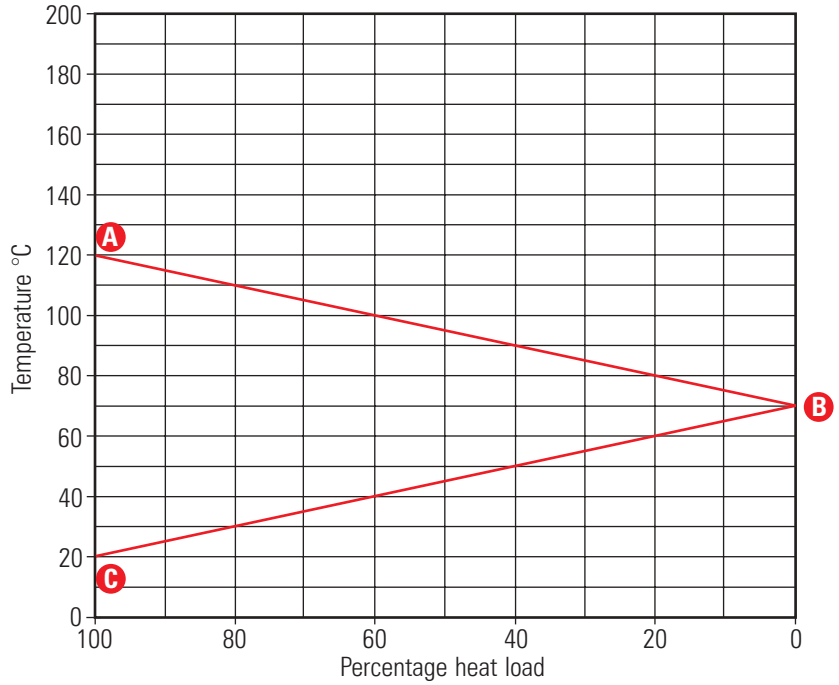


Fig. 13.5.2 Constant flowrate / Varying inlet temperature - Stage 1

It is then necessary to add a horizontal line to represent the equivalent steam saturation temperature of the condensate backpressure. This temperature should be marked on the right vertical axis, as shown in the Figure 13.5.3 (Point **D**). A straight line should then be drawn in to connect this point with the same temperature on the left vertical axis at point **E**.

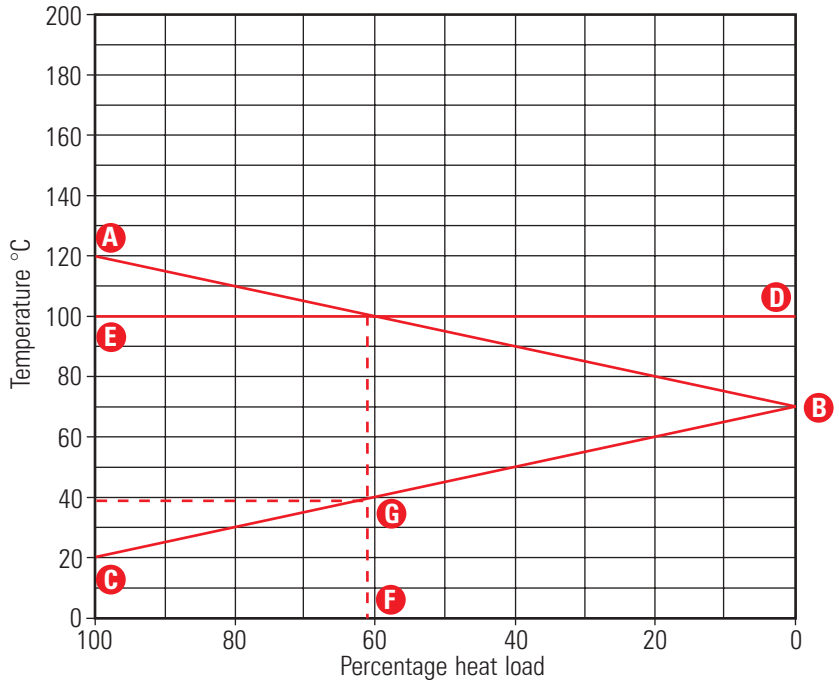


Fig. 13.5.3 Constant flowrate / Varying inlet temperature - Stage 2

The condensate backpressure takes into account the pressure in the condensate system plus any static pressure that may be due to a lift in the condensate discharge line from the bottom of the heat exchanger. A column of liquid will exert a pressure at its base due to its own mass. This is often referred to as 'static lift' when it is exerted on the outlet of the trap.

A 1 metre column of water under atmospheric pressure will exert a pressure at the bottom of the column of approximately 10 kPa or 0.1 bar g (actually 9.806 65 kPa or 0.098 066 5 bar). Any lift in the condensate discharge line will thus exert a static lift due, to the column of condensate held in the line, in addition to any pressure in the condensate system.

The horizontal line **DE** will either intersect the line **AB**, or will be above point **A** on the chart. The point of intersection between the lines **AB** and **DE** represents the 'stall point', where the steam pressure and the backpressure are the same. If the line **DE** is on or above point **A**, the system permanently operates under stall conditions. (In vacuum condensate systems, or when **B** is greater than 100°C, point **D** may also be below point **B**, if this is the case, the system will not stall at any heat load). A vertical line should then be dropped down from the stall point. The point at which this vertical line crosses the bottom horizontal axis (Point **F**) marks the percentage stall load relative to the full heat load. The percentage stall load can also be quickly calculated using Equation 13.5.1.

$$\% \text{ Stall load} = \frac{D - B}{A - B} \times 100 \qquad \textbf{Equation 13.5.1}$$

Where:
A = The steam temperature in the steam space at full-load
B = The secondary fluid outlet temperature
D = The backpressure equivalent saturated steam temperature

The vertical line connecting the stall point with point **F** will also intersect the line **BC**. If a horizontal line is drawn from this intersection point to the left vertical axis, this will mark the secondary inlet temperature at which stall occurs (Point **G**).

Example 13.5.1
The steam pressure in a heat exchanger at full-load is observed to be 7 bar g. Condensate pressure is 1 bar g, and there is a lift after the trap of 10 m. At full-load, the secondary fluid enters the heat exchanger at 25°C and leaves the heat exchanger at 80°C.

1. What is the percentage heat load at stall?
2. What is the secondary inlet temperature at stall?

The saturation temperature of saturated steam at 7 bar g is 170°C. Therefore the steam temperature in the heat exchanger at full-load is 170°C. This can then be plotted as point **A** in Figure 13.5.4:

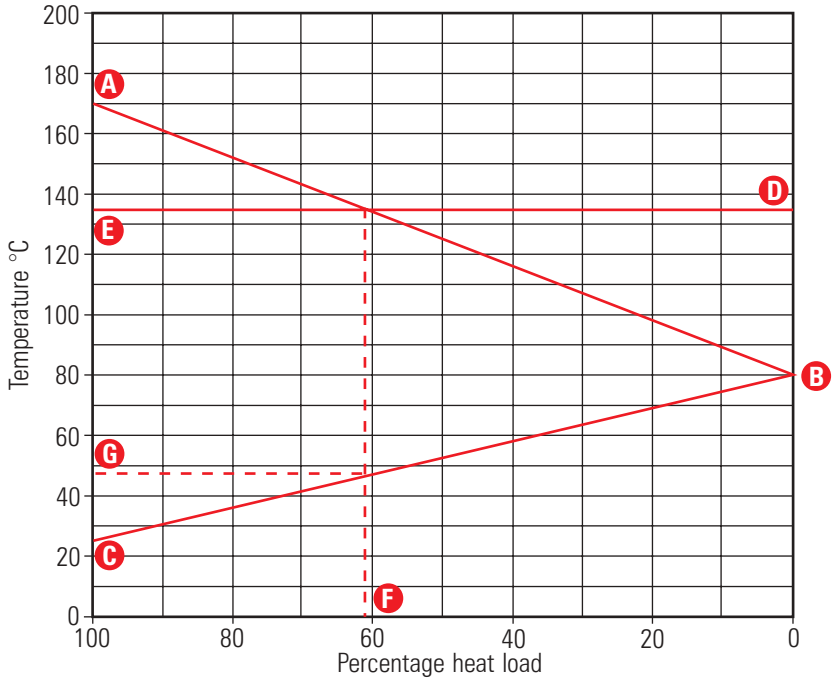


Fig. 13.5.4 Stall chart for Example 13.5.1

1. What is the percentage heat load at stall?

The secondary fluid outlet temperature of 80°C should be plotted as point **B** in Figure 13.5.4, while the secondary fluid inlet temperature at full-load of 25°C should be plotted as point **C**.

The lift in the condensate line of 10 m creates a backpressure of 1 bar, in addition to the 1 bar g pressure in the condensate system. Therefore, the total system backpressure is 2 bar g. As the saturation temperature of steam at 2 bar g is 135°C, the horizontal line **DE** representing the backpressure is added at this temperature.

The stall chart in Figure 13.5.4 shows that the percentage heat load at stall (Point **F**) is approximately 61%. The mathematical calculation can be validated by use of Equation 13.5.1:

$$\% \text{ Stall load} = \frac{D - B}{A - B} \times 100 \qquad \textbf{Equation 13.5.1}$$

Where:

A = The steam temperature in the steam space at full-load = 170°C

B = The secondary fluid outlet temperature = 80°C

D = The backpressure equivalent saturated steam temperature = 135°C

$$\% \text{ Stall load} = \frac{D - B}{A - B} \times 100$$

$$\% \text{ Stall load} = \frac{135 - 80}{170 - 80} \times 100$$

$$\% \text{ Stall load} = \frac{55}{90} \times 100$$

$$\% \text{ Stall load} = 61\%$$

2. What is the secondary inlet temperature at stall?

The stall chart in Figure 13.5.4 also indicates that the inlet temperature at stall (Point **G**) is about 46°C or 47°C. The mathematical calculation can be validated by use of Equation 13.4.1:

$$T_X = [(T_2 - T_1)\ (1 - x)] + T_1 \qquad \textbf{Equation 13.4.1}$$

Where:

T_x = The secondary inlet temperature at any load factor 'x'

T_1 = The secondary inlet temperature at full-load

T_2 = The secondary outlet temperature at full-load

x = The load factor.

For example; the minimum heat load of 61% load is equivalent to a load factor of 0.61

$$T_{(0.61)} = [(T_2 - T_1) \times (1 - x)] + T_1$$

$$T_{(0.61)} = [(80 - 25) \times (1 - 0.61)] + 25$$

$$T_{(0.61)} = [(55) \times (0.39)] + 25$$

$$\mathbf{T_{(0.61)} = 46.5°C}$$

Questions

1. What causes stall?

a| The steam pressure is more than the condensate pressure ☐

b| The condensate pressure less than the steam pressure ☐

c| The condensate pressure is more than or equal to the steam pressure ☐

d| The steam trap is too small ☐

2. What is the relationship between the steam and secondary fluid inlet temperature at full-load?

a| The difference is large ☐

b| The difference is small ☐

c| The difference is zero ☐

d| The secondary inlet temperature is higher than the steam temperature ☐

3. What is the temperature difference between the steam and secondary fluid inlet temperature at 75% load?

a| It is 75% of the temperature difference at full-load ☐

b| It is 25% of the temperature difference at full-load ☐

c| It is at a minimum ☐

d| It is exactly the same ☐

**4. Figure 13.5.4 shows the backpressure at 2 bar g.
What would the stall load be if the condensate pressure was atmospheric?**

a| 10% ☐

b| 22% ☐

c| 30% ☐

d| 80% ☐

5. Also, what would be the secondary inlet temperature?

a| 25°C ☐

b| 45°C ☐

c| 55°C ☐

d| 68°C ☐

6. If, at full-load, the steam pressure were 1 bar g (120°C) instead of 7 bar g, what would be the approximate stall load for an atmospheric backpressure?

a| 20% ☐

b| 30% ☐

c| 40% ☐

d| 50% ☐

Answers

1: c, 2: a, 3: a, 4: b, 5: d, 6: d

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Module 13.6

The Stall Chart
- Varying Flow Secondary
- Constant Inlet Temperature
- Constant Outlet Temperature

The Stall Chart

Varying flowrate with constant inlet/outlet temperature

Not all heat exchangers are required to operate with a constant secondary flow. Typical applications might include the provision of hot water to batch processes such as tanks and vats. The supply of hot water to each tank is controlled either by means of an on-off ball valve or a modulating globe valve; there is no recirculation of water back to the heat exchanger. Cold make-up water is simply heated as per the hot water demand, as depicted in Figure 13.6.1. A modulating control valve on the steam supply to the heat exchanger regulates the temperature of the hot water being drawn off. The cold make-up water might be supplied from a pressurised main, and its temperature may alter seasonally. Its lowest probable temperature should be considered when considering the stall condition.

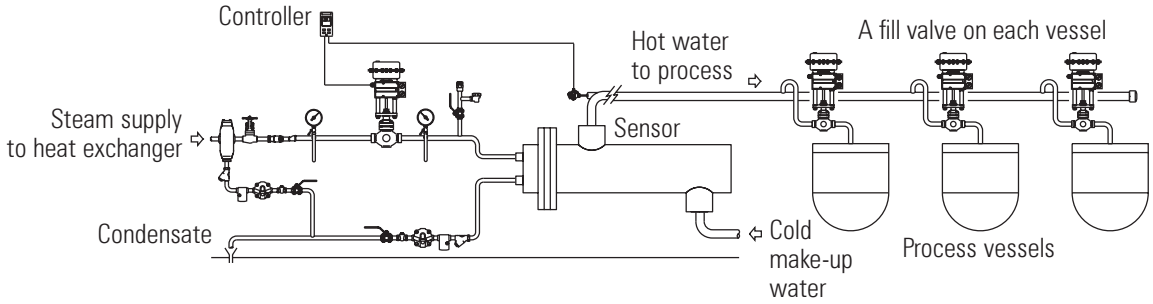


Fig. 13.6.1 Shell and tube heat exchanger with varying secondary flow

The stall chart can also be used in these types of installations, but the construction method for shell and tube heat exchangers is slightly different to that used for constant secondary flow. This method is described below.

The first part of this method is very similar to that shown in Example 13.5.1. With reference to Figure 13.6.2, the steam temperature in the heat exchanger under full-load conditions (Point **A**) should be marked on the left vertical axis. The desired secondary fluid outlet temperature should then be marked on the right vertical axis (Point **B**).

The secondary fluid inlet temperature (Point **C**) should also be marked on the left vertical axis. The horizontal line representing the system backpressure must also be marked on this chart. This temperature should be marked on the right vertical axis at point **D**, with a straight line connecting it to the same temperature on the left vertical axis at point **E**.

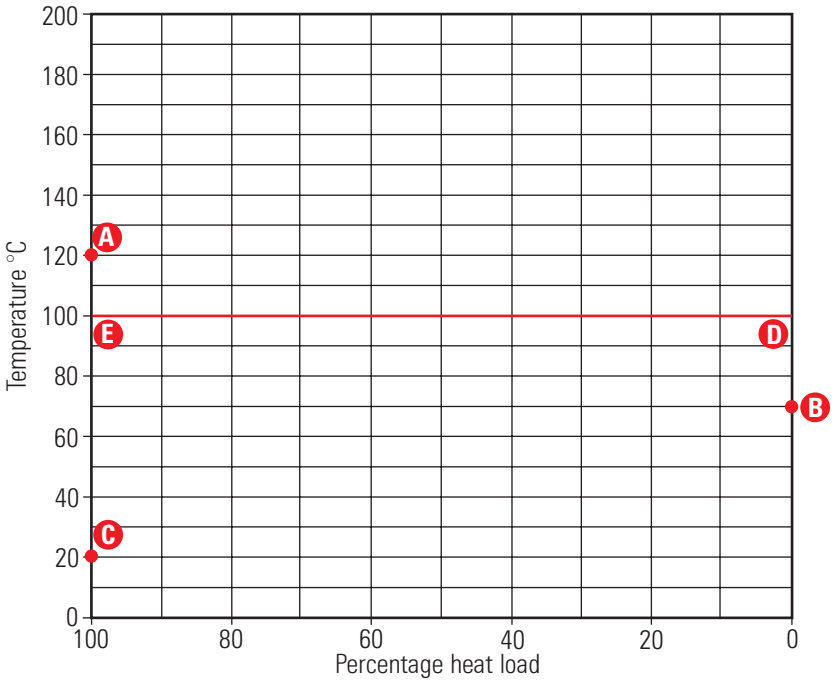


Fig. 13.6.2 Varying flowrate / Constant inlet temperature - Stage 1

With reference to Figure 13.6.3, the secondary load line **BC** should be drawn connecting points **B** and **C**. A horizontal line should then be drawn from where **BC** crosses the 50% load ordinate, to the right axis. This represents the mean secondary fluid temperature, and is shown as point **F**.

The mean secondary fluid temperature point **F** should then be connected by a diagonal straight line to the steam temperature point **A** in the heat exchanger under full-load, creating the line **AF**.

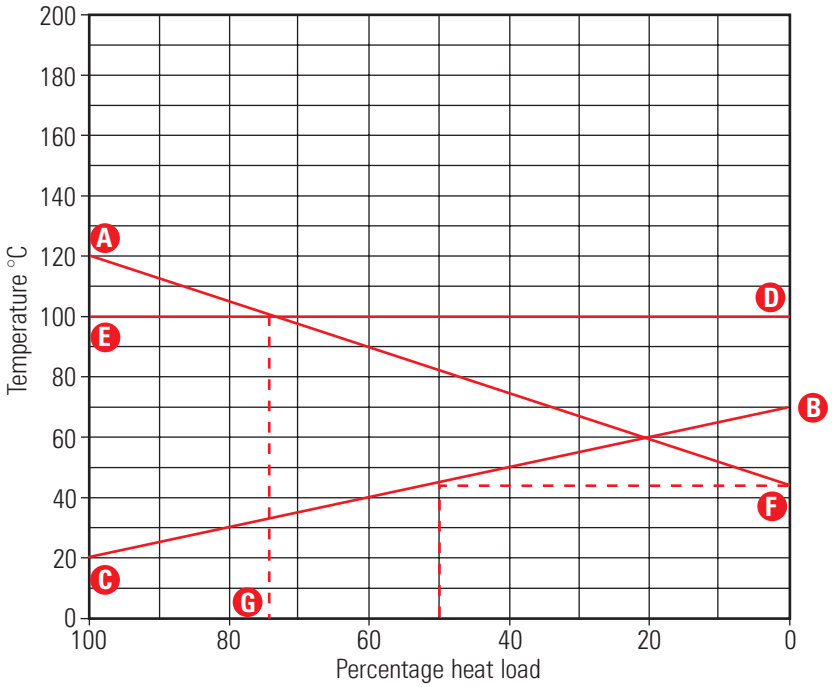


Fig. 13.6.3 Varying flowrate / constant inlet temperature - stage II

The backpressure line **DE** will either intersect the steam line **AF,** or be above point **A** on the chart. The point of intersection between the lines **AF** and **DE** marks the stall point, where the steam pressure and the backpressure are the same. A vertical line may be dropped down from the stall point, to indicate when the stall condition occurs.

The point at which this vertical line crosses the bottom horizontal axis (Point **G**) should mark the percentage load. As in the previous example, if the line **DE** is above the point **A**, stall occurs under all load conditions.

The percentage stall load can also be calculated using Equation 13.6.1:

$$\% \text{ Stall load} = \frac{D - \left(\frac{B + C}{2}\right)}{A - \left(\frac{B + C}{2}\right)} \times 100$$

Equation 13.6.1

Where:

A = Steam temperature at full-load

B = Secondary fluid outlet temperature at full-load

C = Secondary fluid inlet temperature at full-load

D = Equivalent backpressure steam temperature

The minimum steam temperature

It should be noted that the lowest operating steam temperature equals the set point temperature at point **B**. This occurs at 70°C in the stall chart, Figure 13.6.4, and is represented by point **H** on the steam line **AF**.

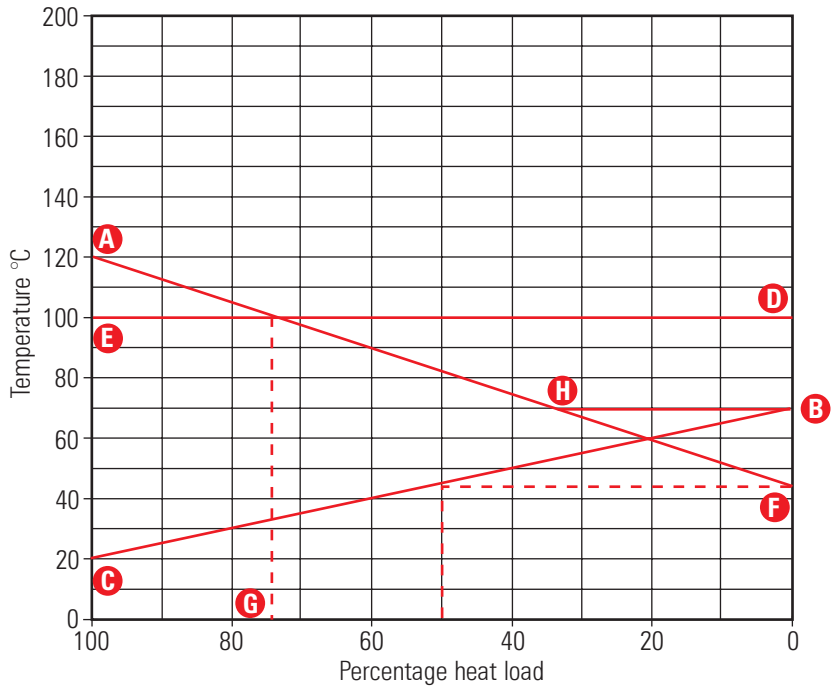


Fig. 13.6.4 Minimum steam temperature equals the set point

In practice, as the heat load decreases, and the steam temperature approaches the secondary control temperature at point **H**, changes in steam temperature occur slowly rather than the rapid step change suggested at point **H** in Figure 13.6.4. The steam temperature will tend to fall in a similar way to that shown in Figure 13.6.5. It is difficult and unnecessary to draw this line on a stall chart, whereas Figure 13.6.4 is practical and easy to use.

Referring to Figure 13.6.4, it can be seen in this example that the steam temperature at any load less than 37% is 70°C. In truth, the gradual fall in steam temperature is more like that depicted in Figure 13.6.5, but the difference is so small as to be insignificant with regard to selecting and sizing the trapping device.

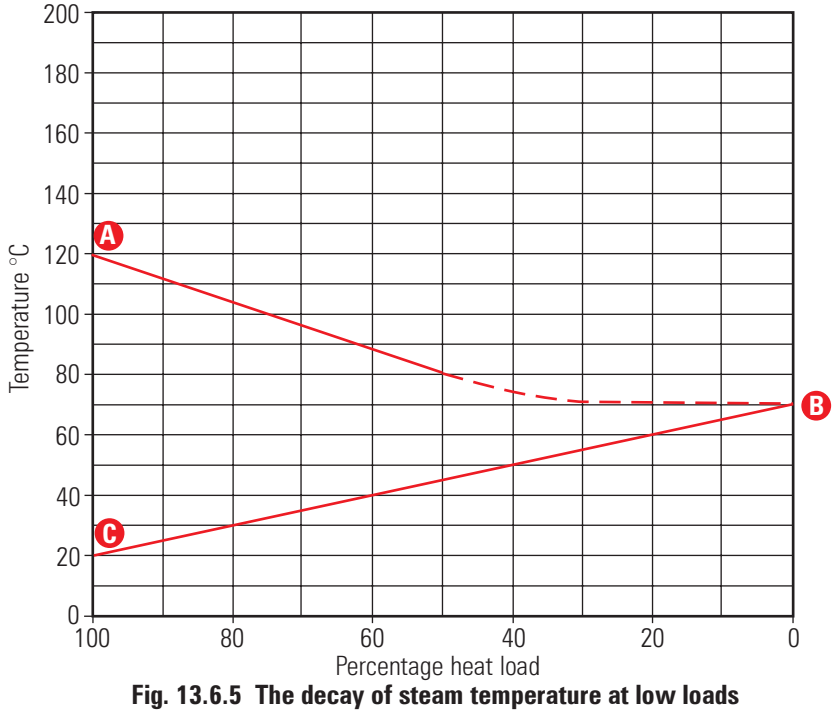


Fig. 13.6.5 The decay of steam temperature at low loads

Example 13.6.1

The steam pressure inside a shell and tube heat exchanger with a varying secondary flowrate at full-load is 8 bar g, the pressure in the condensate line is 0.5 bar g, and there is a lift of 7 metres after the trap. At full-load, the secondary fluid enters the heat exchanger at 30°C and leaves the heat exchanger at 90°C with a flowrate of 3.64 L/s.

What is the percentage load at stall, and what is the secondary flowrate through the heat exchanger at stall?

The saturation temperature of the steam at 8 bar g is 175°C. Therefore the steam temperature in the heat exchanger at full-load is 175°C. This should then be plotted as point **A** in Figure 13.6.6.

The secondary fluid outlet temperature of 90°C should be plotted as point **B**, while the secondary fluid inlet temperature of 30°C should be plotted as point **C**.

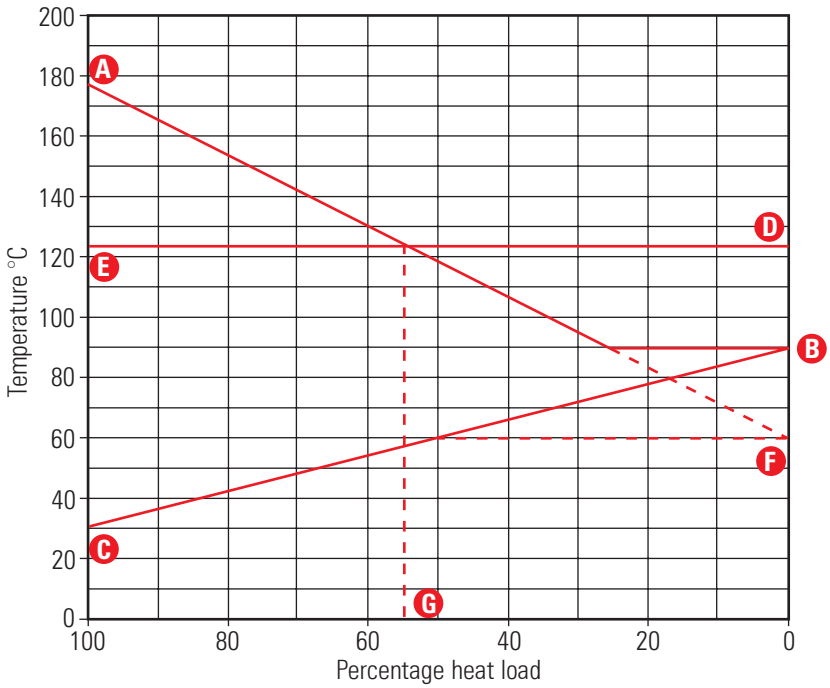


Fig. 13.6.6 Stall chart for varying flow / constant temperature

The lift in the condensate line of 7 m creates a differential pressure of 0.7 bar, in addition to the 0.5 bar g pressure in the condensate line. Therefore, the total system backpressure is 1.2 bar g. As the saturation temperature of steam at 1.2 bar g is 123°C, the horizontal line **DE** representing the backpressure is drawn at this temperature in Figure 13.6.6.

In this example the percentage load (Point **G**) is approximately 55%. This means that the secondary liquid flowrate must reduce to 55% of the maximum flowrate for stall to occur, that is, 55% of 3.64 L/s = 2 L/s. This can be verified mathmatically by using Equation 13.6.1.

$$\% \text{ Stall load} = \frac{D - \left(\frac{B + C}{2}\right)}{A - \left(\frac{B + C}{2}\right)} \times 100$$ **Equation 13.6.1**

Where:

A = Steam temperature at full-load = 175°C

B = Secondary fluid outlet temperature at full-load = 90°C

C = Secondary fluid inlet temperature at full-load = 30°C

D = Equivalent backpressure steam temperature = 123°C

$$\% \text{ Stall load} = \frac{123 - \left(\frac{90 + 30}{2}\right)}{175 - \left(\frac{90 + 30}{2}\right)} \times 100$$

$$\% \text{ Stall load} = \frac{123 - 60}{175 - 60} \times 100$$

$$\% \text{ Stall load} = \frac{63}{115} \times 100$$

% Stall load = 55%

Most heat exchanger applications will either be varying flowrate or varying temperature as described above and in the previous Modules in Block 13.

There may, however, also be instances where both the flowrate and the inlet temperature of the secondary fluid vary. In these examples it becomes more difficult to determine their combined effect by interpretation of the stall chart. Systems such as these can be analysed by comparing the results from both methods shown above and using the worst case.

Questions

1. What is the difference between the constant and variable stall charts?

a | Nothing ☐
b | The steam line is constructed differently ☐
c | The backpressure line is at different pressures ☐
d | The secondary line is constructed differently ☐

2. If the backpressure line is higher than Point A on the steam line what does this mean?

a | The system will never stall ☐
b | The system will constantly stall ☐
c | The system is constantly in vacuum ☐
d | The heat exchanger is too big ☐

3. If the backpressure line is lower than Point B the steam line what does this mean?

a | The system will never stall ☐
b | The heat exchanger is too small ☐
c | The system is constantly in vacuum ☐
d | The system will constantly stall ☐

4. If, in Example 13.6.1, the condensate backpressure were atmospheric, at what percentage load would stall have occurred?

a | 18% ☐
b | 28% ☐
c | 35% ☐
d | 55% ☐

Answers

1: b, 2: b, 3: a, 4: c

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Module 13.7

The Stall Chart
- Constant Flow Secondary
- Constant Inlet Temperature
- Varying Outlet Temperature

The Stall Chart - Constant Flowrate / Varying Outlet Temperature

All systems discussed up to this point assume that the secondary fluid outlet temperature remains constant. In some applications, the outlet temperature may change with time. This will also change the heat load and affect the stall point.

Such changes often occur in process applications, and also heating calorifiers that change their outlet temperature to compensate for changes in ambient conditions.

If the highest heat requirement occurs when the control temperature (the set point) is at a maximum, any reduction in the set point will cause a reduction in the heat load.

A reducing set point will tend to increase the stall load, as shown in the following calculations. Once the design conditions are known, the effect of reducing the set point can either be calculated mathematically as shown below or illustrated on a stall chart by means of proportionality.

Example 13.7.1

Initially, secondary water at a rate of 1.5 L/s enters a heat exchanger at 20°C and leaves at 70°C. It is observed via a pressure gauge on the steam inlet that the pressure in the steam space under these conditions is 5.2 bar g (T_S = 160°C). The condensate drains down to a vented receiver in a plant room below the installation. ($T_{(back)}$ = 100°C).

If the set point is reduced to 60°C, what is the effect on the stall point and the steam load at stall?

Calculating the effect of reducing the set point arithmetically

It is first necessary to establish the heat exchanger TDC from the full-load operating conditions and by use of Equation 13.2.2:

$$TDC = \frac{T_s - T_1}{T_s - T_2} \qquad \textbf{Equation 13.2.2}$$

The full-load conditions are:
T_1 = 20°C
T_2 = 70°C
T_S = 160 °C (steam temperature at 5.2 bar g)

Therefore:

$$TDC = \frac{160 - 20}{160 - 70}$$

$$TDC = \frac{140}{90}$$

$$TDC = \frac{142}{92}$$

$$\mathbf{TDC = 1.555}$$

How does the stall load change with a lowered set point?

Firstly, consider the stall load with the higher set point of 70°C

The design conditions are:
T_1 = 20°C
T_2 = 70°C
T_S = 160°C
$T_{(back)}$ = 100°C

For a constant secondary flowrate, the stall factor can be calculated from Equation 13.5.1:

$$\% \text{ Stall load} = \frac{D - B}{A - B} \times 100 \qquad \textbf{Equation 13.5.1}$$

Where:
A = The steam temperature at full-load with a 70°C set point (T_S)
B = The secondary fluid outlet temperature (T_2)
D = The backpressure equivalent saturated steam temperature ($T_{(back)}$)

$$\% \text{ Stall load} = \frac{D - B}{A - B} \times 100$$

$$\% \text{ Stall load} = \frac{100 - 70}{160 - 70} \times 100$$

$$\% \text{ Stall load} = \frac{30}{90} \times 100$$

$$\textbf{\% Stall load = 33.33 = Stall factor of 0.333 3}$$

With the set point at 70°C

$$\text{Full heat load } (\dot{Q}) = \dot{m}\, Cp\, \Delta T \text{ (kW)}$$

$$\text{Full heat load } (\dot{Q}) = 1.5 \text{ kg/s} \times 4.19 \text{ kJ/kg°C} \times (70 - 20)\text{°C}$$

$$\textbf{Full heat load } (\dot{Q}) = \textbf{314 kW}$$

$$\text{Heat load at stall} = 0.3333 \times 314 \text{ kW}$$

$$\textbf{Heat load at stall} = \textbf{105 kW}$$

The condensate discharges to atmosphere, and the h_{fg} at atmospheric pressure is 2 257 kJ/kg.

$$\text{Steam load at stall} = \frac{105 \text{ kW} \times 3\,600 \text{ s/h}}{2\,257 \text{ kJ/kg}}$$

$$\textbf{Steam load at stall = 168 kg/h with the set point at 70°C}$$

Secondly, consider the stall load with the lower set point of 60°C

The steam temperature can be predicted for any load by use of Equation 13.2.3:

$$T_s = \frac{(T_2 \times TDC) - T_1}{TDC - 1} \qquad \textbf{Equation 13.2.3}$$

Where:
T_S = Steam temperature (°C)
T_2 = Secondary outlet temperature = 60°C
TDC = Temperature Design Constant = 1.555
T_1 = Secondary inlet temperature = 20°C

$$T_s = \frac{(60 \times 1.555) - 20}{1.555 - 1}$$

$$T_s = \frac{(93.3) - 20}{0.555}$$

$$\textbf{T_s = 132°C with the set point at 60°C}$$

At the reduced set point of 60°C, the stall factor can again be calculated by use of Equation 13.5.1:

$$\% \text{ Stall load} = \frac{D - B}{A - B} \times 100 \qquad \textbf{Equation 13.5.1}$$

Where:
A = The steam temperature at full-load with a 60°C set point (T_S)
B = The secondary fluid outlet temperature (T_2)
D = The backpressure equivalent saturated steam temperature ($T_{(back)}$)

$$\% \text{ Stall load} = \frac{D - B}{A - B} \times 100$$

$$\% \text{ Stall load} = \frac{100 - 60}{132 - 60} \times 100$$

$$\% \text{ Stall load} = \frac{40}{72} \times 100$$

% Stall load = 55.55 = Stall factor of 0.5555

With the set point at 60°C

Full heat load $\dot{Q}$ = $\dot{m}$ Cp ΔT (kW)

Full heat load $\dot{Q}$ = 1.5 kg/s x 4.19 kJ/kg°C x (60 - 20) °C

Full heat load $\dot{Q}$ = 251 kW

Heat load at stall = 0.5555 x 251 kW

Heat load at stall = 140 kW

The condensate discharges to atmosphere, and the h_{fg} at atmospheric pressure is 2 257 kJ/kg.

$$\text{Steam load at stall} = \frac{140 \text{ kW} \times 3\,600 \text{ s/h}}{2\,257 \text{ kJ/kg}}$$

Steam load at stall = 223 kg/h at the reduced set point of 60°C

It can be seen from the above calculations that when the set point is reduced from 70°C to 60°C, the stall load increases from 168 kg/h to 223 kg/h.

It is therefore important that if the application is such that the set point will be reduced, the trapping device, i.e. ball float steam trap or pump-trap, is selected on the stall condition at the lower set point.

Illustrating the effect of reducing the set point by the stall chart method

The stall chart in Figure 13.7.1 shows the secondary temperature line **CB** and the corresponding steam line **AB** for this application (Example 13.7.1) with the higher set point of 70°C.

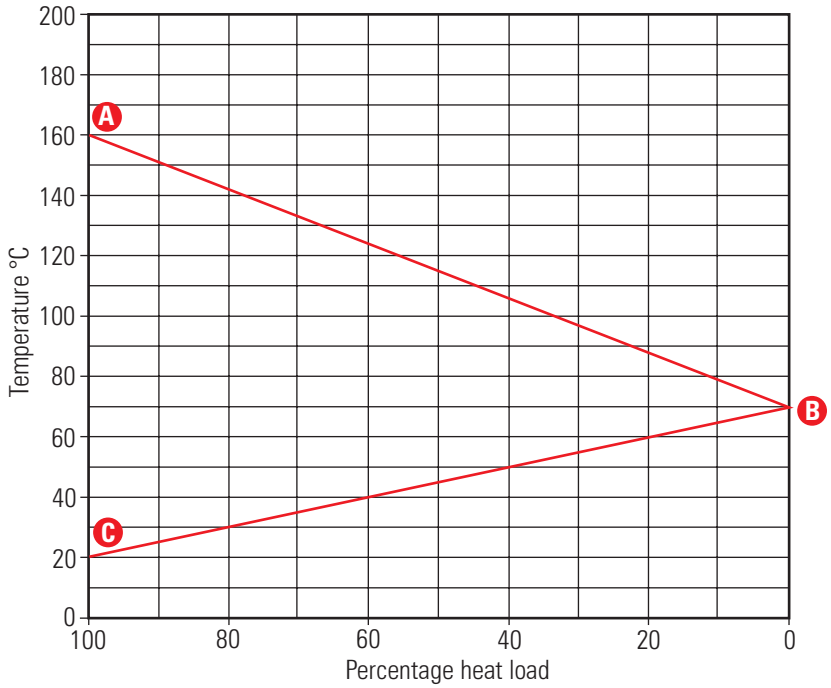


Fig. 13.7.1 The full-load condition with 70°C set point

As mentioned at the beginning of this module, once the operating conditions are known, the effect of reducing the set point can be illustrated on the stall chart by means of proportionality.

This is shown in Figure 13.7.2 by marking the reduced secondary outlet temperature of 60°C (Point **D**) on the secondary load line **CB** and drawing a line **ED** parallel to, and below, the dotted steam line **AB**.

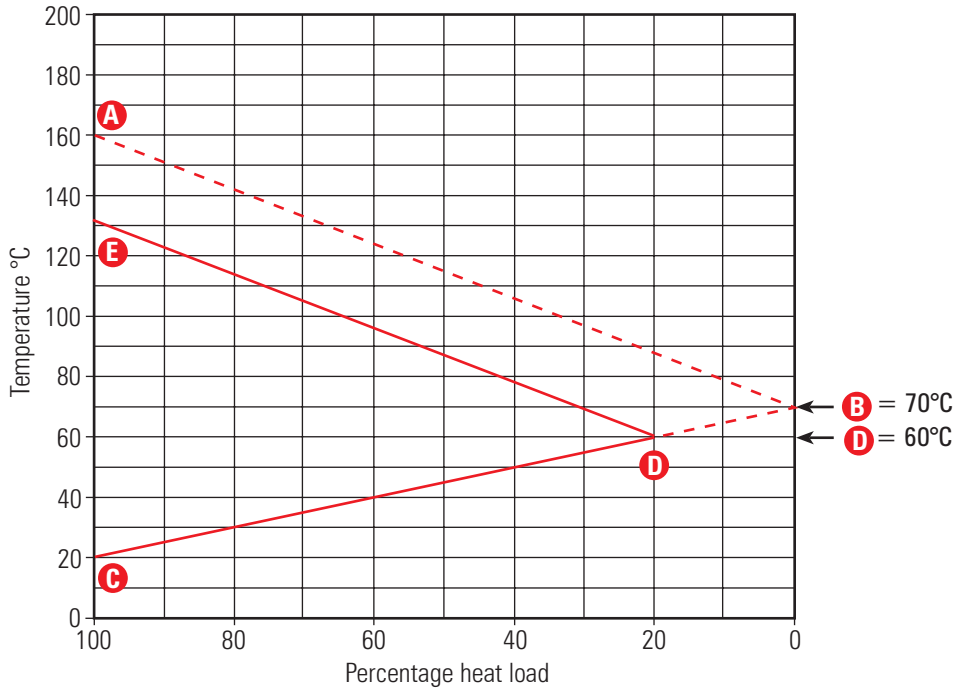


Fig. 13.7.2 Defining the steam temperature for 60°C set point

It is observed that the new steam line **DE** cuts the left hand side of the stall chart at 132°C (Point **E**), and this is the steam temperature when the set point is reduced to 60°C, for a constant secondary flowrate. The steam line **DE** represents steam temperature for reducing heat loads when the set point is 60°C.

Once the new steam temperature of 132°C has been established, it is possible to draw the new steam line **DE** from 132°C to 60°C and the secondary temperature line **CD** from 20°C to 60°C. This stall chart, Figure 13.7.3, represents the steam and secondary inlet temperatures when the set point is at 60°C, consequently zero load now occurs on this stall chart when the secondary temperature is 60°C.

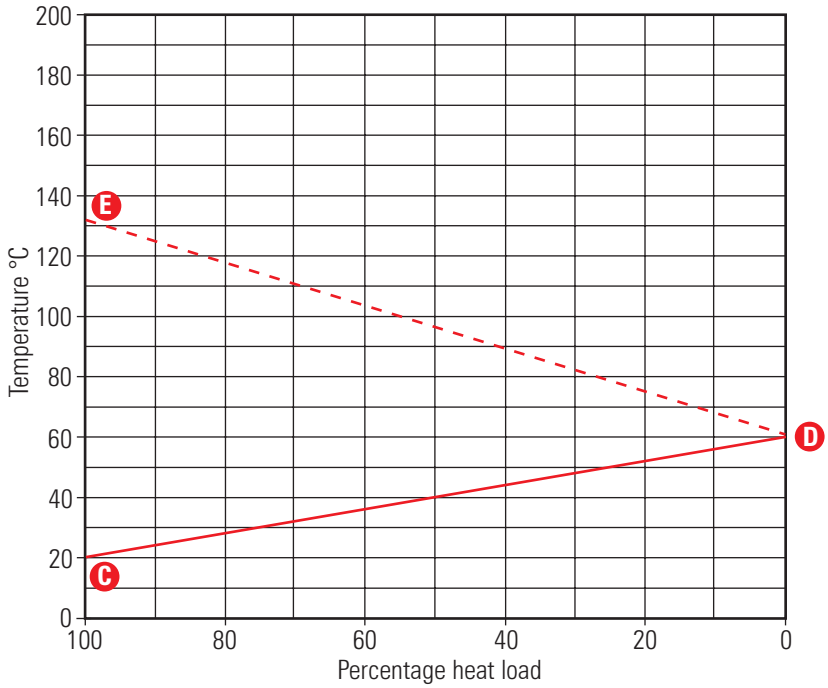


Fig. 13.7.3 The steam line and secondary line for a 60°C set point

By superimposing the backpressure line of 100°C (line **HJ**) onto Figure 13.7.4 it is now possible to depict the new stall load and the corresponding inlet temperature, with a set point of 60°C.

The stall load is approximately 55% (Point **F**) and the inlet temperature at which stall occurs is approximately 38°C (Point **G**).

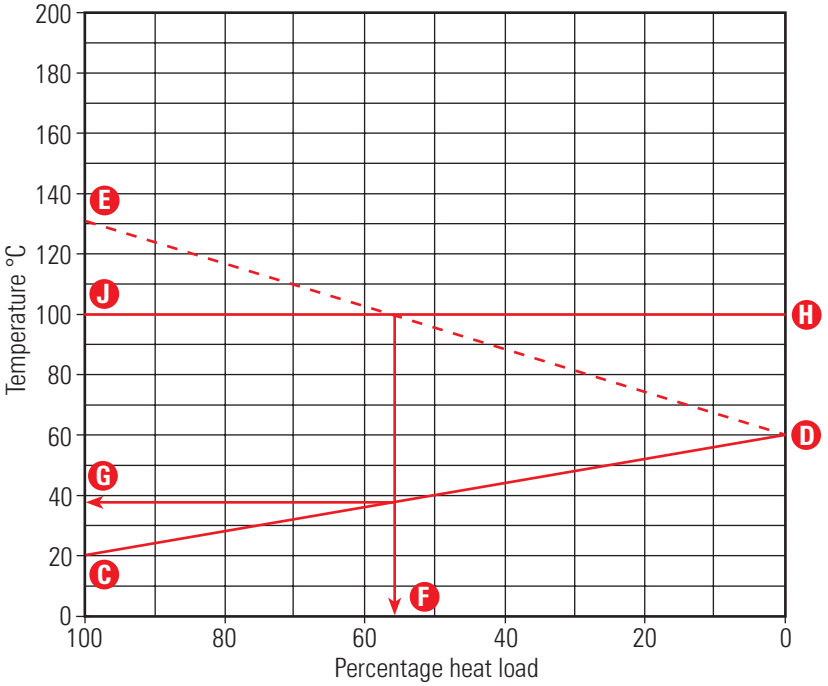


Fig. 13.7.4 The backpressure line is added

By combining Figure 13.7.1 and Figure 13.7.3, it is now possible to observe how the reduction in the outlet temperature from 70°C to 60°C has affected the stall load. In the stall chart below (Figure 13.7.5), it is possible to draw both steam lines **AB** (160°C to 70°C) and **ED** (132°C to 60°C). It can be seen that the backpressure line (**JH**) cuts the two steam lines in different places. The steam line for the higher heat load (with the 70°C set point) is cut at approximately 33% (Point **F1**), whilst the part load line (with the 60°C set point) is cut at approximately 55% (Point **F2**).

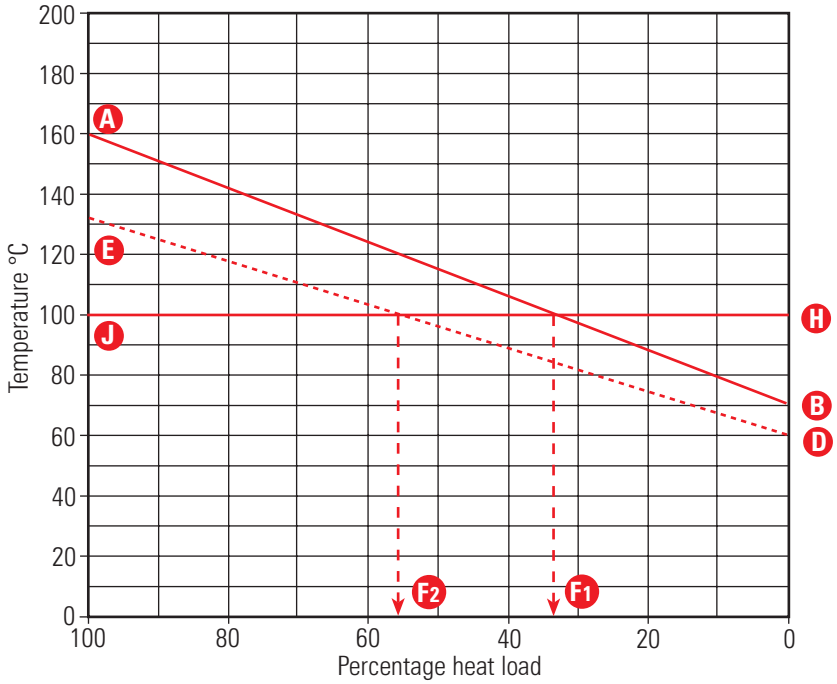


Fig. 13.7.5 The change in stall load

It is important to remember that the above percentages refer to different heat loads. At full-load, the outlet temperature is 70°C and the heat load was calculated in the first part of Example 13.7.1 to be 314 kW, and at the reduced load, when the set point is reduced to 60°C, the heat load was calculated to be 251 kW.

For example

When the set point is 70°C:

The heat load is 314 kW, and stall occurs at 33.33% of this load.

$$\text{Steam load at stall is } 0.333\,3 \times \frac{314 \text{ kW} \times 3\,600 \text{ s/h}}{2\,257 \text{ kJ/kg}} = 167 \text{ kg/h (168 kg/h by calculation)}$$

When the set point is 60°C:

The heat load is 251 kW, and stall occurs at 55.55% of this load.

$$\text{Steam load at stall is } 0.555\,5 \times \frac{251 \text{ kW} \times 3\,600 \text{ s/h}}{2\,257 \text{ kJ/kg}} = 220 \text{ kg/h (223 kg/h by calculation)}$$

It is observed that the steam load at stall increases as the set point is reduced.

The stall chart can also illustrate the inlet temperatures for both stall conditions. This can be useful when carrying out diagnostics on heat exchangers with stall problems. In the stall chart below (Figure 13.7.6), it can be seen how the inlet temperatures can be observed for each of the stall conditions.

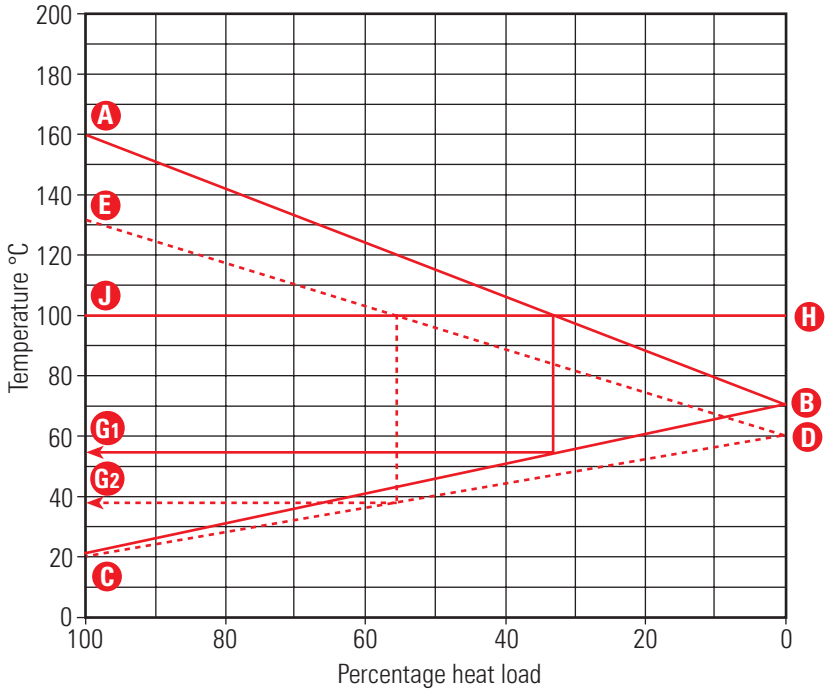


Fig. 13.7.6 The difference in inlet temperatures at the two stall points

With the 70°C set point, an inlet temperature above 53°C (Point **G1**) will produce a stall.

With the 60°C set point, an inlet temperature above 38°C (Point **G2**) will produce a stall.

The validity of these figures can be confirmed by use of the calculation method - Equation 13.2.4:

$$T_1 = T_s - [\ TDC\ (T_s - T_2)\ ] \qquad \textbf{Equation 13.2.4}$$

At the higher set point, $T_2 = 70°C$

$T_1 = 100 - [1.555\ (100 - 70)]$

$T_1 = 100 - [1.555\ (30)]$

$T_1 = 100 - 46.7$

$\mathbf{T_1 = 53.3°C}$

At the lower set point, $T_2 = 60°C$

$T_1 = 100 - [1.555\ (100 - 60)]$

$T_1 = 100 - [1.555\ (40)]$

$\mathbf{T_1} = 100 - 62.2$

$\mathbf{T_1 = 37.8°C}$

Summary

It can be seen from the above information that the stall load will increase as a result of a reducing set point. In fact, stall load will continue to increase to a maximum until the steam pressure falls to equal the condensate backpressure. It is also possible to predict the set point at, and below which permanent stall occurs. The effect can be predicted in the stall chart below (Figure 13.7.7). Stall occurs when the steam temperature is the same as the condensate backpressure, which, in this example, is 100°C, (Point **K**).

In Figure 13.7.7 it is possible to predict the outlet temperature at 100% stall, by projecting the steam temperature line from 100°C (Point **K**) parallel to the full-load steam line **AB**, creating line (**KL**). Where the new steam line **KL** cuts the secondary load line **BC** at point **M**, the outlet temperature can be observed, to be approximately 49°C. If the set point is reduced to (or below) 49°C, stall would be permanent for this example.

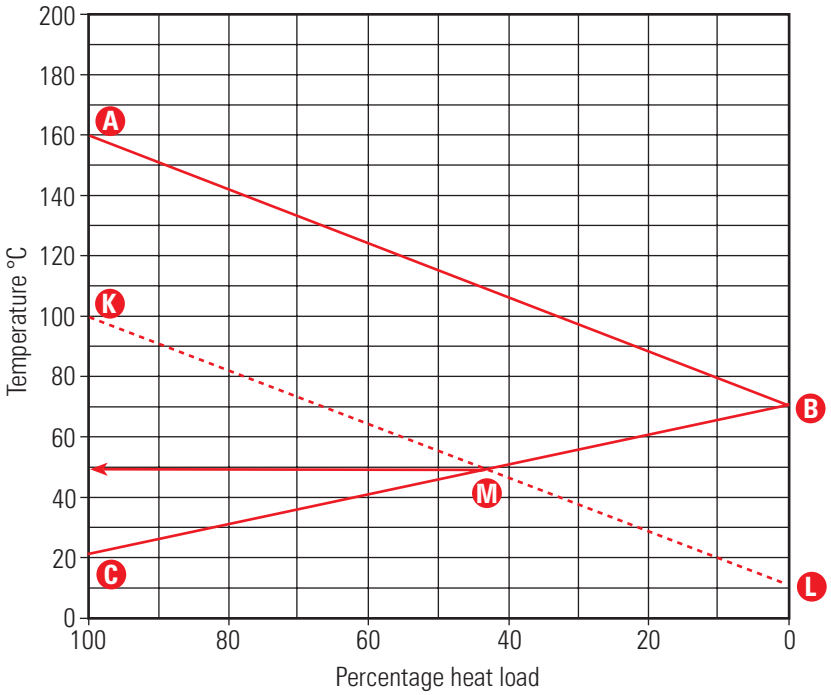


Fig. 13.7.7 The outlet temperature at 100% stall load (for Example 13.7.1) is approximately 49°C

Selecting the correct trapping device

The object of predicting steam pressures and their corresponding steam loads is to enable the selection of the correct trapping device for any application.

In this instance, the trapping device would be selected on the following information.

Maximum steam load = 543 kg/h with the set point at 70°C

Steam pressure at this load = 5.2 bar g

Condensate backpressure = 0 bar g (atmospheric pressure)

∴Trap differential pressure = 5.2 bar at maximum steam load

Stall steam load = 168 kg/h when the set point is 70°C

Stall steam load = 224 kg/h when the set point is 60°C

Differential pressure at stall = 0 bar

A ball float steam trap can be specified if it meets the following two criteria satisfying the initial brief in Example 13.7.1:-

1. It can pass the full-load condition, i.e. 543 kg/h at 5.2 bar differential pressure

2. It can pass the maximum stall load, i.e. 224 kg/h at the 60°C set point

Creating a static head above the ball float steam trap

At the stall condition, with the steam pressure inside the heat exchanger equalling the backpressure, a differential pressure would not exist to push the condensate through a ball float steam trap. Because of this, pressure has to be manufactured on the upstream side of the trap by means of a static head.

Static head must be available between the heat exchanger condensate outlet and the trap inlet to generate enough differential pressure to enable the trap to pass the stall load of 224 kg/h. In order to allow condensate to drain easily from the exchanger, a vacuum breaker is fitted to its steam inlet downstream of the control valve (Figure 13.7.8).

It can be seen in Figure 13.7.9 that a DN25 (1") FT10-10 ball float steam trap will accommodate these criteria. However, the trap requires a minimum of 4 metres head above the trap inlet to pass the stall load.

A 4 metre head might not be available in practice, and, if so, a larger trap would need to be specified. Refer to Figure 13.7.8.

For the purposes of Example 13.7.1, if the available head were only 200 mm then it can be seen from Figure 13.7.10 that a DN40 (1½") FT10-10 ball float steam trap would be required.

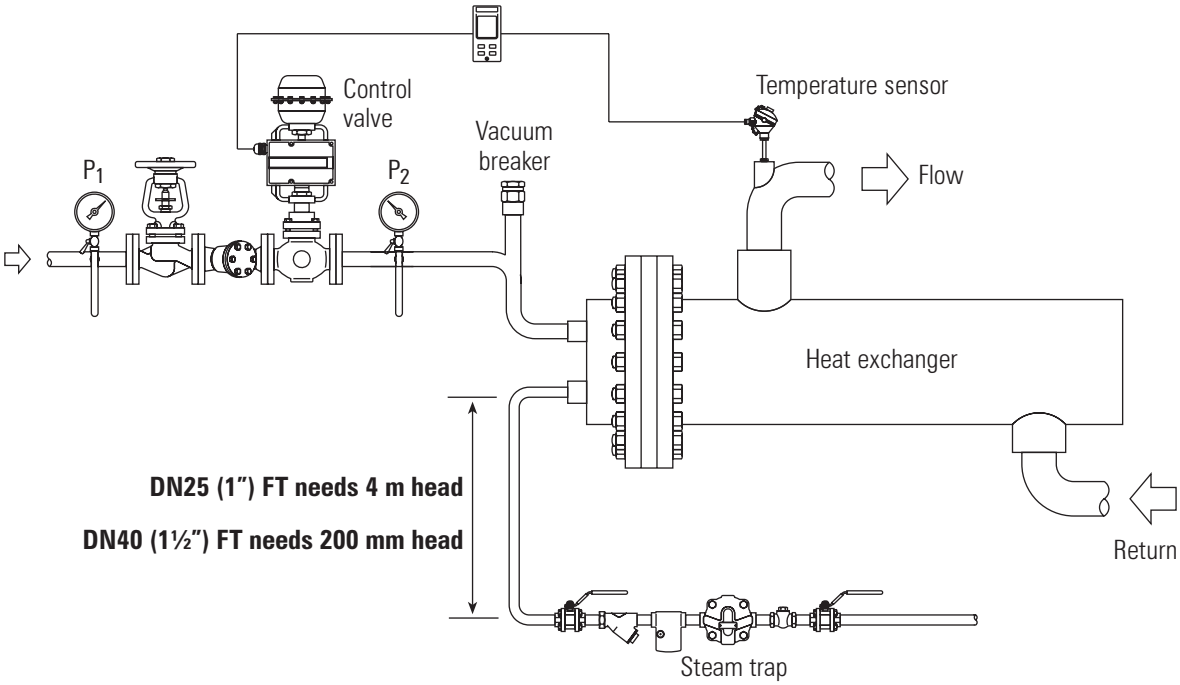


Fig. 13.7.8 The trap size depends on the static head

Footnote:

Should the backpressure have been greater than atmospheric pressure, due perhaps to a lift after the trap and/or a pressurised condensate line, then the same sizing routine could be carried out. Depending upon the amount of backpressure, it may be that even the largest sized steam trap cannot pass the required amount of condensate at stall.

Under these circumstances, a ball float steam trap cannot be specified, as the heat exchanger will flood at part loads. Instead, a pump-trap must be used, which is able to clear the condensate from the heat exchanger into the condensate system at any heat load.

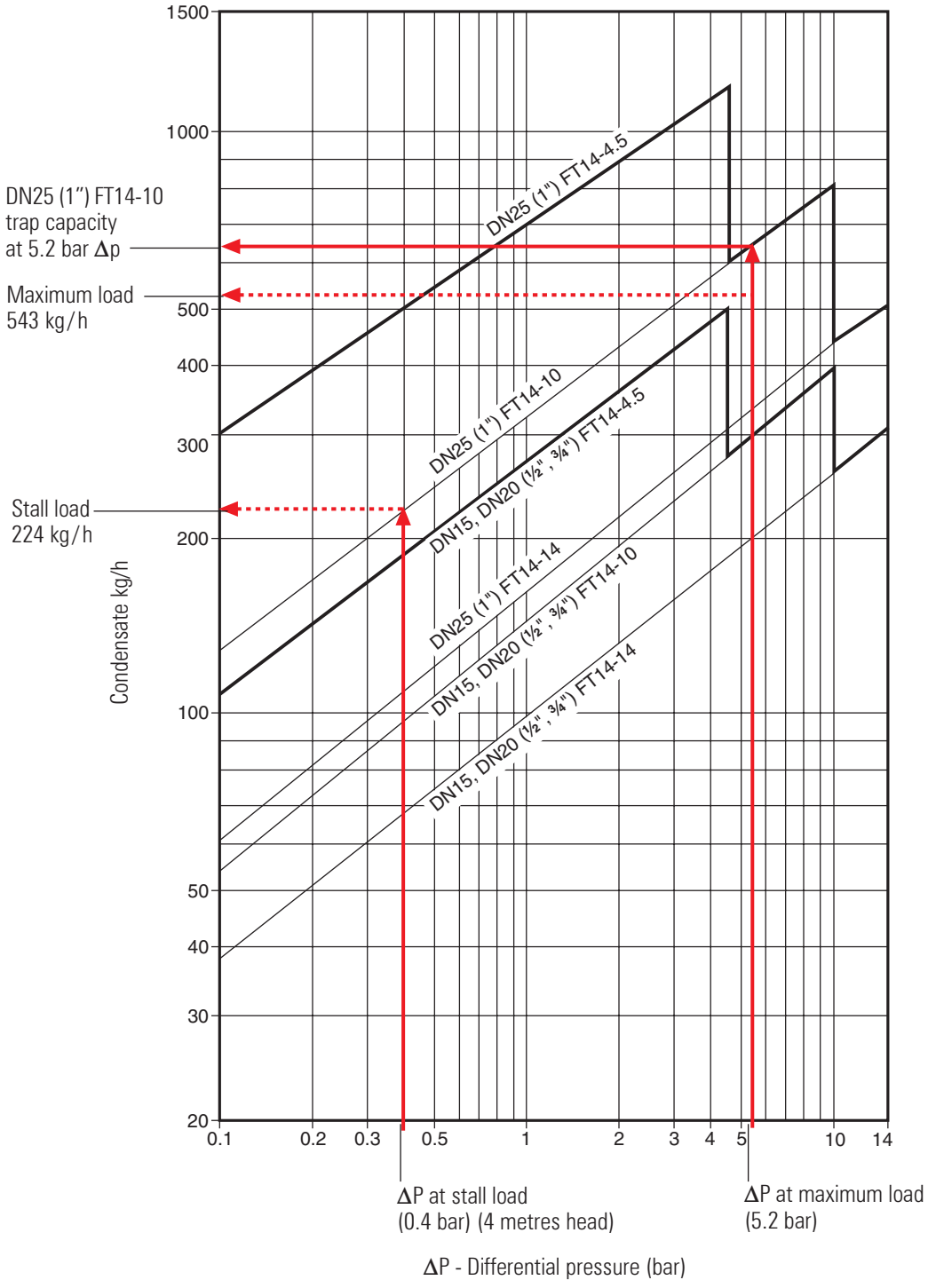

1500
1000
500
400
300
200
100
50
40
30
20
0.1
0.2
0.3
0.5
1
2
3
4
5
10
14
DN25 (1") FT14-10
trap capacity
at 5.2 bar Δp
Maximum load
543 kg/h
Stall load
224 kg/h
Condensate kg/h
DN25 (1") FT14-4.5
DN25 (1") FT14-10
DN15, DN20 (½", ¾") FT14-4.5
DN25 (1") FT14-14
DN15, DN20 (½", ¾") FT14-10
DN15, DN20 (½", ¾") FT14-14
ΔP at stall load
(0.4 bar) (4 metres head)
ΔP at maximum load
(5.2 bar)
ΔP - Differential pressure (bar)

Fig. 13.7.9

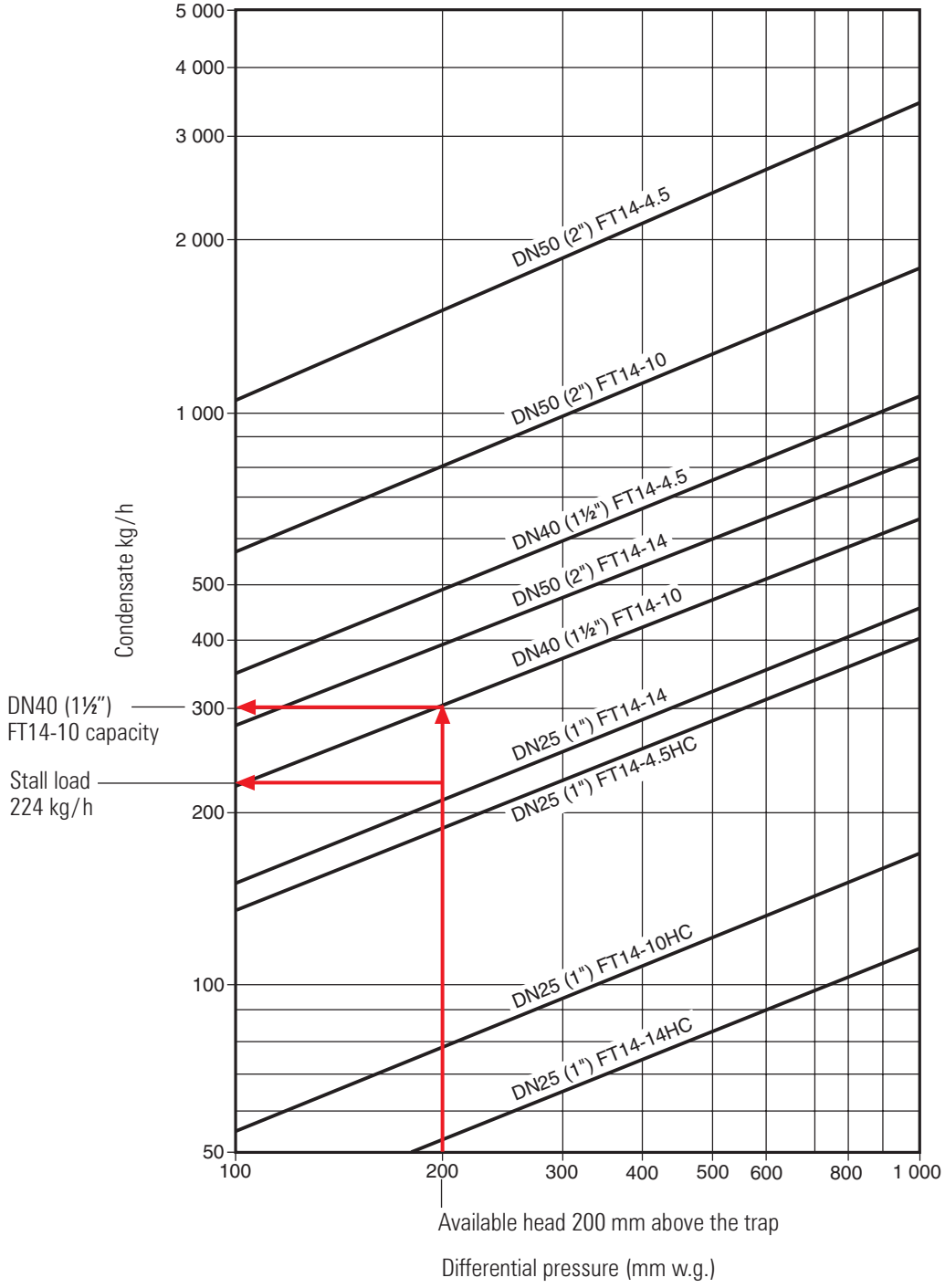

5 000
4 000
3 000
2 000
1 000
500
400
300
200
100
50
Condensate kg/h
DN50 (2") FT14-4.5
DN50 (2") FT14-10
DN40 (1½") FT14-4.5
DN50 (2") FT14-14
DN40 (1½") FT14-10
DN25 (1") FT14-14
DN25 (1") FT14-4.5HC
DN25 (1") FT14-10HC
DN25 (1") FT14-14HC
DN40 (1½")
FT14-10 capacity
Stall load
224 kg/h
100
200
300
400
500
600
800
1 000
Available head 200 mm above the trap
Differential pressure (mm w.g.)

Fig. 13.7.10

Questions

1. How does a reduction in set point affect stall?

a| It does not affect it at all ☐

b| It reduces the percentage stall ☐

c| The system operates under permanent stall conditions ☐

d| It increases the percentage stall ☐

2. How does a reduction in the set point affect the steam pressure?

a| It reduces the steam pressure ☐

b| It does not affect it at all ☐

c| It increases the steam pressure ☐

d| The steam pressure equals the backpressure ☐

3. In Example 13.7.1, with the set point at 60°C what would the steam temperature be if the system were operating at 30% heat load?

a| 100°C ☐

b| 110°C ☐

c| 120°C ☐

d| 80°C ☐

4. In Example 13.7.1, with the set point at 60°C if the minimum possible heat load were 70%, would the system stall?

a| Yes, at all loads ☐

b| Only if the set point were increased ☐

c| No ☐

d| Yes, but only on loads higher than 70% ☐

5. If, in Example 13.7.1, the set point were reduced to 40°C, what would be the approximate steam temperature if the inlet temperature remained at 20°C?

a| 66°C ☐

b| 45°C ☐

c| 55°C ☐

d| 76°C ☐

6. If, in Example 13.7.1, the set point were reduced to 40°C, what would be the approximate steam temperature if the inlet temperature rose to 30°C?

a| 28°C ☐

b| 38°C ☐

c| 48°C ☐

d| 58°C ☐

Answers

1: d, 2: a, 3: d, 4: c, 5: d, 6: d

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Module 13.8

Practical Methods of Preventing Stall

Practical Methods of Preventing Stall

If stall conditions are inevitable, potential problems can be overcome by designing the installation around one of three basic solutions:

1. Ensure the steam pressure in the steam space can never drop below atmospheric pressure, and that the condensate can drain by gravity to and from a ball float steam trap.

2. Accept that the pressure in the steam space may be less than the backpressure, and provide an alternative means of removing condensate, by installing a pump-trap.

3. Ensure the pressure in the steam space is stable and higher than the backpressure. This will entail having the temperature control system on the secondary side of the system.

Taking these three options in turn:

1. Installations that ensure the conditions in the steam space can never drop below atmospheric pressure, and that the condensate can drain by gravity to and from a steam trap:

1a) Condensate removal by vacuum breaker method (see Figure 13.8.1)

The steam trap cannot be subject to any backpressure higher than atmospheric, and must drain condensate either to an open end (which may be wasteful), or to a nearby vented receiver and pump, enabling the energy contained in the condensate to be reclaimed.

There are two criteria that must be satisfied:

- A vacuum breaker must be fitted to the steam inlet to the heat exchanger after the control valve.
- The trap must be installed at a discreet distance below the heat exchanger outlet such that sufficient static head is created to pass the requisite amount of condensate when stall occurs. A distance of between 0.5 to 1 m is usually sufficient; however, smaller distances can be accommodated with larger traps, if less head is available.

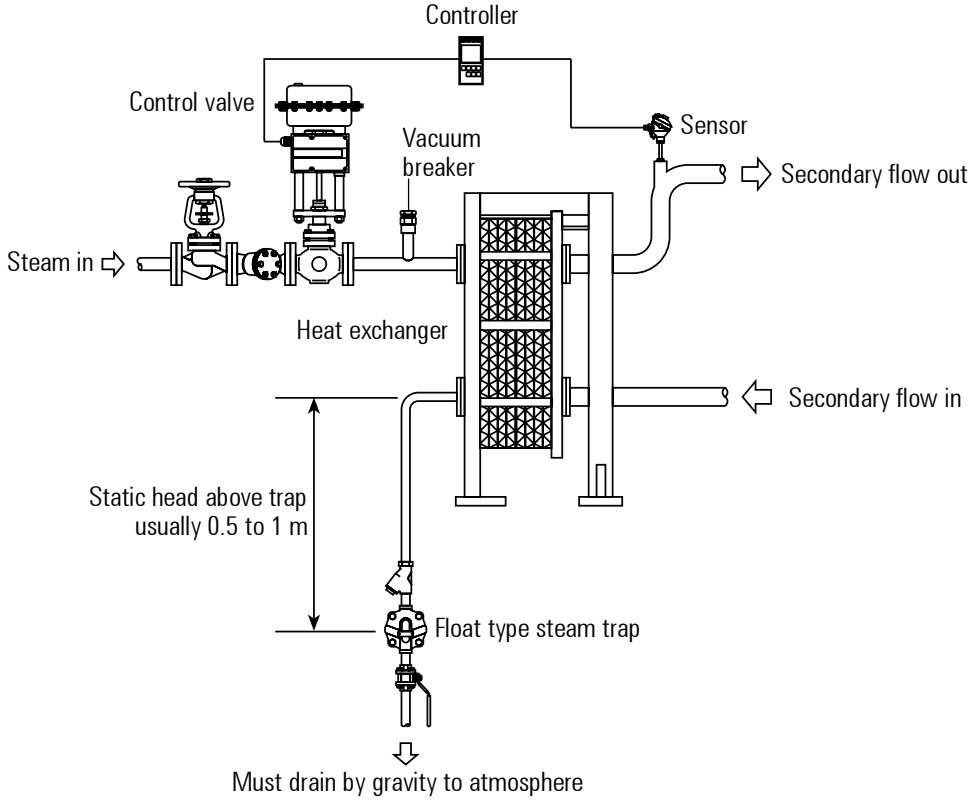


Fig. 13.8.1 Static head and vacuum breaker method of dealing with stall

1b) Auxiliary drain trap method (see Figure 13.8.2)

A standard float trap set is installed with condensate returning to a condensate system, which is either pressurised and/or elevated above the trap. An auxiliary float trap may be fitted, discharging condensate via an open end to drain.

When there is sufficient steam pressure to overcome the backpressure, the main float trap will function, but when stall occurs, condensate will back-up and drain through the auxiliary float trap thus preventing condensate flooding back into the heat exchanger.

As this condensate will drain to waste, this method should only be used if stall occurs infrequently. The auxiliary trap should be sized on static head to pass the stall load as in method 1a, and the 'main' trap should be the same size, but fitted at least 150 mm below the auxiliary take-off tee-piece.

Apart from the obvious disadvantage of energy loss, this method also requires available head between the trap inlets and the heat exchanger outlet.

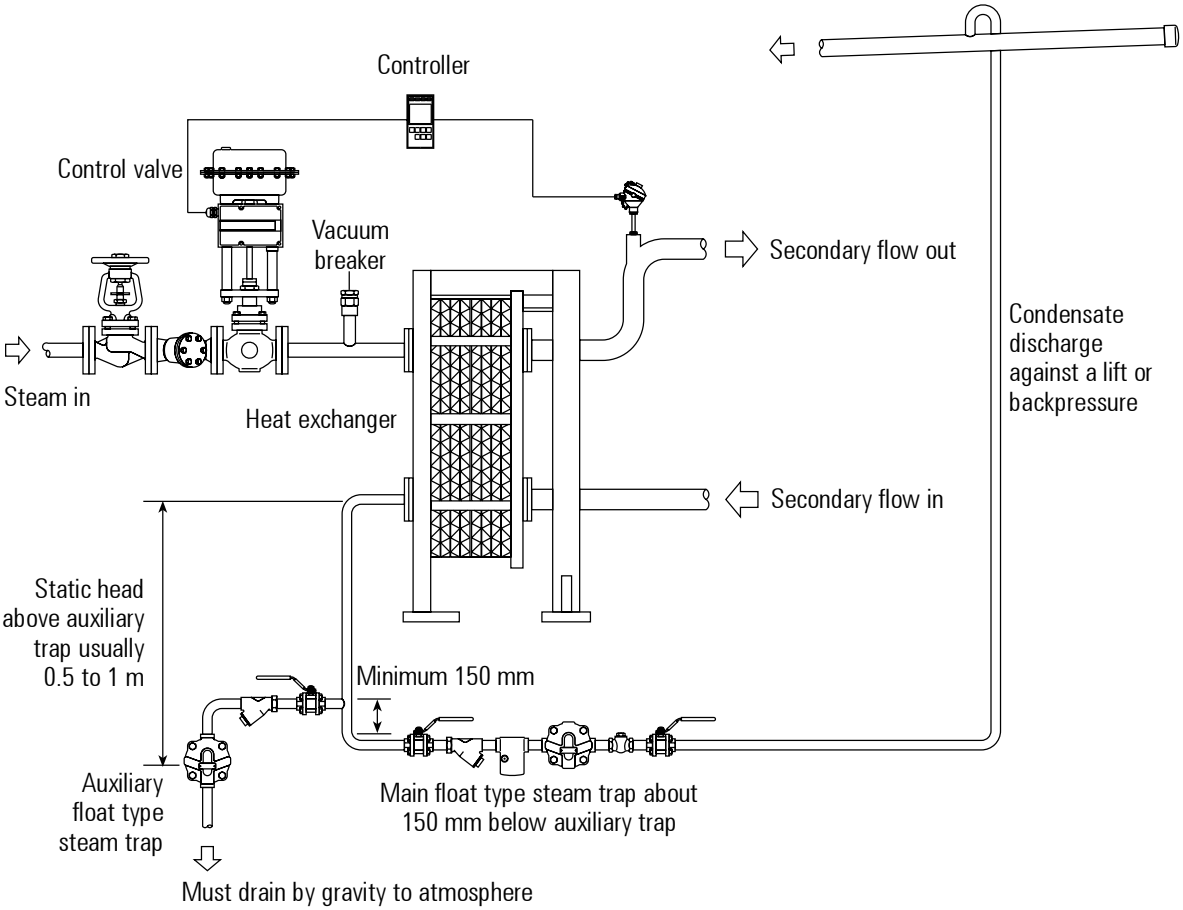


Fig. 13.8.2 Auxiliary drain method of dealing with stall

2. Installations which allow the steam pressure in the steam space to drop below the backpressure, but where the condensate can drain by gravity to a pump-trap arrangement:

2a) A pump and float trap installed in combination (see Figure 13.8.3)
This method uses a pump and float trap installed in combination. It is better suited to heat exchangers with nominal heating capacities in excess of 1.5 MW (nominally 2 500 kg/h of steam).

The steam pressure changes relative to changes in heat load. At high loads the steam pressure will be higher than the backpressure, but at low loads it will be lower.

The pump is a mechanical pressure-powered type, in which an auxiliary steam supply automatically takes over to provide the motive power to discharge the condensate when stall occurs. If the steam space pressure is higher than the backpressure, condensate passes through the pump body to the float trap, which allows the condensate to discharge.

This method is more practical and economical on larger installations; for example, those using condensate drain lines of 40 mm or more.

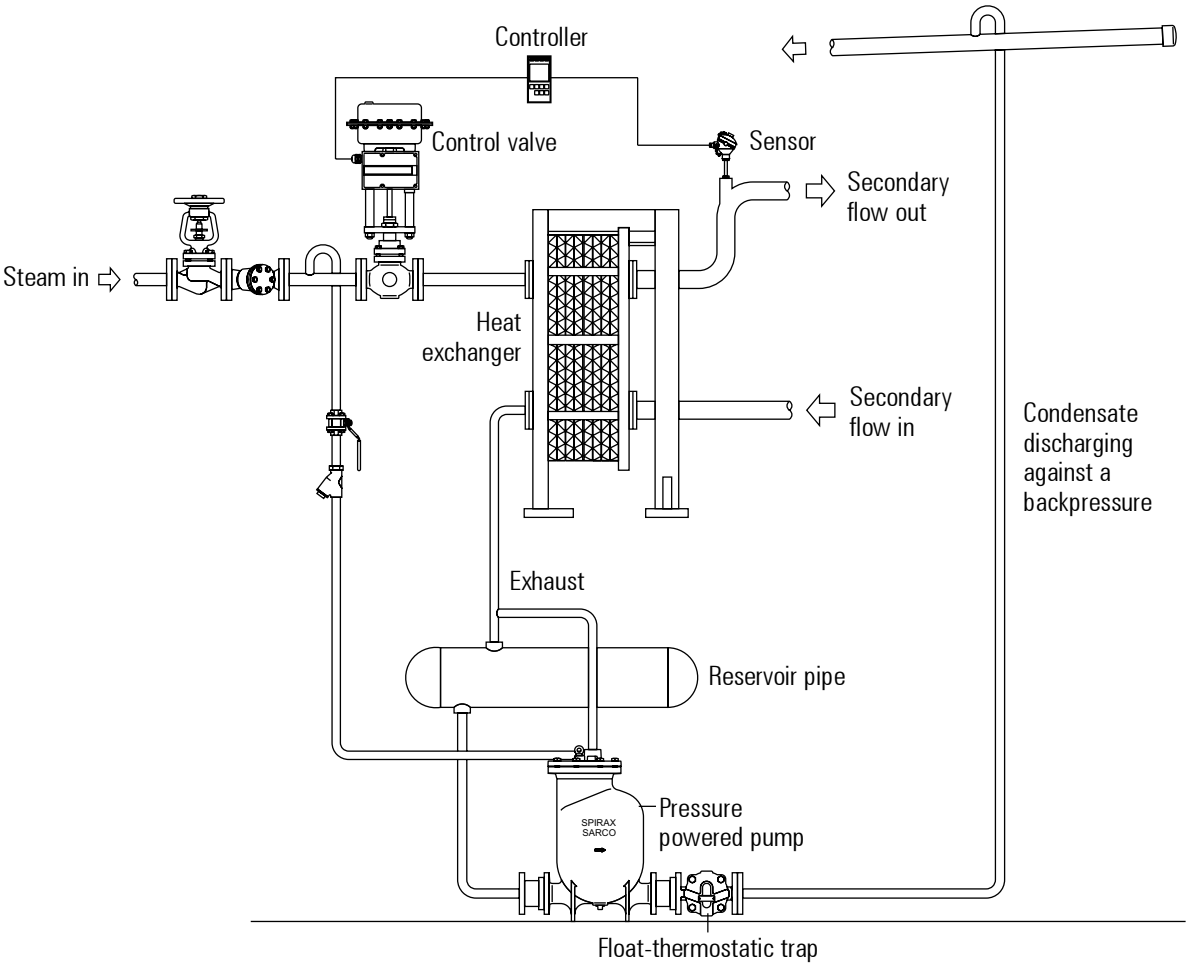


Fig. 13.8.3 Combination pump and steam trap method of dealing with stall

2b) A pump-trap with constant flow heat exchanger (see Figure 13.8.4)
The secondary flowrate does not change as it passes through the heat exchanger, consequently the steam pressure changes relative to changes in the secondary inlet temperature. At high loads the steam pressure will be higher than the backpressure, but at low loads it will be lower.

This method uses a pump-trap device, which offers the functions of a pump, steam trap and check valves in one body.

The Spirax Sarco APT14 automatic pump-trap is designed to occupy a minimum amount of space, and can be fitted to heat exchangers with nominal heating capacity of up to 1.5 MW.

It is most suited to installations with condensate drain lines up to 25 mm, but can be used on drain lines up to 40 mm in some circumstances.

A typical installation is shown on Figure 13.8.4.

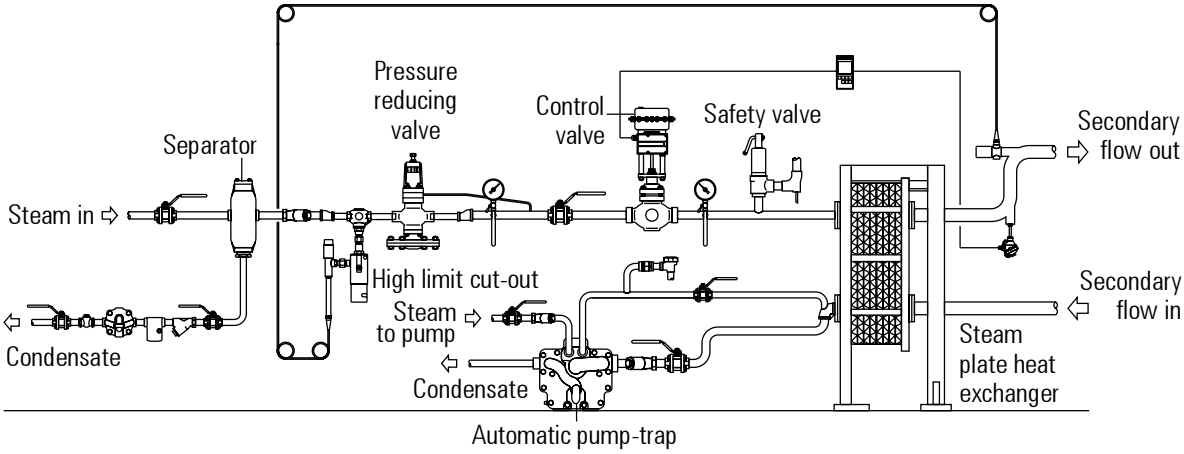


Fig. 13.8.4 Pump-trap method of dealing with stall

2c) A pump-trap device with varying flow heat exchanger (see Figure 13.8.5)
This method is similar to 2b), but the secondary flow through the heat exchanger varies with the heat load, due to the action of the secondary mixing valve.

The heat exchanger delivers a constant temperature water flow which is blended by the secondary mixing valve according to load. As the secondary flow varies, the steam pressure changes to maintain a constant outlet temperature, such that, at high loads, it is above the backpressure, and at low loads it is below.

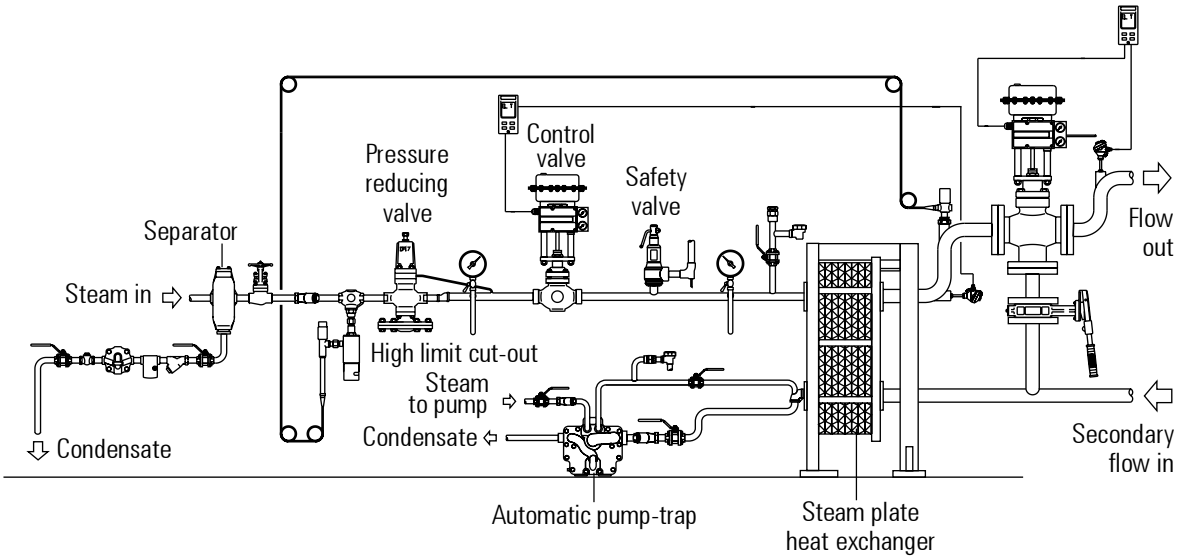


Fig. 13.8.5 Pump-trap method of dealing with stall

3. Installations which ensure the steam pressure is kept constant and can never drop below the backpressure, and that the condensate can drain to and from a steam trap:

3a) Steam trap with temperature control valve in secondary circuit (see Figure 13.8.6)
This method requires temperature control to be carried out with a 3-port mixing or diverting valve in the secondary circuit. The steam supply to the heat exchanger is held at a constant pressure (usually less than 1 bar g) with a pressure control valve, and as such, condensate can always be cleared from the heat exchanger against a lower backpressure.

This method is not always practical or possible. It is unsuitable on steam/air heater batteries or liquid systems where the secondary system is at such a low pressure that it is unable to prevent the liquid from boiling.

Like all methods, it has both advantages and disadvantages, which must be assessed before an option can be chosen.

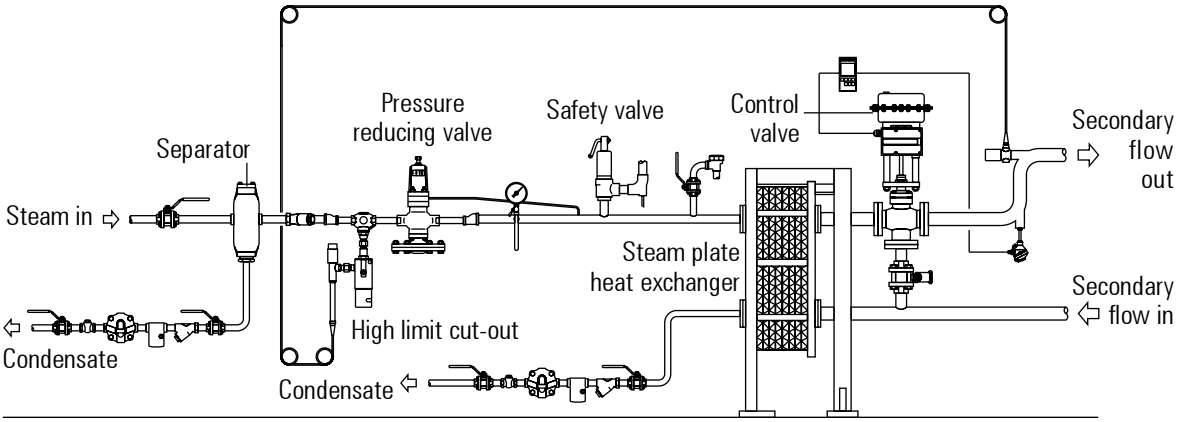


Fig. 13.8.6 Constant steam pressure - Secondary temperature control

3b) Steam trap and modulating valve in condensate drain line (see Figure 13.8.7)
Condensate drainage is achieved with a modulating valve in the condensate drain line. This method also maintains the desired steam pressure in the steam space regardless of load conditions.

However, it encourages (instead of eliminates) waterlogging in the heat exchanger, as control is achieved by deliberately flooding the steam space with condensate as the load reduces. Usually this method is only considered if:

- The heat load is steady or changes very slowly.
- The heat exchanger is designed to withstand the effects of waterlogging.
- The likely stratification of temperatures of the secondary fluid is acceptable.

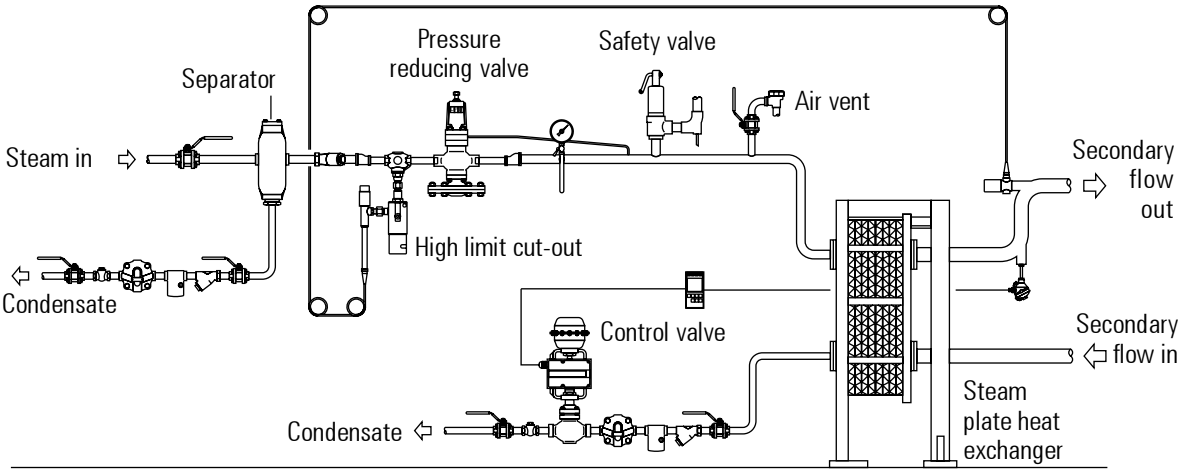


Fig. 13.8.7 Constant steam pressure - Condensate level control

On/off control should not be used with heat exchangers

An on/off temperature control valve does not modulate depending on heat load, but is either fully open or fully closed. An example would be a solenoid valve. When open, full steam pressure will be maintained in the heat exchanger to clear the condensate against the backpressure. At first glance, this method of control would seem to overcome any backpressure problems, but is not recommended on processes such as heat exchangers, where the secondary fluid has to be heated to its required temperature as it passes through. There are three main reasons for this:

- An 'on/off' control system is activated by a thermostat which relies upon a product overtemperature to achieve control. As steam has high heat content, a significant amount of heat can be held in the steam space after the solenoid valve has shut. The overall effect is a higher product temperature than required. Should the thermostat setting be lowered to counteract this effect, the 'on' temperature may be lower than the system parameters may require. It can result in poor control of the system temperature and the potential for product spoilage.
- The continual and rapid changes in pressure and temperature will impose thermal and mechanical stresses upon the heat exchanger which will probably reduce its service life.
- It is never a good idea to subject steam systems to an instantaneous increase in pressure. Any condensate present in the steam space and condensate pipe is instantly pushed, by the sudden inrush of steam, through the system towards the steam trap. This can cause waterhammer, and damage the heat exchanger and steam trap.

On/off control is normally only suitable for 'non-flow' or 'batch' type heat exchange processes, notably tanks with robust heating coils, or jacketed pans, where the desired steam pressure is applied over a long heating up period (usually over many minutes or even hours). The rise in product temperature is much slower than that experienced with flow-type systems that are expected to heat the product in the short time it takes to pass through a heat exchanger.

Conclusion

The most suitable type of steam trap for heat exchange equipment in general, and especially if stall is likely, is a ball float steam trap with integral balanced pressure air vent.

If there is any likelihood of stall, a pump-trap is generally the most effective way of dealing with it, as it benefits from being:

- Simple.
- Cost effective.
- Compact.

Please note: The diagrams in this Module are schematic only, and for simplicity do not contain all the ancillary equipment that would be necessary or advisable for a specific installation. The exception is Figure 13.8.8, which shows a detailed, actual, installation of an APT14 automatic pump-trap.

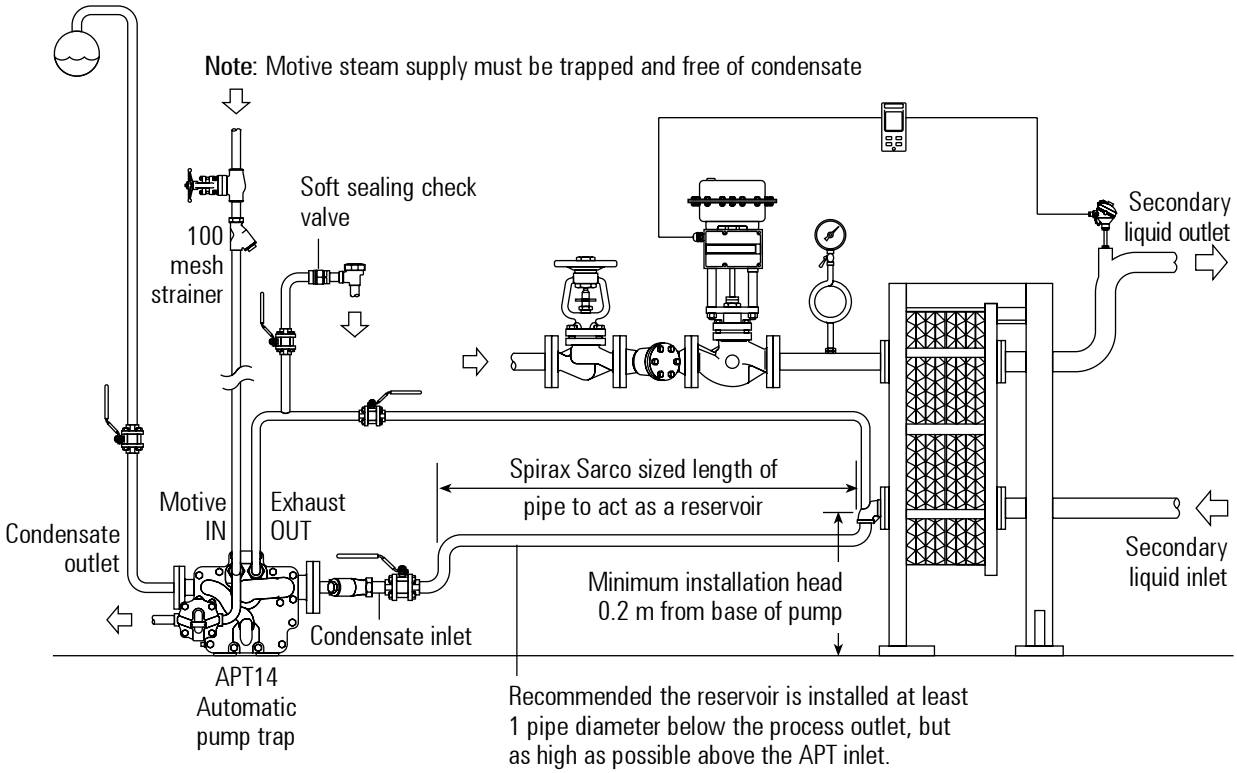


Fig. 13.8.8 Detailed installation of a pump-trap with plate heat exchanger

Footnote

The subject of stall can become somewhat complex, especially when selecting and sizing the most appropriate equipment and designing its installation such that it can be guaranteed to work when commissioned to do so.

This Module is not so much intended to make the reader an expert in the subject of stall, but rather:

- To allow him or her to understand what it is.
- To understand why it exists.
- To know what can be done to prevent it.
- To know who to contact for proper advice.

Questions

1. Which of the following methods can be employed to prevent the effects of stall?

a| Prevent vacuum formation in the steam space and drain to atmosphere ☐

b| Maintain the condensate backpressure below the steam space pressure ☐

c| Ensure condensate removal by installing a pump-trap ☐

d| All of the above ☐

2. When using the vacuum breaker method of preventing stall, which of the following methods is ideal to ensure the backpressure is maintained below atmospheric pressure?

a| Install the auxiliary steam trap ☐

b| Drain the condensate to the atmosphere ☐

c| Drain the condensate to a vented receiver to allow energy recovery ☐

d| Replace the float trap with a balanced pressure thermostatic device ☐

3. Which of the following functions is provided by an automatic pump-trap?

a| Check valve ☐

b| Float type steam trap ☐

c| Pressure powered pump ☐

d| All of the above ☐

4. When should a separate pump, steam trap and reservoir combination be used to prevent stall?

a| In small heat exchangers where backpressure may be greater than steam pressure ☐

b| In large heat exchangers where backpressure may be greater than steam pressure ☐

c| In vacuum breaker systems where the condensate is drained under gravity ☐

d| All of the above ☐

5. What is the advantage of placing a modulating valve in the condensate drain line?

a| The desired steam pressure is maintained regardless of the load conditions ☐

b| It can be used to effectively deal with rapid heat loss changes ☐

c| It is not necessary to control the steam flowrate ☐

d| Stall is prevented regardless of the backpressure ☐

6. Why should on/off control not be used on heat exchangers where the secondary fluid is heated whilst it flows through the heat exchanger?

a| Product damage may result from continued heating after the steam valve is closed ☐

b| Rapid opening of the steam valve may cause waterhammer ☐

c| On/off control cannot accurately maintain product temperature ☐

d| All of the above ☐

Answers

1: d, 2: c, 3: d, 4: b, 5: a, 6: d

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Module 14.1

Introduction to Condensate Recovery

Introduction to Condensate Recovery

Steam is usually generated for one of two reasons:

- To produce electrical power, for example in power stations or co-generation plants.
- To supply heat for heating and process systems.

When a kilogram of steam condenses completely, a kilogram of condensate is formed at the same pressure and temperature (Figure 14.1.1). An efficient steam system will reuse this condensate. Failure to reclaim and reuse condensate makes no financial, technical or environmental sense.

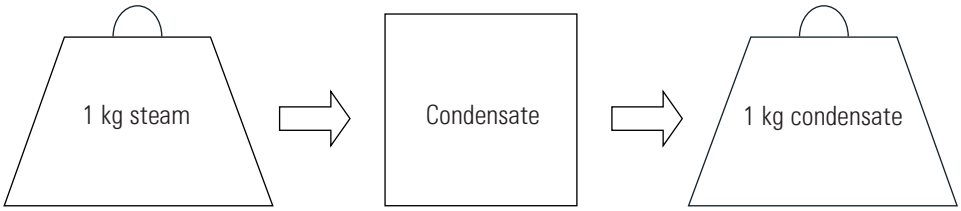


Fig. 14.1.1 1 kg of steam condenses completely to 1 kg of condensate

Saturated steam used for heating gives up its latent heat (enthalpy of evaporation), which is a large proportion of the total heat it contains. The remainder of the heat in the steam is retained in the condensate as sensible heat (enthalpy of water) (Figure 14.1.2).

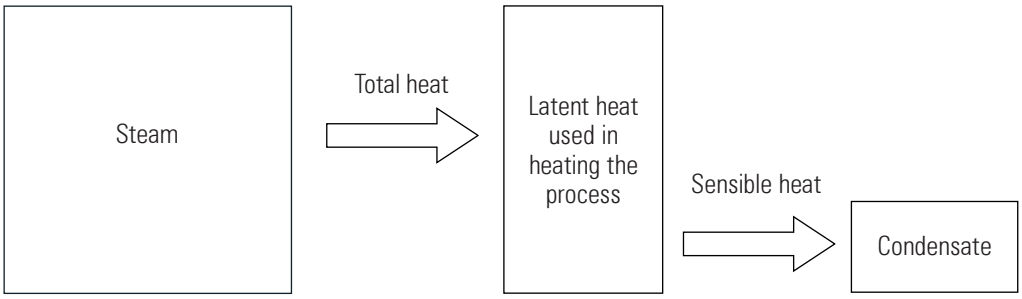


Fig. 14.1.2 After giving up its latent heat to heat the process, steam turns to water containing only sensible heat

As well as having heat content, the condensate is basically distilled water, which is ideal for use as boiler feedwater. An efficient steam system will collect this condensate and either return it to a deaerator, a boiler feedtank, or use it in another process. Only when there is a real risk of contamination should condensate not be returned to the boiler. Even then, it may be possible to collect the condensate and use it as hot process water or pass it through a heat exchanger where its heat content can be recovered before discharging the water mass to drain.

Condensate is discharged from steam plant and equipment through steam traps from a higher to a lower pressure. As a result of this drop in pressure, some of the condensate will re-evaporate into 'flash steam'. The proportion of steam that will 'flash off' in this way is determined by the amount of heat that can be held in the steam and condensate. A flash steam amount of 10% to 15% by mass is typical (see Module 2.2). However, the percentage volumetric change can be considerably more. Condensate at 7 bar g will lose about 13% of its mass when flashing to atmospheric pressure, but the steam produced will require a space some 200 times larger than the condensate from which it was formed. This can have the effect of choking undersized trap discharge lines, and must be taken into account when sizing these lines.

Example 14.1.1 Calculating the amount of flash steam from condensate

Hot condensate at 7 bar g has a heat content of about 721 kJ/kg. When it is released to atmospheric pressure (0 bar g), each kilogram of water can only retain about 419 kJ of heat. The excess energy in each kilogram of the condensate is therefore 721 – 419 = 302 kJ. This excess energy is available to evaporate some of the condensate into steam, the amount evaporated being determined by the proportion of excess heat to the amount of heat required to evaporate water at the lower pressure, which in this example, is the enthalpy of evaporation at atmospheric pressure, 2 258 kJ/kg.

$$\text{Therefore, in this example, the percentage of flash steam evaporated} = \frac{302}{2\,258} \times 100\%$$

Flash steam evaporated = 13.4%

The subject of flash steam is examined in greater depth in Module 2.2, 'What is steam?' A simple graph (Figure 14.1.3) is used in this Module to calculate the proportion of flash steam.

Example:

Proportion of flash steam using Figure 14.1.3:

Pressure on the trap = 4 bar g
Flash steam pressure = 0 bar g
% Flash steam = 10%

The amount of flash steam in the pipe is the most important factor when sizing trap discharge lines.

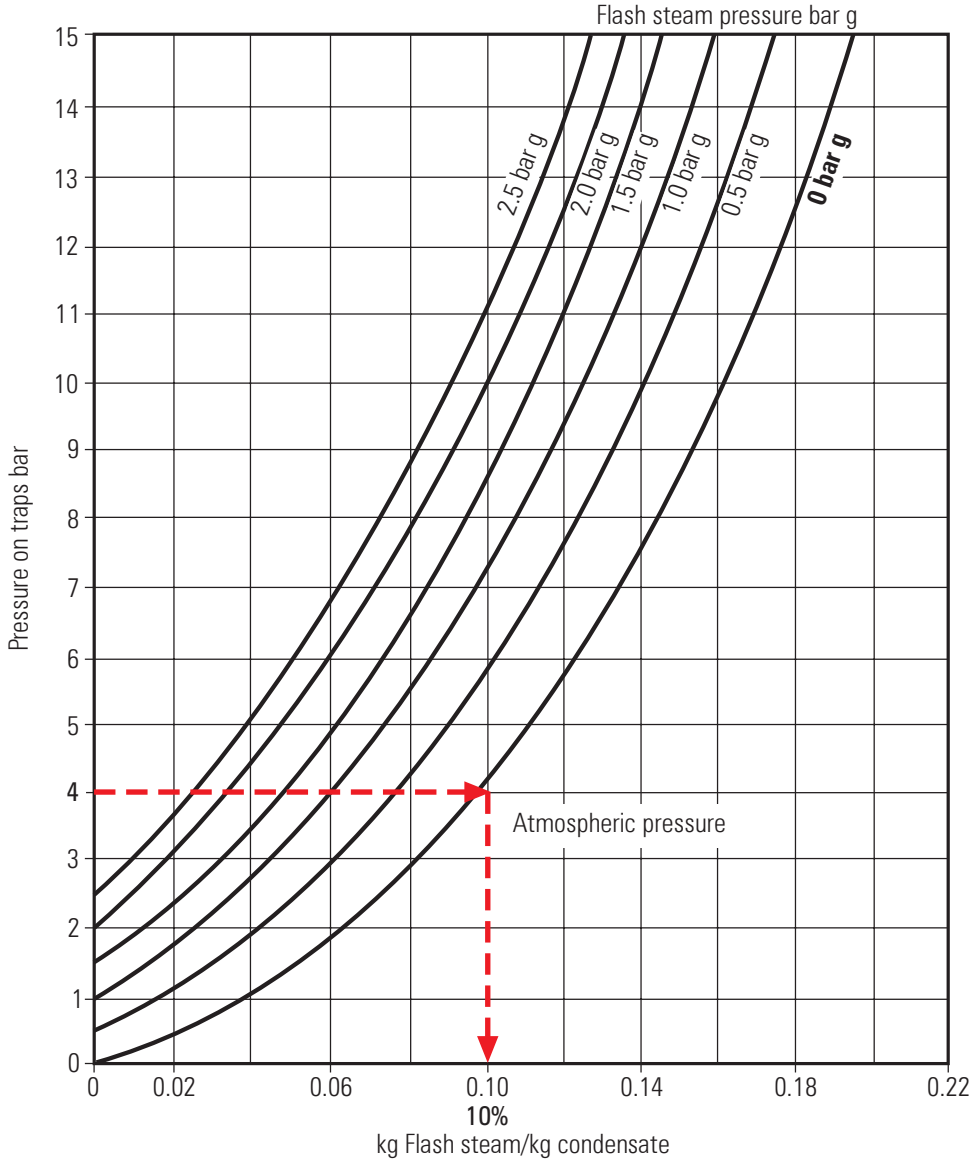


Fig. 14.1.3 Quantity of Flash Steam Graph

Steam produced in a boiler by the process of adding heat to the water is often referred to as live steam. The terms live steam and flash steam are only used to differentiate their origin. Whether steam is produced in a boiler or from the natural process of flashing, it has exactly the same potential for giving up heat, and each is used successfully for this purpose. The flash steam generated from condensate can contain up to half of the total energy of the condensate. An efficient steam system will recover and use flash steam. Condensate and flash steam discharged to waste means more make-up water, more fuel, and increased running costs.

This Module will look at two essential areas – condensate management and flash steam recovery. Some of the apparent problem areas will be outlined and practical solutions proposed.

Note: The term 'trap' is used to denote a steam-trapping device, which could be a steam trap, a pump-trap, or a pump and trap combination. The ability of any trap to pass condensate relies upon the pressure difference across it, whereas a pumping trap or a pump-trap combination will be able to pass condensate irrespective of operational pressure differences (subject to design pressure ratings).

Condensate return

An effective condensate recovery system, collecting the hot condensate from the steam using equipment and returning it to the boiler feed system, can pay for itself in a remarkably short period of time. Figure 14.1.4 shows a simple steam and condensate circuit, with condensate returning to the boiler feedtank.

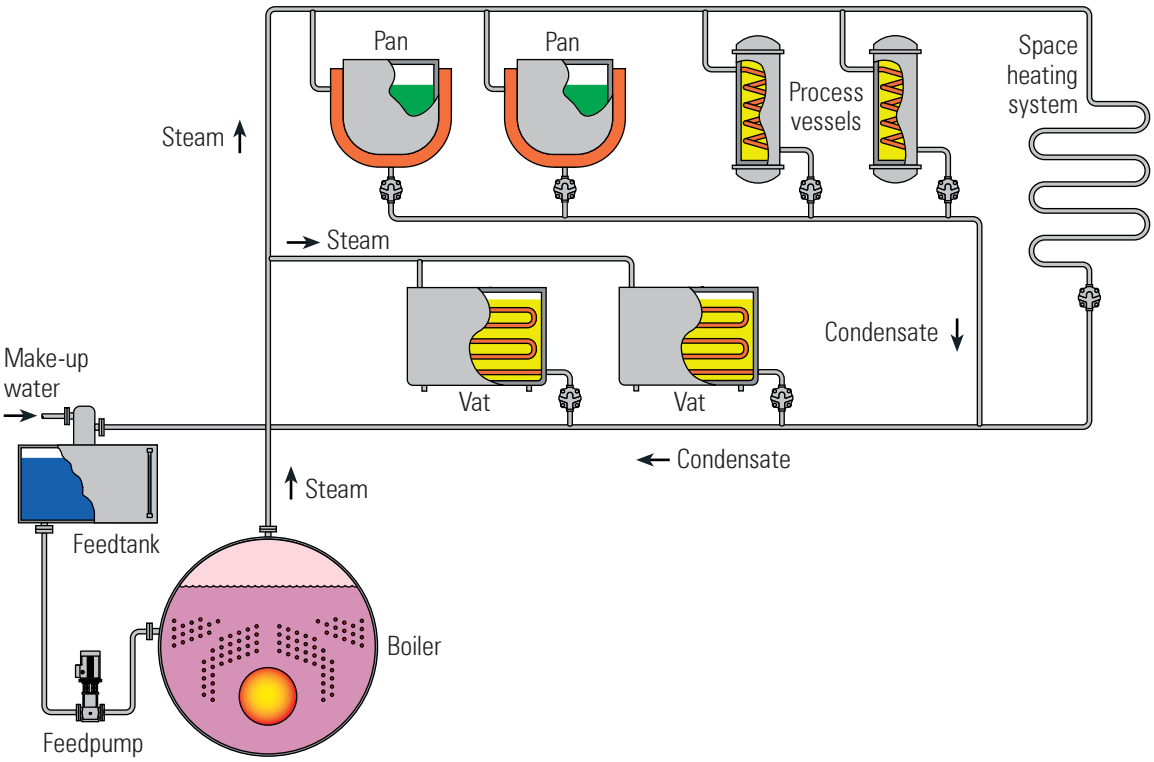


Fig. 14.1.4 A typical steam and condensate circuit

Why return condensate and reuse it?

Financial reasons

Condensate is a valuable resource and even the recovery of small quantities is often economically justifiable. The discharge from a single steam trap is often worth recovering.

Un-recovered condensate must be replaced in the boiler house by cold make-up water with additional costs of water treatment and fuel to heat the water from a lower temperature.

Water charges

Any condensate not returned needs to be replaced by make-up water, incurring further water charges from the local water supplier.

Effluent restrictions

In the UK for example, water above 43°C cannot be returned to the public sewer by law, because it is detrimental to the environment and may damage earthenware pipes. Condensate above this temperature must be cooled before it is discharged, which may incur extra energy costs. Similar restrictions apply in most countries, and effluent charges and fines may be imposed by water suppliers for non-compliance.

Maximising boiler output

Colder boiler feedwater will reduce the steaming rate of the boiler. The lower the feedwater temperature, the more heat, and thus fuel needed to heat the water, thereby leaving less heat to raise steam.

Boiler feedwater quality

Condensate is distilled water, which contains almost no total dissolved solids (TDS). Boilers need to be blown down to reduce their concentration of dissolved solids in the boiler water. Returning more condensate to the feedtank reduces the need for blowdown and thus reduces the energy lost from the boiler.

Summary of reasons for condensate recovery:

- Water charges are reduced.
- Effluent charges and possible cooling costs are reduced.
- Fuel costs are reduced.
- More steam can be produced from the boiler.
- Boiler blowdown is reduced - less energy is lost from the boiler.
- Chemical treatment of raw make-up water is reduced.

Figure 14.1.5 compares the amount of energy in a kilogram of steam and condensate at the same pressure. The percentage of energy in condensate to that in steam can vary from 18% at 1 bar g to 30% at 14 bar g; clearly the liquid condensate is worth reclaiming.

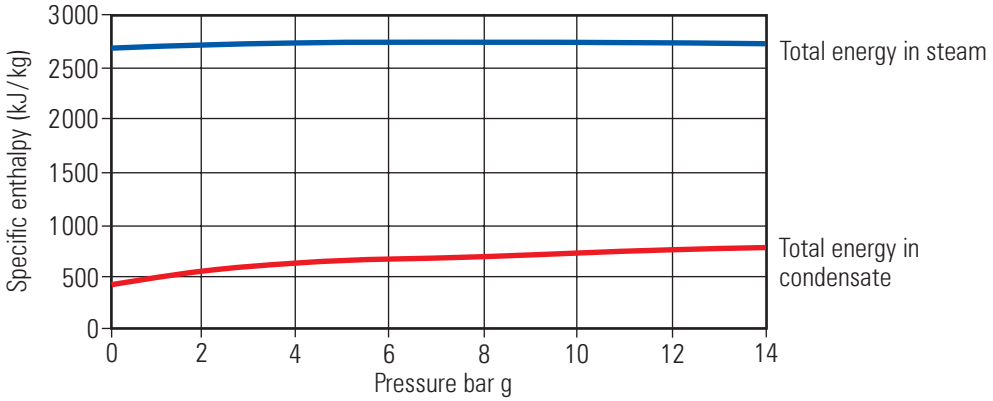


Fig. 14.1.5 Heat content of steam and condensate at the same pressures

The following example (Example 14.1.2) demonstrates the financial value of returning condensate.

Example 14.1.2

A boiler produces:
10 000 kg/h of steam 24 hours/day, 7 days/week and 50 weeks/year (8 400 hours/year).

Raw make-up water is at 10°C. Currently all condensate is discharged to waste at 90°C.

Raw water costs £0.61/m³, and effluent costs are £0.45/m³

The boiler is 85% efficient, and uses gas on an interruptible tariff charged at £0.01/kWh (£2.77/GJ).

Determine the annual value of returning the condensate

Part 1 - Determine the fuel cost
Each kilogram of condensate not returned to the boiler feedtank must be replaced by 1 kg of cold make-up water (10°C) that must be heated to the condensate temperature of 90°C. (ΔT = 80°C).

Calculate the heat required to increase the temperature of 1 kg of cold make-up water by 80°C, by using Equation 2.1.4.

$$Q = m\ c_p\ \Delta T \qquad \textbf{Equation 2.1.4}$$

Where:
Q = Quantity of energy (kJ)
m = Mass of the substance (kg)
c_p = Specific heat capacity of the substance (kJ/kg °C)
ΔT = Temperature rise of the substance (°C)

m is unity; ΔT is the difference between the cold water make-up and the temperature of returned condensate; c_p is the specific heat of water at 4.19 kJ/kg °C.

$$1\ kg\ x\ 4.19\ kJ/kg\ °C\ x\ 80°C\ =\ 335\ kJ/kg$$

Basing the calculations on an average evaporation rate of 10 000 kg/h, for a plant in operation 8 400 h/year, the energy required to replace the heat in the make-up water is:

$$10\ 000\ kg/h\ x\ 335\ kJ/kg\ x\ 8\ 400\ h/year\ =\ 28\ 140\ GJ/year$$

If the average boiler efficiency is 85%, the energy supplied to heat the make-up water is:

$$\frac{28\ 140\ GJ/year}{0.85}\ =\ 33\ 106\ GJ/year$$

With a fuel cost of £2.77/GJ, the value of the energy in the condensate is:

Annual fuel cost = 33 106 GJ/year x £2.77/GJ = **£91 704**

Part 2 - Determine the water cost
Water is sold by volume, and the density of water at normal ambient temperature is about 1 000 kg/m³. The total amount of water required in one year replacing non-returned condensate is therefore:

$$\frac{8400\ h\ x\ 10000\ kg/h}{1000\ kg/m^3}\ =\ 84000\ m^3/year$$

If water costs are £0.61 per m³, the annual water cost is:

Annual water cost = 84 000 m³/year x £0.61/m³ = **£51 240**

Part 3 - Determine the effluent cost
The condensate that was not recovered would have to be discharged to waste, and may also be charged by the water authority.

Total amount of water to waste in one year also equals 84 000 m³

If effluent costs are £0.45 per m³, the annual effluent cost is:

Annual effluent cost = 84 000 m³/year x £0.45/m³ = **£37 800**

Part 4 - Total value of condensate

The total annual value of 10 000 kg/h of condensate lost to waste is shown in Table 14.1.1:

Table 14.1.1 The potential value of returning condensate in Example 14.1.2

Fuel savings =	£ 91 704
Water savings =	£ 51 240
Effluent savings =	£ 37 800
Total value =	**£ 180 744**

On this basis, it follows that for each 1% of condensate returned per 10 000 kg/h evaporated as in Example 14.1.2, a saving of 1% of each of the values shown in Table 14.1.1 would be possible.

Example 14.1.3

If it were decided to invest £50 000 in a project to return 80% of the condensate in a similar plant to Example 14.1.2, but where the total evaporation rate were only 5 000 kg/h, the savings and simple payback term would be:

$$\text{Savings} = £180\,744 \times \frac{80}{100} \times \frac{5000}{10000}$$

$$\textbf{Savings} = \textbf{£72 297 / year}$$

$$\text{Payback} = \frac{£50000}{£72\,297 / \text{year}}$$

$$\textbf{Payback} = \textbf{0.69 year (36 weeks)}$$

This sample calculation does not include a value for savings due to correct TDS control and reduced blowdown, which will further reduce water losses and boiler chemical costs. These can vary substantially from location to location, but should always be considered in the final analysis. Clearly, when assessing condensate management for a specific project, such savings must be determined and included.

TDS control and water treatment have already been discussed in Block 3.

The routines outlined in Examples 14.1.2 and 14.1.3 may be developed to form the basis of a forced path calculation to assign a monetary value to projects intended to improve condensate recovery.

Equation 14.1.1 can be used to calculate the fuel savings per year:

$$\text{Fuel savings/year} = \frac{X\,A\,B\,C\,D}{E\,10^6} \qquad \textbf{Equation 14.1.1}$$

Where:

X = Expected improvement in condensate return expressed as a percentage between 1 and 100

A = Cost of fuel to provide 1 GJ of energy:

If gas on an interruptible tariff costs £0.01/kWh (1 kWh = 3.6 MJ)

$$\text{Cost of 1 GJ of energy} = \frac{£0.01}{3.6\ \text{MJ}} \times 1000 = £2.77$$

Similarly, if oil has a calorific value of 42 MJ/l, and costs £0.15/l

$$\text{Cost of 1 GJ of energy} = \frac{£0.15}{42\ \text{MJ}} \times 1000 = £3.57$$

B = Energy required per kilogram of make-up water to reach condensate temperature (kJ/kg).
This is determined by Q in Equation 2.1.4 ($Q = m\ c_p\ \Delta T$)

C = Average evaporation rate (kg/h)

D = Operational hours per year (h/year)

E = Boiler efficiency (%)

Savings in water costs can be determined using Equation 14.1.2:

$$\text{Savings in water costs /year} = \left(\frac{X\ C\ D}{10^5}\right) \times \text{Cost of water / m}^3 \qquad \textbf{Equation 14.1.2}$$

Savings in effluent costs can be determined using Equation 14.1.3:

$$\text{Savings in effluent costs /year} = \left(\frac{X\ C\ D}{10^5}\right) \times \text{Cost of effluent / m}^3 \qquad \textbf{Equation 14.1.3}$$

Where:
X = Expected improvement in condensate return expressed as a percentage between 1 and 100
C = Average evaporation rate (kg/h)
D = Operational hours per year (h/year)

Example 14.1.4

A major condensate management project costing £70 000 expects to recover an additional 35% of the condensate produced at a plant.

The average boiler steaming rate is 15 000 kg/h, and the plant operates for 8 000 h/year.

The fuel used is gas on a firm tariff of £0.011/kWh, and the boiler efficiency is estimated as 80%.

Make-up water temperature is 10°C and insulated condensate return lines ensure that condensate will arrive back at the boiler house at 95°C.

Consider the water costs to be £0.70/m³ and the total effluent costs to be £0.45/m³.

- Determine the payback period for the project.

Part 1 - Determine the fuel savings

Use Equation 14.1.1:

$$\text{Fuel savings/year} = \frac{X\ A\ B\ C\ D}{E\ 10^6} \qquad \textbf{Equation 14.1.1}$$

Where:
X = Expected improvement in condensate return = 35%

A = Cost of providing 1 GJ of energy $= \dfrac{£0.011}{3.6\ MJ} \times 1\ 000 = £3.055$

B = Energy required per kilogram of make-up water to reach condensate temperature (kJ/kg).
This is determined by Q in Equation 2.1.4 ($Q = m\ c_p\ \Delta T$)
$Q = m \times c_p \times \Delta T$
$Q = 1 \times 4.19 \times (95°C - 10°C)$
$Q = 356.15$ kJ/kg

B = Q in Equation 2.1.4 = 356.15kJ/kg
C = Average evaporation rate = 15 000 kg/h
D = Steaming hours per year = 8 000 h
E = Boiler efficiency = 80%

Substituting the values for X, A, B, C, D, and E into Equation 14.1.1

$$\text{Fuel savings/year} = £\frac{35 \times 3.055 \times 356.15 \times 15\,000 \times 8000}{80 \times 10^6}$$

Fuel savings/year = £57 122

Part 2 - Determine the water and effluent savings

Use Equation 14.1.2 to calculate the savings in water costs/year:

$$\text{Savings in water costs/year} = \left(\frac{X\ C\ D}{10^5}\right) \times \text{Cost of water / m}^3 \qquad \textbf{Equation 14.1.2}$$

Substituting values into Equation 14.1.2:

$$\text{Savings in water costs/year} = \left(\frac{35 \times 15\,000 \times 8\,000}{10^5}\right) \times £0.70 / \text{m}^3$$

Savings in water costs /year = £29 400

Use Equation 14.1.3 to calculate the savings in effluent costs/year:

$$\text{Savings in effluent costs/year} = \left(\frac{X\ C\ D}{10^5}\right) \times \text{Cost of effluent / m}^3 \qquad \textbf{Equation 14.1.3}$$

Substituting values into Equation 14.1.3:

$$\text{Savings in effluent costs/year} = \left(\frac{35 \times 15\,000 \times 8\,000}{10^5}\right) \times £0.45 / \text{m}^3$$

Savings in effluent costs /year = £18 900

Total water and effluent savings/year = £29 400 + £18 900

Total water and effluent savings/year = £48 300

Part 3 - Determine the payback period

Total savings = Fuel savings + Water and effluent savings

Total savings = £57 122 + £ 48 300

Total savings = £105 422/year

$$\text{Simple payback (years)} = \frac{\text{Cost of project}}{\text{Annual savings}}$$

$$\text{Simple payback (years)} = \frac{£70\,000}{£105\,442}$$

Simple payback (years) = 0.66 year (35 weeks)

Questions

1. When 10 kg of steam condenses at 0 bar g, how much condensate is produced?

a | 10 kg ☐
b | 1.5 kg ☐
c | 10% of the mass of the steam ☐
d | 10% of the volume of the steam ☐

2. 10 kg of steam condenses at 14 bar g. What proportion of the total heat in the steam is held in the condensate?

a | 5% ☐
b | 10% ☐
c | 20% ☐
d | 30% ☐

3. A boiler produces 1 000 kg/h of steam at 7 bar g, but none of the condensate is recovered. Approximately at what rate is energy being wasted? (Steam tables are required).

a | 20 kW ☐
b | 40 kW ☐
c | 200 kW ☐
d | 1 000 kW ☐

4. If, in Question 3, it is proposed that 50% of the wasted condensate is to be returned to the boiler feedtank at 90°C, and the fuel cost is £3/GJ, the cold water make-up temperature is 15°C, the water make-up temperature is 15°C, and the water/effluent costs are £0.8/m³, what are the potential total annual condensate savings if the boiler steams at 85% efficiency for 4 000 hours per year?

a | £1 500 ☐
b | £2 218 ☐
c | £10 100 ☐
d | £500 ☐

5. If in Question 4, the cost of this project were £2 000, what would be the simple payback term?

a | 3 weeks ☐
b | 33 weeks ☐
c | 18 months ☐
d | 47 weeks ☐

Answers

1: a, 2: d, 3: c, 4: b, 5: d

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Module 14.2

Layout of Condensate Return Lines

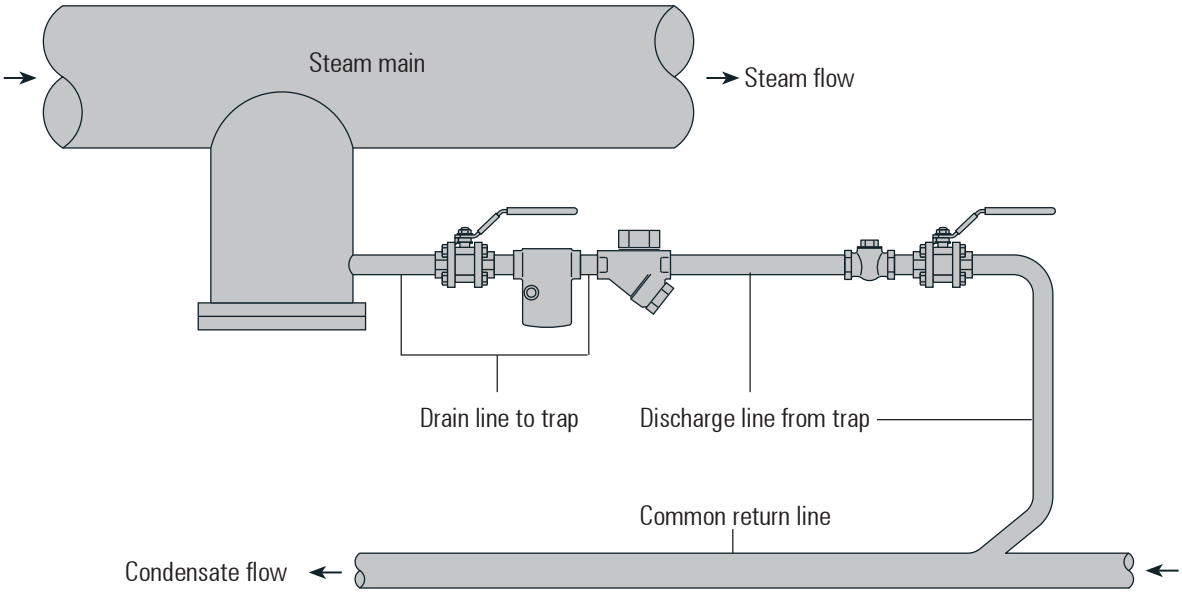
Layout of Condensate Return Lines

No single set of recommendations can cover the layout of condensate pipework. Much depends on the application pressure, the steam trap characteristics, the position of the condensate return main relative to the plant, and the pressure in the condensate return main. For this reason it is best to start by considering what has to be achieved, and to design a layout which will ensure that basic good practice is met.

The prime objectives are that:

- Condensate must not be allowed to accumulate in the plant, unless the steam using apparatus is specifically designed to operate in this way. Generally apparatus is designed to operate non-flooded, and where this is the case, accumulated condensate will inhibit performance, and encourage the corrosion of pipes, fittings and equipment.
- Condensate must not be allowed to accumulate in the steam main. Here it can be picked up by high velocity steam, leading to erosion and waterhammer in the pipework.

The subject of condensate piping will divide naturally into four basic types where the requirements and considerations of each will differ. These four basic types are defined and illustrated in Figure 14.2.1.



Type of condensate line	Condensate line is sized to carry the folllowing:
Drain line to trap	Condensate
Discharge line from trap	Flash steam
Common return line	Flash steam
Pumped return line (not shown)	Condensate

Fig. 14.2.1 A steam main trap set discharging condensate into a common return line

Drain lines to steam traps

In the drain line, the condensate and any incondensable gases must flow from the drain outlet of the plant to the steam trap.

In a properly sized drain line, the plant being drained and the body of the steam trap are virtually at the same pressure and, because of this, condensate does not flash in this line. Gravity is the driving force and is relied upon to induce flow along the pipe. For this reason, it makes sense for the trap to be situated below the outlet of the plant being drained, and the trap discharge pipe to terminate below the trap. (An exception to this is the tank heating coils discussed in Module 2.10).

The type of steam trap used (thermostatic, thermodynamic or mechanical) can affect the piping layout.

Thermostatic steam traps

Thermostatic traps will cool condensate below saturation temperature before discharging. This effectively waterlogs the drain line, often allowing condensate to back-up and flood the plant.

There are some applications where the sub-cooling of condensate has significant advantages and is encouraged. Less flash steam is produced in the trap discharge line, and the introduction of condensate into the condensate main is gentler.

Thermostatic traps discharging via open-ended pipework will waste less energy than mechanical traps because more of the sensible heat in the waterlogged condensate imparts its heat to the process; a typical example is that of a steam tracer line.

Thermostatic traps should not be used to drain steam mains or heat exchangers, unless proper consideration is given to a longer and/or larger drain line to act as a reservoir and dissipate heat to atmosphere. The extra length (or larger diameter) of drain line required to do this is usually impractical, as shown in Example 14.2.1.

Example 14.2.1

A 30 kW air heater is to be fitted with a DN15 thermostatic steam trap, which releases condensate at 13°C below saturation temperature. The normal working pressure is 3 bar g, the ambient temperature is 15°C, and the heat loss from the drain line to the environment is estimated to be 20 W/m² °C.

Determine the minimum required length of 15 mm drain line to the thermostatic trap.

From steam tables, at 3 bar g:

Saturation temperature of steam = 144°C

Trap discharge temperature = 144 - 13°C = 131°C

Enthalpy of evaporation (h_{fg}) = 2133.24 kJ/kg

Equation 2.8.1 can be used to calculate the steam flow from the heat load:

$$\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW x 3600}}{h_{fg} \text{ at operating pressure}} \qquad \textbf{Equation 2.8.1}$$

$$\text{Steam flowrate (kg/h)} = \frac{\text{Heat load (kW) x 3600 (s/h)}}{h_{fg} \text{ at operating pressure (kJ/kg)}}$$

$$\text{Steam flowrate (kg/h)} = \frac{30 \times 3600}{2133.24}$$

Steam flowrate = 50.6 kg/h (= 0.0141 kg/s)

As the trap discharges at 131°C, the drain line has to emit enough heat such that the condensate at the heater outlet is at saturation temperature, and that condensate will not back-up into the heater. The required heat loss from the drain line can be calculated from Equation 2.6.5.

$$\dot{Q} = \dot{m}\ c_p\ \Delta T \qquad \textbf{Equation 2.6.5}$$

Where:
$\dot{Q}$ = Mean heat transfer rate (kW)
$\dot{m}$ = Mean secondary fluid flowrate (kg/s)
c_p = Specific heat capacity of the secondary fluid (kJ/kg K) or (kJ/kg °C) = 4.19 for water
ΔT = Temperature rise of the secondary fluid (K or °C)

ΔT in Equation 2.6.5 is the required temperature drop along the drain line of 13°C.

$$\dot{Q} = 0.014\ 1\ \text{kg/s} \times 4.19\ \text{kJ/kg °C} \times 13\text{°C}$$

$$\dot{Q} = \mathbf{0.768\ kW}$$

This heat loss will be achieved from the mean condensate temperature along the drain line.

Mean condensate temperature in the drain line $= \dfrac{144 + 131}{2} = 137.5\text{°C}$

The surface area of the drain line to provide the required heat loss can be calculated using Equation 2.5.3.

$$\dot{Q} = U\ A\ \Delta T \qquad \textbf{Equation 2.5.3}$$

Where:
$\dot{Q}$ = Heat transferred per unit time (W (J/s))
U = Overall heat transfer coefficient (W/m² K or W/m² °C)
A = Heat transfer area (m²)
ΔT = Temperature difference between the primary and secondary fluid (K or °C)
Note:
$\dot{Q}$ will be a mean heat transfer rate ($\dot{Q}_M$) if ΔT is a mean temperature difference (ΔT_{LM} or ΔT_{AM}).

ΔT in Equation 2.5.3 is the difference between the mean condensate temperature and the ambient temperature = 137.5°C - 15°C = 122.5°C

$\dot{Q}$ = 0.768 kW

U = 20 W/m² °C

From Equation 2.5.3:

$$0.768 \times 10^3\ \text{watts} = 20\ \text{watts/m}^2\text{°C} \times A \times 122.5\text{°C}$$

$$\text{Therefore, } A = 0.313\ \text{m}^2$$

The length of pipe required to provide this surface area can be calculated using information from Table 2.10.3.

Table 2.10.3 Nominal surface areas of steel pipes per metre length

Nominal bore mm	15	20	25	32	40	50	65	80	100
Surface area (m²/m)	0.067	0.085	0.106	0.134	0.152	0.189	0.239	0.279	0.358

The surface area of 15 mm pipe = 0.067 m²/m

$$\text{Therefore, the length of drain line} = \frac{0.313\ \text{m}^2}{0.067\ \text{m}^2/\text{m}}$$

Minimum length of drain line = 4.7 m (for Example 14.2.1)

This length of pipe (4.7 m) is probably impractical in the field. Two alternatives remain. One is to increase the diameter of the drain line, which is still usually impractical; the other is much simpler, to fit the correct trap for this type of application; a float-thermostatic trap which discharges condensate at steam temperature and hence requires no cooling leg.

Should a thermostatic trap be considered essential, and fitted no more than 2 metres away from the heater outlet, it would be necessary to calculate the required diameter of drain line. The heat loss required from the pipe remains the same, along with the total surface area of the pipe, but the surface area per metre length must increase.

$$\text{The surface area required/metre length} = \frac{0.313 \text{ m}^2}{2 \text{ m}}$$

$$\textbf{The surface area required/metre length} = \textbf{0.157 m}^2\textbf{/m}$$

From Table 2.10.3, it can be seen that the minimum sized pipe to give this area per metre is a 50 mm pipe, which, again, may be construed as being impractical and expensive to fabricate.

The moral of this is that it is usually easier and cheaper to select the correct trap for the job, than have the wrong type of trap and fabricate a solution around it.

Thermodynamic steam traps

Traps that discharge intermittently, such as thermodynamic traps, will accumulate condensate between discharges. However, they are extremely robust, will tolerate freezing ambient temperatures and have a relatively small outer surface area, meaning that heat loss to the environment is minimised. They are not suitable for discharging condensate into flooded return lines, as will be explained later in this Block.

Mechanical steam traps

Mechanical steam traps with a continuous discharge characteristic, for example float-thermostatic traps, often prove to be the best option, and have the additional advantage of being able to vent air.

Most float traps are available in two basic flow configurations, either horizontal or vertical flow through the trap. Some inverted bucket traps have bottom inlet and top outlet connections. Clearly, the trap connections will affect the path of connecting pipework.

The drain line should be kept to a minimum length, ideally less than 2 metres. Long drain lines from the plant to the steam trap can fill with steam and prevent condensate reaching the trap. This effect is termed steam locking. To minimise this risk, drain lines should be kept short (see Figure 14.2.2). In situations where long drain lines are unavoidable, the steam locking problem may be overcome using float traps with steam lock release devices. The problem of steam locking should be tackled by fitting the correct length of pipe in the first place, if possible.

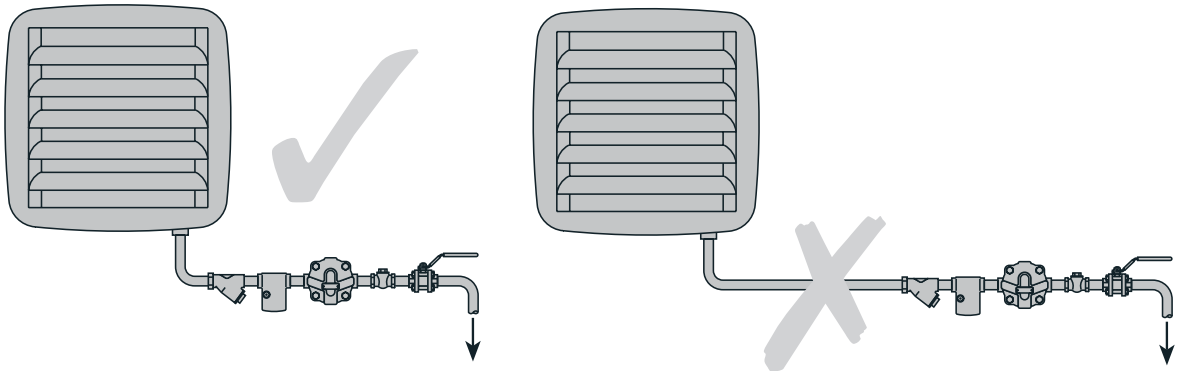


Fig. 14.2.2 Keep drain lines short

The detailed arrangements for trapping steam-using plant and steam mains drainage are different as is explained in the following paragraphs.

With steam-using plant, the pipe from the condensate connection should fall vertically for about 10 pipe diameters to the steam trap. Assuming a correctly sized ball float trap is installed, this will ensure that surges of condensate do not accumulate in the bottom of the plant with its attendant risks of corrosion and waterhammer. It will also provide a small amount of static head to help remove condensate during start-up when the steam pressure might be very low. The pipework should then run horizontally, with a fall in the direction of flow to ensure that condensate flows freely (see Figure 14.2.3).

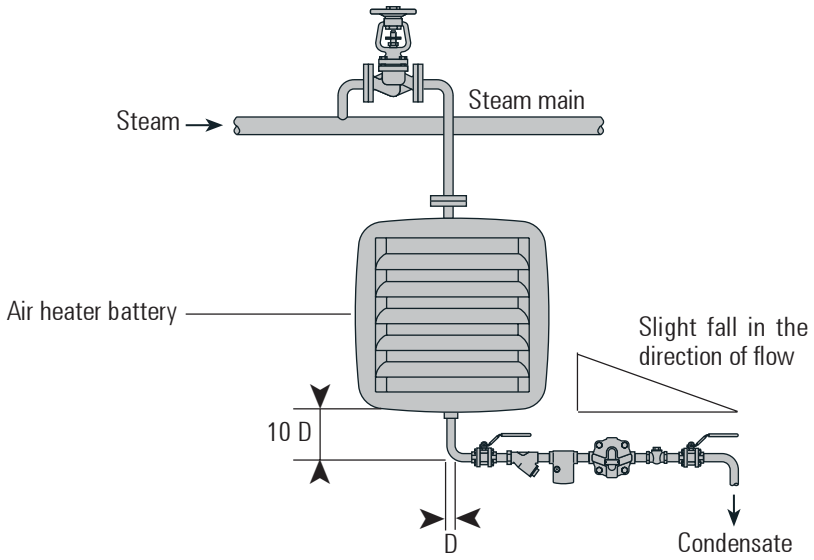
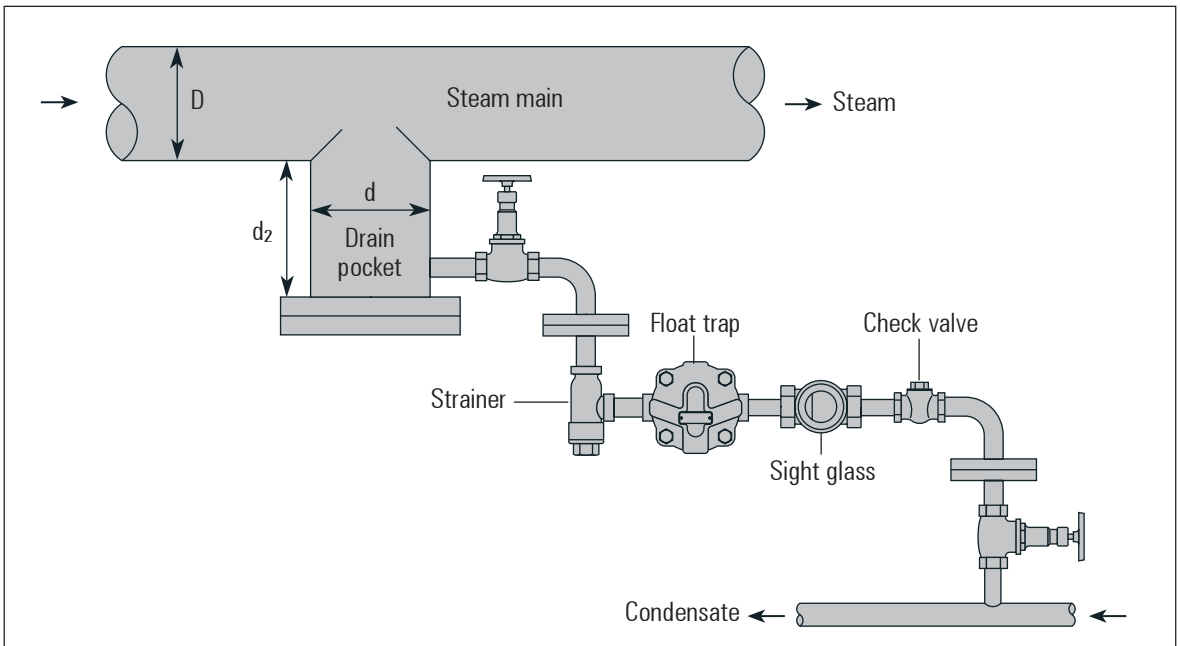


Fig. 14.2.3 Ideal arrangement when draining a steam plant

With steam mains drainage, provided drain pockets are installed as recommended in Module 10.3, then the drain line between the pocket and the steam trap may be horizontal. If the drain pocket is not as deep as the recommendation, then the steam trap should be fitted an equivalent distance below it (see Figure 14.2.4).



Main diameter D	Pocket diameter d_1	Pocket depth d_2
Up to 100 mm	$d_1 = D$	Minimum $d_2 = 100$ mm
125 mm - 200 mm	$d_1 = 100$ mm	Minimum $d_2 = 150$ mm
250 mm and above	$d_1 = D/2$	Minimum $d_2 = D$

Fig. 14.2.4 Ideal arrangement when draining a steam main

Discharge lines from traps

These pipes will carry condensate, incondensable gases, and flash steam from the trap to the condensate return system (Figure 14.2.5). Flash steam is formed as the condensate is discharged from the high-pressure space before the steam trap to the lower pressure space of the condensate return system. (Flash steam is discussed briefly in Module 14.1, and in more detail in Module 2.2).

These lines should also fall in the direction of flow to maintain free flow of condensate. On shorter lines, the fall should be discernible by sight. On longer lines, the fall should be about 1:70, that is, 100 mm every 7 metres.

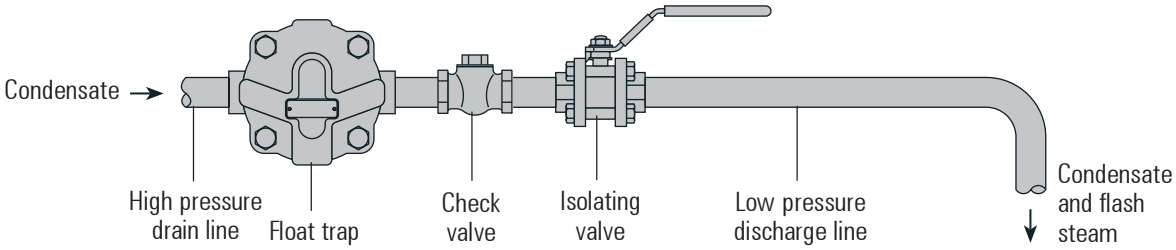


Fig. 14.2.5 Trap discharge lines pass condensate, flash and incondensibles

Discharging into flooded return lines

Discharging traps into flooded return lines is not recommended, especially with blast action traps (thermodynamic or inverted bucket types), which remove condensate at saturation temperature.

Good examples of flooded condensate mains are pumped return lines and rising condensate lines. They often follow the same route as steam lines, and it is tempting to simply connect mains drainage steam trap discharge lines into them.

However, the high volume of flash steam released into long flooded lines will violently push the water along the pipe, causing waterhammer, noise and, in time, mechanical failure of the pipe.

Common return lines

Where condensate from more than one trap flows to the same collecting point such as a vented receiver, it is usual to run a common line into which individual trap discharge lines are connected. Provided the layouts as featured in Figures 14.2.6/7/8 and 10 are observed, and the pipework is adequately sized as indicated in Module 14.3, this is not a problem.

Blast discharge traps

If blast discharge traps (thermodynamic or inverted bucket types) are used, the reactionary forces and velocities can be high. Swept tees will help to reduce mechanical stress and erosion at the point where the discharge line joins the common return line (see Figure 14.2.6).

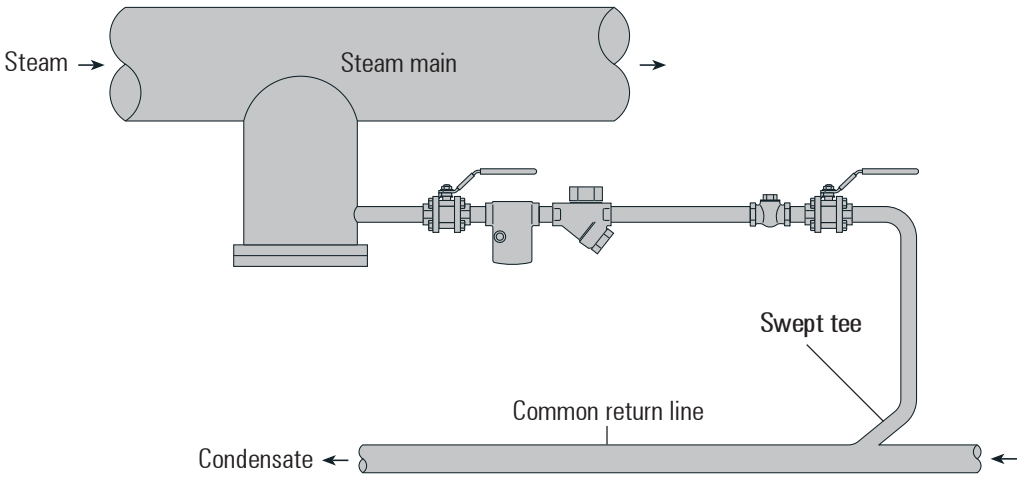


Fig. 14.2.6 A swept tee connection

Continuous discharge traps

If, for some reason, swept tees cannot be used, a float-thermostatic trap with its continuous discharge action is a better option (Figure 14.2.7). The flooded line will absorb the dissipated energy from the (relatively small) continuous flow from the float-thermostatic trap, more easily.

If the pressure difference between the steam and condensate mains is very high, then a diffuser will help to cushion the discharge, reducing both erosion and noise.

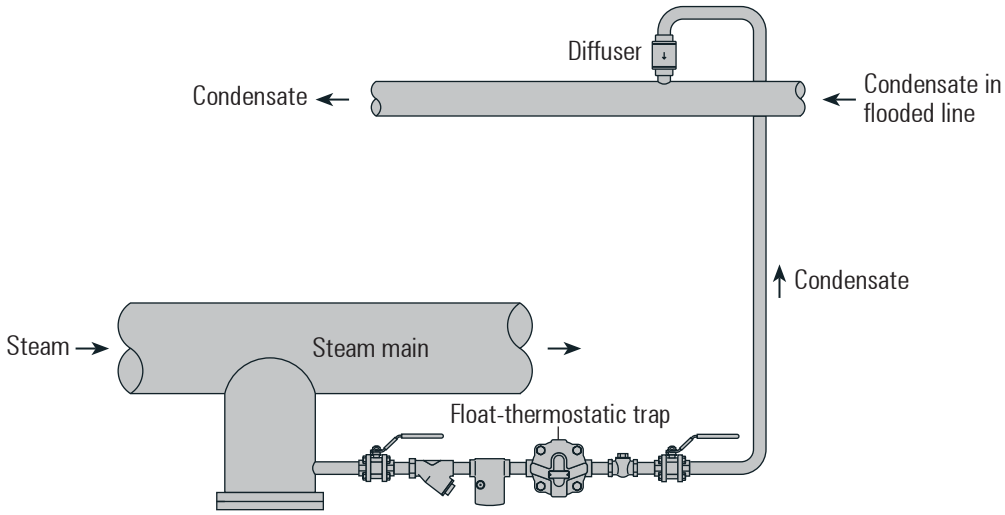


Fig. 14.2.7 Float trap with a diffuser into a flooded line

Another alternative is to use a thermostatic trap that holds back condensate until it cools below the steam saturation temperature; this reduces the amount of flash steam formed (Figure 14.2.8).

To avoid waterlogging the steam main, the use of a generous collecting pocket on the main, plus a cooling leg of 2 to 3 m of unlagged pipe to the trap is essential. The cooling leg stores condensate while it is cooling to the discharge temperature.

If there is any danger of waterlogging the steam main, thermostatic traps should not be used.

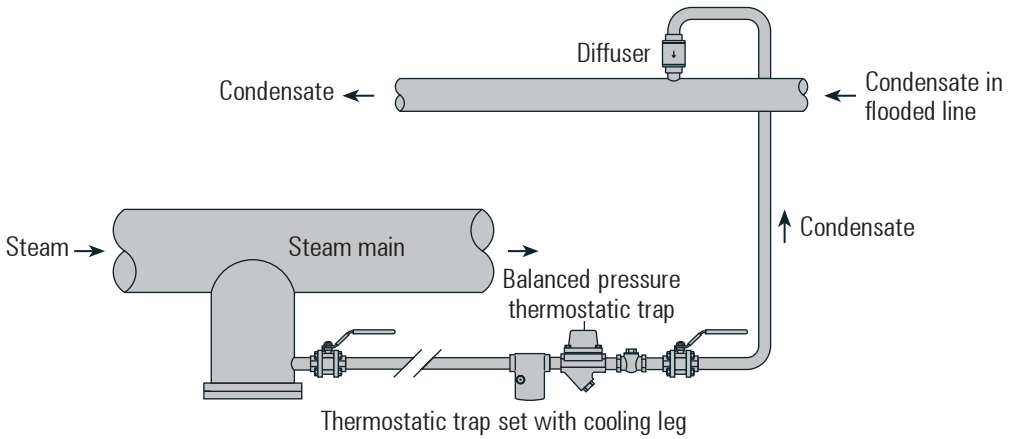


Fig. 14.2.8 Balanced pressure thermostatic trap with cooling leg into a flooded line

Temperature controlled plant with steam traps draining into flooded lines

Processes using temperature control provide an example where the supply steam pressure is throttled across a control valve. The effect of this is to reduce steam trap capacity to a point where the condensate flow can stop completely, and the system is said to have stalled. The subject of stall is discussed in greater depth in Block 13.

Stall occurs as a result of insufficient steam pressure to purge the steam plant of condensate, and is more likely when the plant has a high turndown from full-load to part load.

Not all temperature controlled systems will stall, but the backpressure caused by the condensate system could have an adverse effect on the performance of the trap. This in turn, might impair the heat transfer capability of the process (Figure 14.2.9).

Condensate drain lines should, therefore, be configured so that condensate cannot flood the main into which they are draining as depicted in Figure 14.2.10.

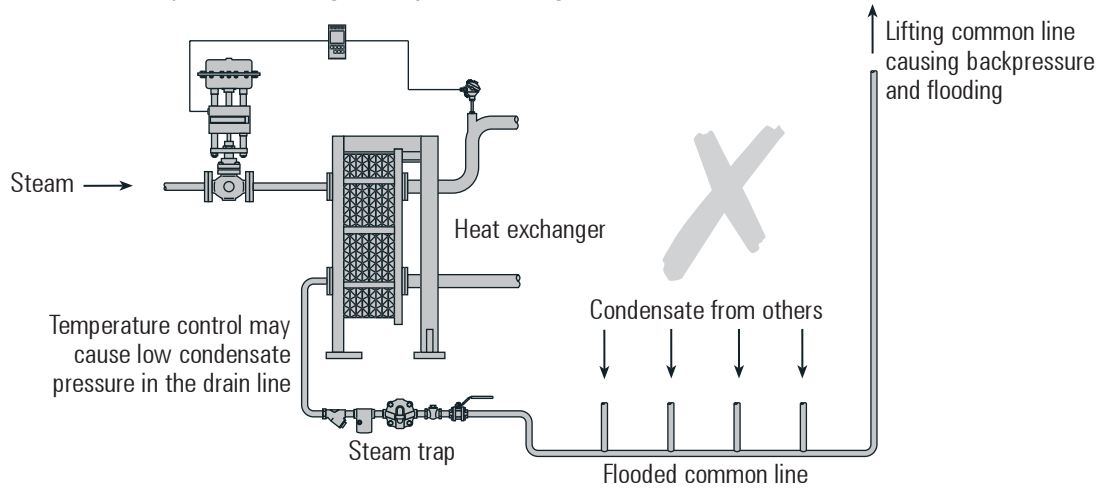


Fig. 14.2.9 Discharge from steam traps on temperature controlled equipment into flooded lines should be avoided if possible

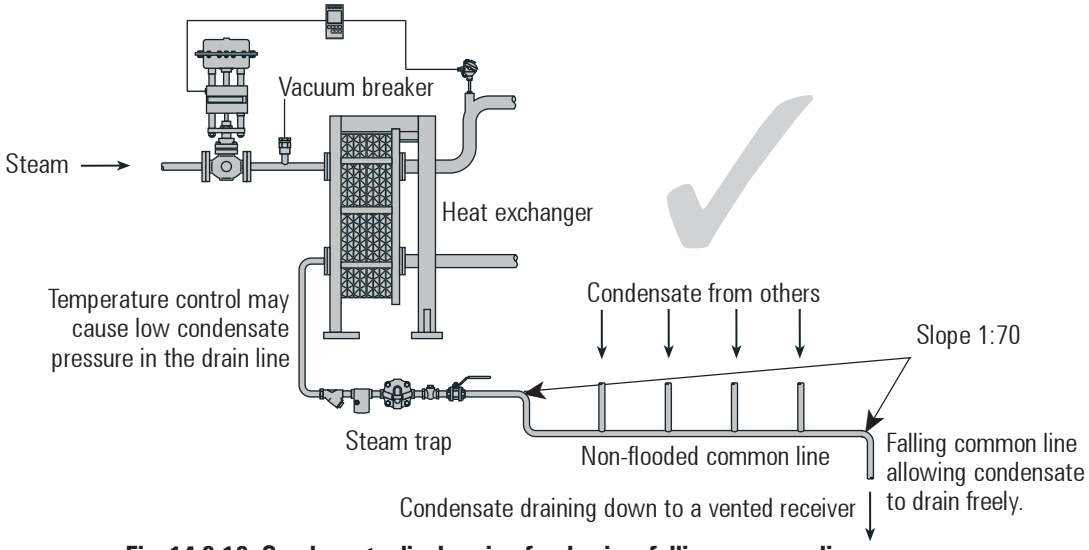


Fig. 14.2.10 Condensate discharging freely via a falling common line

Discharge lines at different pressures

Condensate from more than one temperature controlled process may join a common line, as long as this line is:

- Designed to slope in the direction of flow to a collection point.
- Sized to cater for the cumulative effects of any flash steam from each of the branch lines at full-load.

The concept of connecting the discharges from traps at different pressures is sometimes misunderstood.

If the branch lines and the common line are correctly sized, the pressures downstream of each trap will be virtually the same. However, if these lines are undersized, the flow of condensate and flash steam will be restricted, due to a build up of backpressure caused by an increased resistance to flow within the pipe. Condensate flowing from traps draining the lower pressure systems will tend to be the more restricted.

Each part of the discharge piping system should be sized to carry any flash steam present at acceptable steam velocities. The discharge from a high-pressure trap will not interfere with that from a low-pressure trap if the discharge lines and common line are properly sized and sloped in the direction of flow. Module 14.3, 'Sizing of condensate return lines' gives further details.

Pumped return lines

Flash steam may, at some point, be separated from the condensate and used in a recovery system, or simply vented to atmosphere from a suitable receiver (Figure 14.2.11). The residual hot condensate from the latter can be pumped on to a suitable collecting tank such as a boiler feedtank. When the pump is served from a vented receiver, the pumped return line will be fully flooded with condensate at temperatures below 100°C, which means flash steam is less likely to occur in the line.

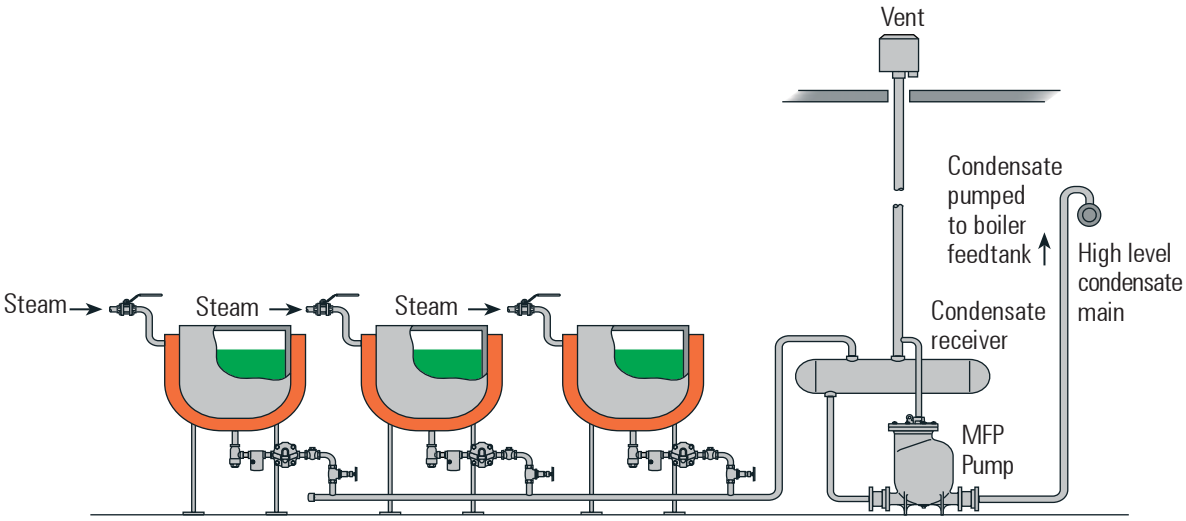


Fig. 14.2.11 Condensate recovery from a vented receiver

Flow in a pumped return line is intermittent, as the pump starts and stops according to its needs. The pump discharge rate will be higher than the rate at which condensate enters the pump. It is, therefore, the pump discharge rate which determines the size of the pump discharge line, and not the rate at which condensate enters the pump.

The pumping of condensate is discussed in further detail in Module 14.4, 'Pumping condensate from vented receivers'.

Questions

1. How many different basic types of condensate lines are there?

a| One ☐
b| Two ☐
c| Three ☐
d| Four ☐

2. Why are thermostatic traps not recommended for draining steam mains?

a| They tend to waterlog the drain line ☐
b| They tend to waterlog the process ☐
c| Long drain lines are necessary to cool the condensate ☐
d| All of the above ☐

3. When might a thermostatic trap be used to drain a steam main?

a| When it is fitted to a correctly sized drain pocket ☐
b| When the difference in pressure between the steam and condensate is high ☐
c| When it is fitted with a cooling leg and draining into a flooded main ☐
d| Never ☐

4. When are thermodynamic traps not recommended for draining steam mains?

a| They are not intended to drain steam mains ☐
b| When draining into flooded condensate lines ☐
c| When fitted outside and there is a danger of freezing ☐
d| When fitted to large drain pockets ☐

5. What will a trap discharge line normally carry that a drain line does not?

a| The weight of the trap ☐
b| Live steam ☐
c| A mixture of live steam and condensate ☐
d| A mixture of flash steam and condensate ☐

6. Upon which criterion is a pump discharge line sized?

a| The condensate discharge rate from the pump ☐
b| The pump filling rate ☐
c| The size of the pump outlet ☐
d| The height of the process above the top of the pump ☐

Answers

1: d, 2: d, 3: c, 4: b, 5: d, 6: a

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Module 14.3

Sizing Condensate Return Lines

Sizing Condensate Lines

The four main types of condensate line, as mentioned in Module 14.2, are shown in Table 14.3.1:

Table 14.3.1 The four basic types of condensate line

Type of condensate line	Condensate line is sized to carry the following
Drain lines to trap	Condensate
Discharge lines from traps	Flash steam
Common return lines	Flash steam
Pumped return lines	Condensate

Sizing of all condensate lines is a function of:

- **Pressure -** The difference in pressure between one end of the pipe and the other. This pressure difference may either promote flow, or cause some of the condensate to flash to steam.
- **Quantity -** The amount of condensate to be handled.
- **Condition -** Is the condensate predominately liquid or flash steam?

With the exception of pumped return lines which will be discussed in Module 14.4, the other three main types of condensate line and their sizing, will be covered in this Module.

Sizing drain lines to traps

It should not be assumed that the drain line (and trap) should be the same size as the plant outlet connection. The plant may operate at a number of different operating pressures and flowrates, especially when it is temperature controlled. However, once the trap has been correctly sized, it is usually the case that the drain line will be the same size as the trap inlet connection, (see Figure 14.3.1).

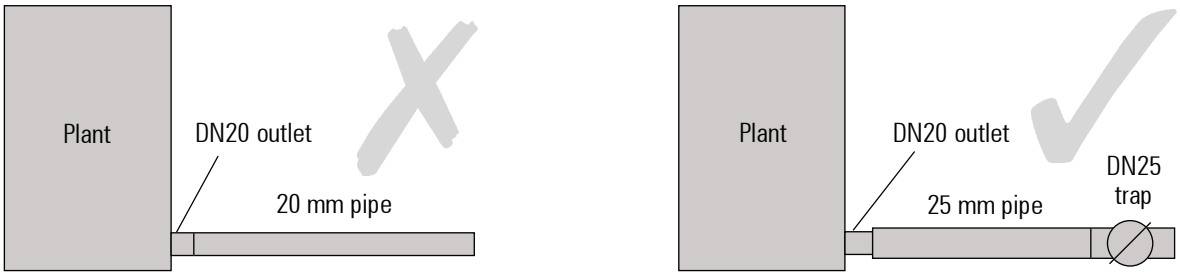


Fig. 14.3.1 The drain line should not be sized on the plant connection

Regarding the conditions inside the drain line, as there is no significant pressure drop between the plant and the trap, no flash steam is present in the pipe, and it can be sized to carry condensate only.

When sizing the drain line, the following will need consideration:

- The condensing rate of the equipment being drained during full-load.
- The condensing rate of the equipment at start-up.

  At plant start-up, the condensing rate can be up to three times the running load – this is where the temperature difference between the steam and colder product is at its maximum.

  The drain line, trap, and discharge line also have to carry the air that is displaced by the incoming steam during this time.

The sizing routine for the steam trap will have to consider both of these variables, however, in general:

- For steam mains drainage, the condensate load for each drain trap is typically 1% of the steam capacity of the main based on drain points at 50 m intervals, and with good insulation.

  For most drain points, sizing the trap to pass twice the running load at the working pressure (minus any backpressure) will allow it to cope with the start-up load.

- On constant steam pressure processes such as presses, ironers, unit heaters, radiant panels and boiling pans, sizing the traps on approximately twice the running load at the working pressure (less any backpressure) will provide sufficient capacity to cope with the start-up load.
- On temperature controlled applications, the steam pressure, the plant turndown, the set temperature and steam trap location need to be considered in detail, and the trap needs to be sized to cater for both the full and minimum load conditions. If these conditions are not known it is recommended that the steam trap be sized on 3 x the running load at the running differential pressure. This should satisfy the start-up condition and provide proper drainage at minimum loads.

  When the trap is sized in this way, it will also cater for the start-up load. Consequently, if the drain line to the trap is sized on the trap size, it will never be undersized.

For practical purposes, where the drain line is less than 10 m, it can be the same pipe size as the steam trap selected for the application. Drain lines less than 10 m long can also be checked against Appendix 14.3.1 and a pipe size should be selected which results in a pressure loss at maximum flowrate of not more than 200 Pa per metre length, and a velocity not greater than 1.5 m/s. Table 14.3.2 is an extract from Appendix 14.3.1.

On longer drain lines (over 10 m), the pressure loss at maximum flowrate should not be more than 100 Pa/m, and a velocity not greater than 1 m/s.

Table 14.3.2 Flow of water in heavy steel pipes

Flowrate		Capacity kg/h									
Pipe size Ø		15 mm	20 mm	25 mm	32 mm	40 mm	50 mm	65 mm	80 mm	100 mm	
Pa/m	mbar/m	<0.15 m/s			0.15 m/s					0.3 m/s	
90.0	0.900	173	403	745	1627	2488	4716	9612	14940	30240	
92.5	0.925	176	407	756	1652	2524	4788	9756	15156	30672	
95.0	0.950	176	414	767	1678	2560	4860	9900	15372	31104	
97.5	0.975	180	421	778	1699	2596	4932	10044	15552	31500	1.0 m/s
100.0	1.000	184	425	788	1724	2632	5004	10152	15768	31932	
120.0	1.200	202	472	871	1897	2898	5508	11196	17352	35100	
140.0	1.400	220	511	943	2059	3143	5976	12132	18792	38160	
160.0	1.600	234	547	1015	2210	3373	6408	12996	20160	40680	
180.0	1.800	252	583	1080	2354	3589	6804	13824	21420	43200	
200.0	2.000	266	619	1141	2488	3780	7200	14580	22644	45720	
220.0	2.200	281	652	1202	2617	3996	7560	15336	23760	47880	
240.0	2.400	288	680	1256	2740	4176	7920	16056	24876	50400	1.5 m/s
260.0	2.600	306	713	1310	2855	4356	8244	16740	25920	52200	
280.0	2.800	317	742	1364	2970	4536	8568	17388	26928	54360	
300.0	3.000	331	767	1415	3078	4680	8892	18000	27900	56160	

Example 14.3.1

An item of plant, using steam at constant pressure, condenses 470 kg of steam an hour at full-load. The pipework between the plant item and the steam trap has an equivalent length of 2 m.

Determine the size of pipe to be used.

Revised load allowing for start-up = 470 kg/h x 2 = 940 kg/h.

As the pipe length is less than 10 metres, the maximum allowable pressure drop is 200 Pa/m.

Using Table 14.3.1, by looking across from 200 Pa/m it can be seen that a 25 mm pipe has a capacity of 1 141 kg/h, and would therefore be suitable for the expected starting load of 940 kg/h.

Checking further up the 25 mm column, it can be seen that a flowrate of 940 kg/h will incur an actual pressure drop of just less than 140 Pa/m flowing through a 25 mm pipe.

Sizing discharge lines from traps

The section of pipeline downstream of the trap will carry both condensate and flash steam at the same pressure and temperature. This is referred to as two-phase flow, and the mixture of liquid and vapour will have the characteristics of both steam and water in proportion to how much of each is present. Consider the following example.

Example 14.3.2

An item of plant uses steam at a constant 4 bar g pressure. A mechanical steam trap is fitted, and condensate at saturation temperature is discharged into a condensate main working at 0.5 bar g.

Determine the proportions by mass, and by volume, of water and steam in the condensate main.

Part 1 - Determine the proportions by mass

From steam tables:

At 4.0 bar g h_f = 640.7 kJ/kg
At 0.5 bar g h_f = 464.1 kJ/kg h_{fg} = 2 225.6 kJ/kg

Equation 2.2.5 is used to determine the proportion of flash steam:

$$\text{Proportion of flash steam} = \frac{(h_f \text{ at } P_1) - (h_f \text{ at } P_2)}{h_{fg} \text{ at } P_2} \qquad \textbf{Equation 2.2.5}$$

Where:
P_1 = Initial pressure
P_2 = Final pressure
h_f = Specific liquid enthalpy (kJ/kg)
h_{fg} = Specific enthalpy of evaporation (kJ/kg)

$$\text{Proportion of flash steam} = \frac{(640.7 - 464.1)}{2225.6} \times \frac{100}{1} = 7.9\%$$

Clearly, if 7.9% is flashing to steam, the remaining 100 – 7.9 = 92.1% of the initial mass flow will remain as water.

Part 2 - Determine the proportions by volume

Based on an initial mass of 1 kg of condensate discharged at 4 bar g saturation temperature, the mass of flash steam is 0.079 kg and the mass of condensate is 0.921 kg (established from Part 1).

Water:

The density of saturated water at 0.5 bar g is 950 kg/m³,

and the volume occupied by 0.921 kg $= \frac{0.921}{950} = 0.001$ m³

Steam:

From steam tables, specific volume (v_g) of steam at 0.5 bar g = 1.15 m³/kg
The volume occupied by the steam is 0.079 kg x 1.15 m³/kg = 0.091 m³
The total volume occupied by the steam and condensate mixture is:
0.001 m³ (water) + 0.091 m³ (steam) = 0.092 m³

By proportion (%):

$$\textbf{The water occupies} = \frac{0.001}{0.092} \times \frac{100}{1} = \textbf{1\% space}$$

$$\textbf{The steam occupies} = \frac{0.091}{0.092} \times \frac{100}{1} = \textbf{99\% space}$$

From this, it follows that the two-phase fluid in the trap discharge line will have much more in common with steam than water, and it is sensible to size on reasonable steam velocities rather than use the relatively small volume of condensate as the basis for calculation. If lines are undersized, the flash steam velocity and backpressure will increase, which can cause waterhammer, reduce the trap capacity, and flood the process.

Steam lines are sized with attention to maximum velocities. Dry saturated steam should travel no faster than 40 m/s. Wet steam should travel somewhat slower (15 to 20 m/s) as it carries moisture which can otherwise have an erosive and damaging effect on fittings and valves.

Trap discharge lines can be regarded as steam lines carrying very wet steam, and should be sized on similarly low velocities.

Condensate discharge lines from traps are notoriously more difficult to size than steam lines due to the two-phase flow characteristic. In practice, it is impossible (and often unnecessary) to determine the exact condition of the fluid inside the pipe.

Although the amount of flash steam produced (see Figure 14.3.2) is related to the pressure difference across the trap, other factors will also have an effect.

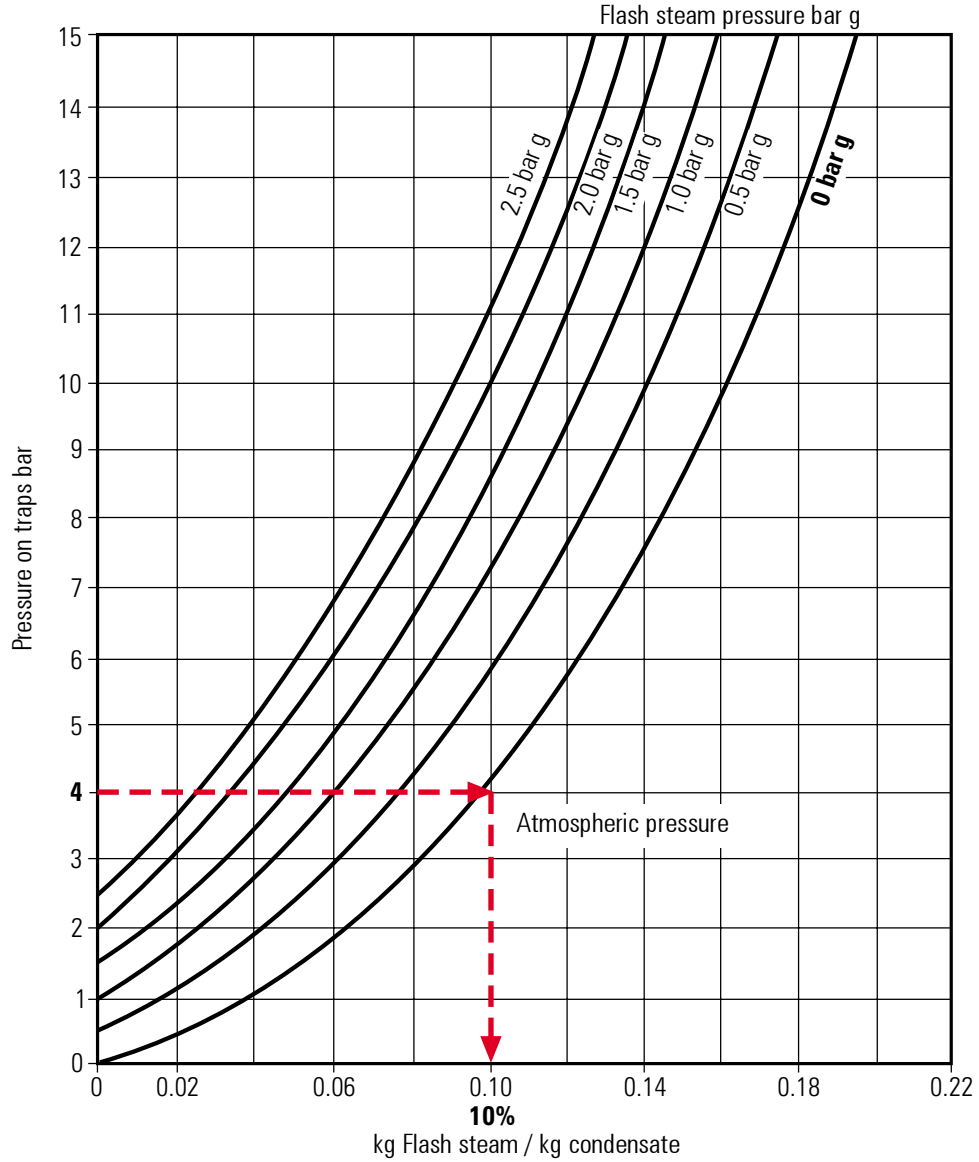


Fig. 14.3.2 Quantity of flash steam graph

Factors having a bearing on two-phase flow inside a pipe, include:

- If the condensate on the upstream side of the trap is cooler than the saturation temperature (for example: a thermostatic steam trap is used), the amount of flash steam after the trap is reduced. This can reduce the size of the line required.
- If the line slopes down from the trap to its termination, the slope will have an effect on the flow of condensate, but to what magnitude, and how can this be quantified?

- On longer lines, radiation losses from the line may condense some of the flash steam, reducing its volume and velocity, and there may be a case for reducing the line size. But at what point should it be reduced and by how much?
- If the discharge line lifts up to an overhead return line, there will be times when the lifting line will be full of cool condensate, and times when flash steam from the trap may evaporate some or all of this condensate. Should the rising discharge line be sized on flash steam velocity or the quantity of condensate?
- Most processes operate some way below their full-load condition for most of their running cycle, which reduces flash steam for most of the time. The question therefore arises: is there a need for the system to be sized on the full-load condition, if the equipment permanently runs at a lower running load?
- On temperature controlled plant, the pressure differential across the trap will itself change depending on the heat load. This will affect the amount of flash steam produced in the line.

Recommendations on trap discharge lines

Because of the number of variables, an exact calculation of line size would be complex and probably inaccurate. Experience has shown that if trap discharge lines are sized on flash steam velocities of 15 to 20 m/s, and certain recommendations are adhered to, few problems will arise.

Recommendations:

1. Correctly sized trap discharge lines which slope in the direction of flow and are open-ended or vented at a receiver, will be non-flooded and allow flash steam to pass unhindered above the condensate (Figure 14.3.3). A minimum slope of 1 in 70 (150 mm drop every 10 m) is recommended. A simple visual check will usually confirm if the line is sloping - if no slope is apparent it is not sloping enough!

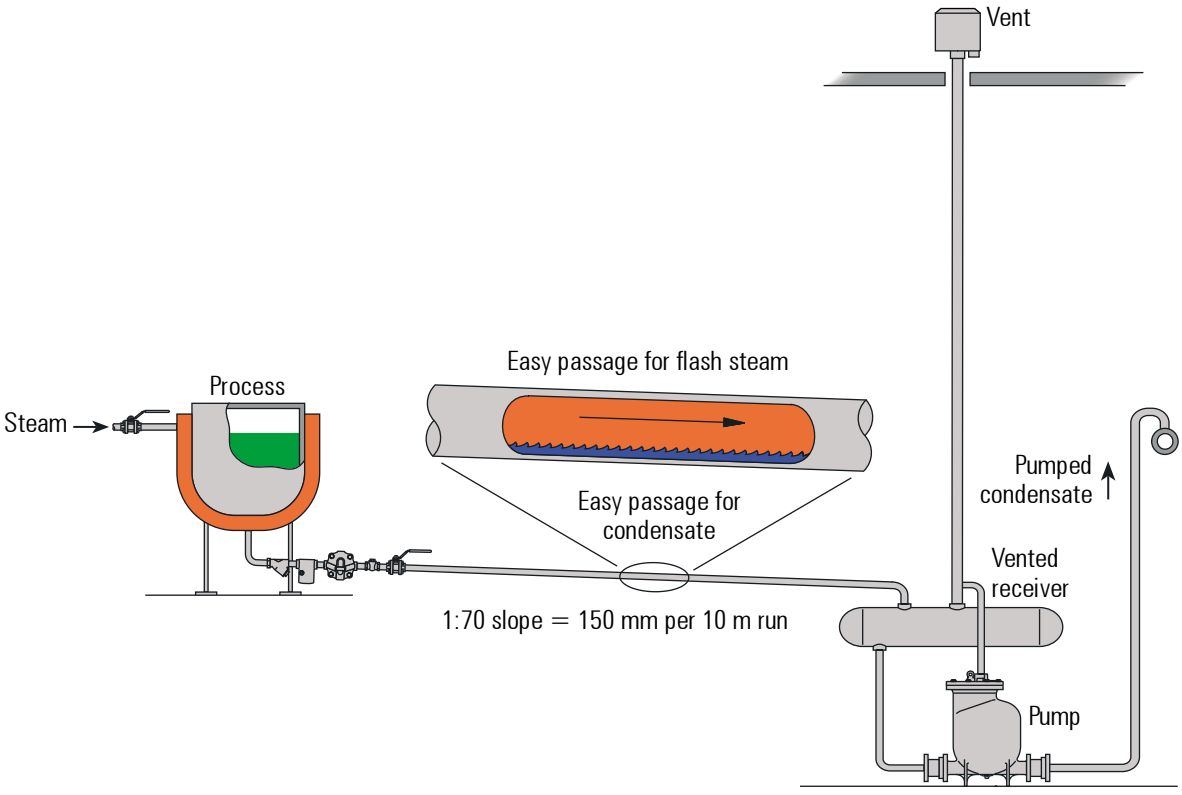


Fig. 14.3.3 Discharge line sloping 1:70 in the direction of flow

2. If it is unavoidable, non-pumped rising lines (Figure 14.3.4) should be kept as short as possible and fitted with a non-return valve to stop condensate falling back down to the trap. Risers should discharge into the top of overhead return lines. This stops condensate draining back into the riser from the return main after the trap has discharged, to assist the easy passage of flash steam up the riser.

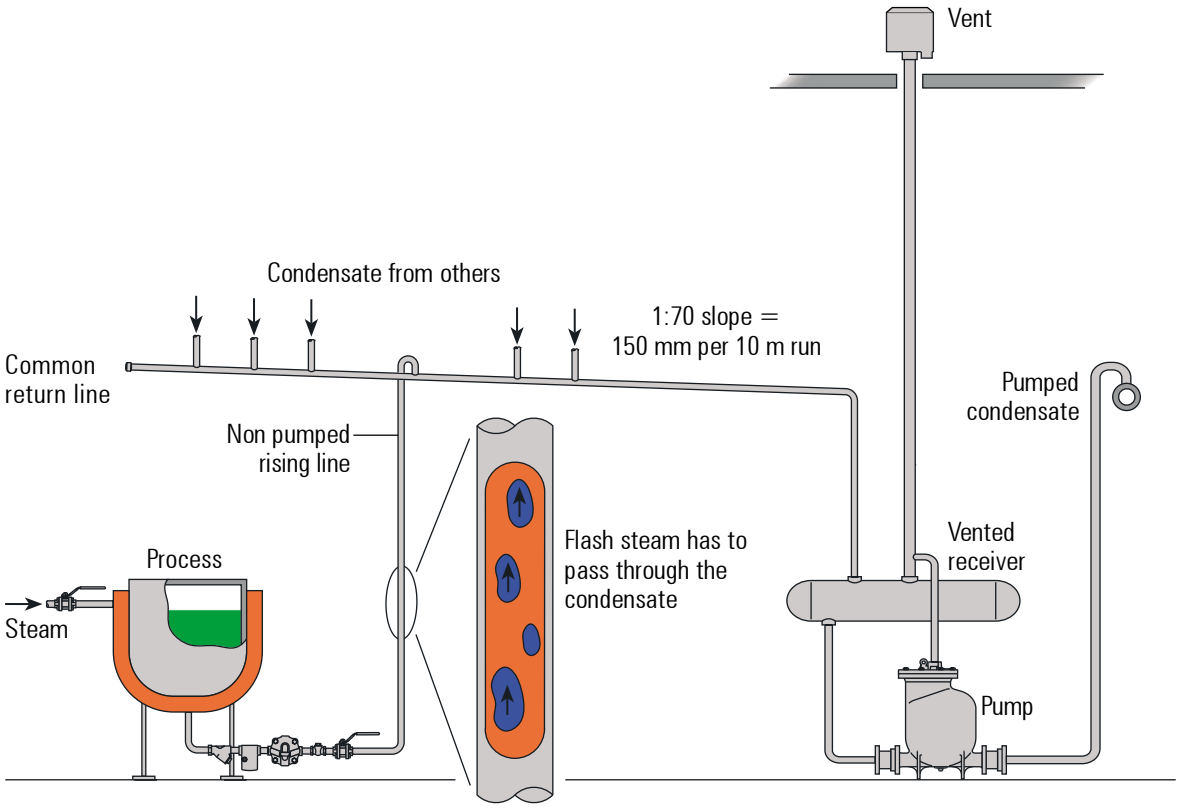


Fig. 14.3.4 Keep rising lines short and connect to the top of return lines

It is sensible to consider using a slightly larger riser, which will produce a lower flash steam velocity. This will reduce the risk of waterhammer and noise caused by steam trying to force a path through the liquid condensate in the riser.

Important: A rising line should only be used where the process steam pressure is guaranteed to be higher than the condensate backpressure at the trap outlet. If not, the process will waterlog unless a pumping trap or pump-trap combination is used to provide proper drainage against the backpressure.

3. Common return lines should also slope down and be non-flooded (Figure 14.3.4). To avoid flash steam occurring in long return lines, hot condensate from trap discharge lines should drain into vented receivers (or flash vessels where appropriate), from where it can be pumped on to its final destination, via a flooded line at a lower temperature.

Condensate pumping is dealt with in more detail in Module 14.4.

The condensate pipe sizing chart

The condensate pipe sizing chart (Figure 14.3.5) can be used to size any type of condensate line, including:

- Drain lines containing no flash steam.
- Lines consisting of two-phase flow, such as trap discharge lines, which are selected according to the pressures either side of the trap.

The chart (Figure 14.3.5):

- Works around acceptable flash steam velocities of 15 - 20 m/s, according to the pipe size and the proportion of flash steam formed.
- Can be used with condensate temperatures lower than the steam saturation temperature, as will be the case when using thermostatic steam traps.
- Is used to size trap discharge lines on full-load conditions. It is not necessary to consider any oversizing factors for start-up load or the removal of non-condensable gases.
- May also be used to estimate sizes for pumped lines containing condensate below 100°C. This will be discussed in Module 14.4.

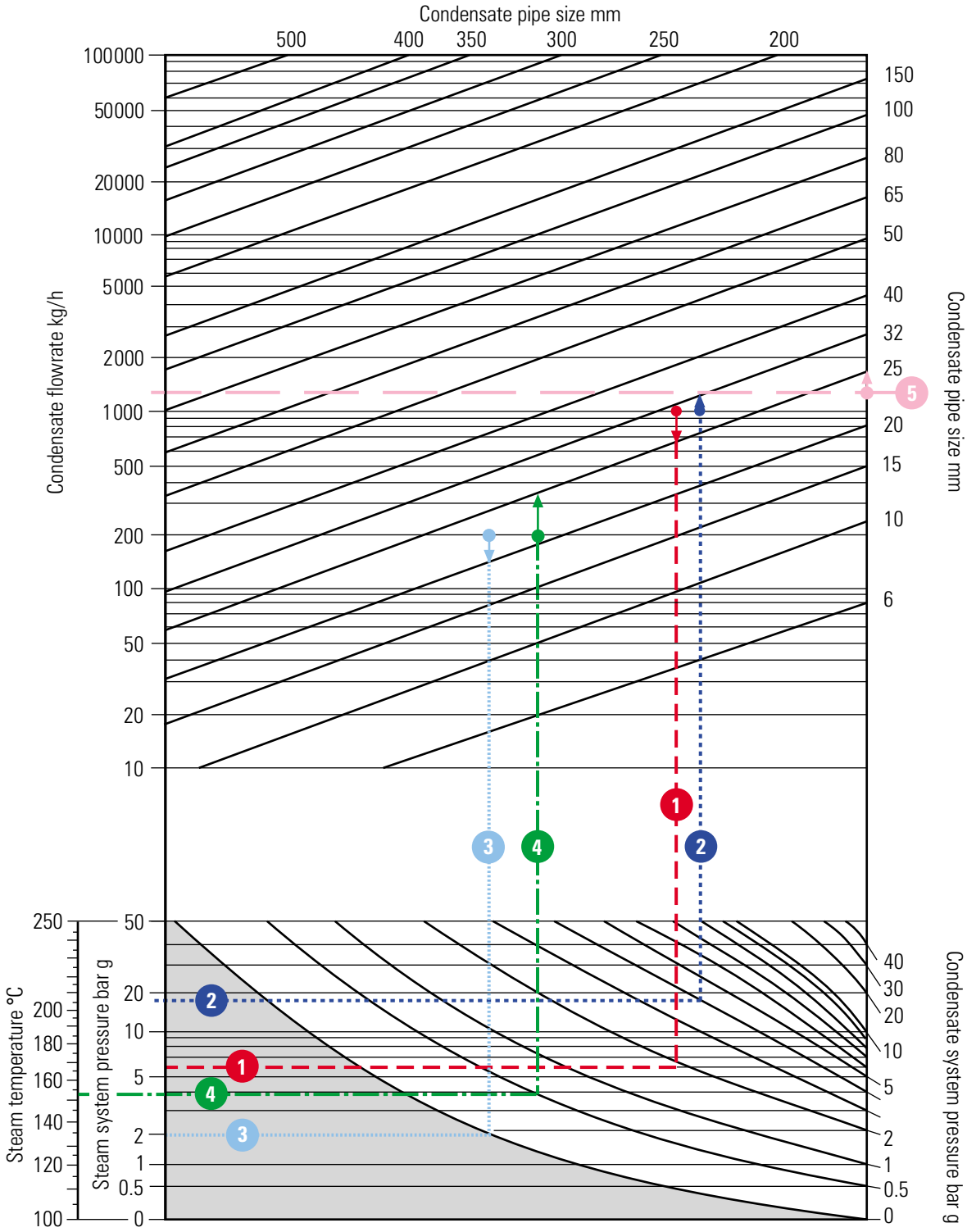


Fig. 14.3.5 Condensate pipe sizing chart

Using the condensate pipe sizing chart (Also available in Appendix 14.3.2)

Establish the point where the steam and condensate pressures meet (lower part of the chart, Figure 14.3.5). From this point, move vertically up to the upper chart to meet the required condensate rate. If the discharge line is falling (non-flooded) and the selection is on or between lines, choose the lower line size. If the discharge line is rising, and therefore likely to be flooded, choose the upper line size.

Note: The reasoning employed for the sizing of a steam trap is different to that used for a discharge line, and it is perfectly normal for a trap discharge line to be sized different to the trap it is serving. However, when the trap is correctly sized, the usual ancillary equipment associated with a steam trap station, such as isolation valves, strainer, trap testing chamber, and check valve, can be the same size as the trapping device selected, whatever the discharge line size.

Example 14.3.3 (1) on the chart (Figure 14.3.6)

A steam trap passing a full-load of 1 000 kg/h at 6 bar g saturated steam pressure through a falling discharge line down to a flash vessel at 1.7 bar g.

As the discharge line is non-flooded, the lower figure of 25 mm is selected from the chart (Figure 14.3.5).

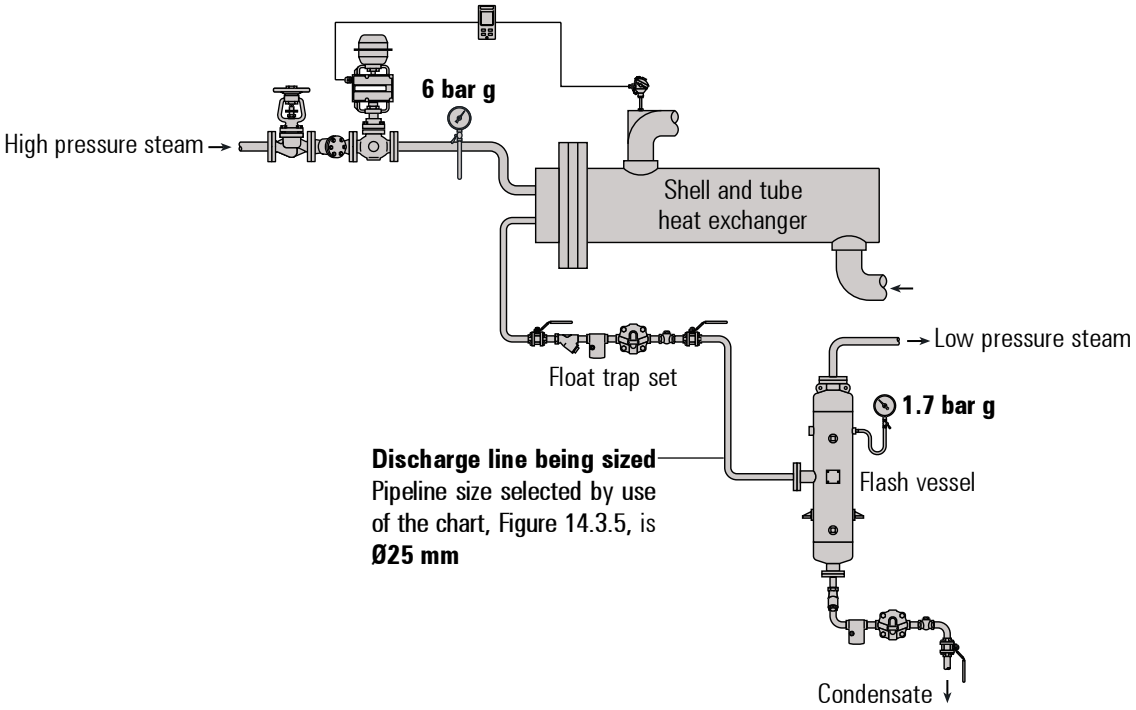


Fig. 14.3.6 A non-flooded pressurised trap discharge line (refer to Example 14.3.3)

Example 14.3.4 (2) on the chart (Figure 14.3.7)

A steam trap passing a full-load of 1 000 kg/h at 18 bar g saturated steam pressure through a discharge line rising 5 m up to a pressurised condensate return line at 3.5 bar g.

Add the 0.5 bar static pressure (5 m head) to the 3.5 bar condensate pressure to give 4 bar g backpressure.

As the discharge line is rising and thus flooded, the upper figure of 32 mm is selected from the chart, (Figure 14.3.5).

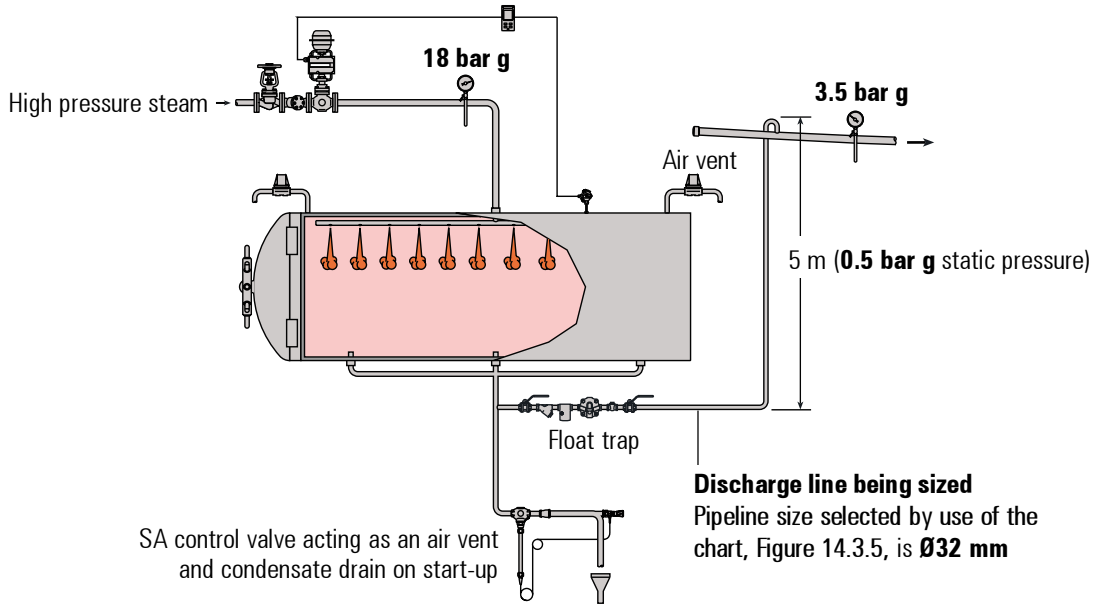


Fig. 14.3.7 A flooded trap discharge line (refer to Example 14.3.4)

Example 14.3.5 3 on the chart (Figure 14.3.8)

A steam trap passing a full-load of 200 kg/h at 2 bar g saturated steam pressure through a sloping discharge line falling down to a vented condensate receiver at atmospheric pressure (0 bar g).

As the line is non-flooded, the lower figure of 20 mm is selected from the chart, (Figure 14.3.5).

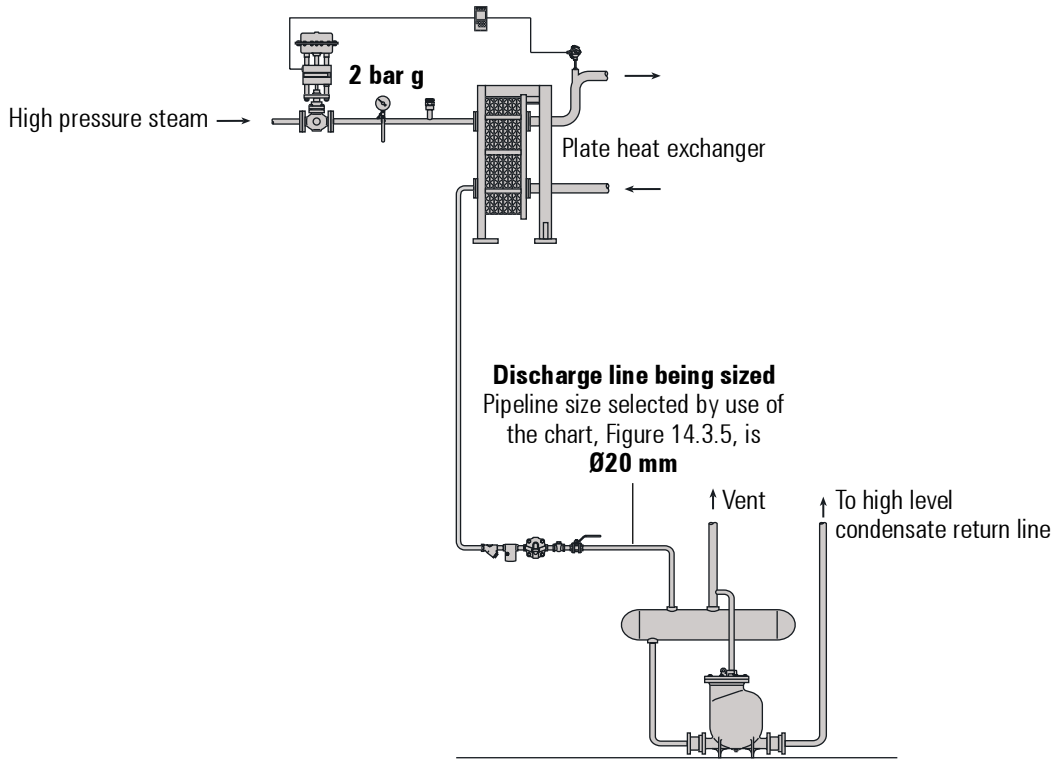


Fig. 14.3.8 A non-flooded vented trap discharge line (refer to Example 14.3.5)

Example 14.3.6 4 on the chart (Figure 14.3.9)

A pump-trap passing a full-load of 200 kg/h at 4 bar g saturated steam space pressure through a discharge line rising 5 m up to a non-flooded condensate return line at atmospheric pressure.

The 5 m static pressure contributes the total backpressure of 0.5 bar g.

As the trap discharge line is rising, the upper figure of 25 mm is selected from the chart, (Figure 14.3.5).

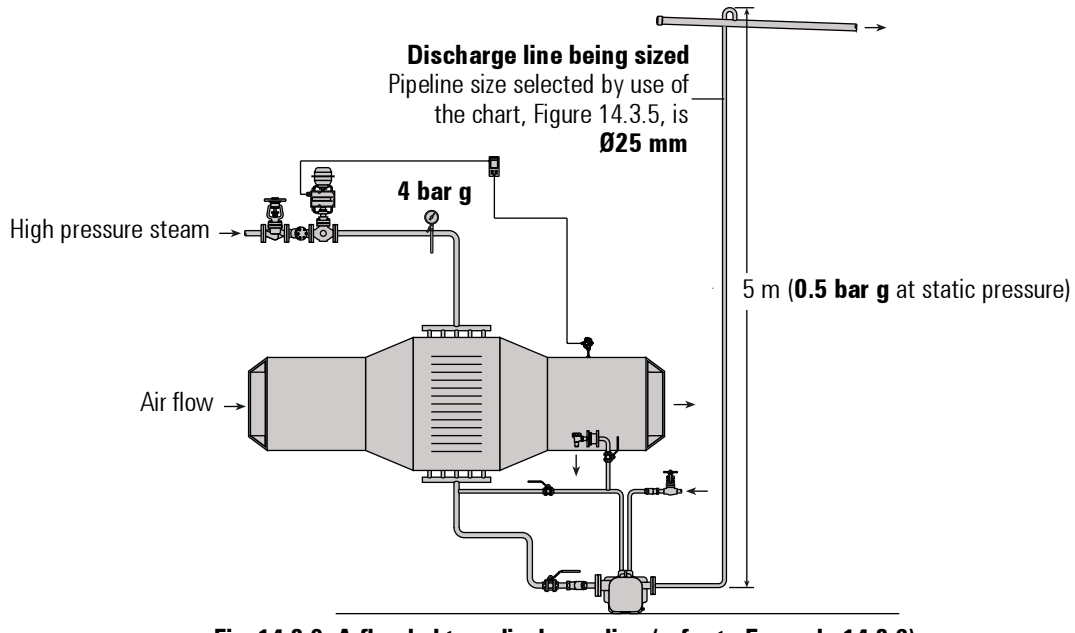


Fig. 14.3.9 A flooded trap discharge line (refer to Example 14.3.6)

Example 14.3.7 5 on the chart (Figure 14.3.10)

Consider a condensate load of 200 kg/h to a receiver and pump. The pump discharge rate for this mechanical type pump is taken as six times the filling rate, hence, the condensate rate taken for this example is 6 x 200 = 1 200 kg/h.

Because the condensate will have lost its flash steam content to atmosphere via the receiver vent, the pump will only be pumping liquid condensate. In this instance, it is only necessary to use the top part of the chart in Figure 14.3.5. As the line from the pump is rising, the upper figure of 25 mm is chosen.

Note: If the pumped line were longer than 100 m, the next larger size must be taken, which for this example would be 32 mm. A useful tip for lines of 100 m or less is to choose a discharge pipe which is the same size as the pump. For further details refer to Module 14.4 'Pumping condensate from vented receivers'.

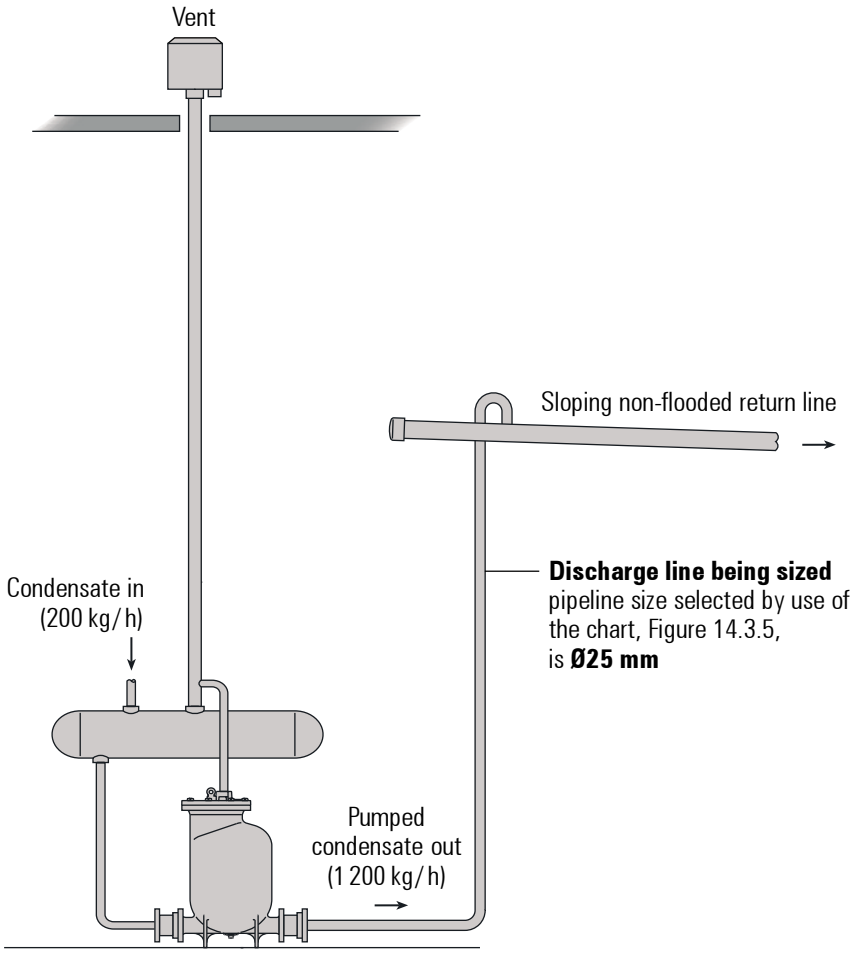


Fig. 14.3.10 A discharge line from the condensate pump (refer to Example 14.3.7)

Common return lines - falling lines

It is sometimes necessary to connect several trap discharge lines from separate processes into a common return line. Problems will not occur if the following considerations are met:

- The common line is not flooded and slopes in the direction of flow to an open end or a vented receiver, or a flash vessel if the conditions allow.
- The common line is sized on the cumulative sizes of the branch lines, and the branch lines are sized from Figure 14.3.5.

Example 14.3.8

Figure 14.3.11 shows three heat exchangers, each separately controlled and operating at the same time. The condensate loads shown are full loads and occur with 3 bar g in the steam space.

The common line slopes down to the flash vessel at 1.5 bar g, situated in the same plant room. Condensate in the flash vessel falls via a float trap down to a vented receiver, from where it is pumped directly to the boiler house.

The trap discharge lines are sized on full-load with steam pressure at 3 bar g and condensate pressure of 1.5 bar g, and as each is not flooded, the lower line sizes are picked from the graph.

Determine the condensate line sizes for the falling discharge lines and common lines.

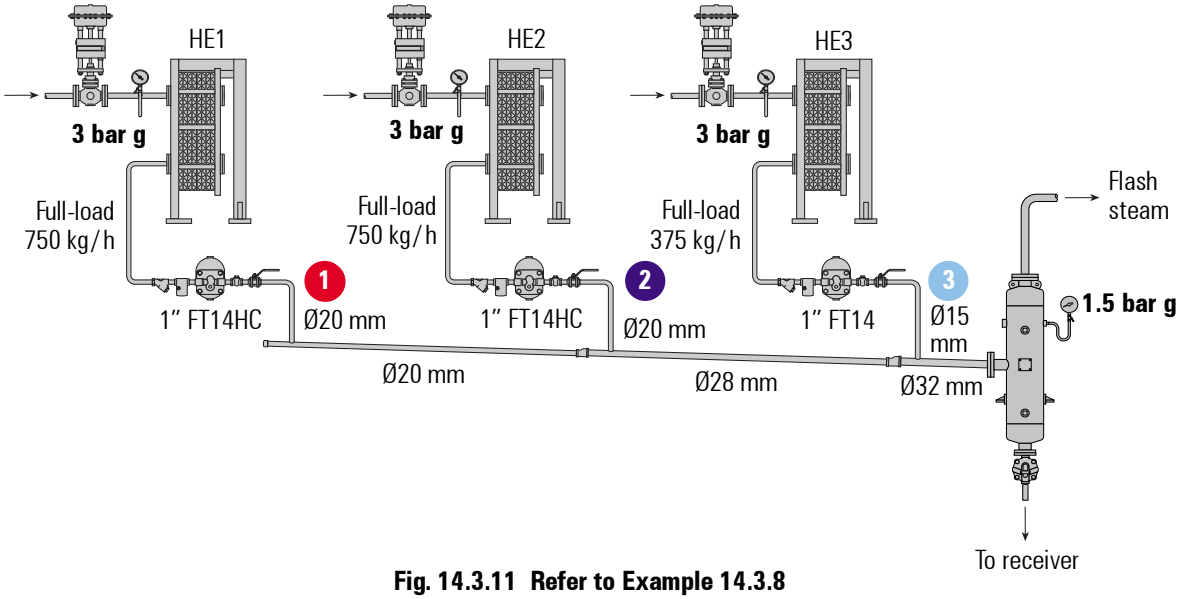


Fig. 14.3.11 Refer to Example 14.3.8

Using Appendix 14.3.2, Condensate pipe sizing chart:

Line 1 picked as 20 mm, 2 picked as 20 mm, 3 picked as 15 mm

The bore of the common line connecting two discharge lines can be found by calculating the square root of the sum of the squares of the bores of the two discharge lines, as shown below:

Common line for 1 + 2 , $= \sqrt{20^2 + 20^2} = 28$ mm : Pick a DN25 pipe (see note below)

Common line for (1 + 2)+ 3 $= \sqrt{28^2 + 15^2} = 32$ mm : Pick a DN32 pipe

Note: The theoretical dimension of 28 mm for the common line 1 + 2 does not exist as a nominal bore in commercial pipe sizes. The internal diameters of pipes can be larger or smaller than the nominal bore depending on the pipe schedule. For example, for a DIN 2448 steel pipe, the internal diameter for a 25 mm pipe is about 28.5 mm, while that for a 25 mm Schedule 40 pipe is about 26.6 mm.

Where the calculated bore is not much greater than the nominal bore, it is practical to choose the next lower size pipe. In this instance, a nominal bore 25 mm pipe may be selected. If, however, the calculated bore is not near the nominal bore, then the next larger nominal bore pipe should be selected. Common sense should be applied.

Common return lines - rising lines

It is sometimes unavoidable for condensate discharge and common lines to rise at some point between the trap and the point of final termination. When this is the case, each discharge line is sized by moving up to the next size on the chart, as previously discussed in this Module.

Example 14.3.9

Figure 14.3.12 shows the same three heat exchangers as in Example 14.3.8.

However, in this instance, the common line rises 15 m and terminates in an overhead non-flooded condensate return main, giving the same backpressure of 1.5 bar as in Example 14.3.8. Each of the discharge lines is sized as a rising line.

Determine the condensate line sizes for the discharge lines and common lines.

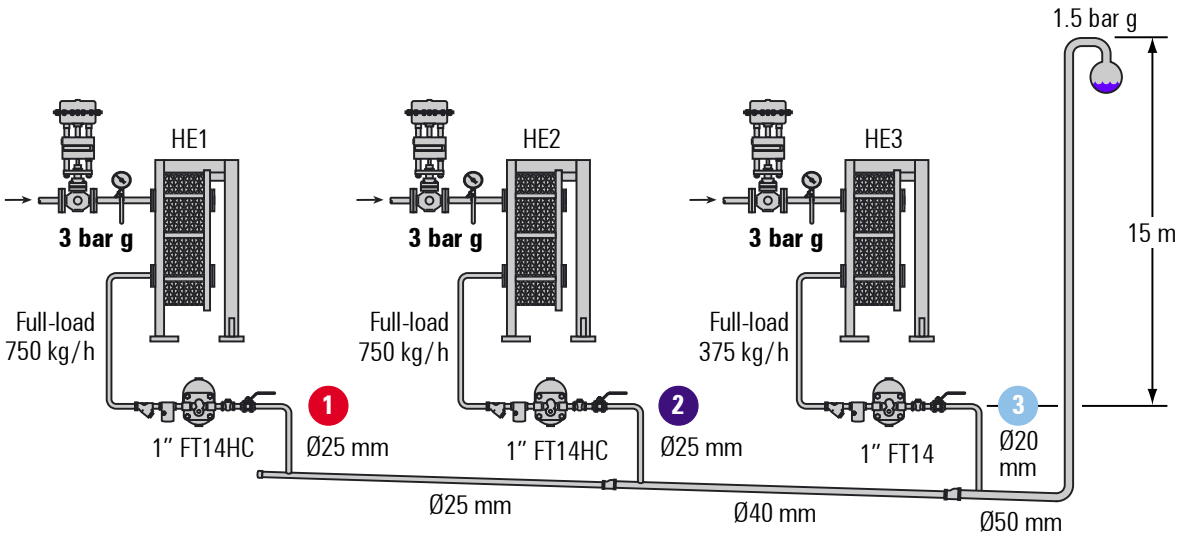


Fig. 14.3.12 Refer to Example 14.3.9

Using Appendix 14.3.2, Condensate pipe sizing chart:

Line 1 picked as 25 mm, 2 picked as 25 mm, 3 picked as 20 mm

Because the common line is rising, it can be seen that each of the discharge lines is a size larger than in Example 14.3.8 even though the backpressure is the same at 1.5 bar g.

The bore of the common line connecting two discharge lines can be found by calculating the square root of the sum of the squares of the bores of the two discharge lines, as shown below:

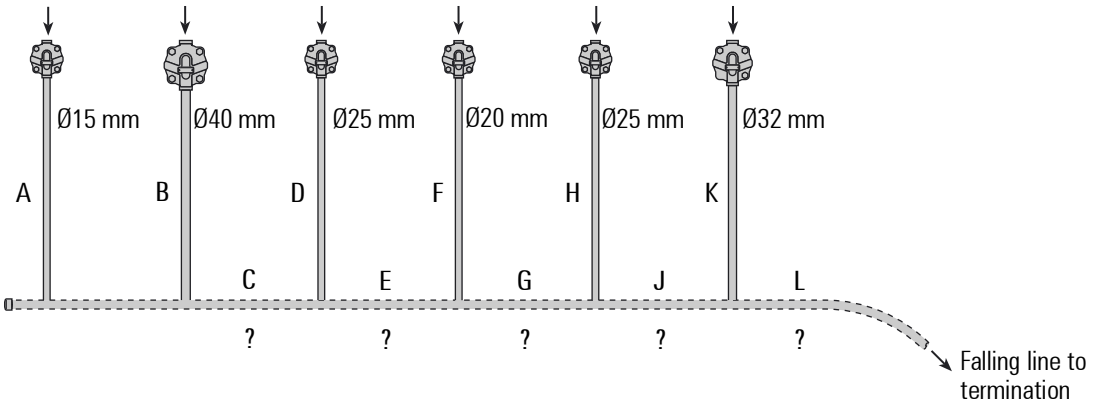
Common line for 1 + 2, $= \sqrt{25^2 + 25^2} = 36$ mm : Pick a DN40 pipe

Common line for (1 + 2)+ 3 $= \sqrt{36^2 + 20^2} = 42$ mm : Pick a DN50 pipe

Note: For rising lines, the chosen nominal bore pipe should always be larger than the calculated bore.

Example 14.3.10 - Falling common line

Calculating the common line sizes for the application shown in Fig. 14.3.12 which falls to a final termination point:



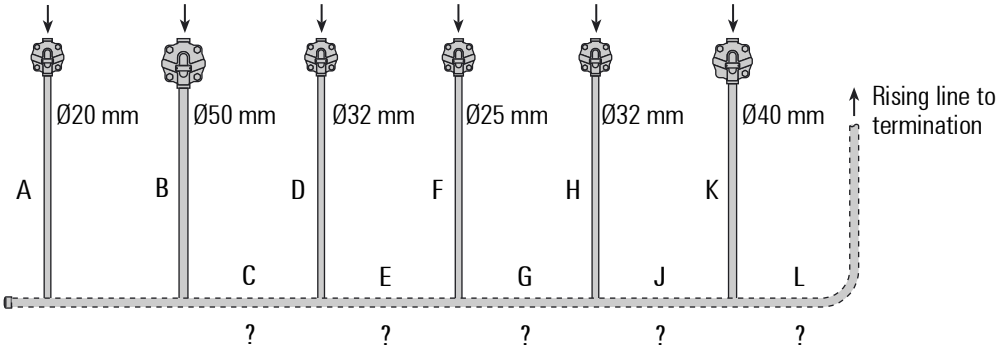
Line	Pipeline diameter (mm)	Commercial pipe size selected (DN)
A	15	
B	40	
C	$\sqrt{40^2+15^2} = 43$*	40*
D	25	
E	$\sqrt{25^2+43^2} = 50$	50
F	20	
G	$\sqrt{20^2+50^2} = 54$	65
H	25	
J	$\sqrt{25^2+54^2} = 60$	65
K	32	
L	$\sqrt{32^2+60^2} = 68$*	65*

Fig. 14.3.13 *Close to nominal bore size

Example 14.3.11 - Rising common line

Calculating the common line sizes for the application shown in Fig. 14.3.14 which rises to a final termination point:

Note that the steam loads are the same as Example 14.3.10, but the discharge lines are one size larger due to the rising common line.



Line	Pipeline diameter (mm)	Commercial pipe size selected (DN)
A	20	
B	50	
C	$\sqrt{50^2+20^2} = 54$*	50*
D	32	
E	$\sqrt{32^2+54^2} = 63$	65
F	25	
G	$\sqrt{25^2+63^2} = 68$*	65*
H	32	
J	$\sqrt{32^2+68^2} = 75$	80
K	40	
L	$\sqrt{40^2+75^2} = 85$*	80*

Fig. 14.3.14 *Close to nominal bore size

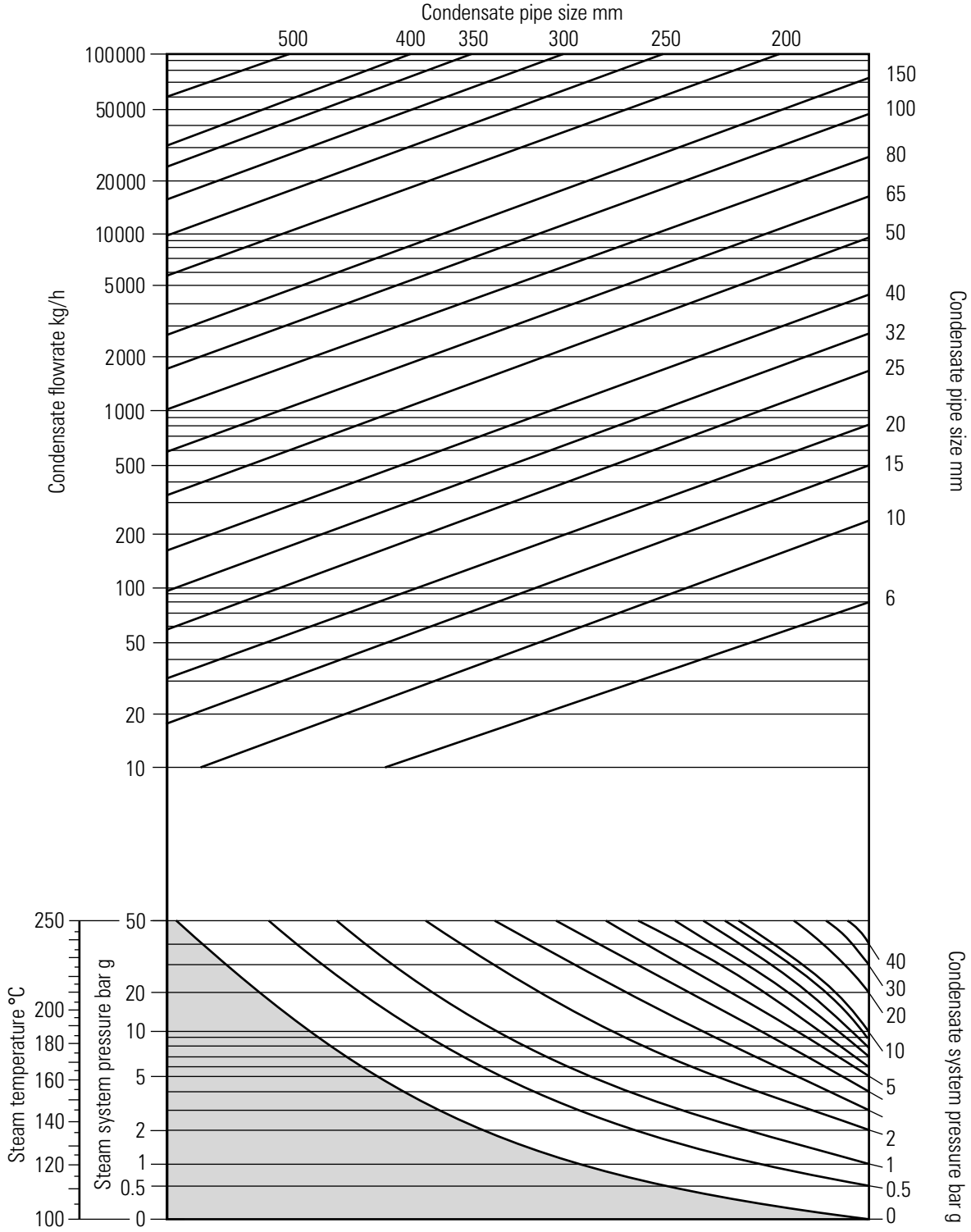
The procedure shown in Examples 14.3.10 and 14.3.11 can be simplified by using Appendix 14.3.3.

For example, where pipes A and B (20 mm and 50 mm) join, the minimum required pipe diameter is shown as 54 mm. Clearly, the user would fit the next largest size of commercial pipe available, unless the calculated bore is close to a nominal bore size pipe.

Appendix 14.3.1 Flow of water in heavy steel pipes

Flowrate		kg/h									
Pipe size Ø		15 mm	20 mm	25 mm	32 mm	40 mm	50 mm	65 mm	80 mm	100 mm	
Pa/m	mbar/m	<0.15 m/s			0.15 m/s					0.3 m/s	
10.0	0.100	50	119	223	490	756	1 447	2 966	4 644	9 432	
12.5	0.125	58	133	252	554	853	1 634	3 348	5 220	10 656	
15.0	0.150	65	151	277	616	943	1 807	3 708	5 760	11 736	
17.5	0.175	68	162	302	670	1 026	1 966	4 032	6 264	12 744	
20.0	0.200	76	176	328	720	1 105	2 113	4 320	6 732	13 680	
22.5	0.225	79	187	349	770	1 177	2 254	4 608	7 164	14 580	0.5
25.0	0.250	83	198	371	814	1 249	2 387	4 860	7 596	15 408	m/s
27.5	0.275	90	209	389	857	1 314	2 513	5 112	7 992	16 200	
30.0	0.300	94	220	410	900	1 379	2 632	5 364	8 352	16 956	
32.5	0.325	97	230	428	940	1 440	2 747	5 616	8 712	17 712	
35.0	0.350	101	241	446	979	1 498	2 858	5 832	9 072	18 432	
37.5	0.375	104	248	464	1 015	1 555	2 966	6 048	9 396	19 116	
40.0	0.400	112	259	479	1 051	1 609	3 071	6 264	9 720	19 764	
42.5	0.425	115	266	497	1 087	1 663	3 175	6 480	10 044	20 412	
45.0	0.450	119	277	511	1 123	1 717	3 272	6 660	10 368	21 024	
47.5	0.475	122	284	526	1 156	1 768	3 370	6 876	10 656	21 636	
50.0	0.500	126	292	540	1 188	1 814	3 463	7 056	10 944	22 212	
52.5	0.525	130	299	558	1 220	1 865	3 553	7 236	11 232	22 788	
55.0	0.550	130	306	572	1 249	1 912	3 636	7 416	11 520	23 364	
57.5	0.575	133	317	583	1 282	1 958	3 744	7 596	11 808	23 904	
60.0	0.600	137	324	598	1 310	2 002	3 816	7 776	12 060	24 444	
62.5	0.625	140	331	612	1 339	2 048	3 888	7 920	12 312	24 984	
65.0	0.650	144	338	626	1 368	2 092	3 996	8 100	12 600	25 488	
67.5	0.675	148	346	637	1 397	2 131	4 068	8 280	12 852	25 992	
70.0	0.700	151	353	652	1 422	2 174	4 140	8 424	13 068	26 496	
72.5	0.725	151	356	662	1 451	2 218	4 212	8 568	13 320	27 000	
75.0	0.750	155	364	677	1 476	2 257	4 284	8 748	13 572	27 468	
77.5	0.775	158	371	688	1 505	2 297	4 356	8 892	13 788	27 972	
80.0	0.800	162	378	698	1 530	2 336	4 464	9 036	14 040	28 440	1
82.5	0.825	166	385	709	1 555	2 372	4 536	9 180	14 256	28 872	m/s
85.0	0.850	166	389	724	1 580	2 412	4 608	9 324	14 472	29 340	
87.5	0.875	169	396	734	1 606	2 448	4 680	9 468	14 724	29 772	
90.0	0.900	173	403	745	1 627	2 488	4 716	9 612	14 940	30 240	
92.5	0.925	176	407	756	1 652	2 524	4 788	9 756	15 156	30 672	
95.0	0.950	176	414	767	1 678	2 560	4 860	9 900	15 372	31 104	
97.5	0.975	180	421	778	1 699	2 596	4 932	10 044	15 552	31 500	
100.0	1.000	184	425	788	1 724	2 632	5 004	10 152	15 768	31 932	
120.0	1.200	202	472	871	1 897	2 898	5 508	11 196	17 352	35 100	
140.0	1.400	220	511	943	2 059	3 143	5 976	12 132	18 792	38 160	
160.0	1.600	234	547	1 015	2 210	3 373	6 408	12 996	20 160	40 680	
180.0	1.800	252	583	1 080	2 354	3 589	6 804	13 824	21 420	43 200	1.5
200.0	2.000	266	619	1 141	2 488	3 780	7 200	14 580	22 644	45 720	m/s
220.0	2.200	281	652	1 202	2 617	3 996	7 560	15 336	23 760	47 880	
240.0	2.400	288	680	1 256	2 740	4 176	7 920	16 056	24 876	50 400	
260.0	2.600	306	713	1 310	2 855	4 356	8 244	16 740	25 920	52 200	
280.0	2.800	317	742	1 364	2 970	4 536	8 568	17 388	26 928	54 360	
300.0	3.000	331	767	1 415	3 078	4 680	8 892	18 000	27 900	56 160	

Appendix 14.3.2 Condensate pipe sizing chart



Appendix 14.3.3 Common pipe sizing table

D1 = Connecting branch size (N.B.)
D2 = Common pipe size

D2	D1 - Connecting branch size (NB)								
	15	20	25	32	40	50	65	80	100
15	21	25	29	35	43	52	67	81	101
16	22	26	30	36	43	52	67	82	101
17	23	26	30	36	43	53	67	82	101
18	23	27	31	37	44	53	67	82	102
19	24	28	31	37	44	53	68	82	102
20	25	28	32	38	45	54	68	82	102
21	26	29	33	38	45	54	68	83	102
22	27	30	33	39	46	55	69	83	102
23	27	30	34	39	46	55	69	83	103
24	28	31	35	40	47	55	69	84	103
25	29	32	35	41	47	56	70	84	103
26	30	33	36	41	48	56	70	84	103
27	31	34	37	42	48	57	70	84	104
28	32	34	38	43	49	57	71	85	104
29	33	35	38	43	49	58	71	85	104
30	34	36	39	44	50	58	72	85	104
31	34	37	40	45	51	59	72	86	105
32	35	38	41	45	51	59	72	86	105
33	36	39	41	46	52	60	73	87	105
34	37	39	42	47	52	60	73	87	106
35	38	40	43	47	53	61	74	87	106
36	39	41	44	48	54	62	74	88	106
37	40	42	45	49	54	62	75	88	107
38	41	43	45	50	55	63	75	89	107
39	42	44	46	50	56	63	76	89	107
40	43	45	47	51	57	64	76	89	108
41	44	46	48	52	57	65	77	90	108
42	45	47	49	53	58	65	77	90	108
43	46	47	50	54	59	66	78	91	109
44	46	48	51	54	59	67	78	91	109
45	47	49	51	55	60	67	79	92	110
46	48	50	52	56	61	68	80	92	110
47	49	51	53	57	62	69	80	93	110
48	50	52	54	58	62	69	81	93	111
49	51	53	55	59	63	70	81	94	111
50	52	54	56	59	64	71	82	94	112
51	53	55	57	60	65	71	83	95	112
52	54	56	58	61	66	72	83	95	113
53	55	57	59	62	66	73	84	96	113
54	56	58	60	63	67	74	85	97	114
55	57	59	60	64	68	74	85	97	114
56	58	59	61	64	69	75	86	98	115
57	59	60	62	65	70	76	86	98	115

D2	D1 - Connecting branch size (NB)								
	15	20	25	32	40	50	65	80	100
58	60	61	63	66	70	77	87	99	116
59	61	62	64	67	71	77	88	99	116
60	62	63	65	68	72	78	88	100	117
61	63	64	66	69	73	79	89	101	117
62	64	65	67	70	74	80	90	101	118
63	65	66	68	71	75	80	91	102	118
64	66	67	69	72	75	81	91	102	119
65	67	68	70	72	76	82	92	103	119
66	68	69	71	73	77	83	93	104	120
67	69	70	72	74	78	84	93	104	120
68	70	71	72	75	79	84	94	105	121
69	71	72	73	76	80	85	95	106	121
70	72	73	74	77	81	86	96	106	122
71	73	74	75	78	81	87	96	107	123
72	74	75	76	79	82	88	97	108	123
73	75	76	77	80	83	88	98	108	124
74	76	77	78	81	84	89	98	109	124
75	76	78	79	82	85	90	99	110	125
76	77	79	80	82	86	91	100	110	126
77	78	80	81	83	87	92	101	111	126
78	79	81	82	84	88	93	102	112	127
79	80	81	83	85	89	93	102	112	127
80	81	82	84	86	89	94	103	113	128
81	82	83	85	87	90	95	104	114	129
82	83	84	86	88	91	96	105	115	129
83	84	85	87	89	92	97	105	115	130
84	85	86	88	90	93	98	106	116	131
85	86	87	89	91	94	99	107	117	131
86	87	88	90	92	95	99	108	117	132
87	88	89	91	93	96	100	109	118	133
88	89	90	91	94	97	101	109	119	133
89	90	91	92	95	98	102	110	120	134
90	91	92	93	96	98	103	111	120	135
91	92	93	94	96	99	104	112	121	135
92	93	94	95	97	100	105	113	122	136
93	94	95	96	98	101	106	113	123	137
94	95	96	97	99	102	106	114	123	137
95	96	97	98	100	103	107	115	124	138
96	97	98	99	101	104	108	116	125	139
97	98	99	100	102	105	109	117	126	139
98	99	100	101	103	106	110	118	127	140
99	100	101	102	104	107	111	118	127	141
100	101	102	103	105	108	112	119	128	141

Questions

1. As a simple rule, what can condensate drain lines be sized on?

a| The plant condensate outlet connection ☐

b| The plant steam inlet connection ☐

c| The trap inlet connection with the correct sized trap ☐

d| It is unimportant to size drain lines correctly ☐

2. For steam mains and constant pressure processes, how is start load estimated?

a| Twice the running load at the rated pressure ☐

b| Three times the running load at a third of the rated pressure ☐

c| Ten times the running load at half the rated pressure ☐

d| The running load at twice the rated pressure ☐

3. On which pressure loss should drain lines be sized?

a| 100 Pa/m ☐

b| They need only be sized on velocity ☐

c| 200 Pa/m ☐

d| 200 Pa/m for lines less than 10 m and 100 Pa/m for lines over 10 m ☐

4. What is the major factor that influences the size of the trap discharge lines?

a| The size of the trap ☐

b| The size of the drain line ☐

c| The amount of flash steam produced in the discharge line ☐

d| The amount of condensate flowing ☐

5. Using Appendix 14.3.1, which size of drain line 1.5 m long should be chosen for a constant pressure process with a maximum running load of 450 kg/h?

a| 20 mm ☐

b| 32 mm ☐

c| 25 mm ☐

d| 15 mm ☐

6. Three discharge lines 25 mm, 50 mm, 65 mm are to branch into a common line discharging into a vented receiver. What should be the nominal size of the common line into the receiver?

a| 100 mm ☐

b| 80 mm ☐

c| 65 mm ☐

d| 50 mm ☐

Answers

1: c, 2: a, 3: d, 4: c, 5: a, 6: a

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Module 14.4

Pumping Condensate from Vented Receivers

Pumping Condensate from Vented Receivers

The justification for returning condensate has already been made and, often, this will entail lifting condensate by a pump into the boiler feedtank.

Before looking at the types of pump available for returning condensate, it may be helpful to discuss some basic pumping terminology.

Pumping terminology

Vapour pressure - This term is used to define the pressure corresponding to the temperature at which a liquid changes into vapour. In other words, it is the pressure at which a liquid will boil.

- At 100°C, water will boil at atmospheric pressure.
- At 170°C, water will boil at a pressure of 7 bar g.
- At 90°C, water will boil at a pressure of 0.7 bar a.

The vapour pressure is a very important consideration when pumping condensate. Condensate is usually formed at a temperature close to its boiling point, which may cause difficulties where a centrifugal pump is concerned. This is because centrifugal pumps have an area of lower pressure at the centre, or eye, of the impeller. This produces the suction effect, which draws the liquid into the pump. Although the drop in pressure is small, if the condensate is already very close to its vapour pressure, a proportion of the liquid will flash to steam in the form of small bubbles. These steam bubbles occupy a significantly greater volume than the equivalent mass of water, and have a high ratio of surface area to mass.

As the bubbles travel through the impeller passageways towards its outer edge, they experience increasing pressure. At some point during this journey, the vapour pressure is exceeded, and the steam bubbles implode with considerable force. This is termed 'cavitation' and the implosions are both noisy and destructive. The noise is similar to gravel being shovelled and the implosions will, in time, damage the pump internals.

For this reason, it is recommended that condensate be pumped by electrical pumps specifically built for the task, and that condensate temperatures in atmospheric systems do not exceed 98°C. Some pumps will have limits as low as 94°C or 96°C, depending on the design of the pump, the speed of rotation and the height of the receiver above the pump.

Head (h) - Head is a term used to describe the potential energy of a fluid at a given point. There are several ways that head can be measured: pressure head, static head and friction head. Pressure head and static head are essentially the same thing, but tend to be measured in different units. Pressure head is measured in pressure units such as pascal or bar g; whilst static head is referred to in terms of height, usually in metres (or metres head).

For water, a static head of 10 metres is approximately equivalent to a pressure head of 1 bar g (see Figure 14.4.1).

Pressure head (h_p) - Pressure head is the fluid pressure at the point in question. For example: A pump is required to discharge water against a static head of 30 metres, which approximately equals a pressure head of 3 bar g. The pump fills from a static head of 1 metre, which equals a pressure head of 0.1 bar g. (See Figure 14.4.2).

Static head (h_s) - Static head is the equivalent vertical height of fluid above a datum. The following example explains the measure of static head. Example: the pump inlet in Figure 14.4.2 is subjected to a static head (known as the suction or filling head) of 1 m, and discharges against a static head (known as the static delivery head) of 30 m. Note that in this case, the water being pumped is above the pump inlet (this situation is called a flooded suction).

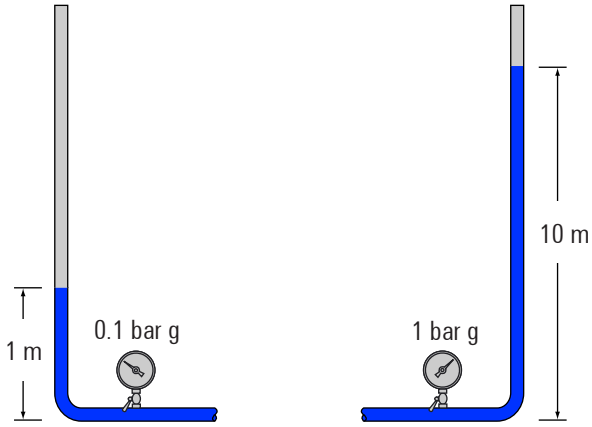


Fig. 14.4.1 Pressure of water in terms of head

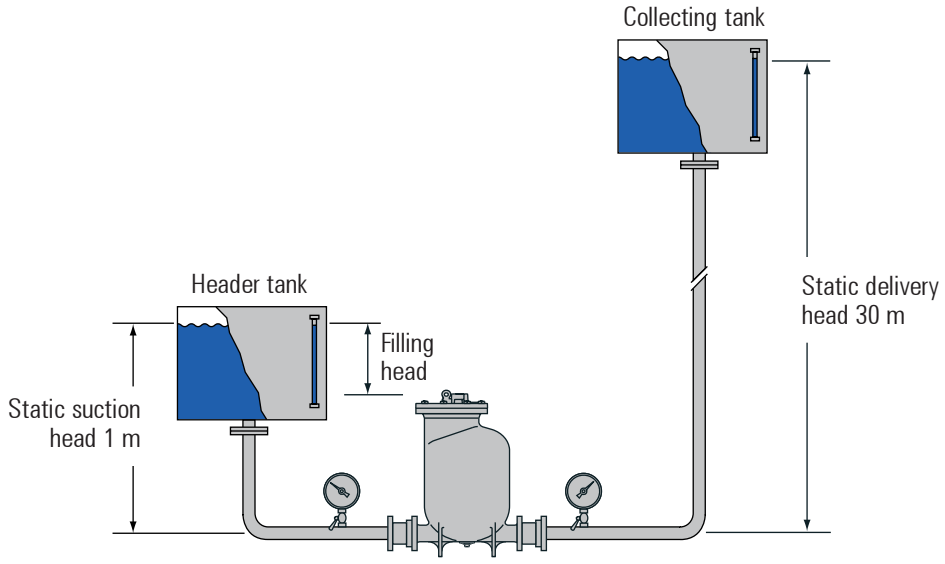


Fig. 14.4.2 Suction, delivery, and filling heads

Net static head - This depends upon whether the pump is a centrifugal type pump or a positive displacement, mechanical type pump.

With an electrical centrifugal pump (Figure 14.4.3), the pressure exerted by the suction head is always present in the pump. The net static head, against which the pump has to work, is the difference between the suction head and the delivery head.

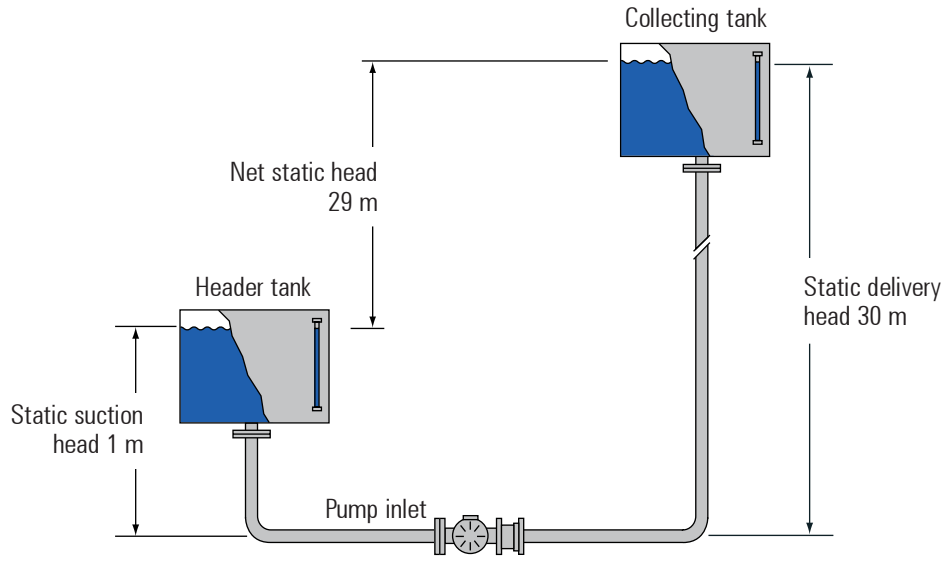


Fig. 14.4.3 Net static head for an electrical pump

With a mechanical displacement pump (Figure 14.4.4), the suction head only provides the energy to fill the pump during the filling cycle. It is not present in the pump body during pumping and has no effect on the delivery head against which the pump has to operate. The net static head is simply the delivery head.

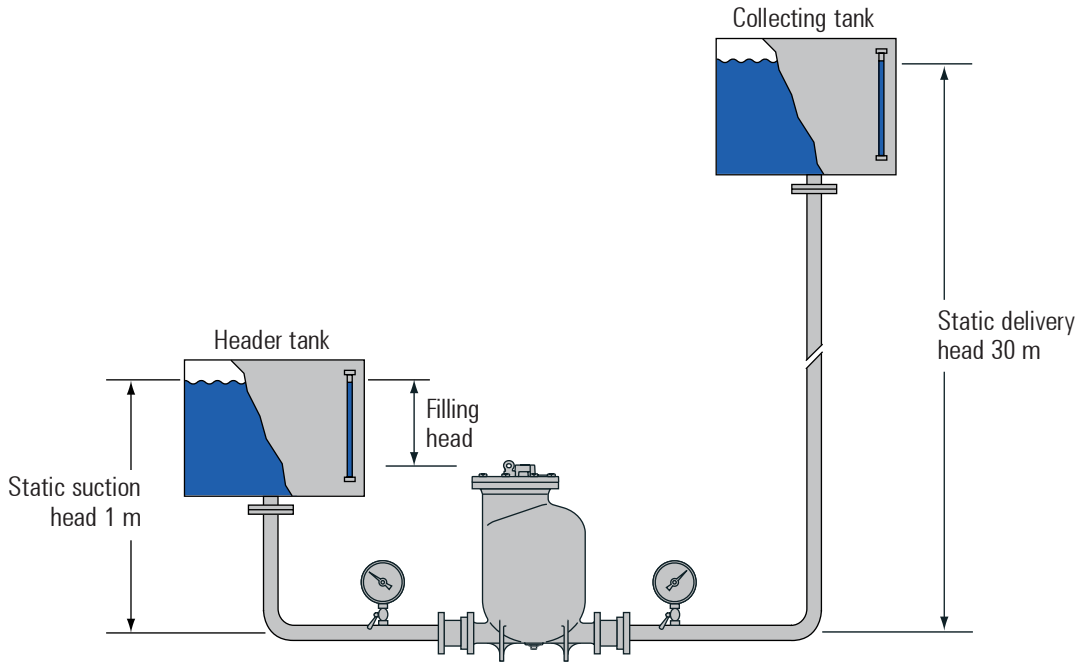


Fig. 14.4.4 Net static head for a mechanical pump equals static delivery head

Friction head (h_f) - The friction head (or head loss to friction) is more accurately defined as the energy required to move the fluid through the pipe. This is discussed in further detail in Module 10.2, 'Pipes and pipe sizing'.

Pressure loss can be calculated using the procedures shown in Block 4, 'Flowmetering' and Block 10, 'Steam distribution', but is more usually found from tables that correlate liquid flowrate, pipe diameter and velocity. To be precise, the resistance to flow encountered by the various pipeline fittings must also be taken into account. Tables are available to calculate the equivalent length of straight pipe exerted by various pipe fittings.

This extra 'equivalent length' for pipe fittings is then added to the actual pipe length to give a 'total equivalent length'. However, in practice, if the pipe is correctly sized, it is unusual for the pipe fittings to represent more than an additional 10% of the actual pipe length.

A general rule, which can be applied, is:

Total equivalent length (l_e) = Actual length + 10%

In most cases, the Steam Plant Engineer will be designing a system with a proprietary manufactured pump arrangement, which has appropriate factors built in. Bearing this in mind, the figure of 10% will be used in this Block as the equivalent length for calculating pressure loss due to friction.

This pressure loss due to friction is greatly dependent on the velocity of the water in the pipe. In simple terms, the pressure loss due to friction increases by a factor proportional to the square of the velocity.

Tables are available which give head loss per metre of pipe for various flowrates and pipe diameters.

Table 14.4.1 Flow of water in black steel pipes (kg/h)

Pressure drop		Pipe size (mm)								
Pa/m	mbar/m	15	20	25	32	40	50	65	80	100
100	**1.00**	184	425	788	1 724	2 632	5 004	10 152	15 768	31 932
114	**1.14**	194	450	845	1 832	2 790	5 366	10 841	16 828	34 247
118	**1.18**	198	457	857	1 890	2 830	5 443	11 022	17 055	34 746

Example 14.4.1

The 50 mm discharge pipework on a pumped condensate line rises vertically for 29 metres to a vented tank. The line is 150 m long and the pumping rate is 5 000 kg/h of water. What is:
(**A**) the pressure head loss due to friction (the friction head), and
(**B**) the total delivery head?

A - Calculate the pressure head loss due to friction (the friction head)

Total equivalent length (l_e) = 150 + 10 % = 165 metres

From Table 14.4.1, it can be seen that a 50 mm pipe carrying 5 004 kg/h of water will experience a pressure drop of 1.0 mbar/m. The flowrate in this example is marginally less, and, although a more accurate estimate could be obtained by interpolation, take the pressure drop as 1 mbar/m. Pressure head loss due to friction is therefore:

165 metres x 1 mbar/m = 165 mbar (**0.165 bar**)

Taking 1 bar to be equivalent to 10 metres of water head the equivalent friction head loss in terms of metres is:

0.165 bar x 10m/bar = 1.65 metres.

B - The total delivery head

Total delivery head (h_d) - The total delivery head h_d against which the pump needs to operate is the sum of three components as can be seen in Equation 14.4.1:

$$\text{Total delivery head } (h_d) = h_s + h_f + h_p \qquad \textbf{Equation 14.4.1}$$

Where:
h_d = Total delivery head
h_s = Pressure required to raise the water to the desired level (static head)
h_f = Pressure required to move the water through the pipes (friction head)
h_p = Pressure in the condensate system (zero in this example as the condensate tank is vented to atmosphere).

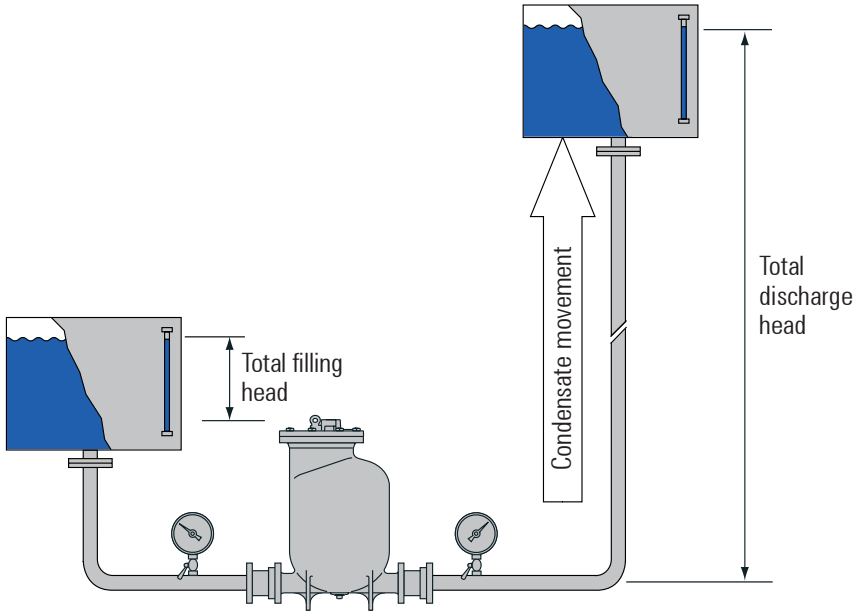


Fig. 14.4.5 Net static head for a mechanical pump equals static delivery head

From the information above:

Total delivery head (h_d) required = static head + equivalent loss in static head due to friction

h_d = 29 m + 1.65 m

h_d = 30.65 metre

Electrical centrifugal condensate pumps

Pump operation

Liquid entering the pump is directed into the centre, or eye, of the rotating impeller vanes. The liquid will then gain velocity as it travels towards the outside of the impeller.

Pump application

The electrical pump is well suited to applications where large volumes of liquid need to be transported.

Electrical pumps are usually built into a unit, often referred to as a condensate recovery unit (CRU). A CRU will usually include:

- A receiver.
- A control system operated by probes or floats.
- One or two pumps.

The instantaneous flow from the CRU can be up to 1.5 times greater than the rate at which condensate returns to the receiver. It is this pumping rate that must be considered when calculating the friction loss in the discharge line.

On twin pump units, a cascade control system may also be employed which allows either pump to be selected as the lead pump and the other as a stand-by pump to provide back-up if the condensate returning to the unit is greater than one pump can handle. This control arrangement also provides back-up in the case of the one pump failing to operate; the condensate level in the tank will increase and bring the stand-by pump into operation. Cascade type units usually pump at a rate of 1.1 times the return rate to the receiver, allowing a smaller discharge line to be considered.

It is very important to follow the manufacturer's literature regarding the discharge pumping rate. Failure to do so could result in undersizing the pump discharge pipework.

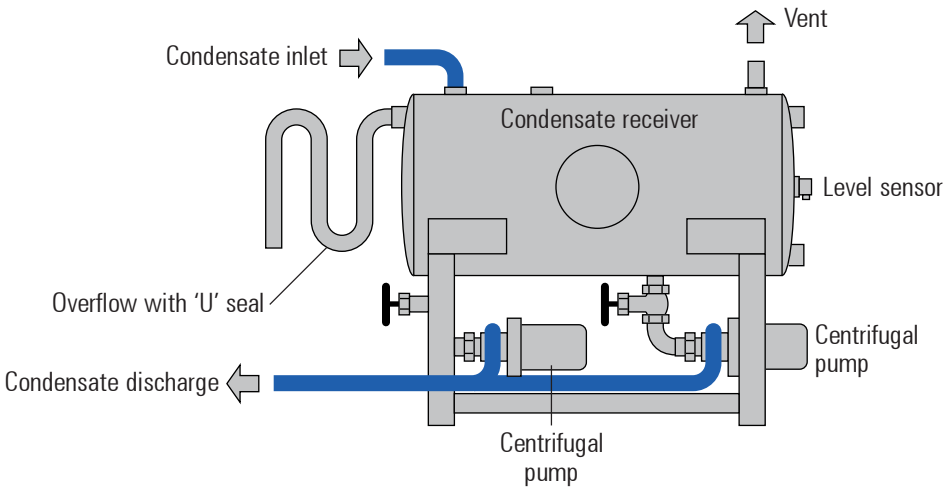


Fig. 14.4.6 A typical electrical condensate recovery unit (CRU)

Sizing an electrical condensate recovery unit

To size an electric condensate recovery unit, it is necessary to know:

- The amount of condensate reaching the receiver at running load.
- The temperature of the condensate. This must not exceed the manufacturer's specified ratings to avoid cavitation, however, manufacturers usually have different impellers to suit different temperature ranges, for example, 90°C, 94°C and 98°C.
- The total discharge head the pump has to pump against - To be determined from the site conditions.
- The pump discharge rate in order to size the return pipework - It is necessary to read the manufacturer's data properly to determine this.

Example 14.4.2 Sizing discharge pipework for an electric condensate recovery unit

Where:

Temperature of condensate = 94°C
Condensate to be handled = 1 000 kg/h
Static lift (h_s) = 30 m
Length of pipework = 150 m
Condensate backpressure = friction losses only (h_f)

An initial selection of a condensate recovery unit can be made by using the manufacturer's sizing chart (an example of which is shown in Figure 14.4.7). From the chart, CRU1 should be the initial choice subject to frictional losses in the delivery pipework.

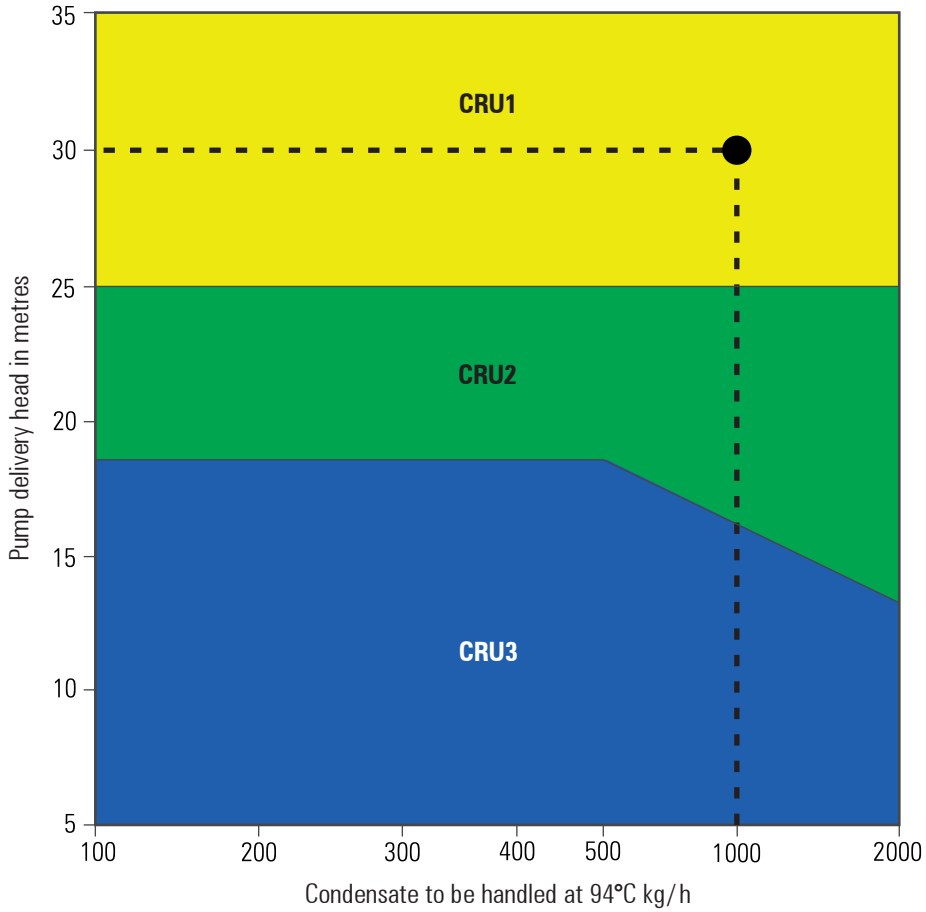


Fig. 14.4.7 A typical electrical condensate recovery unit (CRU) sizing chart (see Example 14.4.2)

From the chart in Figure 14.4.7, it can be seen that CRU1 is actually rated to handle 2 000 kg/h of condensate against a maximum delivery head of 35 m.

However, on CRUs with pumps that work intermittently, in order to be able to handle the rated amount of condensate, the pump has to actually move the condensate at some higher flowrate during the time it is pumping. It is important to know this to be able to size the discharge pipe correctly.

Consider that the manufacturer's data shows that the CRU will actually pump at a rate of 1.5 times the amount of condensate being handled as shown on the sizing chart i.e.:

Actual pumping rate = 1.5 x 2 000 kg/h = 3 000 kg/h

It is this figure, 3 000 kg/h, that must be used to size the discharge pipework.

It is now possible to calculate the optimum size for the return line.

Actual length of pipework = 150 m

Equivalent length of pipework = 150 m + 10% = 165 m

Estimating the friction loss in the pipe (h_f)

To size a pumped discharge line it is usually a good idea to begin the friction loss calculation with an arbitrary pressure drop of between 100 and 200 Pa/m

From the pressure drop Table 14.4.2 (extract shown below), it can be seen that, for a flowrate of 3 000 kg/h, and for a pressure drop of between 100 and 200 Pa/m, a 40 mm discharge pipe will suffice.

Extract from Table 14.4.2

Flowrate		kg/h									
Pipe size Ø		15 mm	20 mm	25 mm	32 mm	40 mm	50 mm	65 mm	80 mm	100 mm	
Pa/m	mbar/m	<0.15 m/s				0.15 m/s				0.3 m/s	
100.0	1.000	184	425	788	1 724	2 632	5 004	10 152	15 768	31 932	
120.0	1.200	202	472	871	1 897	2 898	5 508	11 196	17 352	35 100	
140.0	1.400	220	511	943	2 059	3 143	5 976	12 132	18 792	38 160	
160.0	1.600	234	547	1 015	2 210	3 373	6 408	12 996	20 160	40 680	
180.0	1.800	252	583	1 080	2 354	3 589	6 804	13 824	21 420	43 200	1.5
200.0	2.000	266	619	1 141	2 488	3 780	7 200	14 580	22 644	45 720	m/s

It can be interpolated from Table 14.4.2 that a flowrate of 3 000 kg/h will correspond to a pressure drop of 128 Pa/m, for 40 mm pipework,

The head loss to friction can now be calculated for 40 mm pipework.

Head loss to friction (h_f) = 128 Pa/m x 165 m

h_f = 21 000 Pa

h_f = Approximately 2.1 metres

Establishing the total delivery head

The total delivery head against which the pump has to discharge is therefore $h_s + h_f = h_d$, where:

h_s = static lift of 30 m (given)

h_f = 2.1 metres

h_d = 30 m + 2.1 m = 32.1 metres

The delivery head of 32.1 metres needs to be checked against the CRU manufacturer's sizing chart to confirm that the unit can pump against this amount of head. It can be seen from Figure 14.4.7 that this CRU can actually pump against a 35 metre head. Had the design head of 35 metres been exceeded, then the options are to re-calculate using a larger pipe, or to select a CRU with a greater lifting capacity.

An alternative way to size the delivery pipework

With an actual static head (h_s) of 30 m, and a CRU design head of 35 m, a 5 m head is available for pipe friction losses (h_f). It might be possible to install a smaller diameter pipe and have a larger friction loss. However, the designer must weigh this initial cost saving against the extra running power (and hence cost) required to pump against a larger head.

Velocity also needs to be checked against a typical maximum of about 3 m/s allowable for pumped water at temperatures below 100°C.

Table 14.4.2 will show that, if the next lower sized pipe (32 mm) were chosen, the unit friction loss (h_f) to pass 3 000 kg/h is interpolated to be 286 Pa/m, and the velocity is about 1 m/s, which is below 3 m/s and therefore suitable for the application.

h_f is 286 Pa/m x 165 m = 47 190 Pa (or 4.72 m)

Therefore, total delivery head (h_d) = $h_s + h_f$

h_d = 30 + 4.72 m

h_d = 34.72 m

The conclusion is that a 32 mm pipe could be used, as the CRU1 pump can handle up to 35 m total delivery head. However, from a practical viewpoint, it might not be reasonable to design a system to operate so close to its limits, and that, in this instance, 40 mm pipe would probably be the better solution.

Table 14.4.2 A section of a typical friction loss table for fully flooded pipelines (flowrates in kg/h)

Flowrate		kg/h									
Pipe size Ø		15 mm	20 mm	25 mm	32 mm	40 mm	50 mm	65 mm	80 mm	100 mm	
Pa/m	mbar/m	<0.15 m/s			0.15 m/s					0.3 m/s	
10.0	0.100	50	119	223	490	756	1 447	2 966	4 644	9 432	
12.5	0.125	58	133	252	554	853	1 634	3 348	5 220	10 656	
15.0	0.150	65	151	277	616	943	1 807	3 708	5 760	11 736	
17.5	0.175	68	162	302	670	1 026	1 966	4 032	6 264	12 744	
20.0	0.200	76	176	328	720	1 105	2 113	4 320	6 732	13 680	
22.5	0.225	79	187	349	770	1 177	2 254	4 608	7 164	14 580	0.5
25.0	0.250	83	198	371	814	1 249	2 387	4 860	7 596	15 408	m/s
27.5	0.275	90	209	389	857	1 314	2 513	5 112	7 992	16 200	
30.0	0.300	94	220	410	900	1 379	2 632	5 364	8 352	16 956	
32.5	0.325	97	230	428	940	1 440	2 747	5 616	8 712	17 712	
35.0	0.350	101	241	446	979	1 498	2 858	5 832	9 072	18 432	
37.5	0.375	104	248	464	1 015	1 555	2 966	6 048	9 396	19 116	
40.0	0.400	112	259	479	1 051	1 609	3 071	6 264	9 720	19 764	
42.5	0.425	115	266	497	1 087	1 663	3 175	6 480	10 044	20 412	
45.0	0.450	119	277	511	1 123	1 717	3 272	6 660	10 368	21 024	
47.5	0.475	122	284	526	1 156	1 768	3 370	6 876	10 656	21 636	
50.0	0.500	126	292	540	1 188	1 814	3 463	7 056	10 944	22 212	
52.5	0.525	130	299	558	1 220	1 865	3 553	7 236	11 232	22 788	
55.0	0.550	130	306	572	1 249	1 912	3 636	7 416	11 520	23 364	
57.5	0.575	133	317	583	1 282	1 958	3 744	7 596	11 808	23 904	
60.0	0.600	137	324	598	1 310	2 002	3 816	7 776	12 060	24 444	
62.5	0.625	140	331	612	1 339	2 048	3 888	7 920	12 312	24 984	
65.0	0.650	144	338	626	1 368	2 092	3 996	8 100	12 600	25 488	
67.5	0.675	148	346	637	1 397	2 131	4 068	8 280	12 852	25 992	
70.0	0.700	151	353	652	1 422	2 174	4 140	8 424	13 068	26 496	
72.5	0.725	151	356	662	1 451	2 218	4 212	8 568	13 320	27 000	
75.0	0.750	155	364	677	1 476	2 257	4 284	8 748	13 572	27 468	
77.5	0.775	158	371	688	1 505	2 297	4 356	8 892	13 788	27 972	
80.0	0.800	162	378	698	1 530	2 336	4 464	9 036	14 040	28 440	1
82.5	0.825	166	385	709	1 555	2 372	4 536	9 180	14 256	28 872	m/s
85.0	0.850	166	389	724	1 580	2 412	4 608	9 324	14 472	29 340	
87.5	0.875	169	396	734	1 606	2 448	4 680	9 468	14 724	29 772	
90.0	0.900	173	403	745	1 627	2 488	4 716	9 612	14 940	30 240	
92.5	0.925	176	407	756	1 652	2 524	4 788	9 756	15 156	30 672	
95.0	0.950	176	414	767	1 678	2 560	4 860	9 900	15 372	31 104	
97.5	0.975	180	421	778	1 699	2 596	4 932	10 044	15 552	31 500	
100.0	1.000	184	425	788	1 724	2 632	5 004	10 152	15 768	31 932	
120.0	1.200	202	472	871	1 897	2 898	5 508	11 196	17 352	35 100	
140.0	1.400	220	511	943	2 059	3 143	5 976	12 132	18 792	38 160	
160.0	1.600	234	547	1 015	2 210	3 373	6 408	12 996	20 160	40 680	
180.0	1.800	252	583	1 080	2 354	3 589	6 804	13 824	21 420	43 200	1.5
200.0	2.000	266	619	1 141	2 488	3 780	7 200	14 580	22 644	45 720	m/s
220.0	2.200	281	652	1 202	2 617	3 996	7 560	15 336	23 760	47 880	
240.0	2.400	288	680	1 256	2 740	4 176	7 920	16 056	24 876	50 400	
260.0	2.600	306	713	1 310	2 855	4 356	8 244	16 740	25 920	52 200	
280.0	2.800	317	742	1 364	2 970	4 536	8 568	17 388	26 928	54 360	
300.0	3.000	331	767	1 415	3 078	4 680	8 892	18 000	27 900	56 160	

Mechanical (positive displacement) condensate pumps

Pump operation

A mechanical pump consists of a body shell, into which condensate flows by gravity. The body contains a float mechanism, which operates a set of changeover valves.

Condensate is allowed to flow into the body, which raises the float. When the float reaches a certain level, it triggers a vent valve to close, and an inlet valve to open, to allow steam to enter and pressurise the body to push out the condensate. The condensate level and the float both fall to a preset point, at which the steam inlet valve shuts and the vent valve re-opens, allowing the pump body to refill with condensate.

Check valves are fitted to the pump inlet and discharge ports to ensure correct directional flow through the pump.

The cyclic action of the pump means that a receiver is required to store condensate while the pump is discharging (see Figure 14.4.8).

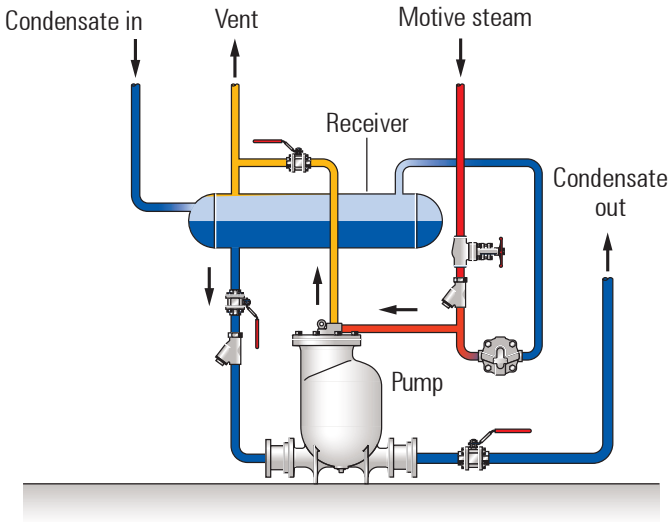


Fig. 14.4.8 A typical mechanical condensate recovery unit

Pump application

Generally, mechanical pumps handle smaller amounts of condensate than electrical pumps. They are however, particularly valuable in situations where:

- High condensate temperatures will cause cavitation in electrical pumps.
- Condensate is in vacuum.
- Plant room space is at a premium.
- Low maintenance is an issue.
- The environment is hazardous, humid or wet.
- Electrical supplies are not at hand.
- Condensate has to be removed from individual items of temperature controlled equipment, which may be subject to stall conditions (see Block 13 'Condensate Removal', for further details).

As with electrically driven pumps, positive displacement mechanical pumps are sometimes, but not always, specified as packaged condensate recovery units. A mechanical condensate recovery unit will comprise a condensate receiver and the pump unit. No additional control system is required as the pump is fully automatic and only operates when needed. This means that the pump is self-regulating.

With mechanical pumps, the pump cycles as the receiver fills and empties. The instantaneous flowrate while the pump is discharging can often be up to six times the filling rate and it is this instantaneous discharge flowrate, which must be used to calculate the size of the discharge pipe. Always refer to the pump manufacturer for data on sizing the pump and discharge line. A typical mechanical pump sizing chart is shown in Figure 14.4.10.

Sizing a mechanical condensate pump

To size a mechanical condensate pump, the following information is required:

- The maximum condensate flowrate reaching the receiver.
- The motive pressure of steam or air available to drive the pump. The selection of steam or air will depend on the application and site circumstances.
- The filling head available between the receiver and pump.
- The total delivery head of the condensate system.

The method of sizing mechanical pumps varies from manufacturer to manufacturer, and is usually based on empirical data, which are translated into factors and nomographs. The following example gives a typical method for sizing a mechanical pump. (The pipe length is less than 100 m consequently friction loss is ignored):

Example 14.4.3 How to size a mechanical condensate pump

Where:

Condensate handling load = 2 100 kg/h
Steam pressure available for operating pump = 5.2 bar g
Vertical lift from pump to return piping = 9.2 m
Pressure in the return piping (piping friction negligible) = 1.7 bar g
Available filling head on the pump = 0.3 m

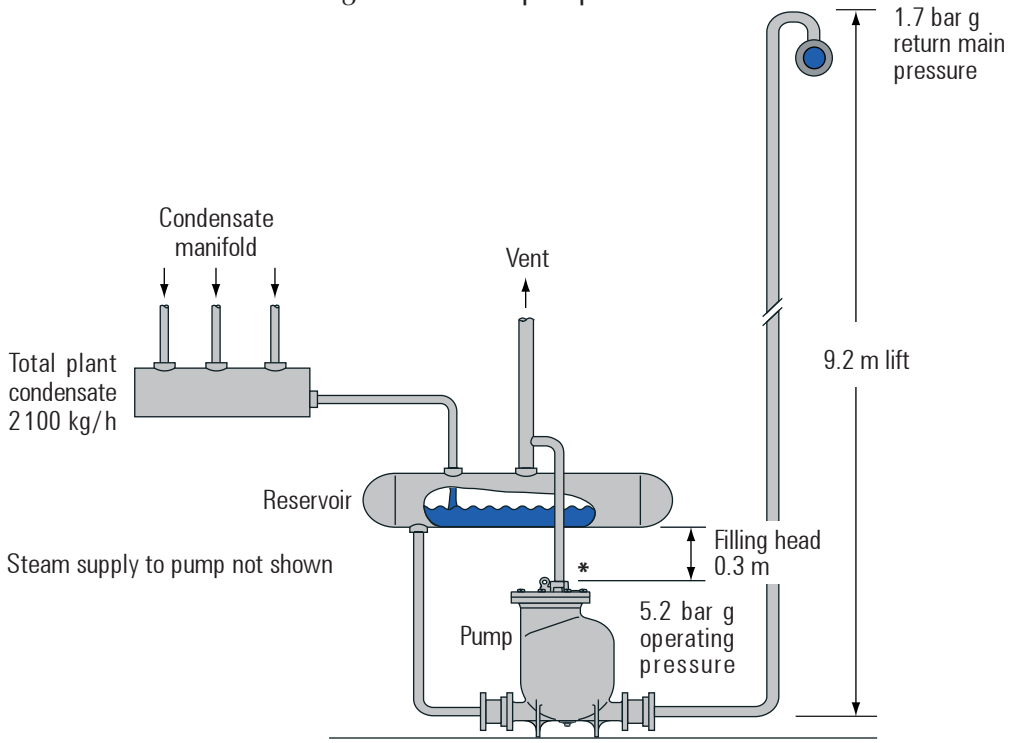


Fig. 14.4.9 Sizing a mechanical condensate recovery unit (see Example 14.4.3)

Calculate the total backpressure (delivery head h_d), against which the condensate must be pumped:

Total backpressure (h_d) = lift (h_s) + condensate pressure (h_p)

Note: The friction loss is neglected because the pipeline is shorter than 100 m.

Condensate lift (h_s) = 9.2 m
Condensate pressure (h_p) = 1.7 bar g = 17 m head
Total delivery head (h_d) = 9.2 m + 17 m
Total delivery head (h_d) = 26 m

With reference to the sizing chart shown in Figure 14.4.10: **a DN50 pump at 5.2 bar g motive pressure will pump 2 600 kg/h against a 26 m head**. A DN50 pump will thus be an adequate choice for this example, where the condensate handling load is 2100 kg/h.

Sizing the discharge pipework for a mechanical condensate pump

The discharge pipe from a mechanical pump can usually be taken to be the same size as the pump outlet when it is below 100 m long. The frictional resistance of the pipe is relatively small compared to the backpressure caused by the lift and condensate return pressure, and can usually be disregarded. For discharge pipes longer than 100 m, the general rule would be to select one pipe size larger than the pump outlet check valve, but for such longer lines, the size should be checked as shown in Example 14.4.4

Delivery lines longer than 100 metres

On delivery lines over 100 m, and/or where the condensate flow is near the pump capacity, it is advisable to check the pipe size to ensure that the total friction loss (including inertia loss) does not exceed the pump's capability. Inertia loss is explained in Example 14.4.4

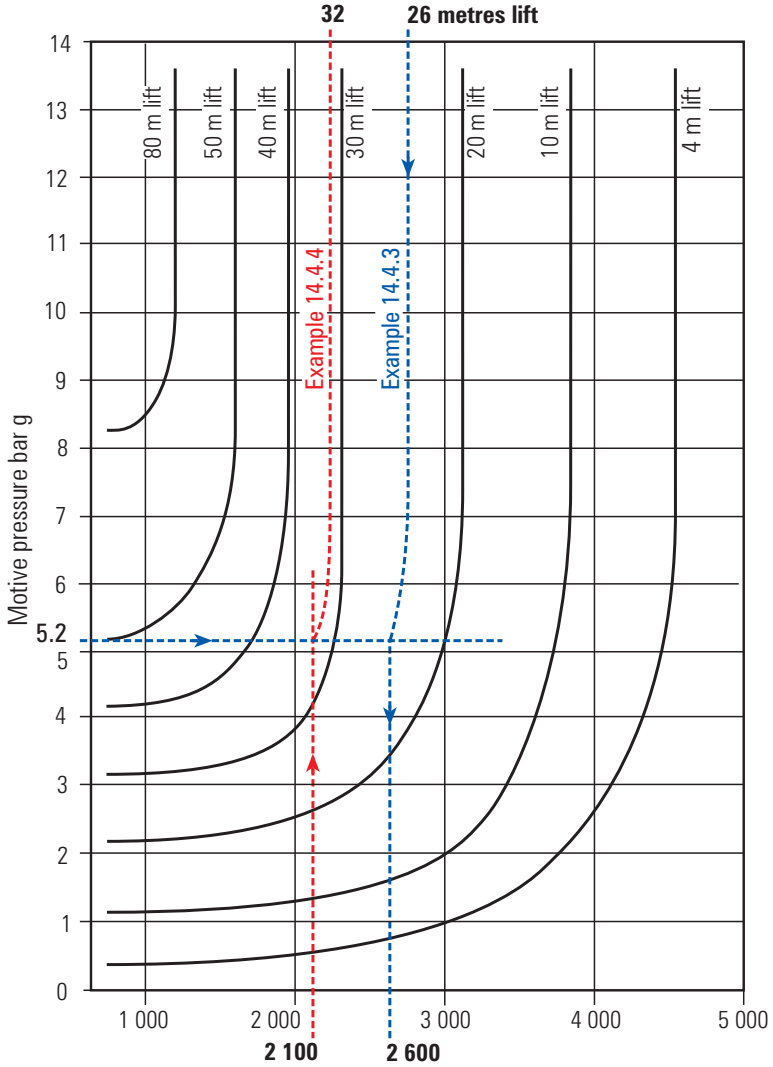
Consider the same condensate pumping requirement as in Example 14.4.3 but with a delivery line 250 metres long.

Example 14.4.4 Sizing a delivery line 250 m long (refer to Figure 14.4.10):

For a DN50 pump, with 5.2 bar g motive steam and 26 m delivery head, the maximum pump capacity = 2 600 kg/h.
From Figure 14.4.10, the following can be determined:

The actual condensate flowrate into pump = 2 100 kg/h.
Maximum backpressure permissible at 2 100 kg/h = 32 m
Therefore, maximum frictional resistance allowable = 32 - 26 m

Maximum frictional resistance allowable = 6 m (approximately 60 000 Pa)



Note: The pump is sized on the filling rate

Fig. 14.4.10 Mechanical condensate recovery unit sizing chart - DN50 pump

The effect of inertia loss on pump delivery lines longer than 100 metres.

On lines over 100 m, a considerable volume of liquid will be held within the pump discharge pipe. The sudden acceleration of this mass of liquid at the start of the pump discharge can absorb some part of the pump energy and result in a large amount of waterhammer and noise. This needs to be considered within the calculation by reducing the allowable friction loss of 60 000 Pa in Example 14.4.4 by 50%, thus:

$$\text{Total allowable friction loss} = 50\% \times 60\,000 \text{ Pa} = 30\,000 \text{ Pa}$$

$$\text{Consider delivery pipe length to be 250 m + 10\% for additional fittings} = 275 \text{ m}$$

$$\text{Consequently, maximum frictional resistance allowable / metre} = \frac{30\,000 \text{ Pa}}{275 \text{ m}}$$

$$\textbf{Maximum frictional resistance} \approx \textbf{109 Pa/m}$$

$$\text{For this type of pump the delivery flowrate is taken as 6 times the filling rate} = 6 \times 2\,100 \text{ kg/h}$$

$$\textbf{Therefore, the delivery rate of condensate from the pump} = \textbf{12\,600 kg/h}$$

Total allowable friction loss

With a frictional resistance of 109 Pa/m, Table 14.4.2 reveals that an 80 mm pipe (minimum) is required to give an acceptable flowrate of 12600 kg/h. In fact, Table 14.4.2 indicates that an 80 mm pipe will pass 16480 kg/h with a frictional resistance of 109 Pa/m.

By rising up the '80 mm column' in the table, it can be seen that, by interpolation, the flowrate of 12 600 kg/h actually induces a frictional loss of 65 Pa/m in an 80 mm pipe.

Fully loaded pumps and longer lines

In Example 14.4.4, Figure 14.4.10 shows that the maximum pump filling rate with a motive pressure of 5.2 bar g and a delivery head of 26 metres is 2 600 kg/h. Had the filling rate been close to this maximum, (perhaps 2 500 kg/h), then less delivery head would have been available for friction loss. For the same size DN50 pump, this would mean a larger delivery pipeline as shown in Example 14.4.5

Example 14.4.5 Consider the same DN50 pump as described in Example 14.4.4, but having a condensate filling rate of 2 500 kg/h. Now determine the size of the delivery pipeline.

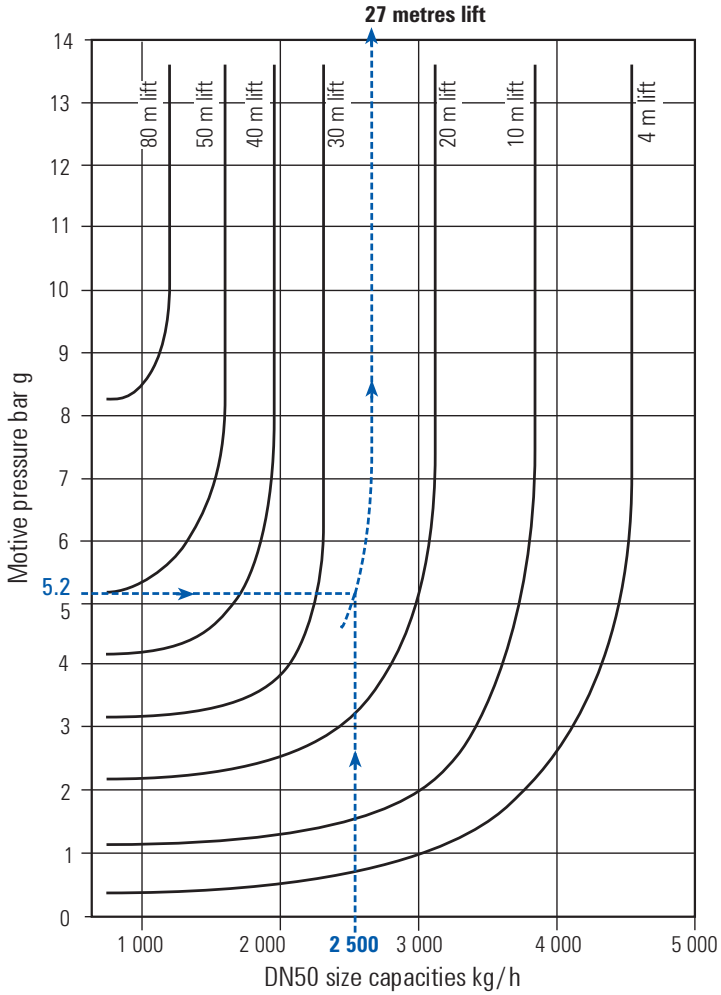


Fig. 14.4.11 Mechanical condensate recovery unit sizing chart (DN50 pump)

Sizing on a filling rate of 2 500 kg/h, and a steam pressure of 5.2 bar, referring to Figure 14.4.11, for the DN50 pump, it can be seen that a condensate filling rate of 2 500 kg/h equates to a maximum backpressure of about 27 m, so in this instance:
With an actual delivery head of 26 m:

$$\text{Available head left for friction losses} = 27 - 26 \text{ m}$$
$$\text{Available head left for friction losses} = 1 \text{ m}$$

Conversion tables reveal that a 1 m head of water is equivalent to 9 806.65 Pa.

For an equivalent length line of 275 m:

$$\text{The frictional resistance allowable} = \frac{9806.65 \text{ Pa}}{275 \text{ m}}$$
$$= 35.7 \text{ Pa/m}$$
$$\text{Minus allowance of 50\% for inertia loss} = 50\% \times 35.7 \text{ Pa/m}$$
$$\text{Maximum frictional resistance allowable} = 18 \text{ Pa/m}$$

As before, the discharge pipework has to be sized on the instantaneous flowrate from the pump outlet, which is taken as 6 × the filling rate. In this instance, the pipe would have been sized on 6 × 2 500 kg/h = 15 000 kg/h with a friction loss of 18 Pa/m.

Table 14.4.2 shows that this would require a pipe larger than 100 mm (actually 125 mm) to allow the pump to operate within its capability.

Although the system would certainly work with this arrangement, it is probably more economical to consider a larger pump in conjunction with a smaller pipeline.

Considerations of a larger pump and smaller pipeline

Consider the same pumping conditions as Example 14.4.4, but with a larger DN80 pump. As a larger unit can pump against a higher delivery head, a smaller delivery line can be used

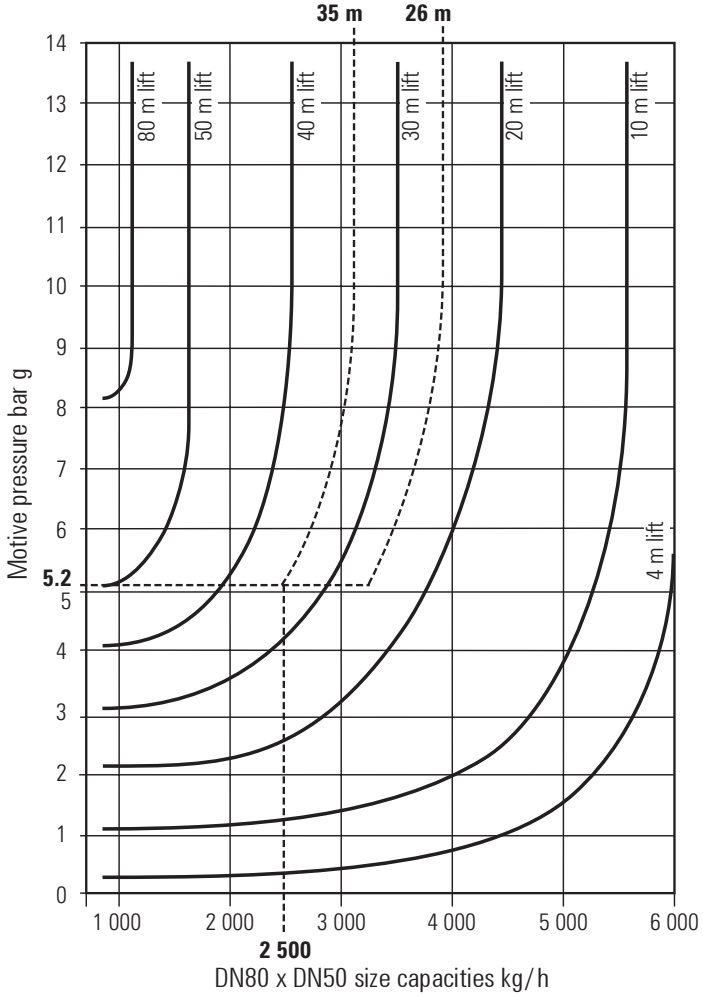


Fig. 14.4.12 Mechanical condensate recovery unit sizing chart (DN80 pump)

Figure 14.4.12 shows that a DN80 pump under the same conditions of 5.2 bar g motive steam and 2 500 kg/h flowrate would allow a maximum delivery head of 35 m.

From Example 14.4.4, the actual delivery head = 26 m
At a filling rate of 2 500 kg/h, maximum allowed = 35 m
Head available for friction loss = 35 m - 26 m = 9 metres

Conversion tables reveal that a 9 m head of water is equivalent to 88 259.9 Pa.

$$\text{Therefore 88 259.9 Pa over 275 m and including inertia loss} = 50\% \times \frac{88\,259.9 \text{ Pa}}{275 \text{ m}}$$

Maximum frictional resistance allowable = 160 Pa / m

The delivery pipe is again sized to carry 6 x 2 500 kg/h = 15 000 kg/h of condensate. By interpolation, Table 14.4.2 shows that an 80 mm pipe will accommodate 20 160 kg/h with a friction loss of 160 Pa/m, flowing at about 1 m/s.

In this instance, the larger DN80 pump will comfortably allow a pipe two sizes smaller than that for the smaller pump, and with a velocity of about 1 m/s, which is within recommendations. The 80 mm pipe is therefore suitable for the DN80 pump.

Note: The DN80 pump would cost about 10% more than the DN50 pump, but the extra cost would be justified by the difference in installation costs on long delivery lines; which in this instance would mean the difference in cost between a 80 mm and 125 mm pipe; installation, fittings, and insulation.

Condensate velocities

Equation 14.4.2 can be used to check the condensate velocity.

$$\text{Condensate velocity (m/s)} = \frac{\text{Condensate flowrate (kg/h)} \times \text{Condensate specific volume (m}^3\text{/kg)}}{(\text{pipe bore[mm]})^2 \times 0.002827}$$

Equation 14.4.2

$$\text{Condensate velocity (m/s)} = \frac{\text{kg/h} \times \text{m}^3\text{/kg}}{(\text{mm})^2 \times 0.002\,827}$$

In Equation 14.4.2, the specific volume of water is taken to be 0.001 m³/kg. This value varies slightly with temperature but not enough to make any significant difference on condensate lines.

The condensate velocity can be checked for the 80 mm pipework in Example 14.4.4.

$$\text{The pumping rate} = 15\,000 \text{ kg/h}$$
$$\text{Condensate specific volume} = 0.001 \text{ m}^3\text{/kg}$$
$$\text{Pipe bore} = 80 \text{ mm}$$

$$\text{Condensate velocity} = \frac{15000 \times 0.001}{80^2 \times 0.002827}$$

$$\textbf{Condensate velocity} = \textbf{0.83 m/s}$$

From Table 14.4.3 the maximum velocity for an 80 mm bore pipe is 1.84 m/s.

Table 14.4.3 Maximum recommended velocities for pipe bores (based on a maximum friction loss of 450 Pa/m)

Pipe bore, mm	15	20	25	32	40	50	65	80	100
Velocity, m/s	0.62	0.8	1.0	1.23	1.27	1.5	1.8	1.84	2.4

Best practice for long delivery lines

The momentum of the moving contents of a long delivery line may keep the water in motion for some time after a mechanical pump has completed its discharge stroke.

When the water in the discharge pipe comes to rest, the backpressure in the line will attempt to reverse the initial flow of water, back towards the outlet check valve. The result is noise and pipe movement due to waterhammer, which can be both alarming and serious. Installing another check valve in the discharge pipe one pipe length from the pump will usually alleviate the problem.



Fig. 14.4.13 An additional check valve 1 pipe length from the pump body to reduce the effect of backflow

If there is any choice, it is always best to lift immediately after the pump to a height allowing a gravity fall to the end of the line (Figure 14.4.14). If the fall is enough to overcome the frictional resistance of the pipe (Table 14.4.4), then the only backpressure onto the pump is that formed by the initial lift. A vacuum breaker can be installed at the top of the lift not only to assist the flow along the falling line but also to prevent any tendency for backflow at the end of the stroke.

Should the falling line have to fall anywhere along its length to overcome an obstruction, then an automatic air vent fitted at the highest point will reduce air locking and assist flow around the obstruction, see Figure 14.4.14.

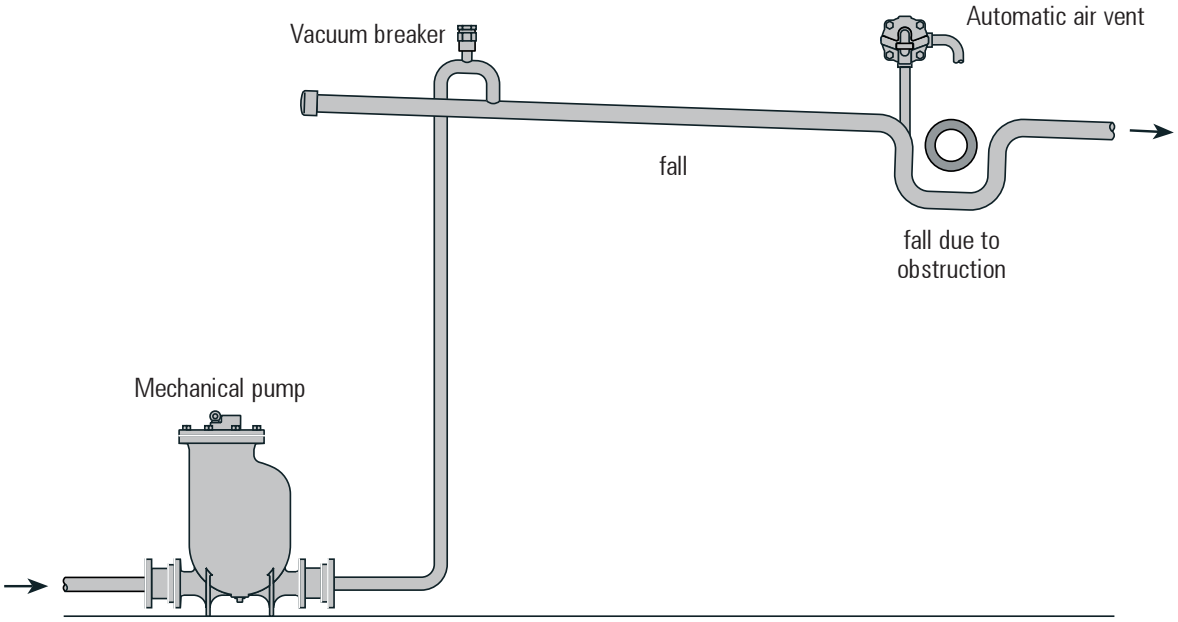


Fig. 14.4.14 Best choice - lift after the pump

Table 14.4.4 Pipefall to overcome frictional losses

Pipefall needed to overcome pipe friction	Pipe size (DN mm)										
	15	20	25	32	40	50	65	80	100	125	150
	Litres of water per hour										
25 mm in 15 m	48	140	303	580	907	1 950	3 538	5 806	12 610	22 906	37 284
25 mm in 10 m	59	177	381	694	1 134	2 449	4 445	7 257	15 680	28 576	46 492
25 mm in 8 m	69	204	442	800	1 310	2 834	5 148	8 391	18 159	33 089	53 862
25 mm in 6 m	79	231	503	907	1 487	3 220	5 851	9 525	20 638	37 602	61 223
25 mm in 5 m	86	256	553	1 007	1 642	3 551	6 441	10 568	22 770	41 821	67 538
25 mm in 4 m	93	279	598	1 093	1 778	3 878	7 030	11 521	24 811	45 994	73 571
25 mm in 3 m	113	338	730	1 329	2 168	4 672	8 527	13 925	30 073	54 073	89 356
25 mm in 2 m	140	419	907	1 655	2 694	5 851	10 614	17 327	37 421	68 039	111 128
25 mm in 1.75 m*	152	454	984	1 793	2 923	6 327	11 498	18 756	40 573	73 708	120 426
25 mm in 1.5 m	165	490	1 061	1 932	3 152	6 804	12 383	20 185	43 726	79 378	129 725
25 mm in 1 m	206	612	1 324	2 404	3 923	8 482	15 422	25 174	54 431	99 019	161 476

*A fall of 25 mm in 1.75 m is equivalent to a fall of 1:70.

Alternatively, any question of backpressure caused by the horizontal run can be entirely eliminated by an arrangement as in Figure 14.4.15 in which the pump simply lifts into a vented break tank. The pipe from the tank should fall in accordance with Table 14.4.4.

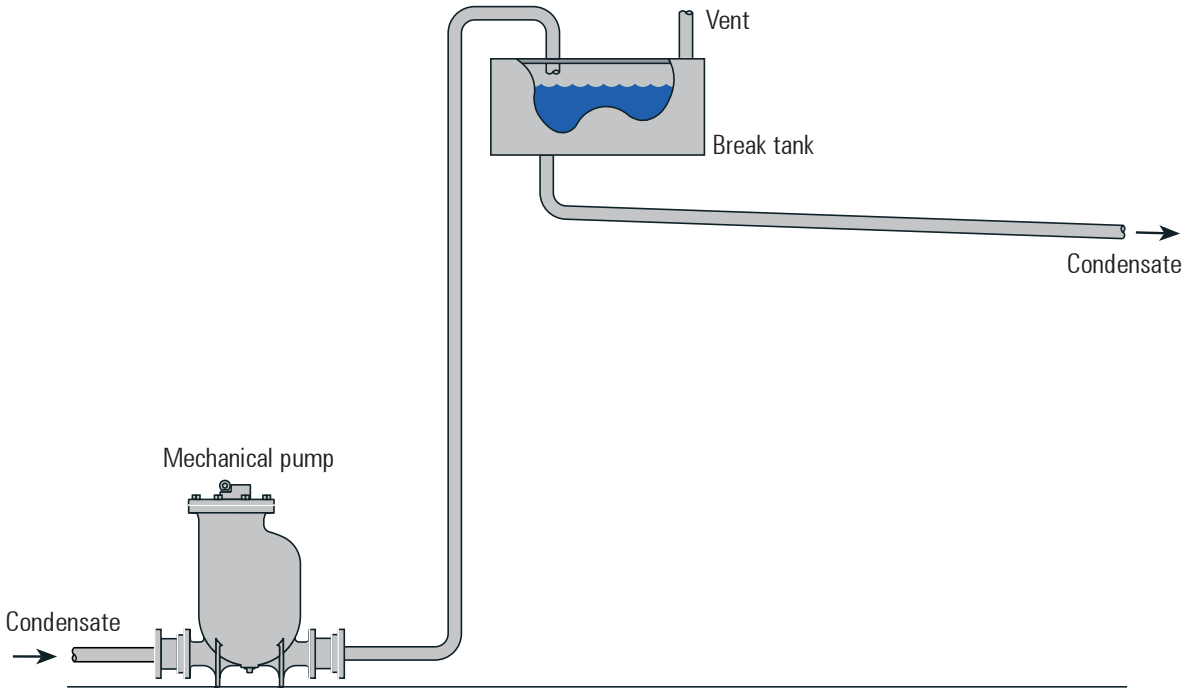


Fig. 14.4.15 Alternative choice - lift after the pump to a break tank

Vented pumps, pumping traps and pump-trap installations

Discharge lines from pumps vented to atmosphere are sized on the discharge rate of the pump.

Condensate passing through pumping traps and pump-trap combinations in closed loop applications will often be at higher pressures and temperatures and flash steam will be formed in the discharge line.

Because of this, discharge lines from pumping traps and pump-trap combinations are sized on the trapping condition at full-load and not the pumping condition, as the line has to be sized to cater for flash steam. Sizing on flash steam will ensure the line is also able to cope with the pumping condition.

Questions

1. For pumping condensate, what is the total delivery head?

a| Pressure required to raise the condensate to the required level ☐

b| Pressure required to move the condensate through the pipes ☐

c| Pressure in the condensate system ☐

d| All of the above ☐

2. What is the important factor to consider when sizing a pump discharge line?

a| The pump filling rate ☐

b| The pump discharge rate ☐

c| The size of the pump discharge connection ☐

d| The size of the pump inlet connection ☐

3. For a mechanical pump, what is the net static head?

a| The static delivery head ☐

b| The static delivery head less the filling head ☐

c| The static delivery head less the static suction head ☐

d| All of the above ☐

4. As a general rule, what equivalent length is added to pipe length to account for pipe fittings?

a| 5% ☐

b| 10% ☐

c| 15% ☐

d| 20% ☐

5. What is a good arbitrary pressure drop to choose to initially size a pumped delivery line?

a| 10 to 20 Pa/m ☐

b| 50 to 100 Pa/m ☐

c| 500 to 1000 Pa/m ☐

d| 100 to 200 Pa/m ☐

6. In Figure 14.4.7, what is the maximum capacity of a CRU3 pumping unit against a 15 metre delivery head?

a| 2000 kg/h ☐

b| 100 kg/h ☐

c| 500 kg/h ☐

d| 1400 kg/h ☐

Answers

1: d, 2: b, 3: a, 4:b, 5: d, 6: d

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Module 14.5

Lifting Condensate and Contaminated Condensate

Lifting Condensate and Contaminated Condensate

Lifting condensate from a steam main

It is sometimes necessary to lift condensate from a steam trap to a higher level condensate return line (Figure 14.5.1). The condensate will rise up the lifting pipework when the steam pressure upstream of the trap is higher than the pressure downstream of the trap.

The pressure downstream of the trap is generally called backpressure, and is made up of any pressure existing in the condensate line plus the static lift caused by condensate in the rising pipework. The upstream pressure will vary between start-up conditions, when it is at its lowest, and running conditions, when it is at its highest.

Backpressure is related to lift by using the following approximate conversion:
1 metre lift in pipework = 1 m head static pressure ≅ 0.1 bar backpressure

If a head of 5 m produces a backpressure of 0.5 bar, then this reduces the differential pressure available to push condensate through the trap; although under running conditions the reduction in trap capacity is likely to be significant only where low upstream pressures are used.

In steam mains at start-up, the steam pressure is likely to be very low, and it is common for water to back-up before the trap, which can lead to waterhammer in the space being drained. To alleviate this problem at start-up, a liquid expansion trap, fitted as shown in Figure 14.5.1, will discharge any cold condensate formed at this time to waste.

As the steam main is warmed, the condensate temperature rises, causing the liquid expansion trap to close. At the same time, the steam pressure rises, forcing the hot condensate through the 'working' drain trap to the return line.

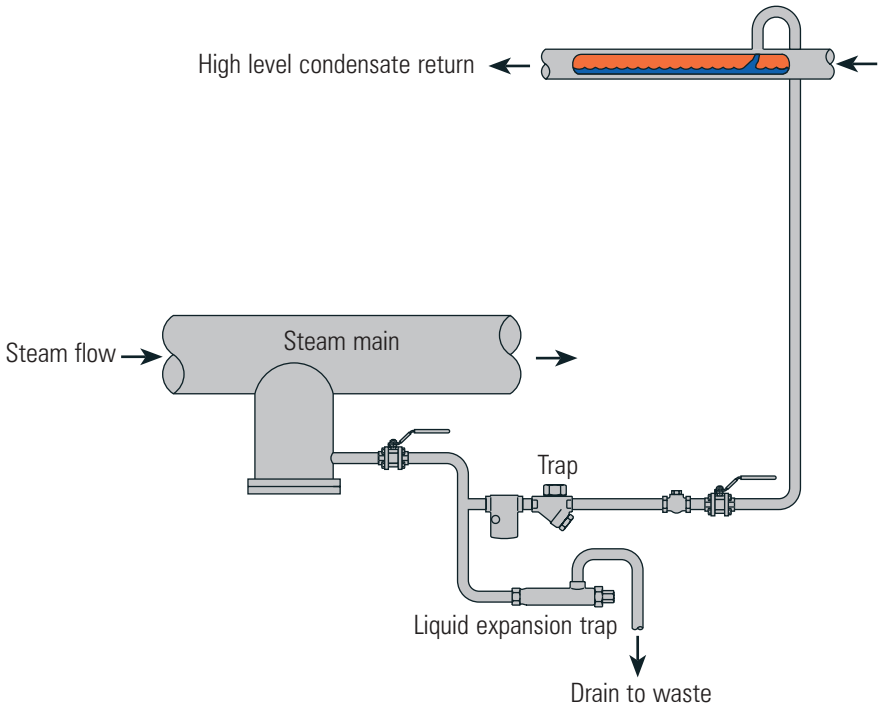


Fig. 14.5.1 Use of a liquid expansion trap

The discharge line from the trap to the overhead return line, preferably discharges into the top of the main rather than simply feed to the underside, as shown in Figure 14.5.1. This assists operation, because although the riser is probably full of water at start-up, it sometimes contains little more than flash steam once hot condensate under pressure passes through. If the discharge line were fitted to the bottom of the return line, it would fill with condensate after each discharge and increase the tendency for waterhammer and noise.

It is also recommended that a check valve be fitted after any steam trap from where condensate is lifted, preventing condensate from falling back towards the trap.

The above general recommendations apply not just to traps lifting condensate from steam mains, but also to traps draining any type of process running at a constant steam pressure.

Temperature controlled processes will often run with low steam pressures. Rising condensate discharge lines should be avoided at all costs, unless automatic pump-traps are used.

Contaminated condensate

Occasionally, condensate is discharged from sources where it might have become contaminated by corrosive process liquids. This is unsuitable for boiler feedwater because of the dangers of foaming, scaling, and corrosion which it can cause in the boiler and distribution pipes.

However, although contaminated, the condensate still carries the same useful heat as clean condensate which could be recovered if proper contamination detection equipment were employed.

Such equipment detects changes in condensate conductivity. When a change from the desired conductivity occurs then this may mean that the condensate is contaminated. A controller signals a dump valve to open, allowing the condensate to flow to drain.

In some countries, continuous monitoring of condensate is a legal requirement.

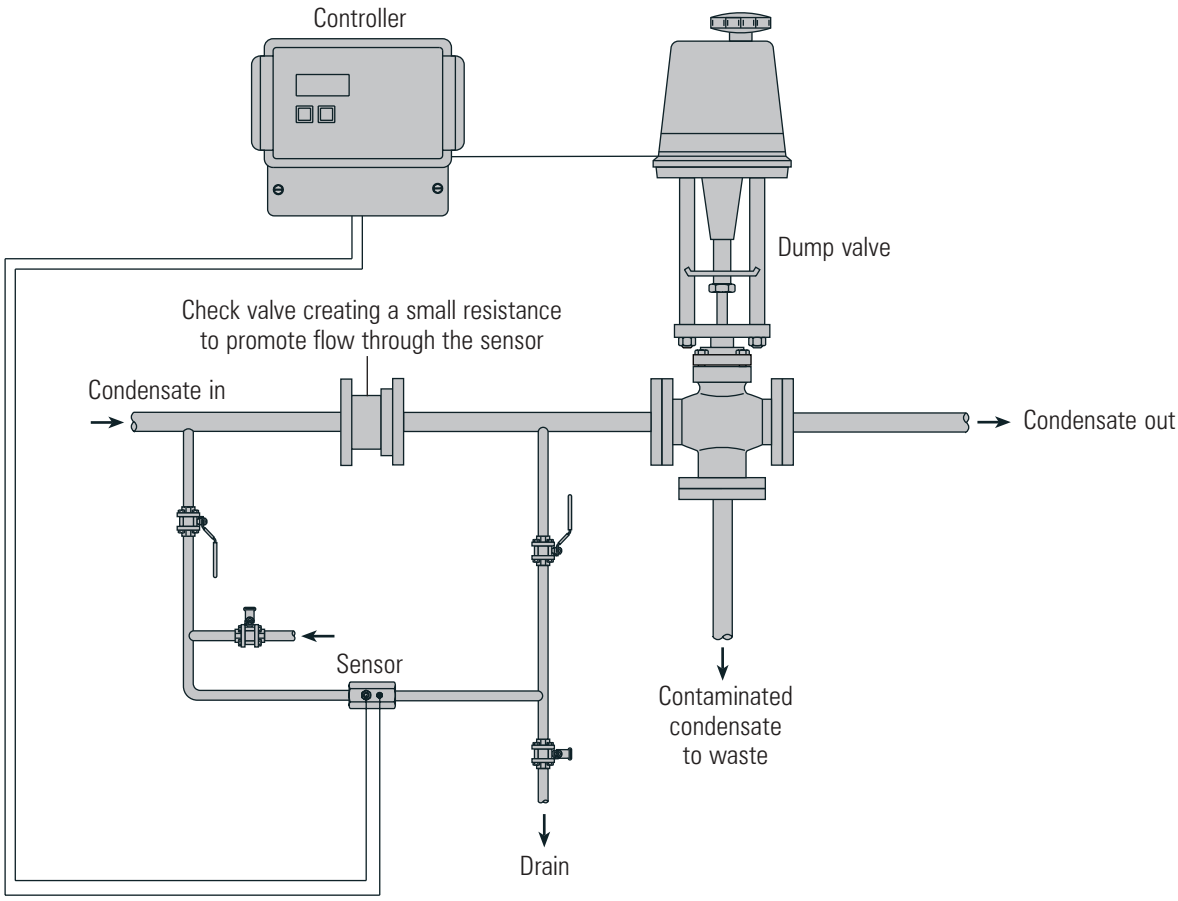


Fig. 14.5.2 Condensate contamination detection equipment

Questions

1. Approximately how much backpressure will 15 m head of water produce?

a | 0.15 bar ☐

b | 1.5 bar ☐

c | 15 bar ☐

d | 15 000 Pa ☐

2. What type of steam trap can assist in draining steam mains at start-up?

a | Thermodynamic type ☐

b | Float-thermostatic type ☐

c | Thermostatic type ☐

d | Liquid expansion type ☐

3. Why is it sensible to dump contaminated condensate?

a | It can corrode steam boilers and distribution pipework ☐

b | It can cause scale in steam boilers and distribution pipework ☐

c | It can cause the boiler water to foam and create carryover ☐

d | All of the above ☐

4. Why is it good practice to run a trap discharge line into the top of any condensate return main?

a | It is cheaper ☐

b | It removes the backpressure ☐

c | It helps to keep the rising line free of residual condensate ☐

d | It removes the static lift ☐

Answers

1: b 2: d, 3: d, 4: c

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Module 14.6

Flash Steam

Flash Steam

The formation of flash has already been discussed in Module 2.2, 'What is steam', and a major flash steam application has been covered in Module 3.13, 'Heat recovery from boiler blowdown'. This Module will provide a brief reminder of these earlier Modules; discuss how flash steam is formed, and focus on how flash steam can be used effectively to improve steam plant efficiency.

What is flash steam and why should it be used?

'Flash steam' is released from hot condensate when its pressure is reduced. Even water at an ambient room temperature of 20°C would boil if its pressure were lowered far enough. It may be worth noting that water at 170°C will boil at any pressure below 6.9 bar g. The steam released by the flashing process is as useful as steam released from a steam boiler.

As an example, when steam is taken from a boiler and the boiler pressure drops, some of the water content of the boiler will flash off to supplement the 'live' steam produced by the heat from the boiler fuel. Because both types of steam are produced in the boiler, it is impossible to differentiate between them. Only when flashing takes place at relatively low pressure, such as at the discharge side of steam traps, is the term flash steam widely used. Unfortunately, this usage has led to the erroneous conclusion that flash steam is in some way less valuable than so-called live steam.

In any steam system seeking to maximise efficiency, flash steam will be separated from the condensate, and used to supplement any low pressure heating application. Every kilogram of flash steam used in this way is a kilogram of steam that does not need to be supplied by the boiler. It is also a kilogram of steam not vented to atmosphere, from where it would otherwise be lost.

The reasons for the recovery of flash steam are just as compelling, both economically and environmentally, as the reasons for recovering condensate.

How much flash steam is available?

If use is to be made of flash steam, it is helpful to know how much of it will be available. The quantity is readily determined by calculation, or can be read from simple tables or charts.

Example 14.6.1 - Consider the jacketed vessel shown in Figure 14.6.1

The condensate enters the steam trap as saturated water, at a gauge pressure of 7 bar g and a temperature of 170°C. The specific amount of heat in the condensate at this pressure is 721 kJ/kg.

After passing through the steam trap, the pressure in the condensate return line is 0 bar g. At this pressure, the maximum amount of heat each kilogram of condensate can hold is 419 kJ and the maximum temperature is 100°C. There is an excess of 302 kJ of heat which evaporates some of the condensate into steam. The quantity of steam is calculated in the following text.

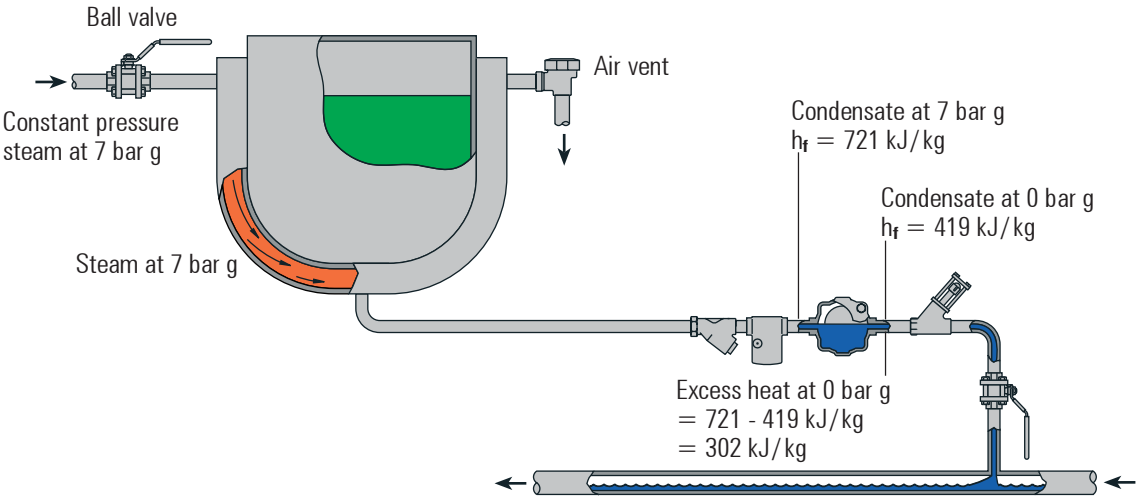


Fig. 14.6.1 Excess heat in condensate produces flash steam

The heat needed to produce 1 kg of saturated steam from water at the same temperature, at 0 bar gauge, is 2 257 kJ. An amount of 302 kJ can therefore evaporate:

$$\frac{302 \text{ kJ}}{2257 \text{ kJ}} \approx 0.134 \text{ kg of steam per kg of condensate}$$

From each kilogram of condensate in this example, the proportion of flash steam generated therefore equals 13.4% of the initial mass of condensate.

If the equipment using steam at 7 bar g were condensing 250 kg/h, then the amount of flash steam released by the condensate at 0 bar g would be:

$$0.134 \times 250 \text{ kg/h of condensate} = 33.5 \text{ kg/h of flash steam}$$

Alternatively, the chart in Figure 14.6.2 can be read directly for the moderate and low pressures encountered in many plants.

The example shown in Figure 14.6.1 is depicted in Figure 14.6.2 and shows that 0.134 kg of flash steam is produced per kg of condensate passing through the trap.

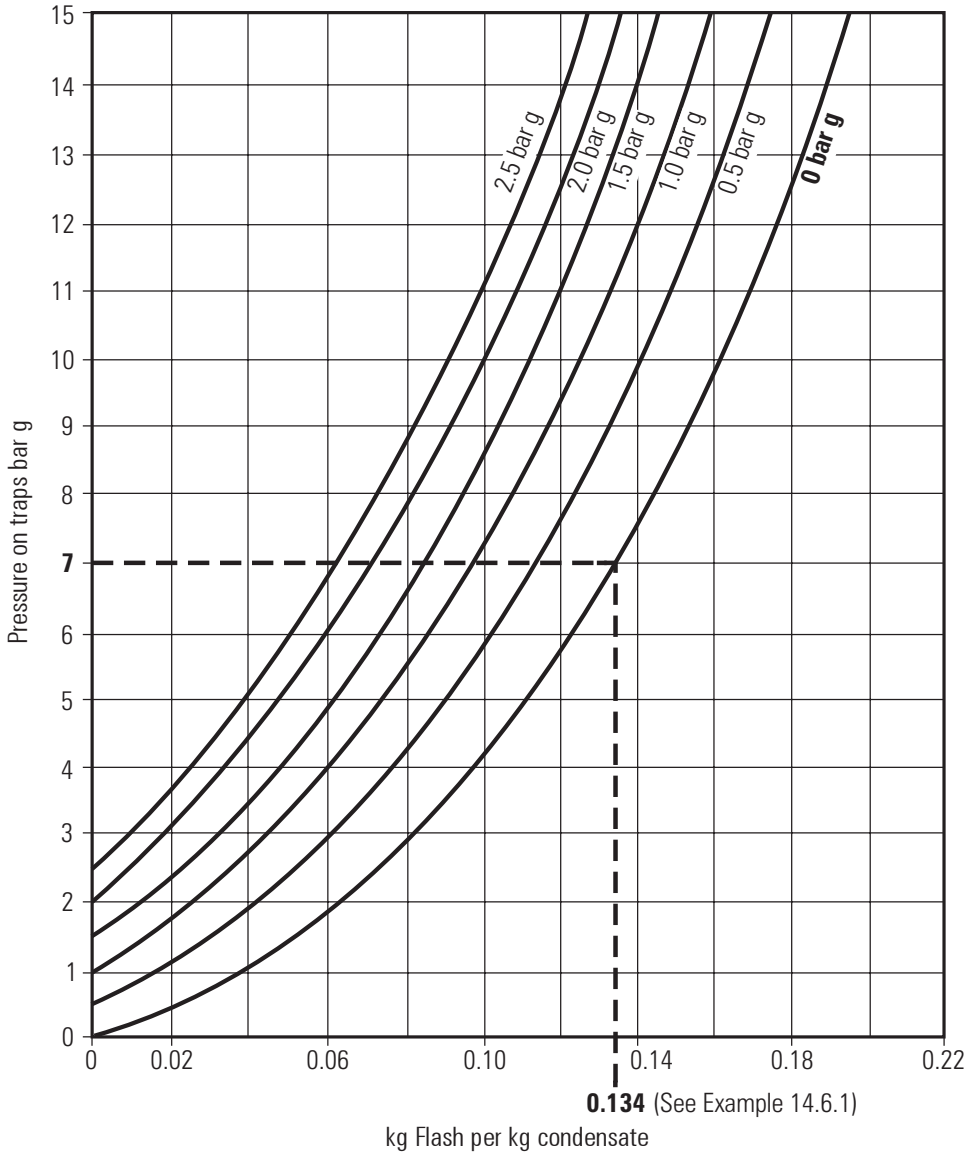


Fig. 14.6.2 Flash steam graph

Sub-cooled condensate

If the steam trap is of a thermostatic type, the discharged condensate is sub-cooled below saturation temperature. The heat in the cooler condensate will be slightly less, and the amount of flash steam produced would be less.

If the trap in Example 14.6.1 discharged condensate at 15°C below the steam saturation temperature, then the available heat in the condensate would be less.

Example 14.6.2 Consider condensate discharging at 7 bar g and with 15°C of subcooling

Temperature of saturated condensate at 7 bar g = 170°C

Amount of sub cooling = 15°C

Temperature of sub-cooled condensate at 7 bar g = 155°C

From steam tables:

Amount of heat in condensate at 155°C = 654 kJ/kg

At 0 bar g, saturated condensate can only hold = 419 kJ/kg

Surplus heat in saturated condensate at 0 bar g = 235 kJ/kg

Heat in steam at 0 bar g = 2 257 kJ/kg

$$\text{Proportion of flash steam} = \frac{235 \text{ kJ/kg}}{2257 \text{ kJ/kg}}$$

Proportion of flash steam from the condensate = 0.104 (10.4%)

Therefore, in this example, condensate discharging at a temperature lower than the saturation temperature has reduced the proportion of flash steam from 13.4% to 10.4%.

Pressurised condensate

Example 14.6.3 Consider the condensate in Example 14.6.1 discharging to a flash vessel pressurised at 1 bar g

If the return line were connected to a vessel at a pressure of 1 bar g, then it could be seen from steam tables that the maximum heat in the condensate at the trap discharge would be 505 kJ/kg and the enthalpy of evaporation at 1 bar g would be 2 201 kJ/kg.

The proportion of the condensate flashing off at 1 bar g can then be calculated as follows:

Heat in condensate at 7 bar g = 721 kJ/kg

At 1 bar g saturated condensate can only hold = 505 kJ/kg

Surplus heat in saturated condensate at 1 bar g = 216 kJ/kg

Heat in steam at 1 bar g = 2 201 kJ/kg

$$\text{Proportion of flash steam} = \frac{216 \text{ kJ/kg}}{2\,201 \text{ kJ/kg}}$$

Proportion of flash steam from the condensate = 0.098 (9.8%)

In this example, if the equipment using steam at 7 bar g were condensing 250 kg/h of steam, then the amount of flash steam released by the condensate at 1 bar g would be 0.098 x 250 kg/h = 24.5 kg/h of flash steam.

Therefore, the amount of flash steam produced can depend on the type of steam trap used, the steam pressure before the trap, and the condensate pressure after the trap.

The flash steam recovery vessel (flash vessel)

Flash vessels are used to separate flash steam from condensate. Figure 14.6.3 shows a typical flash vessel constructed in compliance with the European Pressure Equipment Directive 97/23/EC.

After condensate and flash steam enter the flash vessel, the condensate falls by gravity to the base of the vessel, from where it is drained, via a float trap, usually to a vented receiver from where it can be pumped. The flash steam in the vessel is piped from the top of the vessel to any appropriate low pressure steam equipment.

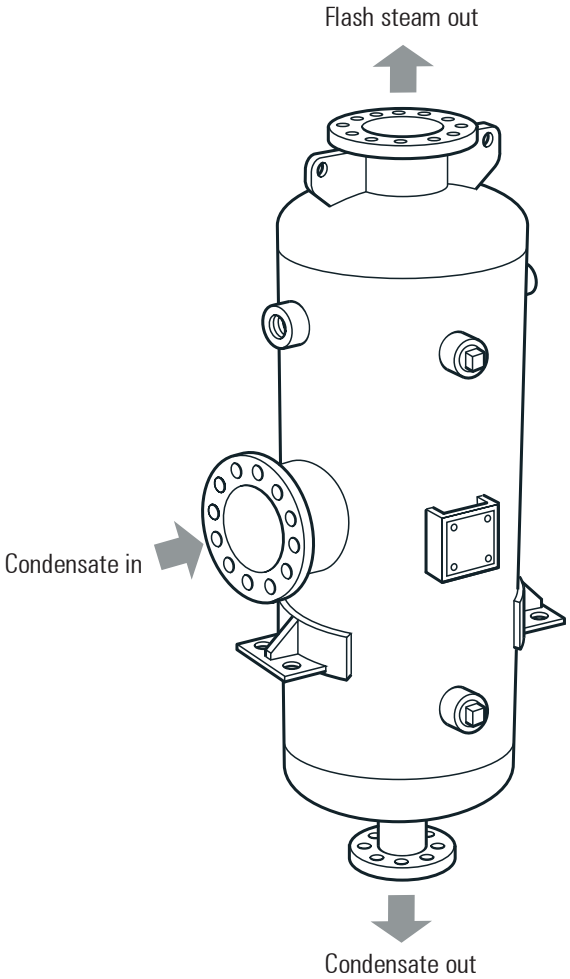


Fig. 14.6.3 A typical flash vessel constructed to European standards

Sizing flash steam recovery vessels

To size a flash vessel, the following information is required:

- The steam pressure before the steam trap(s) supplying the vessel.
- The total condensate flowrate into the flash vessel.
- The flash steam pressure in the flash vessel.

Using this information, together with a flash vessel sizing chart (see Figure 14.6.4), the size of the vessel can be determined. Example 14.6.4 demonstrates flash vessel sizing, using a chart.

Example 14.6.4 Determine the size of a flash vessel to suit the following conditions:
The pressure onto the steam traps is 12 bar g with a total condensate flow of 2 500 kg/h. The flash steam from the vessel is to be supplied to equipment using low pressure steam at 1 bar g.

Method:
1. From the 'Pressure on steam traps' axis at 12 bar g, move horizontally to the 1 bar g flash steam pressure curve at point **A**.
2. Drop down vertically to the condensate flowrate level of 2 500 kg/h, point **B**, and follow the curved line to point **C**.
3. Move right from point **C** to meet the 1 bar g flash line at point **D**.
4. Move upwards to the flash vessel size and select the vessel.

For this example, an FV8 flash vessel would be selected.

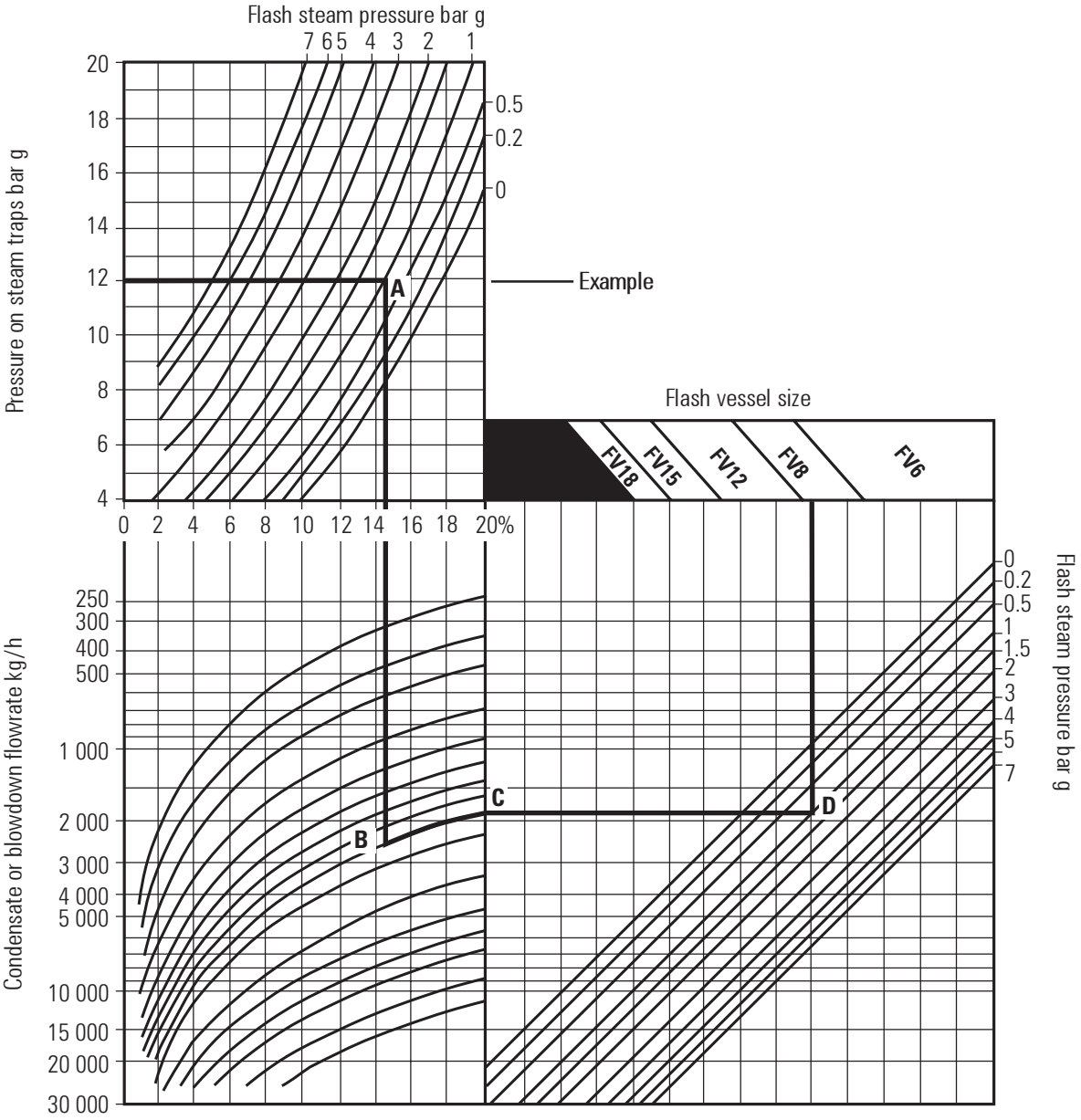


Fig. 14.6.4 Flash vessel sizing chart

Requirements for successful flash steam applications

If full use is to be made of flash steam, some basic requirements must be satisfied:

- It is essential to have a continual supply of sufficient condensate from applications operating at higher pressures, to ensure that enough flash steam can be released for economic recovery.
- The steam traps and the equipment they are draining must be able to function satisfactorily against the backpressure applied by the flash system.

- Care must be taken when attempting flash steam recovery with condensate from temperature controlled equipment. At less than full-load, the steam space pressure will be lowered by the closing action of the steam control valve. If the steam pressure in the equipment approaches or falls below the specified flash steam pressure, the overall amount of flash steam formed will be marginal, and one must question whether recovery is worthwhile in this instance.
- It is important that there is a demand for low pressure flash steam that either equals or exceeds the flash steam being produced. Any deficit of flash steam can be made up by live steam from a pressure reducing valve. If the supply of flash steam exceeds its demand, surplus pressure will be created in the flash steam distribution system, which will then have to be vented to waste through a surplussing valve.
- It is possible to utilise the flash steam from condensate on a space heating installation - but savings will only be achieved during the heating season. When heating is not required, the recovery system becomes ineffective. Wherever possible, the best arrangement is to use flash steam from process condensate to supply process loads - and flash steam from heating condensate to supply heating loads. Supply and demand are then more likely to remain in-step.
- It is preferable to actually use the flash steam close to the high pressure condensate source. Relatively large diameter pipes are used for low pressure steam, to reduce pressure loss and velocity, which can mean costly installation if the flash steam has to be piped any distance.

Control of flash steam pressure

Another consideration is a method of controlling the pressure of the flash steam.

In some cases, flash pressure will find its own level and nothing more needs to be done. When supply and demand are always in-step, and particularly if the low pressure steam is used on the same equipment producing the high pressure condensate, it is only neccessary to pipe the flash steam to the low pressure plant without any other control.

Figure 14.6.5 shows the application of flash steam recovery to a multi-bank air heater battery, which is supplying high temperature air to a process. Condensate from the high pressure sections is taken to the flash vessel, from where the low pressure flash steam is used, to preheat the cold air entering the battery via the frost coil (preheater). The surface area of the preheater section, and the relatively low temperature of the incoming air, will mean that the low pressure flash steam is readily condensed.

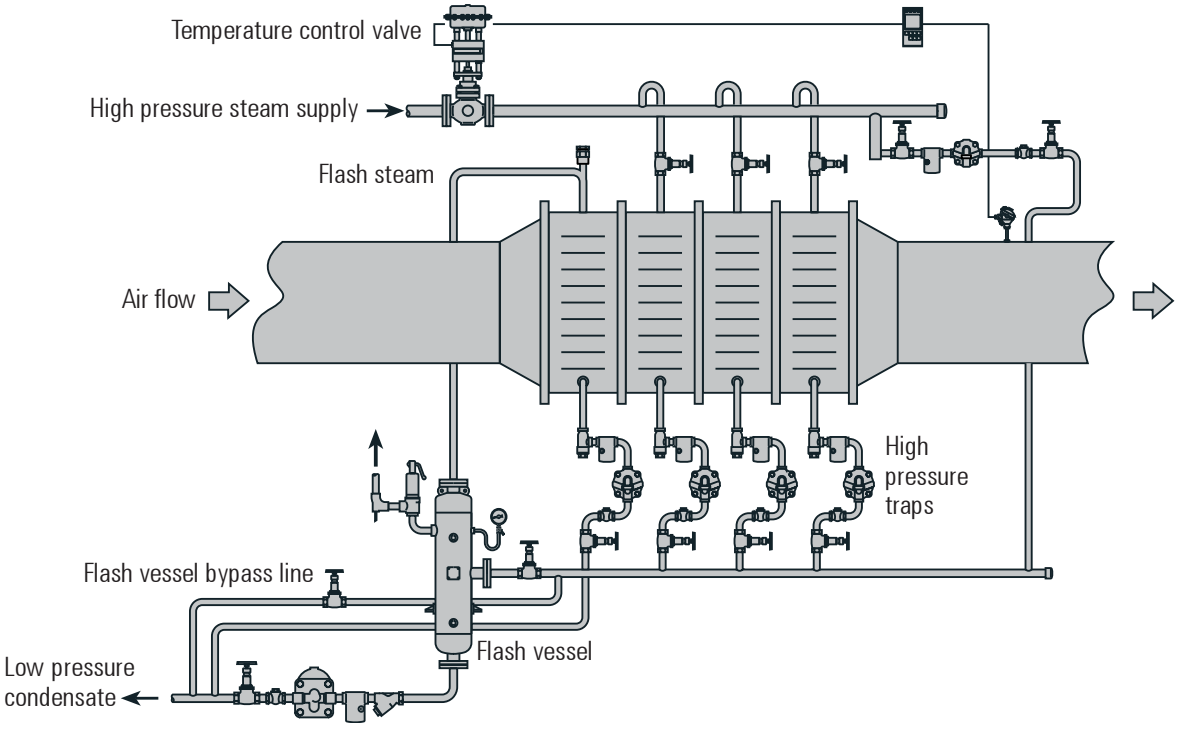


Fig. 14.6.5 Flash steam recovery on a multi-bank air heater battery

Depending on operating temperatures, the flash steam will condense at some low pressure, perhaps even sub-atmospheric. If site conditions and layout permit, the flash vessel and the steam trap draining the preheater should be located far enough below the preheater condensate outlet to give enough hydrostatic head to push the condensate through the trap. If this is not possible, pumping traps can be used to drain both the preheater coil and the flash vessel.

Steam condensing in the preheater at sub-atmospheric pressure will generally mean that a vacuum breaker is required on the flash steam supply to the preheater. This will prevent the pressure in the battery becoming sub-atmospheric, thereby assisting condensate flow to the trap. Drainage from the preheater trap is induced by gravity flow.

Figure 14.6.6 shows an application where the flash steam system is kept at a specified constant pressure by steam fed from a reducing valve. This ensures a reliable source of steam to the low pressure system if there is a lack of flash steam to meet the load.

Typical applications for flash steam

Flash steam supply and demand in-step

This gives maximum utilisation of the available flash steam. The air heater battery discussed in Figure 14.6.5 is one such system, but similar arrangements are practical with many other applications such as space heating installations using either radiant panels, or unit heaters.

Figure 14.6.6 depicts a system where a number of heaters are supplied with high pressure steam. The condensate from approximately 90% of the heaters is collected and taken to a flash recovery vessel. This supplies low pressure steam to the remaining 10% of the heaters.

With this system, the total heat output of the system is marginally reduced, as 10% of the heaters are operating at a lower steam pressure. However, it is rare to find an installation that does not have a sufficient margin of output above the normal load to accept this small reduction.

Sometimes a problem arises where the use of available flash steam may require more than one heater but less than two. It would be better in this case to connect two heaters to the flash steam supply, rather than vent the excess flash steam off to waste. Two heaters together will usually pull the flash pressure down to a lower level, even to sub-atmospheric levels. To cope with this, the supply of flash steam can be supplemented with live steam from a pressure reducing valve.

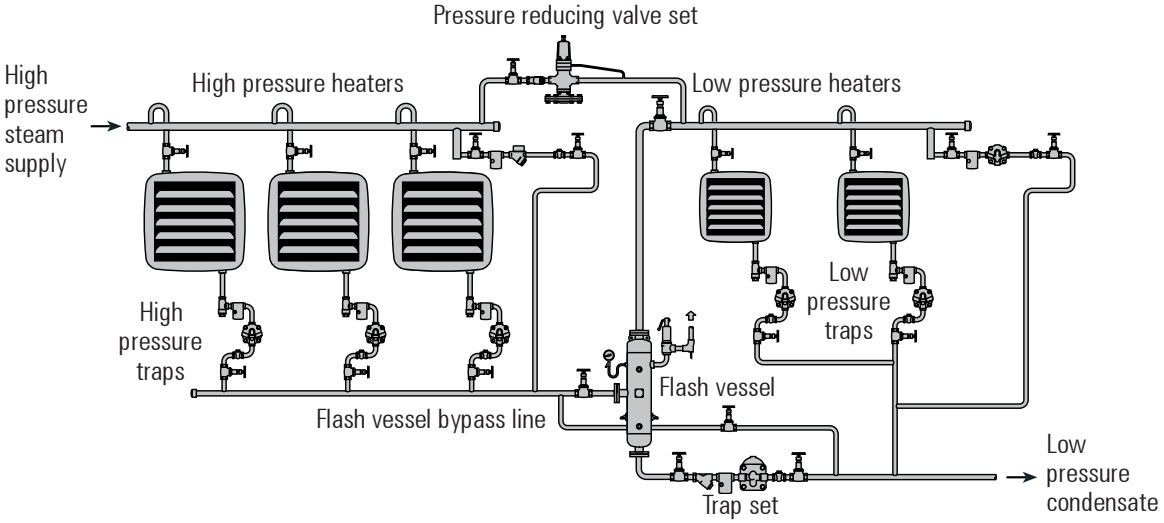


Fig. 14.6.6 Flash steam supply and demand in step

Another example where supply and demand are 'in step' is the steam heated hot water storage calorifier. Some of these incorporate a second coil, fitted close to the bottom of the vessel adjacent to where the cold feedwater enters.

Condensate and flash steam from the trap on the primary coil is passed directly to the secondary coil. Here, any flash steam produced by the drop in pressure across the trap is condensed, while giving up its heat to the feedwater. A typical arrangement is shown in Figure 14.6.7.

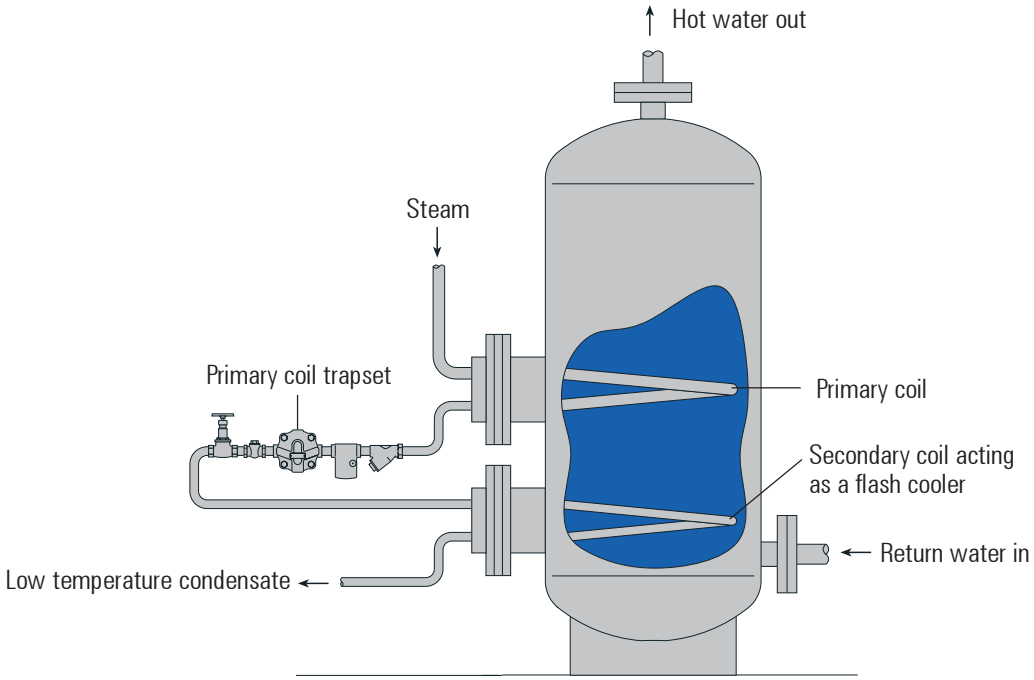


Fig. 14.6.7 Secondary flash steam coil in a storage calorifier

Another example of this idea is shown in Figure 14.6.8. Here, a normal steam-to-water calorifier drains condensate through a float trap to a smaller shell-and-tube heat exchanger (called a flash condenser), in which the flash steam is condensed to sub-cooled condensate. The unit is fitted such that the secondary flow pipework is in series with both calorifier and condenser. This enables the secondary return water to be preheated by the condenser, thereby reducing the demand for live steam in the first instance.

If the condensate in the flash condenser is likely to be sub-atmospheric, a mechanical pump is required to lift the condensate to any higher return line. The motive steam exhausting from the pump is itself condensed in the flash condenser. The pumping of the condensate is then achieved at virtually no cost.

Consideration must be given to the pump filling head in that it needs to be greater than the pressure drop across the flash condenser tubes under full-load conditions. A minimum head of 600 mm will usually achieve this.

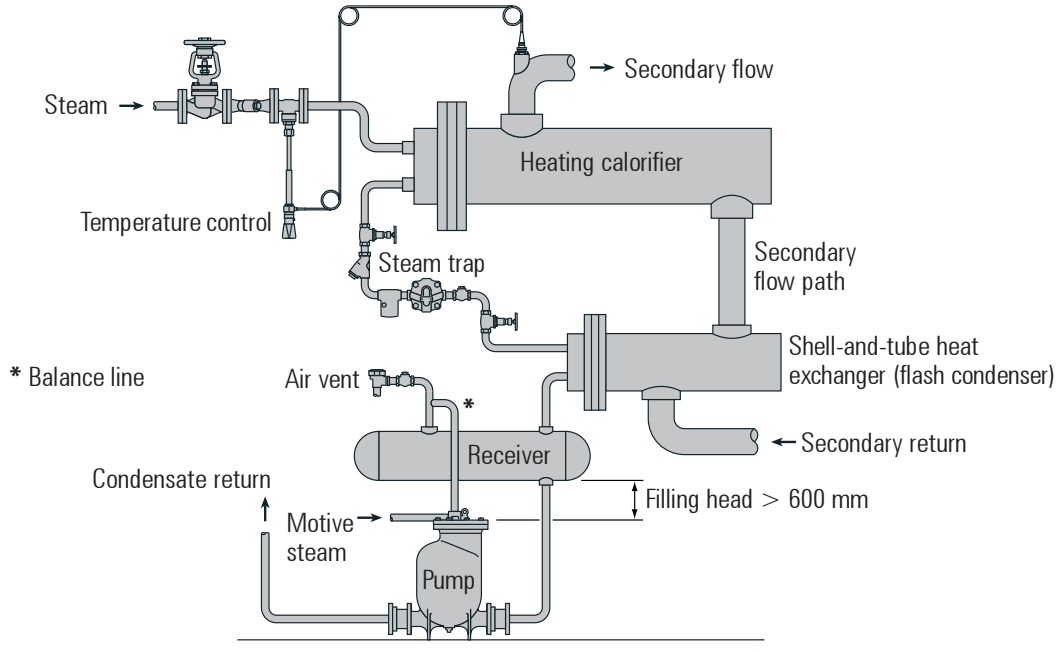


Fig. 14.6.8 Packaged calorifier and flash condenser unit

Flash steam supply and demand not in-step

The arrangement in Figure 14.6.9 is an example of flash steam recovery where the supply and demand are not always 'in-step'.

Condensate from three jacketed pans and a drain pocket releases flash steam, but it can only be used to augment the supply of steam to the space heating installation. This is quite satisfactory during the heating season, as long as the heating load exceeds the availability of flash steam.

During the summer season the heating equipment will not be in use, and even during spring and autumn the heating load may not be able to use all the available flash steam. The arrangement is not ideal, although it is quite possible for the steam savings made during the winter to justify the cost of the flash steam recovery equipment.

Sometimes, surplus flash steam must be vented to atmosphere, and, as indicated, a surplussing valve is more suitable for this purpose than a safety valve, which usually has a 'pop' or 'on / off' action and a seat arrangement designed for infrequent operation. The surplussing valve will be set so that it begins to open slightly above the normal pressure in the system. When the heating load falls and the pressure in the system begins to increase, the pressure reducing valve supplying the make-up steam closes down. A further increase of pressure, perhaps of 0.15 to 0.2 bar, is then allowed before the surplussing valve begins to open to release the excess flash steam.

A safety valve may still be required if the surplussing valve fails. It must be set to open at a pressure between the surplussing valve set pressure and the system design pressure. It is usually convenient to fit the safety valve onto the flash vessel.

Occasionally, during summer conditions it may be preferable to bypass the flash system with a manual valve (not shown in Figure 14.6.9). The condensate and its associated flash steam will then pass directly to a condensate receiver, where the flash steam will be vented to atmosphere.

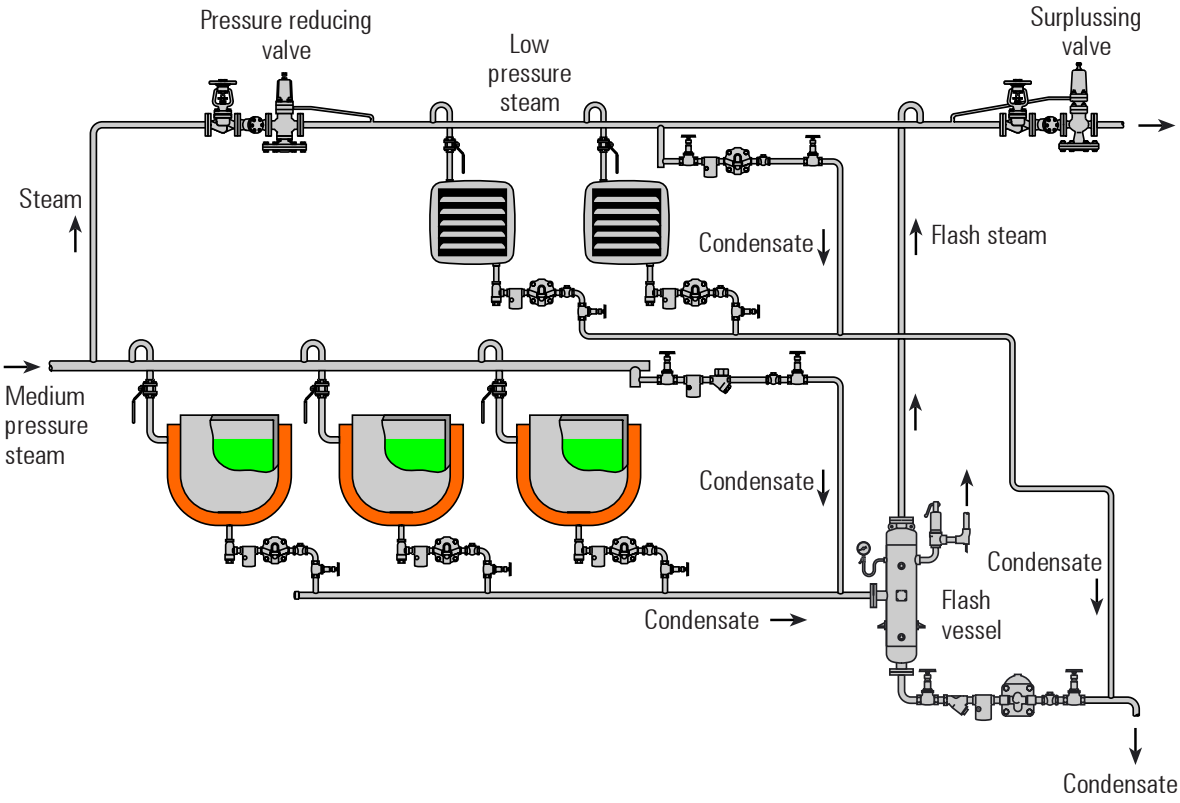


Fig. 14.6.9 Flash steam supply and demand not in-step

Boiler blowdown heat recovery applications

Continuous blowdown of boiler water is necessary to control the level of TDS (Total Dissolved Solids) within the boiler. Continuous blowdown lends itself to the recovery of the heat content of the blowdown water and can enable considerable savings to be made.

Boiler blowdown contains massive quantities of heat, which can easily be recovered as flash steam. After it passes through the blowdown control valve, the lower pressure water flows to a flash vessel. At this point, the flash steam is free from contamination and is separated from the condensate, and can be used to heat the boiler feedtank (see Figure 14.6.10).

The residual condensate draining from the flash vessel can be passed through a plate heat exchanger in order to reclaim as much heat as possible before it is dumped to waste. Up to 80% of the total heat contained in boiler continuous blowdown can be reclaimed in this way.

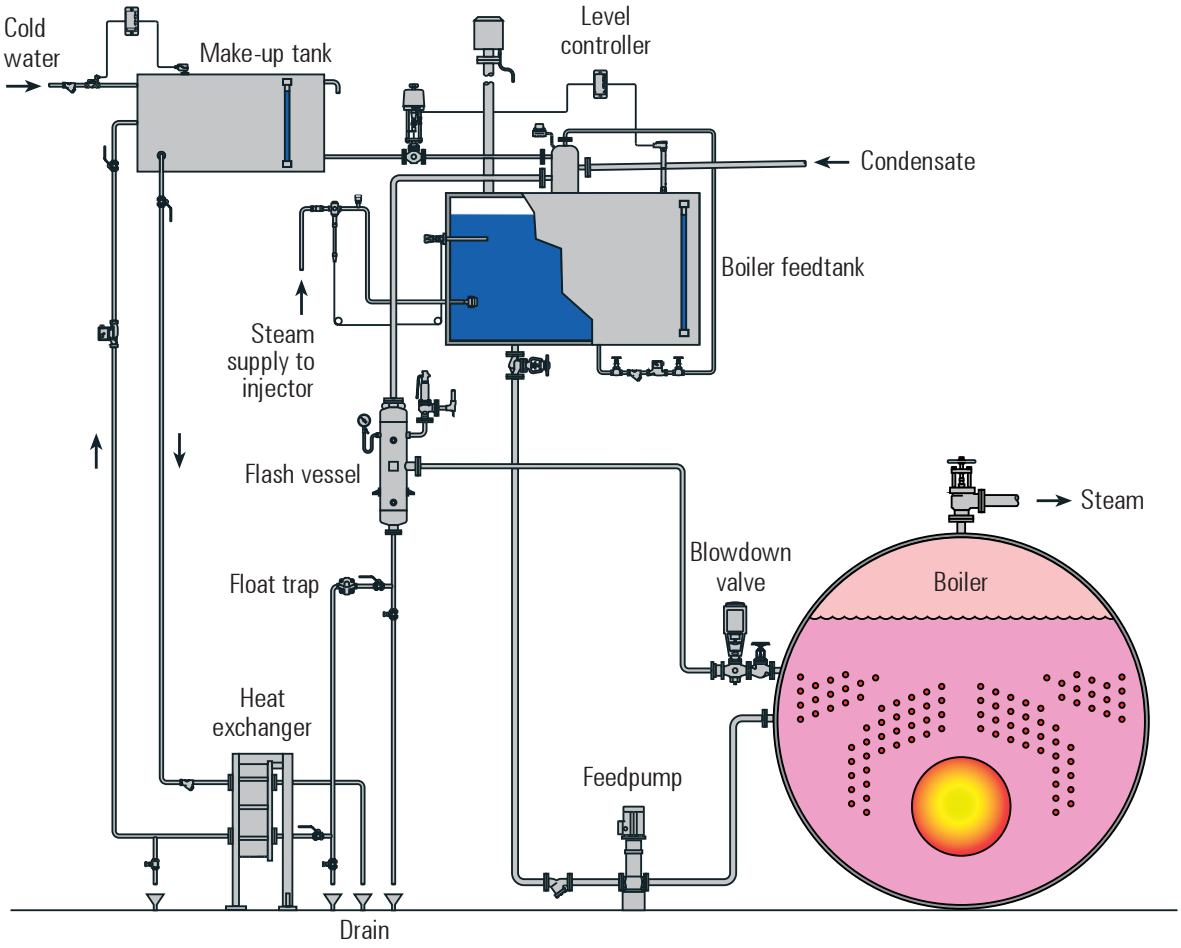


Fig. 14.6.10 Typical heat recovery from boiler blowdown

Spray condensing

Finally, consideration should be given to those cases where flash steam is unavoidably generated at low pressure, but where no suitable load is available which can make use of it.

Rather than simply discharge the flash steam to waste, the arrangement in Figure 14.6.11 can often be adopted.

This arrangement can be useful where the condensate receiver vent cannot be piped to outside, and where the presence of flash steam would be detrimental if left to discharge in a plant room.

A lightweight stainless steel chamber is fitted to the receiver tank vent. Cold water is sprayed into the chamber in sufficient quantities to just condense the flash steam. The flow of cooling water is controlled by a simple self-acting temperature control, adjusted so that minimal amounts of flash steam appear from the vent. The process will use roughly 6 kilograms of cooling water per kilogram of flash steam condensed.

If the cooling water is of boiler feed quality, then the warmed water is added to the condensate in the receiver and re-used. This will continue to make water savings throughout the year.

If the cooling water is not suitable for recovery, the spray pipework can be installed as shown by the dotted arrangement. The cooling water and condensed flash will then fall to waste.

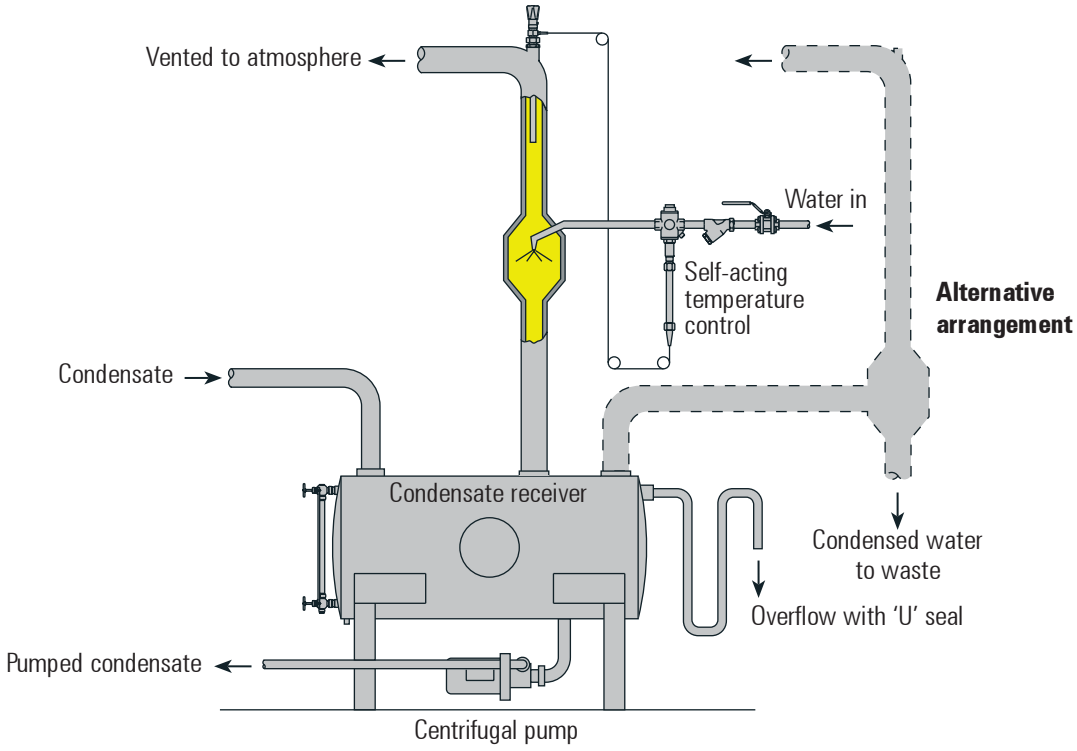


Fig 14.6.11 Flash steam condensing and water saving by spray

Questions

1. What is the difference between live steam and flash steam?

a| Live steam is made from water, flash steam is made from condensate ☐

b| Live steam is always hotter than flash steam ☐

c| Live steam is made by adding heat to water, flash steam is made from heat already contained in water ☐

d| Live steam is always at a higher pressure than flash steam ☐

2. What percentage of flash steam is made from condensate at 10 bar g passing into a flash vessel at 0.5 bar g?

a| 12% ☐

b| 13% ☐

c| 14% ☐

d| 5% ☐

3. What is the effect on the production of flash steam from sub-saturated condensate?

a| The flash steam produced is less than that with saturated condensate ☐

b| The flash steam produced is more than that with saturated condensate ☐

c| There is no effect at all ☐

d| Live steam is always at a higher pressure than flash steam ☐

4. With reference to Example 14.6.1, what would be the proportion of flash steam produced if the flash pressure were 2.5 bar g?

a| 3% ☐

b| 6% ☐

c| 8% ☐

d| 10% ☐

5. In a steam system, the trap pressure is 15 bar g, the flash pressure is 0.5 bar g, and the condensate flowrate is 1300 kg/h. Which flash vessel is required?

a| FV6 ☐

b| FV8 ☐

c| FV12 ☐

d| FV16 ☐

6. What is used to top-up the flash pressure?

a| A safety valve ☐

b| A larger condensate flow ☐

c| A pressure surplussing valve ☐

d| A pressure reducing valve ☐

Answers

1: c 2: c, 3: a, 4: b, 5: b, 6: d

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Module 15.1

Basic Desuperheating Theory

Desuperheating

Superheated steam is steam that is at a temperature higher than the saturation temperature for the steam pressure. For example, steam at a pressure of 3 bar g has a saturation temperature of 143.762°C. If further heat were to be added to this steam and the pressure remained at 3 bar g, it would become superheated. This extra heat results in steam which:

- Is higher than saturation temperature.
- Contains more energy than saturated steam.
- Has a greater specific volume than saturated steam.

The relationships between these three properties are well documented and can be found in most texts relating to the thermodynamic properties of steam.

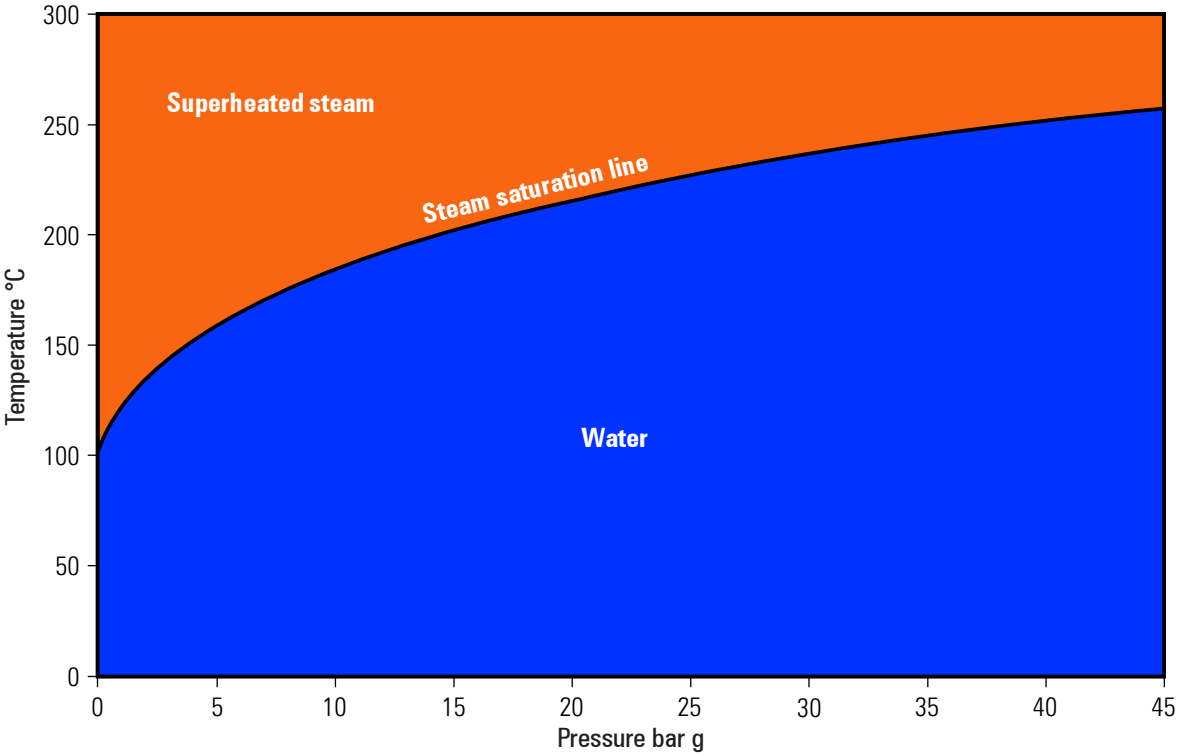


Fig. 15.1.1 Steam saturation diagram

Superheated steam is principally used in power generation plants as the driving force for turbines. A review of the Rankine gas cycle will demonstrate that, for driving turbines, superheated steam is more thermally efficient than saturated steam.

Superheating the steam has further important advantages:

- Wet steam within a turbine would result in water droplets and erosion of the turbine blades, as well as increased friction.
- Higher pipeline velocities (up to 100 m/s) can be used. This means that smaller distribution pipelines can be used (provided that the pressure drop is not excessive).
- For continuously running plants, superheated steam means there is no condensation in the pipework, therefore, there is only a requirement for steam trapping during start-up.

The use of superheated steam has a number of disadvantages:

- Although superheated steam contains a large amount of heat energy, this energy is in three forms; enthalpy of water, enthalpy of evaporation (latent heat) and enthalpy of superheat. The bulk of the energy is in the enthalpy of evaporation, and the energy in the superheat represents a smaller proportion.

  For example, take superheated steam at 10 bar a and 300°C, then:
  Enthalpy of water = 763 kJ/kg
  Enthalpy of evaporation = 2 015 kJ/kg
  Enthalpy of superheat = 274 kJ/kg

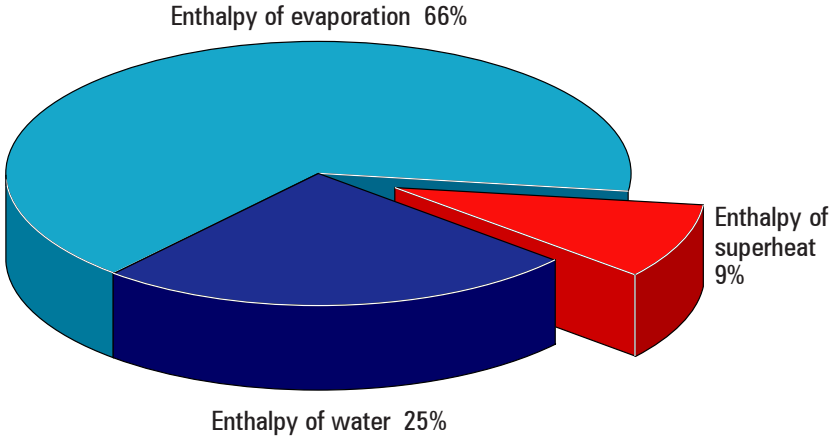


Fig. 15.1.2 Enthalpy in superheated steam

- The coefficient of heat transfer when using superheated steam as the heating medium is variable, low and difficult to quantify accurately. This makes accurate sizing and control of heat transfer equipment difficult, and will also result in a larger and more expensive heat exchanger.

  Once the superheated steam is cooled to saturation temperature, the heat transfer coefficient increases dramatically, and the temperature at which the steam condenses back into water is constant. This greatly assists accurate sizing and control of heat transfer equipment.

  The presence of high heat transfer coefficients associated with saturated steam leads to smaller and cheaper heat exchangers than those which utilise superheated steam.

- Some processes (for example, distillation columns) perform less efficiently when supplied with superheated steam.
- The higher temperatures of superheated steam may mean that higher rated, and hence more expensive equipment is required.
- The higher temperature of superheated steam may damage sensitive equipment.

These disadvantages mean that superheated steam is generally undesirable for thermal process applications. However, sites exist where superheated steam is raised for power generation, and it makes economic sense to desuperheat some of this steam from some point in the power generation cycle, and then use it for process applications. (More information on superheated steam can be found in Module 2.3).

Sites also exist where large quantities of waste are used as fuel for the boiler. If the quantity of waste is sufficiently large, then superheated steam may be produced for power generation. Examples of this type of plant can be found in the papermaking and sugar refining industries.

In plants that have superheated steam available for process use, it makes sense to distribute the superheated steam to remote points in the plant, as this will ensure that the steam remains dry. This becomes significant if there are long lengths of pipe separating the point of generation and the point of use.

Basic steam desuperheating

Desuperheating is the process by which superheated steam is restored to its saturated state, or the superheat temperature is reduced.

Most desuperheaters used to restore the saturated state produce discharge temperatures approaching saturation (typically to within 3°C of the saturation temperature as a minimum).

Designs for discharge temperatures in excess of 3°C above saturation are also possible and often used.

There are basically two broad types of desuperheater:

- **Indirect contact type -** The medium used to cool the superheated steam does not come into direct contact with it. A cooler liquid or gas may be employed as the cooling medium, for example, the surrounding air. Examples of this type of desuperheater are shell and tube heat exchangers. Here the superheated steam is supplied to one side of the heat exchanger and a cooler medium is supplied to the other side. As the superheated steam passes through the heat exchanger, heat is lost from the steam, and gained by the cooling medium.

  The temperature of the desuperheated steam could be controlled by either the inlet superheated steam pressure or the flowrate of the cooling water. Control of the superheated steam flow for this purpose is not normally practical and most systems adjust the flow of the cooling medium.

- **Direct contact type -** The medium used to cool the superheated steam comes into direct contact with it. In most cases, the cooling medium is the same fluid as the vapour to be desuperheated, but in the liquid state. For example, in the case of steam desuperheaters, water is used. A typical direct contact desuperheating station is shown in Figure 15.1.3.

  When the desuperheater is operational, a measured amount of water is added to the superheated steam via a mixing arrangement within the desuperheater. As it enters the desuperheater, the cooling water evaporates by absorbing heat from the superheated steam. Consequently, the temperature of the steam is reduced.

  Control of the amount of water to be added is usually achieved by measuring the temperature of the steam downstream of the desuperheater. The set temperature of the desuperheated steam would typically be 3°C above that at saturation. Therefore, in such arrangements the inlet pressure of the superheated steam should be kept constant.

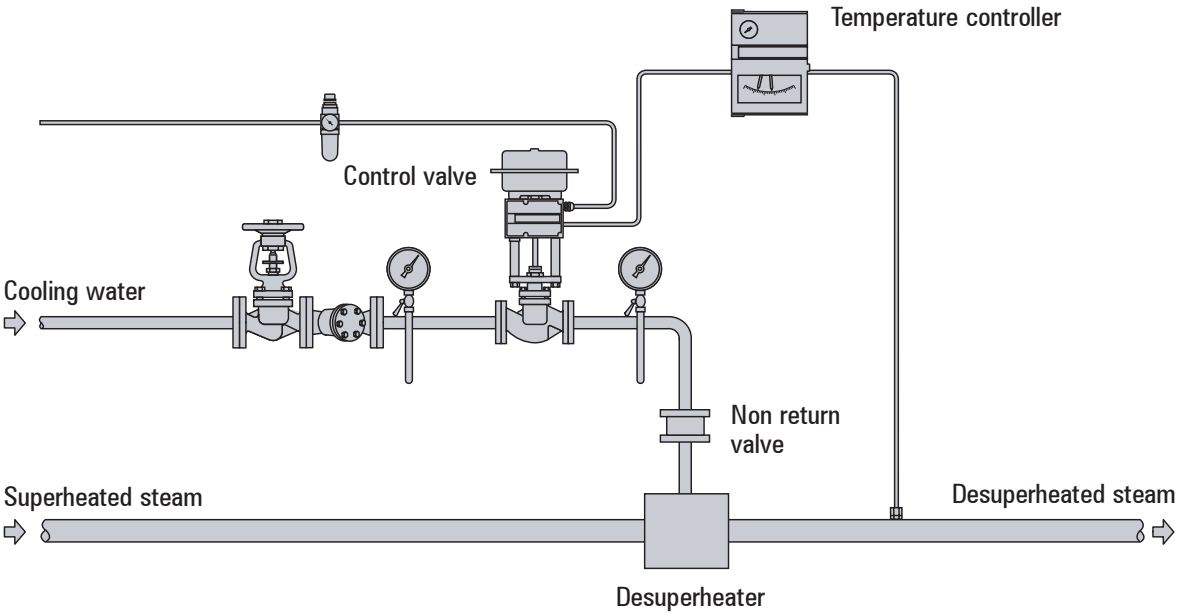


Fig. 15.1.3 A typical direct contact desuperheating station

Desuperheating calculations

The amount of water added must be sufficient to cool the steam to the desired temperature; too little water and the steam will not have been cooled enough, too much and wet saturated steam will be produced which will require drying through a separator.

Using Equation 15.1.1, which is based on the conservation of energy, the cooling liquid requirement can be easily and quickly determined:

$$\text{Enthalpy into process} = \text{Enthalpy out of process}$$

$$\dot{m}_{cw}\ h_{cw} + \dot{m}_s\ h_s = \dot{m}_s\ h_d + \dot{m}_{cw}\ h_d$$

$$\dot{m}_s\ h_s - \dot{m}_s\ h_d = \dot{m}_{cw}\ h_d - \dot{m}_{cw}\ h_{cw}$$

$$\dot{m}_s(h_s - h_d) = \dot{m}_{cw}(h_d - h_{cw})$$

$$\dot{m}_{cw} = \frac{\dot{m}_s(h_s - h_d)}{(h_d - h_{cw})}$$

$$\dot{m}_{cw} = \frac{\dot{m}_s\ (h_s - h_d)}{(h_d - h_{cw})} \qquad \textbf{Equation 15.1.1}$$

Where:
$\dot{m}_{cw}$ = Mass flowrate of cooling water (kg/h)
$\dot{m}_s$ = Mass flowrate of superheated steam (kg/h)
h_s = Enthalpy at superheat condition (kJ/kg)
h_d = Enthalpy at desuperheated condition (kJ/kg)
h_{cw} = Enthalpy of cooling water at inlet connection (kJ/kg)

Example 15.1.1

Determine the required cooling water flowrate for the conditions in the following Table:

Steam supply	Pressure	10 bar a
	Temperature	300°C
	Mass flowrate	10 000 kg/h
Cooling water supply	Pressure	15 bar a
	Temperature	150°C
Required steam conditions	Pressure	10 bar a
	Temperature	Saturation temperature + 5°C

Solution:
The necessary information can be obtained or interpolated from hard copy steam tables; the relevant extracts are shown in Table 15.1.1 and Table 15.1.2. Alternatively, the Spirax Sarco online steam tables can be used.

Table 15.1.1 Extract from steam tables – Saturated water and steam

P	Ts	vg	uf	ug	hf	hfg	hg	sf	sfg	sg
bar a	°C	m³/kg	kJ/kg		kJ/kg			kJ/kg K		
9	175.4	0.214 9	742	2 581	743	2 031	2 774	2.094	4.529	6.623
10	179.9	0.194 4	762	2 584	763	2 015	2 778	2.138	4.448	6.586
11	184.1	0.177 4	780	2 586	781	2 000	2 781	2.179	4.375	6.554
12	188.0	0.163 2	797	2 588	798	1 986	2 784	2.216	4.307	6.523

Table 15.1.2 Extract from steam tables – Superheated steam

Pressure (temperature) bar a		T °C	200	250	300	350	400	450	500	600
9 (175)	v_g 0.214 9	v	0.230 5	0.259 7	0.287 4	0.314 4	0.341 0	0.367 4	0.393 7	0.445 8
	u_g 2 581	u	2 628	2 714	2 796	2 877	2 959	3 041	3 126	3 298
	h_g 2 774	h	2 835	2 948	3 055	3 160	3 266	3 372	3 480	3 699
	s_g 6.623	s	6.753	6.980	7.176	7.352	7.515	7.667	7.811	8.077
10 (180)	v_g 0.194 4	v	0.206 1	0.232 8	0.258 2	0.282 5	0.306 5	0.330 3	0.354 0	0.401 0
	u_g 2 584	u	2 623	2 711	2 794	2 875	2 957	3 040	3 124	3 297
	h_g 2 778	h	2 829	2 944	3 052	3 158	3 264	3 370	3 476	3 698
	s_g 6.586	s	6.695	6.926	7.124	7.301	7.464	7.617	7.761	8.028
15 (198)	v_g 1 317	v	0.132 4	0.152 0	0.169 7	0.186 5	0.202 9	0.219 1	0.235 1	0.266 7
	u_g 2 595	u	2 597	2 697	2 784	2 868	2 952	3 035	3 120	3 294
	h_g 2 792	h	2 796	2 925	3 039	3 147	3 526	3 364	3 473	3 694
	s_g 6.445	s	6.452	6.711	6.919	7.102	7.268	7.423	7.569	7.838

The information required to satisfy Equation 15.1.1 is therefore:

$\dot{m}_s$ = Mass flowrate of superheated steam = 10 000 kg/h

h_s = Enthalpy at superheat condition (From steam tables 300°C at 10 bar a) = 3 052 kJ/kg

h_{cw} = Enthalpy of the cooling liquid = 4.2 kJ/kg°C x 150°C =630 kJ/kg

Determining the enthalpy at the desuperheated condition, h_d:
From steam tables, the saturation temperature (T_s) at 10 bar a is 180°C, therefore at the required desuperheated condition, the temperature will be:

$$T_s + 5°C = 185°C$$

Interpolating between the enthalpy of steam at 10 bar a and its saturation temperature, and at 10 bar a and 200°C:

Enthalpy at 10 bar a, T_s (saturated steam tables) = 2 778 kJ/kg

Enthalpy at 10 bar a, 200°C (superheated steam tables) = 2 829 kJ/kg

Interpolating for enthalpy at 10 bar a and 185°C:

$$h_d = 2778 + \left((2829 \text{ kJ/kg} - 2778 \text{ kJ/kg}) \times \frac{185°C - 180°C}{200°C - 180°C}\right)$$

$$h_d = 2778 + \left(\frac{51 \times 5}{20}\right)$$

Therefore : h_d = 2 791kJ/kg

Finally, applying Equation 15.1.1:

$$\dot{m}_{cw} = \frac{\dot{m}_s (h_s - h_d)}{(h_d - h_{cw})} \qquad \textbf{Equation 15.1.1}$$

$$\dot{m}_{cw} = \frac{10\,000 \times (3052 - 2\,791)}{(2\,791 - 630)}$$

$$\dot{m}_{cw} = 1\,208 \text{ kg/h}$$

Note that the desuperheated steam is supplied at a rate of:

$$10\,000 + 1\,208 \text{ kg/h} = 11\,208 \text{ kg/h}$$

Had the requirement been for 10 000 kg/h of the desuperheated steam, the initial superheated steam flowrate can be determined using a simple proportional method:

$$\frac{10\,000}{11\,208} = \frac{\dot{m}_s}{10\,000}$$

$$\dot{m}_s = \mathbf{8923 \text{ kg/h}}$$

Questions

1. Which of the following are properties of superheated steam?

a| The specific enthalpy of superheated steam is greater than that of saturated steam ☐

b| It has a greater specific volume than saturated steam ☐

c| It is at a higher temperature than saturated steam ☐

d| All of the above ☐

2. What is the main disadvantage of using superheated steam in heat exchanger applications?

a| The superheated steam contains less heat than saturated steam ☐

b| The coefficient of heat transfer is lower than that of saturated steam ☐

c| Superheated steam has a high dryness fraction, typically unity ☐

d| Superheated steam has a low Rankine efficiency ☐

3. Using steam tables (Table 15.1.1 and Table 15.1.2) determine the additional amount of energy contained in superheated steam at 9 bar a, with 74.6°C of superheat, compared with saturated steam at the same pressure.

a| 1% ☐

b| 6% ☐

c| 11% ☐

d| 24% ☐

4. What is the primary function of a desuperheater?

a| To reduce the pressure of the steam ☐

b| To increase the specific volume of the steam ☐

c| To condense the steam into water so that it can be disposed of ☐

d| To reduce the temperature of the steam ☐

5. Determine, by interpolating from steam tables, the enthalpy of desuperheated steam at 9 bar a, 8°C above saturation temperature (T_S + 8°C).

a| 2 795 kJ/kg ☐

b| 2 806 kJ/kg ☐

c| 2 810 kJ/kg ☐

d| 2 815 kJ/kg ☐

6. A boiler in a power station produces 108 kg/s of superheated steam. Given that the steam downstream of the turbine is at 110 bar a and has a temperature of 500°C, how much cooling water (at 200°C) would be required to desuperheat the steam by 150°C?

a| 5 kg/s ☐

b| 15 kg/s ☐

c| 25 kg/s ☐

d| 30 kg/s ☐

Answers

1: d, 2: b, 3: b, 4: d, 5: a, 6: c

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Module 15.2

Basic Desuperheater Types

Basic Desuperheater Types

Desuperheaters

The simplest type of desuperheater is an unlagged section of pipe, where heat can be radiated to the environment. However, apart from the obvious risk of injury to personnel from such a hot item of plant, and the expensive energy wastage, this approach does not adjust to compensate for changes in the environmental conditions, steam temperature or steam flowrate.



Fig. 15.2.1 Typical multi-nozzle spray desuperheater

Several designs of desuperheater are available and it is recommended that the following properties be considered when sizing and selecting a suitable station for a given application:

- Turndown ratio - 'turndown' is used to describe the range of flowrates over which the desuperheater will operate, as shown in Equation 4.2.1.

$$\text{Turndown} = \frac{\text{Maximum flow}}{\text{Minimum flow}} \qquad \textbf{Equation 4.2.1}$$

This is an important parameter, as any variation in inlet pressure, temperature or flowrate will cause a variation in the requirement of cooling liquid.

In general, the two turndown values may be specified for a particular desuperheater:

Steam turndown ratio - This reflects the range of steam flowrates that the device can effectively desuperheat.

Cooling water turndown ratio - This reflects the range of cooling flowrates that can be used. Although this directly affects the steam turndown ratio, the relationship depends on the temperatures of the superheated steam, the cooling water and the resulting desuperheated steam.

Equation 15.1.1 is the mass/heat balance equation for this application:

$$\dot{m}_{cw} = \frac{\dot{m}_s\ (h_i - h_d)}{(h_d - h_{cw})} \qquad \textbf{Equation 15.1.1}$$

Where:

$\dot{m}_{cw}$ = Mass flowrate of cooling water (kg/h)
$\dot{m}_s$ = Mass flowrate of superheated steam (kg/h)
h_i = Enthalpy at superheat condition (kJ/kg)
h_d = Enthalpy at desuperheated condition (kJ/kg)
h_{cw} = Enthalpy of cooling water at inlet connection (kJ/kg)

It should be noted that the steam and water flowrates are directly proportional to each other; the constant of proportionality 'k' depends on the enthalpies of the superheated steam, the cooling water and the required desuperheated steam. Mathematically: $\dot{m}_{cw} \propto k\dot{m}_s$

Where: $k = \frac{(h_i - h_d)}{(h_d - h_{cw})}$

If the required turndown cannot be achieved using a single desuperheater, two desuperheaters can be installed in parallel, with operation switching from one to another; or both can be in operation depending on steam demand.

It should be noted that the desuperheater itself is only one part of a desuperheating station, which will include the necessary control system for correct operation.

- Operating pressures and temperatures.
- Steam and water flowrate.
- Amount of superheat before, and amount of desuperheated steam required after, the process.
- The water pressure available (a booster pump may be required).
- The required accuracy of the final temperature.
- In the case of in-line desuperheaters, the distance travelled by the steam before complete desuperheating has occurred is also an important consideration. This is referred to as the absorption length.

The following Sections include descriptions of the common types of desuperheater available, their limitations and typical applications.

Indirect contact desuperheaters

Tube bundle type desuperheaters

This type of desuperheater (Figure 15.2.2) consists of a heat exchanger, typically a shell and tube, with superheated steam on one side, and the cooling medium on the other.

The shell of the first heat exchanger (containing the cooling water) is fixed at both ends on the inlet side, whereas on the outlet side, it is fixed at the bottom and open at the top. The floating head allows the pressure in the two sections of the shell to equalise.

The cooling medium is water at saturation temperature and pressure. As superheated steam enters the first and then the second set of tubes, it gives up heat to the water, some of which will be evaporated by this addition of energy. Any evaporated cooling water passes through the floating head and will accumulate in the outlet side of the shell. It then passes through the open end of the shell where it is mixed with the desuperheated steam.

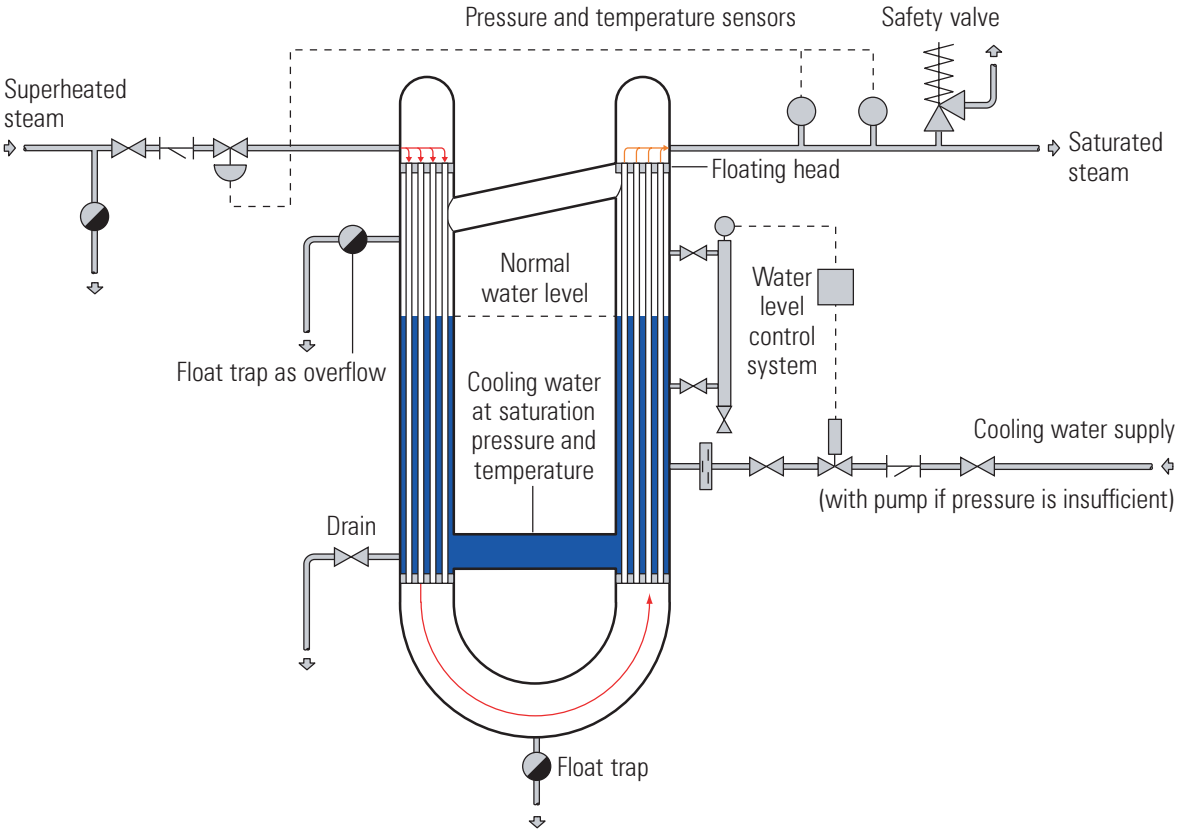


Fig. 15.2.2 A tube and bundle type desuperheater

Advantages:

1. Turndown is only limited by the controls that are fitted.
2. This design is capable of producing desuperheated steam to within 5°C of the saturation temperature.
3. High maximum operating temperatures and pressures, typically around 60 bar and 450°C.
4. Fast response.

Disadvantages:

1. Bulky - because there are now a number of in-line devices available, they have been largely superseded.
2. Cost.
3. An important concern with this type of desuperheater is the efficiency of the heat exchange process. The build up of air or scale films on the heat exchange surface can act as an extremely effective barrier to heat transfer.

Applications:

1. Those applications that experience wide variations in load.

Direct contact desuperheaters

Water bath type desuperheater

This is the simplest form of direct contact desuperheater. The superheated steam is injected into a bath of water. This additional heat will cause saturated steam to evaporate from the surface of the bath. A pressure controller maintains a constant pressure in the vessel, and hence the temperature and pressure of the saturated steam in the downstream pipe.

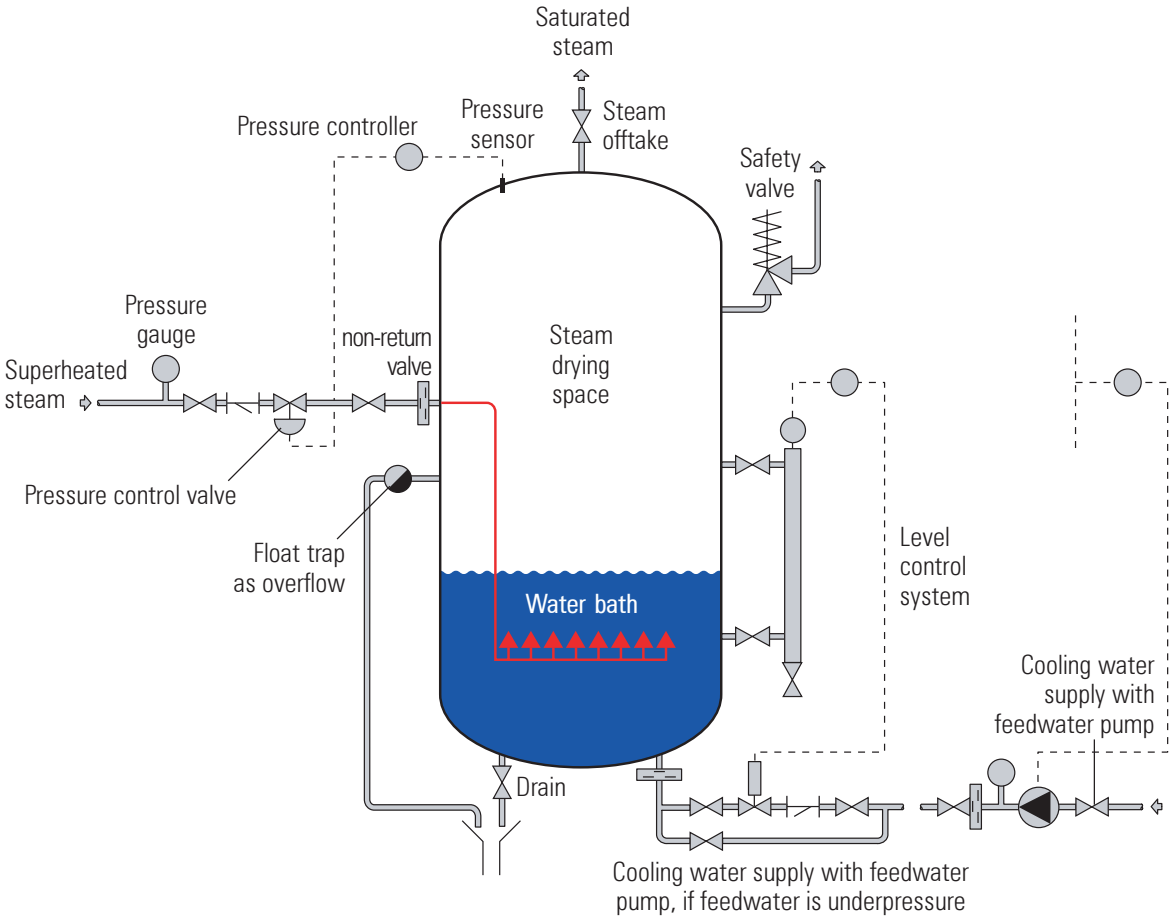


Fig. 15.2.3 Water bath type desuperheater (schematic)

Since the superheated steam has more energy per unit mass than the saturated steam, more steam will be evaporated than actually enters the desuperheater. Consequently, the water level will fall and therefore provision must be made to maintain this level. This usually requires a pump of similar design to a boiler feedwater pump, as the water must be pumped against the vessel pressure.

A good non-return valve is required in the superheated steam supply to avoid any water from the bath being drawn into the superheated steam system should the pressure in the superheated main drop.

Advantages:

1. Simple
2. Steam is produced at saturation temperature.
3. Steam with a dryness fraction of 0.98 can be produced.
4. Turndown is only limited by the controls that are fitted.

Disadvantages:

1. Bulky.
2. Not practical for high temperatures.

Applications:

1. Wide variations in the flowrate.
2. Where no residual superheat can be tolerated.

Water spray desuperheating

This type of desuperheating represents the vast majority of desuperheating applications. In water spray desuperheaters, superheated steam is passed through a section of pipe fitted with one or more spray nozzles. These inject a fine spray of cooling water into the superheated steam, which causes the water to be converted into steam, reducing the quantity of superheat.

The cooling water may be introduced into the superheated steam in a number of ways; consequently, there are a number of different types of water spray desuperheater.

Despite this, most water spray desuperheaters are affected by the following factors:

- **Particle size -** The smaller the water particle size, the greater the ratio of surface area to mass, and the higher the rates of heat transfer. Since the water is being directly injected into the moving superheated steam, the smaller the particle size, the shorter the distance required for heat exchange to take place.

  The water is broken into small particles using either a mechanical device (such as a variable or fixed orifice nozzle) or steam atomising nozzles.

- **Turbulence -** As the flow within the pipeline becomes more turbulent, the individual entrained water particles reside longer in the desuperheater, allowing for greater heat transfer. In addition, turbulence encourages the mixing of the cooling water and the superheated steam. Increased turbulence results in a shorter distance being required for complete desuperheating to occur.

  Turbulence can be created in two ways:

  Pressure drop across the nozzle - Subjecting the cooling water to a higher pressure drop will increase its velocity and induce greater turbulence.

  Velocity - By increasing the overall velocity of the water and steam mixture, the amount of turbulence is inherently increased. The increase in velocity is usually achieved by creating a restriction in the steam path, which further generates turbulence by vortex shedding.

  In addition to these high velocities, if poor piping design practices are used, the speed of the superheated steam could in theory approach Mach 1. At such speeds a number of problems would occur (including the generation of shock waves). However, this would be far in excess of the velocities used in good piping design. Typical velocities of steam entering a desuperheater should be around 40 to 60 m/s.

- **Cooling water flowrate -** The rate at which cooling water can be added to the superheated steam is affected by a number of factors, which are related by Equation 4.2.11:

$$q_v = C\,A\,\sqrt{2\,g\,h} \qquad \textbf{Equation 4.2.11}$$

  Where:
  q_v = Cooling water volumetric flowrate (m^3/s)
  C = Coefficient of discharge for the nozzle
  A = Area of the nozzle (m^2)
  g = Gravitational constant (9.81 m/s^2)
  h = Pressure drop over the orifice (m head)

  Bearing in mind that C and g are constants, reviewing Equation 4.2.11 shows that only two factors can be manipulated to alter the cooling water flowrate, q_v:

  Changing the pressure drop over the orifice (nozzle), h - Expressing flowrate as a function of pressure drop over the nozzle:

$$V \propto \sqrt{h}$$

  This means that if, for example, flow is increased by a factor of 5, the available pressure must increase by a factor of $5^2 = 25$. The effect of this relationship is to severely hamper the turndown ratio.

In addition to affecting the cooling water flowrate, there are two other important considerations when determining the required cooling water pressure:

1. The cooling water pressure must be greater than the superheated steam pressure at the point of injection.
2. The greater the pressure drop across the nozzle, the better the atomisation of the cooling water.

Changing the area of the orifice, A - Expressing flowrate as a function of the area of the orifice:

$$V \propto A$$

This direct relationship means that if, for example, flow is to be increased by a factor of 5, the available area must also increase by a factor of 5. This change may simply be achieved by an orifice, which has the ability to change in area (see Figure 15.2.4), or alternatively by altering the number of orifices passing the coolant.

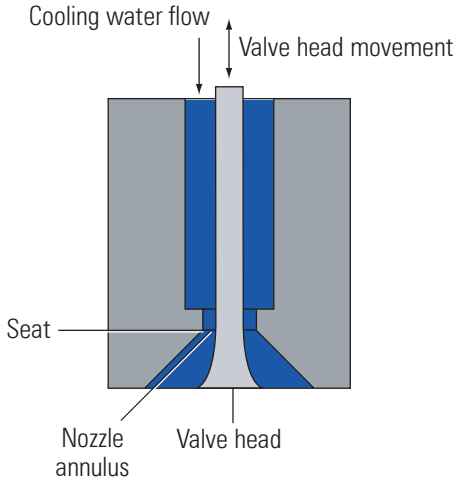


Fig. 15.2.4 Variable area orifice

- **Thermal sleeves -** Careful control of the spray is required to ensure that the water does not fall out of suspension as this can result in thermal stresses being generated in the pipeline and cracking may occur. However, in some cases, an inner thermal sleeve can be used to provide protection from this.

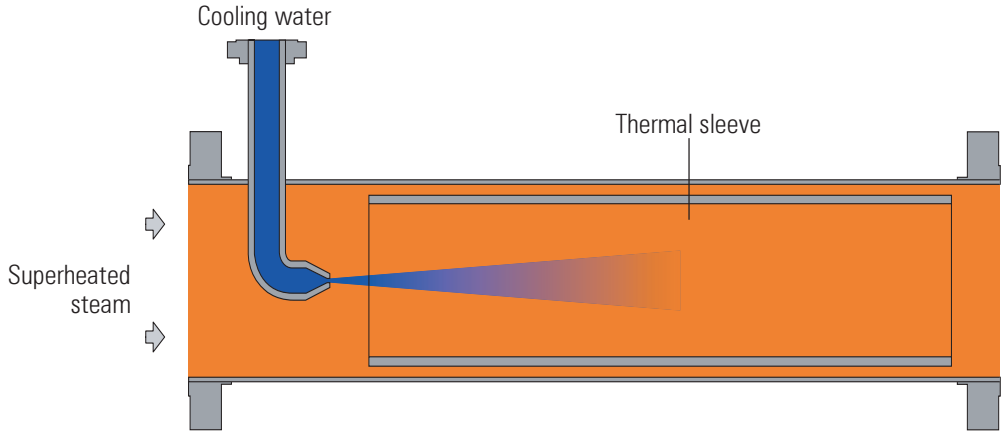


Fig. 15.2.5 A thermal sleeve inserted in an in-line spray desuperheater

The thermal sleeve also allows the circulation of superheated steam around the annular area between the sleeve and the inside diameter of the pipe. This provides a hot surface upon which the injected water can evaporate, as opposed to the walls of the desuperheater, which are inevitably cooler.

Water spray type desuperheaters

Single point radial injection spray desuperheaters

The simplest method of injecting cooling water is to introduce a nozzle through the pipe wall.

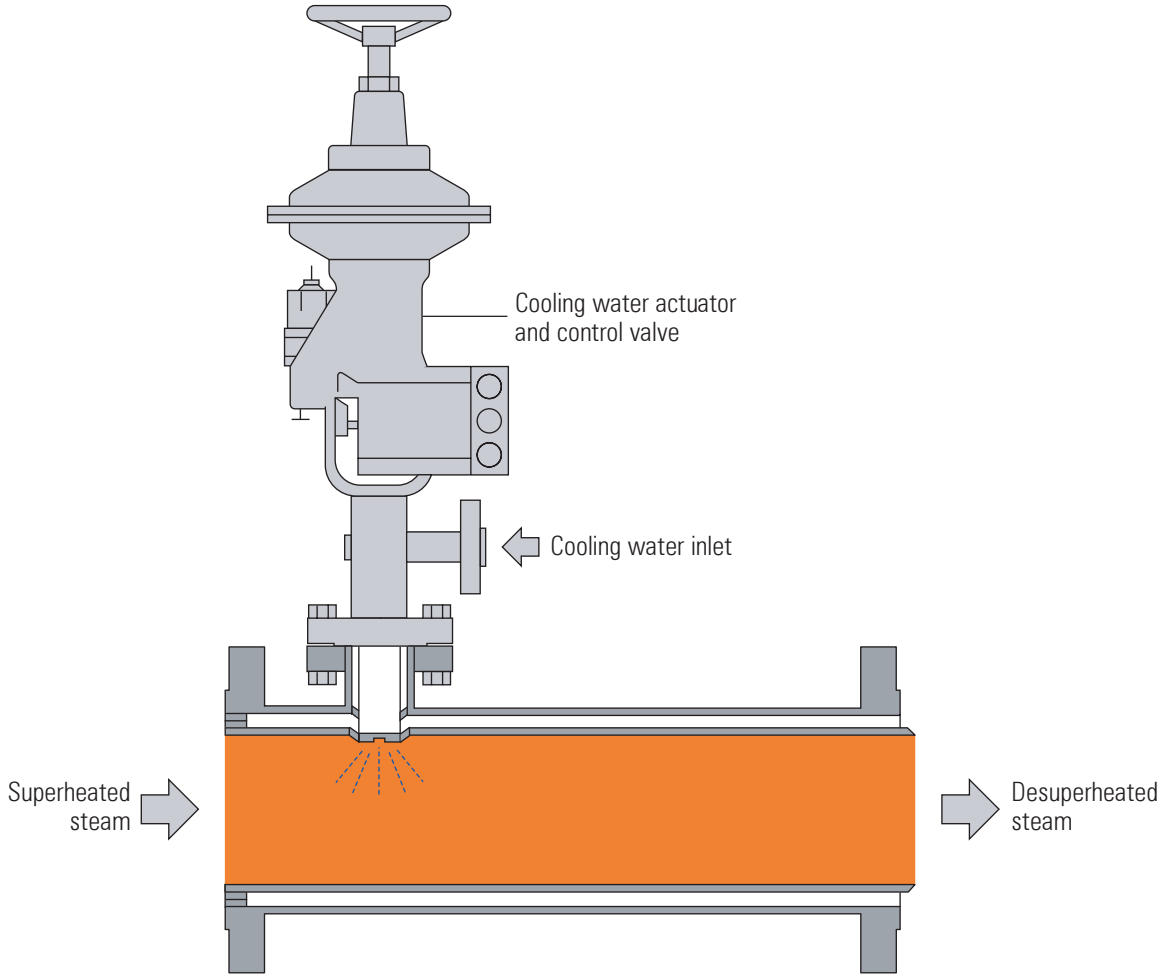


Fig. 15.2.6 Single point radial injection spray desuperheater

The cooling water particles are sprayed across the flow of the superheated steam. The quantity of cooling water injected is controlled by varying the position of the valve in the centre of the nozzle.

Advantages:

1. Simple in operation.
2. Cost effective.
3. Minimum steam pressure drop.

Disadvantages:

1. Low turndown ratio, typically a maximum of 3:1 on both steam and cooling water flow.
2. Desuperheated steam temperature can only be reduced to 10°C above saturation temperature.
3. Longer absorption length than the steam atomising type.
4. Most prone to cause erosion damage to the internal pipework. This can be overcome by the use of a thermal sleeve.
5. Limited pipe sizes.

Applications:

1. Constant steam load.
2. Constant steam temperature.
3. Constant coolant temperature.

All of which mean a relatively constant cooling water requirement.

Multiple point radial injection spray desuperheaters

This is a progression of the single point radial injection spray desuperheater. Cooling water is sprayed in from a number of orifices around the perimeter of the pipe.

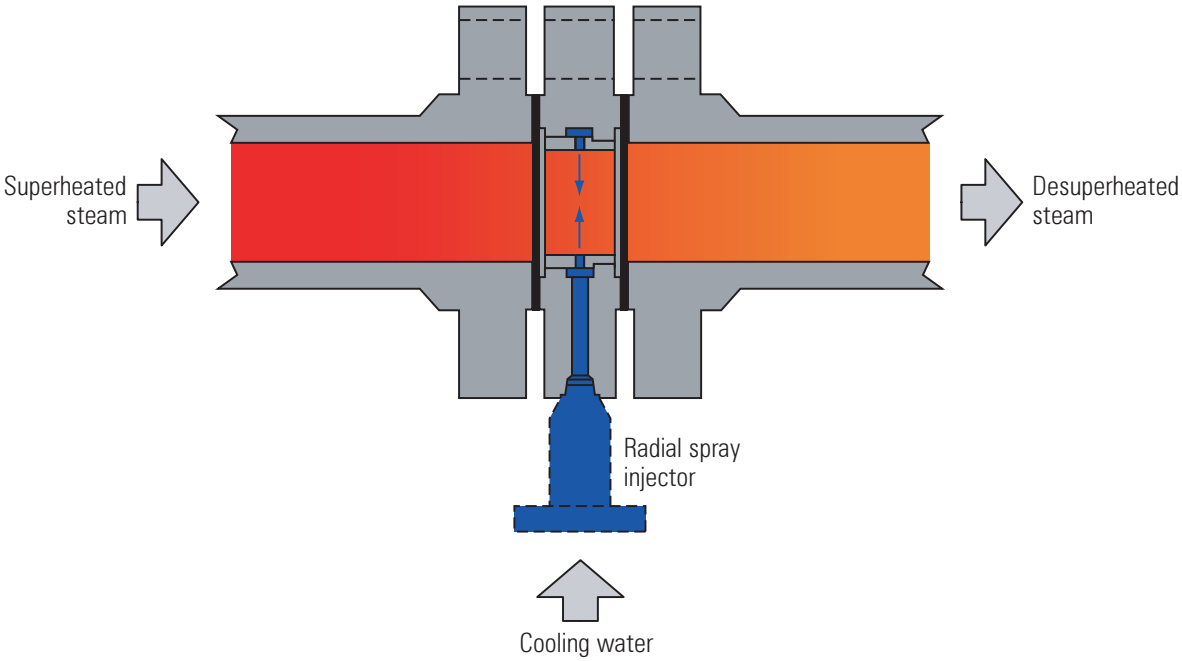


Fig. 15.2.7 Multiple radial injection point desuperheater

Advantages:

1. The pressure of the cooling liquid is less than that in the single point version; therefore, it is not necessary to use a thermal sleeve.
2. The absorption length is shorter compared with that of the single point version due to better mixing of the water and the superheated steam. The absorption length is still significantly longer than other types of water spray desuperheater.

Other advantages, disadvantages and applications are similar to those of single point radial injection spray desuperheaters.

Axial injection spray desuperheaters

This is also a simple in-line injection spray desuperheater, but the point of injection is moved to the axis of the pipeline. The cooling water is injected into the steam flow via one or more atomising nozzles (see Figure 15.2.8). The unit usually employs a thermal sleeve.

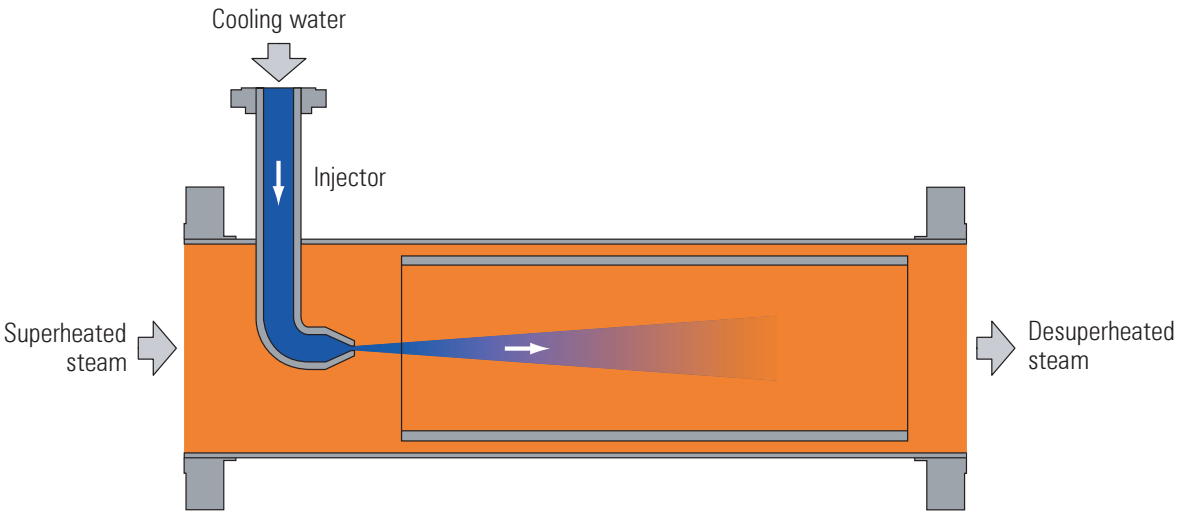


Fig. 15.2.8 Axial injection spray desuperheater

Axial injection of the cooling water improves the mixing of the water and the superheated steam by two methods:

1. As the water is injected along the centre of the pipeline, it will be more evenly distributed throughout the superheated steam.
2. The cooling water delivery pipe that is inserted in the pipeline acts as an obstruction, creating additional turbulence at the point of water injection due to vortex shedding.

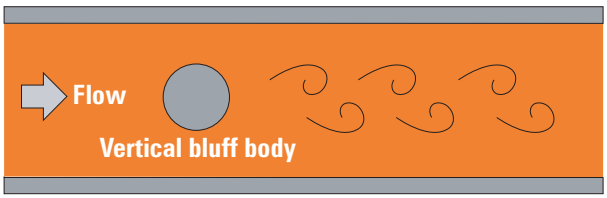


Fig. 15.2.9 Vortex shedding around the cooling water delivery pipe

A modification of this basic arrangement involves turning the nozzle so that the cooling water is sprayed upstream, against the steam flow. The high velocity of the superheated steam reverses the spray water flow pattern and sends it back through a mixing chamber. This achieves more efficient mixing of the water and steam over a short absorption length.

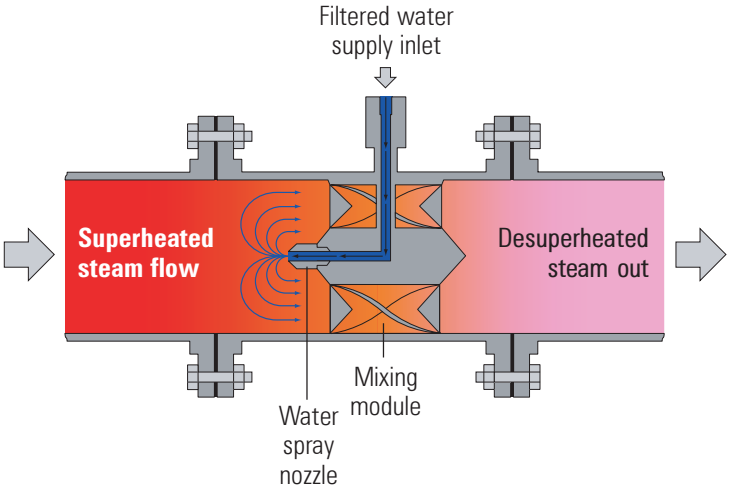


Fig. 15.2.10 Reverse flow type axial desuperheater

Advantages:

1. Simple in operation.
2. No moving parts.
3. Cost effective across the entire range of sizes.
4. Minimal steam pressure drop.

Disadvantages:

1. Low turndown ratio, typically a maximum of 3:1 on both steam and cooling water flow.
2. Desuperheated steam temperature can only be reduced to 10°C above saturation temperature.
3. Longer absorption length than the steam atomising type, but less than the radial type desuperheaters.
4. Most prone to cause erosion damage to the internal pipework. This can be overcome by the use of a thermal sleeve.

Applications:

1. Constant steam load.
2. Constant steam temperature.
3. Constant coolant temperature.

All of which mean a relatively constant cooling water requirement.

Multiple nozzle axial injection desuperheaters

Rather than a single nozzle, the multiple nozzle axial injection desuperheater provides a number of nozzles across the flow of superheated steam. This gives good dispersion of the water droplets. There are three main types of multiple nozzle axial injection desuperheater:

1. **Fixed area type -** All the nozzles are open when the desuperheater is operating, and the cooling water is regulated by a spray water control valve.

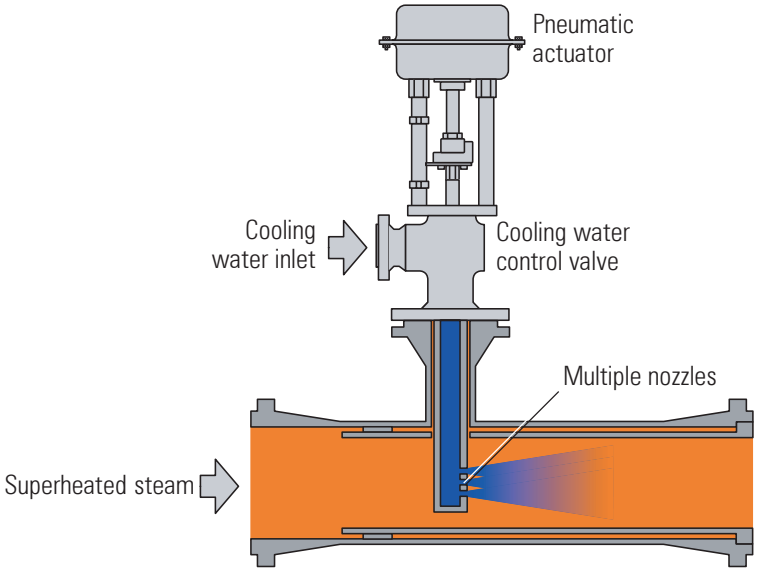


Fig.15.2.11 A multiple nozzle desuperheater

2. **Variable spray type -** The downstream temperature determines the number of exposed nozzles. Cooling water enters the desuperheater through the water jacket to the sealing area above the disc (see Figure 15.2.12). When an increase in the downstream steam temperature is detected by the associated temperature control system, the actuator moves the stem down, progressively exposing more nozzles. When the demand for the cooling water changes, the stem and disc arrangement moves up and down as required. This has the effect of changing the overall orifice area.

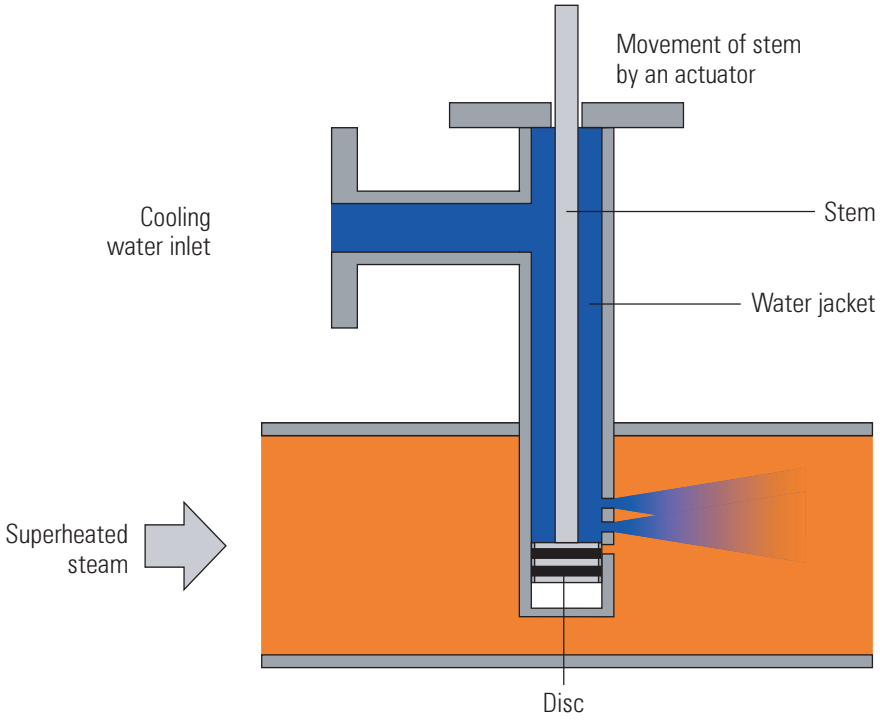


Fig.15.2.12 A variable area type multiple nozzle desuperheater

3. **Spring-assisted type -** This is essentially a combination of the two previous types. Instead of the stem and disc arrangement being controlled by an actuator, the spring-assisted type contains a spring-loaded flow plug, which moves in response to a change in the differential pressure between the coolant and the superheated steam. The moving plug changes the number of open nozzles, thereby adjusting the flow into the main pipeline. In addition, the cooling water is regulated by a spray water control valve.

   Being able to control both the pressure and flow of the cooling water enables accurate control over the amount of water injected into the superheated steam. This type does, however, require a high cooling water pressure.

Advantages:

1. Turndown ratios of up to 8:1 are possible with the fixed area type, up to 9:1 with the spring assisted type and 12:1 for the variable area type.
2. Better dispersion of the water droplets means that the absorption length is less than that of single nozzle devices.
3. Minimal steam pressure drop.

Disadvantages:

1. The desuperheated steam temperature can only be reduced to 8°C above saturation temperature.
2. Longer absorption length than the steam atomising type.
3. Most prone to cause erosion damage to the internal pipework, if a thermal sleeve is not used.
4. Not suitable for small pipe sizes.
5. Requires high pressure cooling water (particularly true of the spring assisted type).
6. Variable area and spring assisted types can be expensive.

Applications:

1. Applications with a requirement for a higher turndown ratio than that offered by single nozzle devices, but where the expense of more sophisticated devices is not justified.
2. Constant steam load.
3. Constant steam temperature.
4. Constant coolant temperature.

All of which require a relatively constant desuperheating load.

Questions

1. Which type of desuperheater should be used if it is essential that steam at saturation temperature be produced?

a | Spring assisted type ☐
b | Reverse flow axial type ☐
c | Single nozzle radial injection type ☐
d | Water bath type ☐

2. Which of the following parameters can be altered in order to increase the rate at which cooling water is injected into superheated steam, using a spray type desuperheater?

i. Pressure drop over the orifice
ii. Area of the orifice
iii. Cooling water temperature

a | i only ☐
b | ii only ☐
c | i and ii ☐
d | i, ii and iii ☐

3. What advantage does a multiple point radial injection spray desuperheater have over a single point desuperheater?

a | Improved approach to saturation temperature ☐
b | Increased turndown ratio ☐
c | Shorter absorption length ☐
d | It can be used on smaller pipe sizes ☐

4. What is the primary function of a thermal sleeve?

a | To reduce the temperature of the superheated steam ☐
b | To prevent erosion of pipework downstream of the desuperheater ☐
c | To reduce the energy losses from the pipework ☐
d | To reduce the amount of cooling water required ☐

5. What advantage do variable area axial type desuperheaters have over fixed area types?

a | Improved approach to saturation temperature ☐
b | Increased turndown ratio ☐
c | Shorter absorption length ☐
d | It can be used on smaller pipe sizes ☐

6. What are the main applications of spray type desuperheaters?

a | Applications with fairly constant cooling water requirements ☐
b | Applications with high turndown requirements ☐
c | Applications where the steam flowrates will vary widely ☐
d | Applications requiring the elimination of all superheat ☐

Answers

1: d, 2: c, 3: c, 4: b, 5: b, 6: a

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Module 15.3

Other Types of Desuperheaters

Desuperheaters

Venturi type desuperheaters

The venturi type desuperheater employs a restriction in the superheated steam pipeline to create a region of high velocity and turbulence where the cooling water is injected. This helps to establish intimate contact between the steam and the cooling water, improving the efficiency of the desuperheating process.

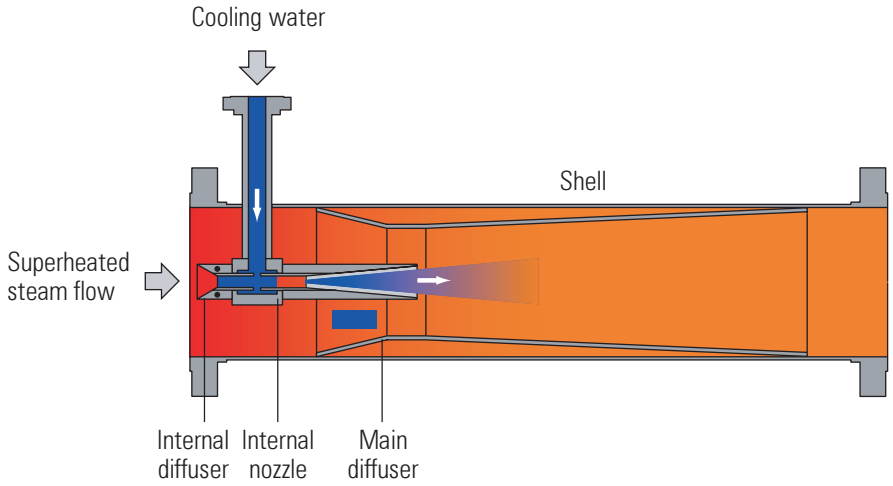


Fig. 15.3.1 Venturi type desuperheater

The desuperheating process is carried out in two separate phases:

1. The first stage of desuperheating occurs in the internal diffuser. A portion of the steam is accelerated in the internal nozzle and the velocity is used to atomise the incoming water. The cooling water is injected into the diffuser through a number of small jets, which helps to further atomise the water.
2. In the second stage of desuperheating, a saturated mist or fog emerges from the internal diffuser into the main diffuser where it mixes with the remainder of the steam.

The main diffuser itself creates a restriction to the remainder of the steam thereby increasing its velocity in this region. Thus, there is a region of turbulence where the second stage of desuperheating occurs. This mechanism minimises cooling water contact with the sidewalls, combining maximum desuperheating effectiveness with minimum pipe wear.

The steam flow turndown ratio does vary depending on the actual conditions, but 4:1 is typical. In applications where there is a dedicated pressure reducing station upstream of the desuperheater, the available steam turndown can be improved to over 5:1.

The cooling water turndown is usually satisfactory for most plant applications, with 20:1 possible depending on the actual operating conditions. At cooling water turndowns beyond 20:1, the need for a cooling water booster pump also increases.

Venturi type desuperheaters can be installed either horizontally or vertically with the steam flow upwards. When installed vertically, better mixing occurs which can result in an improved turndown ratio of over 5:1. The main problem with this is ensuring that there is enough vertical space to install the desuperheater, as it will be more than several metres long.

A modification to the standard venturi type desuperheater is the attemperator desuperheater. This essentially uses the same method of injecting the coolant into the superheated steam, but does not utilise the venturi shaped mixing section. Attemperator desuperheaters are used in place of the venturi type where there is sufficient space available to install a long absorption pipe, especially where slightly higher turndown is required, but where the additional cost of a steam atomising type cannot be justified.

The term attemperator is also generally used to refer to a desuperheater that is installed after a boiler or superheater to give accurate control over temperature and pressure.

Advantages:

1. Steam turndown ratios of up to 5:1 and cooling water turndowns of over 20:1.
2. Simple operating principle (although more complex than the spray type).
3. No moving parts.
4. Accurate control of desuperheated steam temperature; typically within 3°C of the saturation temperature.
5. Suitable for operation under steady or variable steam conditions.
6. There is reduced wear in the downstream pipework compared to a spray type desuperheater, as the cooling water emerges as a mist rather than as a spray.

Disadvantages:

1. Pressure drop is incurred (although this is generally small and within acceptable limits).
2. The absorption length is still longer than the steam atomising type; so more space is required for installation.
3. A minimum cooling water flow rate is required.

Applications:

1. Suitable for most general plant applications, except where high turndowns on steam flowrate are required.

Steam atomising desuperheaters

Steam atomising desuperheaters employ a high-pressure auxiliary steam supply to atomise the incoming cooling water.

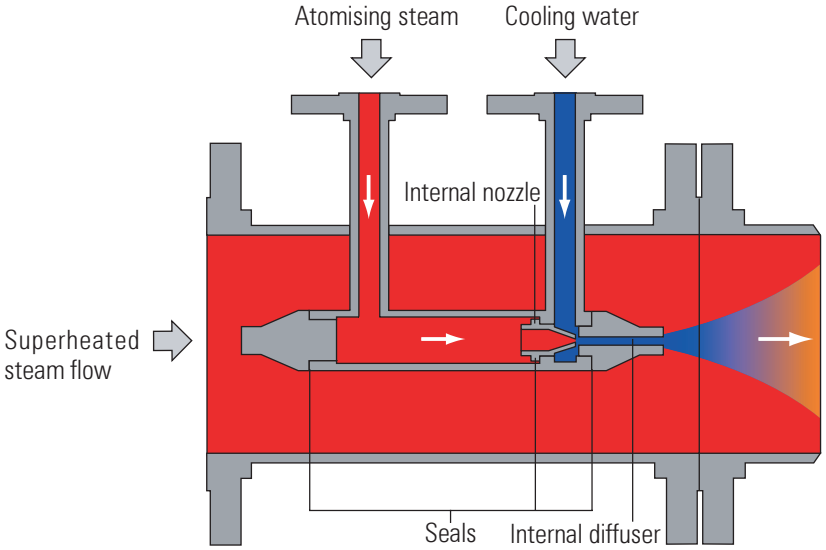


Fig. 15.3.2 Steam atomising desuperheater

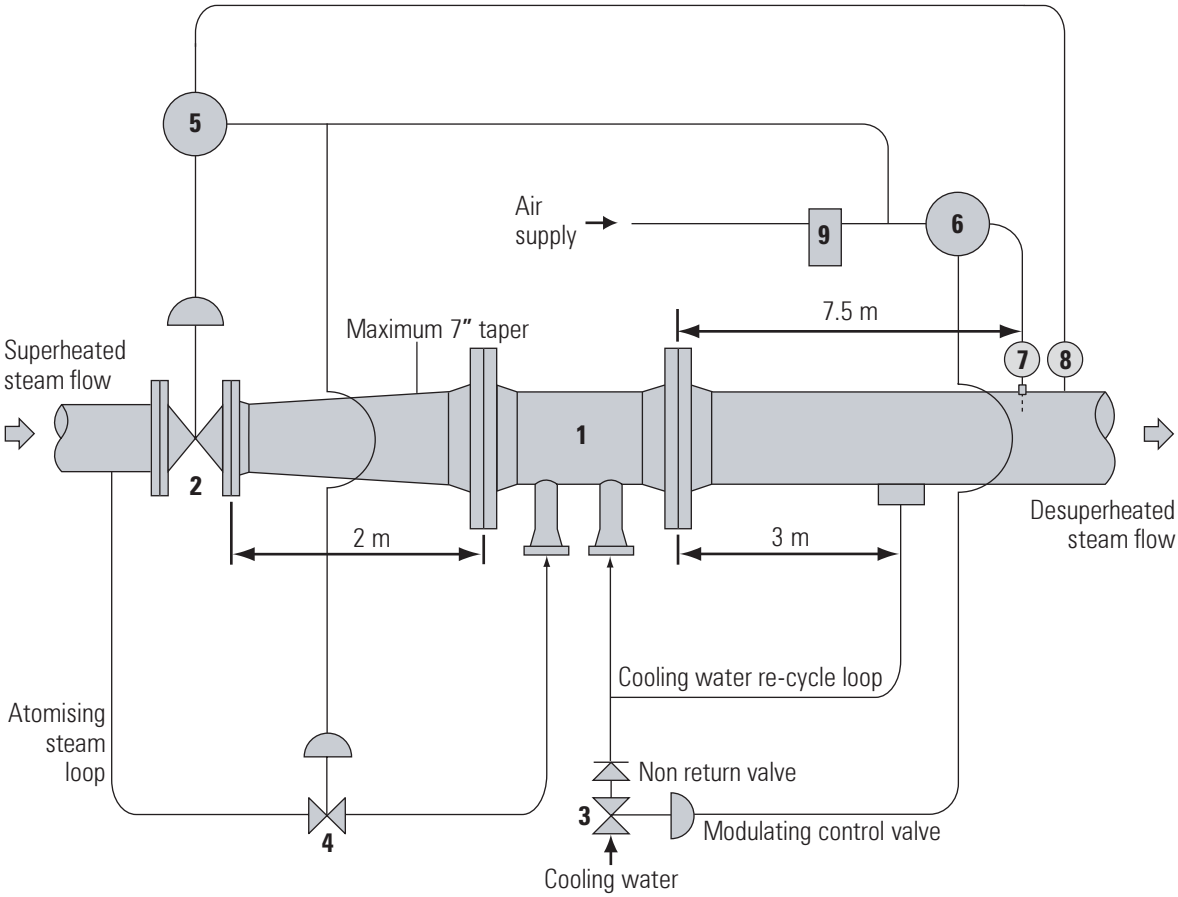
The desuperheating process occurs in two stages:

1. The first stage occurs in the diffuser, where the cooling water is atomised by the high velocity atomising steam. The auxiliary steam pressure needs to be at least 1.5 times the desuperheater inlet pressure, typically with a minimum pressure of 4 bar a. The flowrate of atomising steam is normally between 2% and 5% of the mainline flow. The use of atomising steam means that the cooling water can be introduced into the diffuser at lower pressures. In general, the only requirement is that the pressure must be greater than that of the superheated steam.
2. In the second stage, a wet mist or fog emerges from the diffuser where it mixes with the mainline steam in the pipeline. Evaporation occurs in the pipework immediately downstream of the desuperheater, where the remaining water droplets remain suspended in the steam and gradually evaporate.

Using steam to atomise the cooling water produces finely atomised water particles, which ensures efficient heat transfer and evaporation.

This arrangement allows for high steam turndown ratios; ratios of up to 50:1 are possible. It should however be noted that at turndowns greater than 20:1, low pipeline velocities may result in the 'settling out' of water, caused by the decreasing momentum of the water droplets. In this case, a drainage and recycle arrangement is required (see Figure 15.3.3). If such a recycle arrangement cannot be fitted, the turndown ratio will be reduced.

The typical installation of a steam atomising desuperheater is illustrated in Figure 15.3.3.



1. Steam atomising desuperheater
2. Steam flow control valve
3. Cooling water control valve
4. Automatic steam on/off valve
5. Pressure controller
6. Temperature controller
7. Temperature sensor
8. Pressure sensor
9. Air filter/regulator

Fig. 15.3.3 Typical steam atomising type desuperheater installation

Advantages:

1. Good turndown – steam turndown of up to 50:1 is possible, but operation and control is most efficient for a turndown of around 20:1.
2. Very compact – with a short absorption length relative to the other types.
3. Pressure drop is negligible.
4. The cooling water used can be cold, as the atomising steam will preheat it.
5. Low approach to saturation temperature – typically to within 6°C of saturation temperature.

Disadvantages:

1. Auxiliary high pressure steam is required.
2. The amount of extra equipment required and the additional pipework is relatively expensive.

Applications:

1. Suitable for applications where the steam flowrates will vary widely, for example in combined pressure reducing and desuperheating stations.

Variable orifice desuperheater

The variable orifice desuperheater controls the flow of cooling water into the mainline by a free-floating plug placed in the flow.

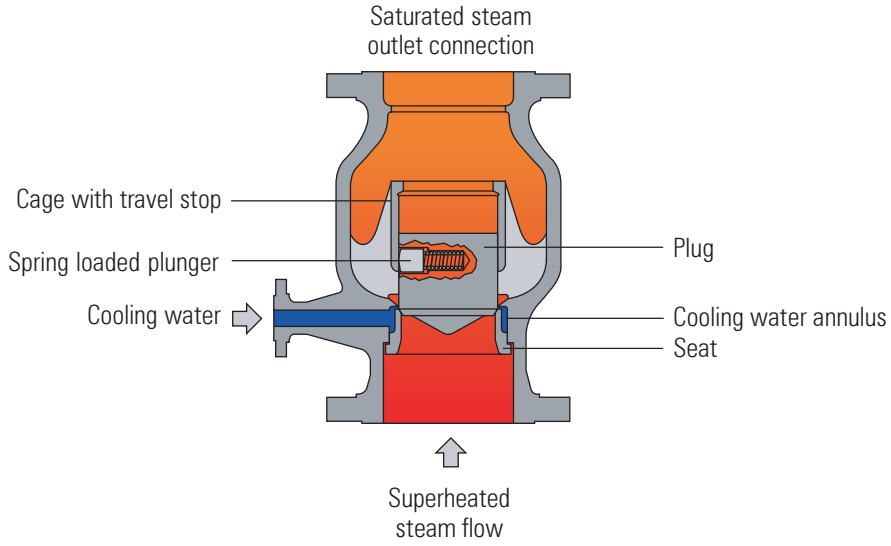


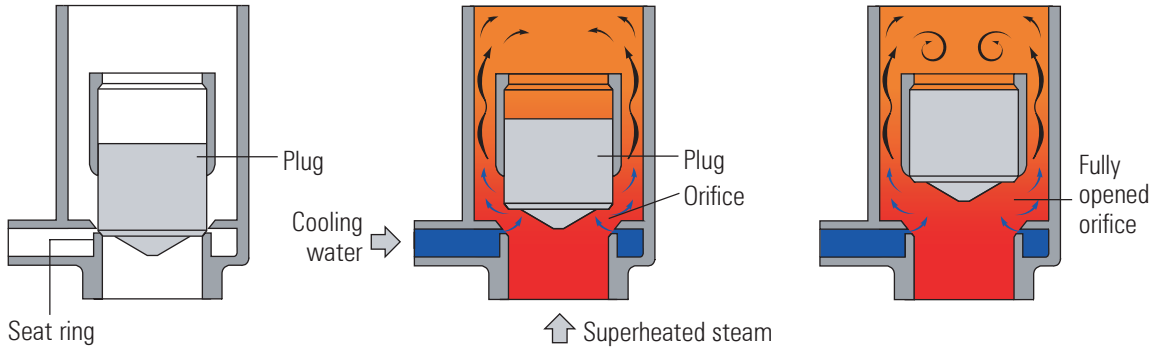
Fig. 15.3.4 Variable orifice desuperheater

The variable orifice desuperheater consists of a plug that moves up and down in a cage. This movement is limited by a travel stop incorporated into the top of the cage. Its position within the cage depends on the flow of superheated steam in the mainline.

Under no-flow conditions, the plug rests on a seat ring, surrounded by an annulus of cooling water. When superheated steam starts to flow through the desuperheater, the plug is forced off the seat by the steam pressure. As the flow increases, the plug is lifted further away from the seat, thereby creating a variable orifice between the plug and the seat. The increase in velocity between the plug and the seat creates a pressure drop across the annulus, drawing water into the superheated steam flow.

The low pressure drawing the water into the pipeline also helps to atomise the water into a fine mist. The turbulence associated with the change in velocity and direction of the steam assists in mixing the coolant and the steam. Vortices created immediately upstream of the plug ensure that the coolant is completely mixed with the steam.

The efficient mixing of the coolant and the superheated steam within the desuperheater body means that the absorption length is relatively short, and the temperature sensing element may be installed within 4 or 5 metres of the desuperheater body.



No-load
The plug is seated on the seat ring. There is no flow of superheated steam and no flow of cooling water.

Light-load
The plug lifts and opens the orifice slightly. Cooling water is instantly and thoroughly mixed with the superheated steam.

Full-load
The orifice is fully opened. There is equal pressure drop and high turbulence at all loads and over the full range.

Fig. 15.3.5 Operation of variable orifice desuperheater installation

The rate at which cooling water enters the annulus is varied by a control valve that is regulated as a function of the downstream temperature.

The plug is typically fitted with a spring-loaded plunger, which increases the friction between the plug and the cage, effectively damping the plug's movement. Given a fixed pressure drop across the valve this effectively enables the amount of cooling water to be varied when mixing with the flow of superheated steam.

The plunger also provides stability under light load conditions.

The fact that the coolant is not sprayed into the desuperheater, and that virtually all the desuperheating occurs in the body of the device, means that there is little wear of associated pipework or the desuperheater itself. Therefore, thermal sleeves are unnecessary.

A typical installation of a variable orifice desuperheater is illustrated in Figure 15.3.6

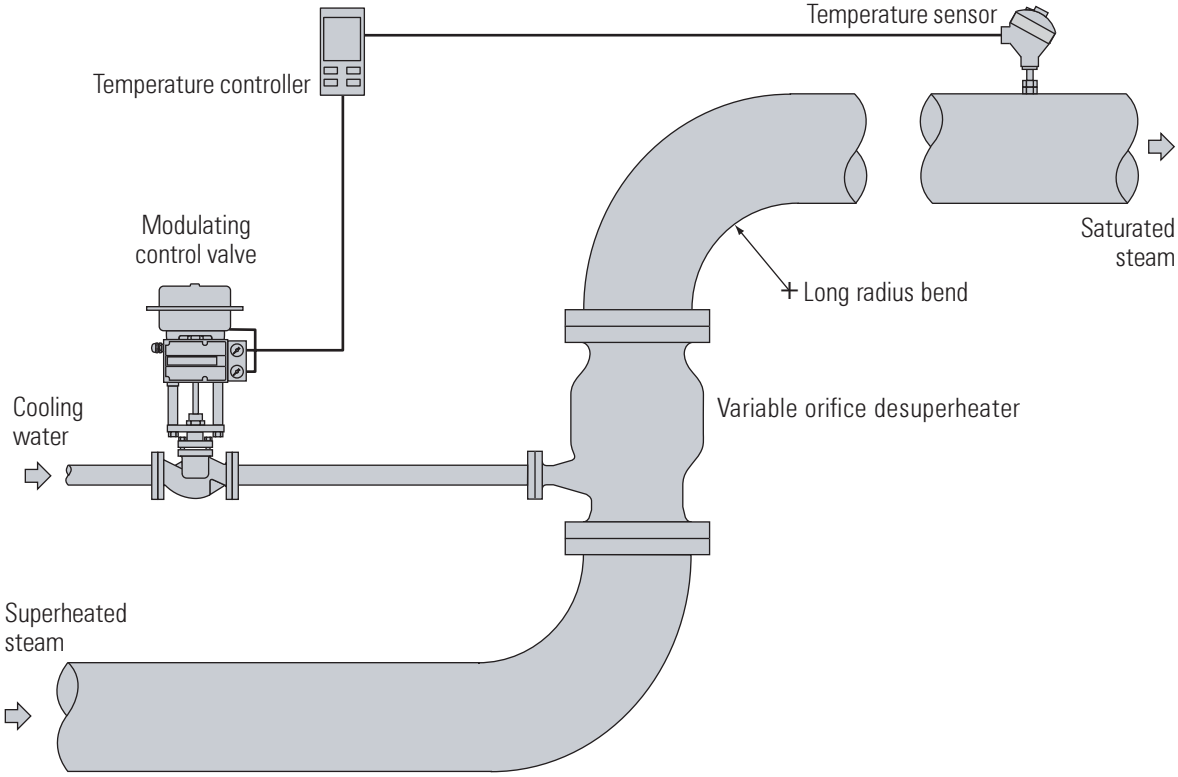


Fig. 15.3.6 Typical variable orifice type desuperheater installation

Advantages:

1. The turndown is only limited by the cooling water control valve, and steam turndown ratios of up to 100:1 can be readily achieved.
2. Low approach to saturation temperature – typically to within 2.5°C of saturation temperature.
3. Short absorption length.
4. The cooling water pressure need only be 0.4 bar superior to the superheated stem pressure.
5. Superheated steam velocities may be very low.

Disadvantages:

1. Significant pressure drop across the desuperheater.
2. Relatively higher cost.
3. The desuperheater has to be installed vertically. If a bend is located immediately after the outlet, it must have a long radius.

Applications:

1. Suitable for applications where the steam flowrate will vary widely and a relatively high pressure drop is not critical.
2. Where the steam velocity is likely to be very low.

Combined pressure control valve and desuperheater

In some instances, it is convenient to integrate the pressure control valve and the desuperheater into one unit.

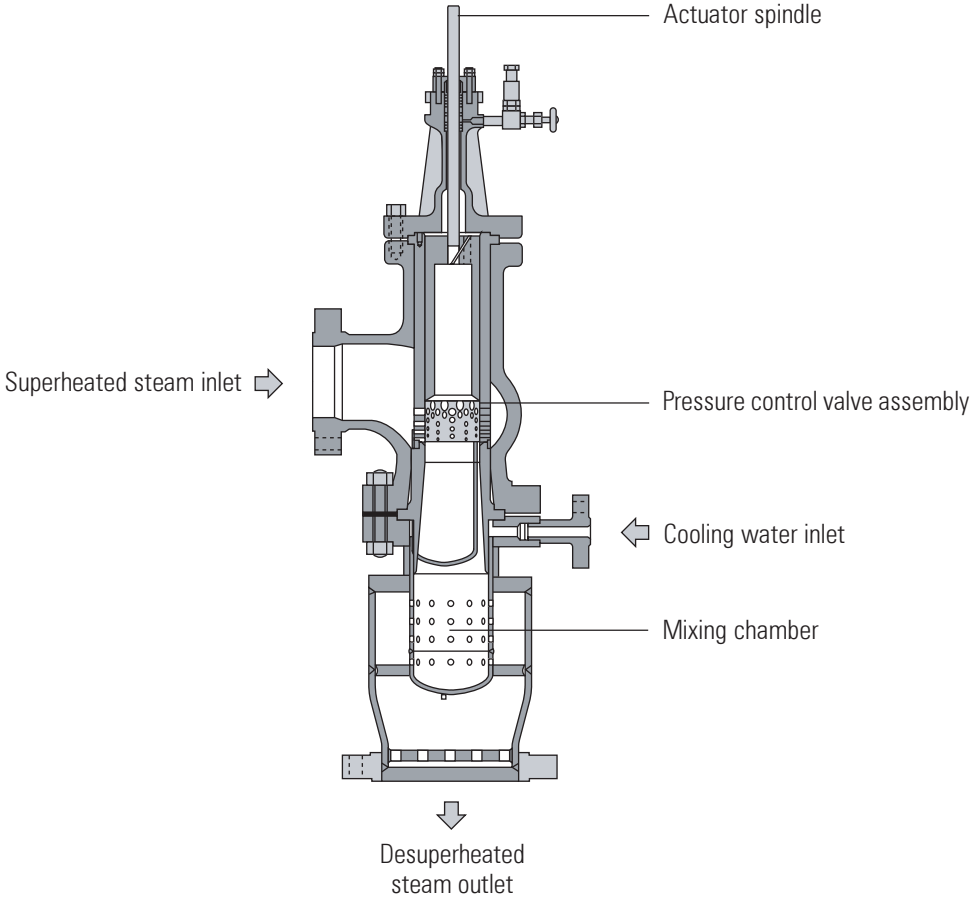


Fig. 15.3.7 Combined pressure control valve and desuperheater (cut section)

The pressure reducing aspect is similar to a standard pressure reducing valve. Although a number of different designs of pressure reducing valve could be used, angle or globe configurations are most commonly used. In addition, the valve is typically of the balanced type (with either a balancing plug or a balanced bellows arrangement) to reduce the required actuator force.

As accurate pressure control is usually important in desuperheater applications, pneumatic actuation of the valve is virtually universal, and so is the use of positioners. In addition, because quite substantial pressure drops may be involved, the manufacturer will often offer a noise reduction trim for the pressure control valve (see Figure 15.3.8).



Fig. 15.3.8 Typical valve noise reduction trim

The desuperheating aspect will also vary depending on the application, but it is common for a multiple point radial injection type to be used. The mixing of the coolant and the steam is improved due to the high velocity of the superheated steam after the pressure-reducing valve. Radial injection type desuperheaters have the advantage that they can be easily combined with the pressure reducing valve to produce a single unit.

In some combined pressure control and desuperheating stations, there are a number of baffle plates installed immediately after the desuperheating station. These plates induce further pressure drop and improve mixing of the steam and coolant.

Combined pressure control valve and desuperheating stations are commonly used in turbine bypasses, where the valve dumps the flow directly to the condenser or to 'cold reheat'.

Comparison of types of desuperheater

Table 15.3.1 compares the typical performance and installation characteristics of the different desuperheater types. It should be noted that these properties may vary between different manufacturers, and indeed, they may depend on the particular operating conditions of the system.

Table 15.3.1 Comparison of desuperheater types

Type	Steam turndown ratio	Minimum temperature above T_s (°C)	Minimum cooling water pressure* (bar)	Velocity at minimum flow (m/s)	Pipeline sizes (mm)	
					Minimum	Maximum
Tube bundle	Depends on coolant control valve	5.0	-	-	-	-
Water bath	Depends on coolant control valve	0	-	-	-	-
Multiple point radial injection	3:1	10.0	1.0	6.0	20	600
Single nozzle axial injection	3:1	10.0	0.5	6.0	50	1200
Multiple nozzle axial injection (fixed area)	8:1	8.0	4.0	6.0	150	1500
Multiple nozzle axial injection (spring assisted)	9:1	8.0	15.5	9.0	150	600
Multiple nozzle axial injection (variable area)	12:1	8.0	3.5	9.0	150	1500
Venturi	5:1	3.0	1.0	6.0	50	1270
Steam atomising	50:1	6.0	Greater than steam pressure	1.5	100	1500
Depends on Variable orifice control valve	coolant	2.5	0.4	3.0	80	800

* Typical minimum cooling water pressure above the superheated steam pressure

Questions

1. What is the main function of the venturi restriction in a venturi type desuperheater?

a| To create high velocity and turbulence ☐
b| To increase the pressure of the resulting desuperheated steam ☐
c| To contain the expansion of the superheated steam when cooling water is added ☐
d| To reduce the required pressure of the cooling water ☐

2. How does the cooling water system in the venturi type desuperheater differ from that in the steam atomising type?

a| Cooling water in the venturi type desuperheater is injected directly into the main flow of superheated steam ☐
b| The venturi type uses an external source of saturated steam to atomise the water ☐
c| The steam atomising type uses an external source of saturated steam to atomise the water ☐
d| The steam atomising type uses superheated steam to atomise the water ☐

3. In comparison with most other types of desuperheaters, what is the main disadvantage of spring assisted type desuperheaters?

a| Very low turndown ratio ☐
b| High pressure cooling water is required ☐
c| A close approach to the saturation temperature ☐
d| All of the above ☐

4. What is the function of the spring loaded plunger in a variable orifice type desuperheater?

a| To alter the amount of cooling water added to the superheated steam for a given flowrate ☐
b| To improve stability under high loads ☐
c| To limit the movement of the plug ☐
d| To act as a bluff body, increasing the amount of turbulence ☐

5. Which of the following are advantages of the steam atomising type desuperheater?

a| High turndown ratio ☐
b| Short absorption length ☐
c| The cooling water does not have to be pre-heated ☐
d| All of the above

6. Why is noise reduction trim commonly incorporated in combined pressure reduction and desuperheating stations?

a| To increase the velocity and turbulence of the superheated steam ☐
b| To improve the mixing of the coolant and the steam ☐
c| Due to the pressure drop across the valve being typically large ☐
d| None of the above ☐

Answers

1: a, 2: c, 3: b, 4: a, 5: d, 6: c

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Module 15.4

Typical Desuperheater Installations

Typical Desuperheater Installations

Installation

There are a number of important considerations to take into account when installing a desuperheater, namely:

- The properties of the cooling water.
- The installation of the desuperheater itself.
- The ancillary components required.
- The control valves used on the cooling water line and the superheated steam line.

A generalised installation of an in-line desuperheater is shown in Figure 15.4.1.

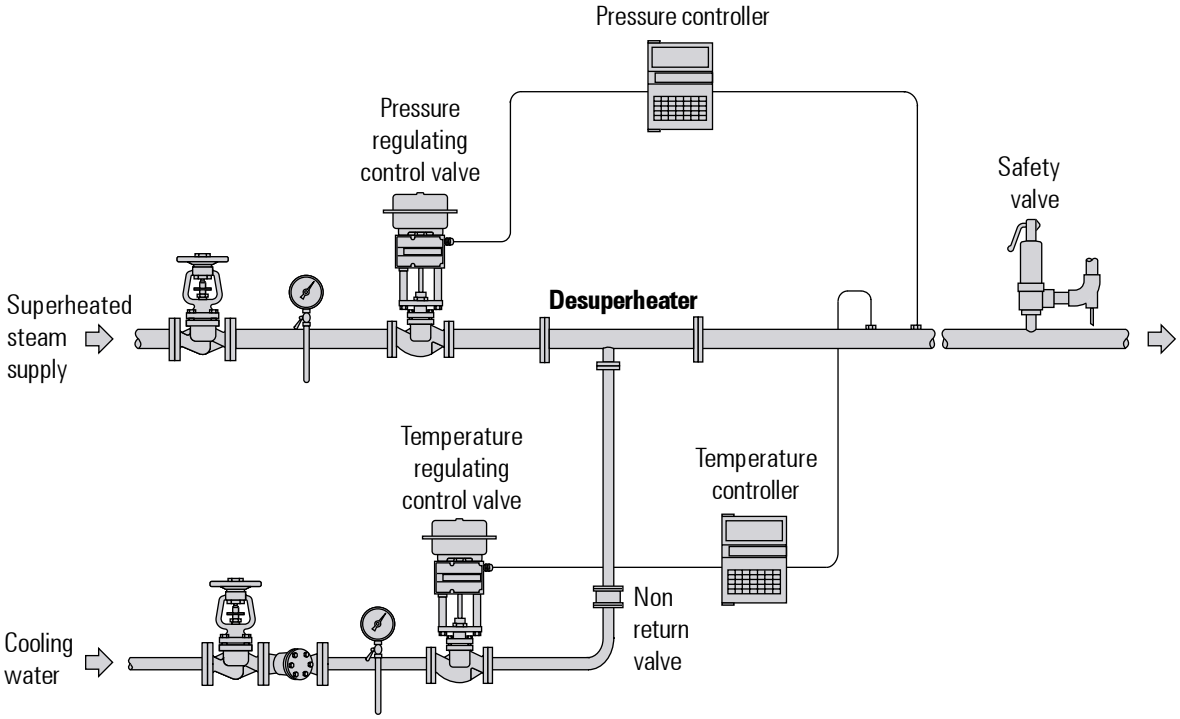


Fig. 15.4.1 A typical in-line desuperheater installation

Properties of the cooling water:

- **Temperature -** The most effective desuperheating will be achieved using cooling water that is hot, preferably as close to the saturation temperature as possible. However, cooling water temperatures as low as 5°C could be used if absolutely necessary.

  The use of hot water has the following advantages:

  - It minimises the time period for which water particles are suspended in the steam.
  - It evaporates more quickly.
  - It minimises the amount of water falling to the inside walls of the pipework.

  There are however, two disadvantages to using high temperature cooling water:

  1. The higher the temperature of the cooling water, the greater the required flowrate due to its lower cooling effect.
  2. Unless a supply of the water at the required temperature is available, additional heating mechanisms may have to be incorporated.

  Due to the benefits of using hot water, it is logical to insulate the hot water supply pipes to minimise heat loss, and to protect personnel.

- **Quality -** The quality of the injected water is important. The Total Dissolved Solids (TDS) content of the injection water should be as low as possible, as any solids that come out of solution will be deposited on:
  - The faces of valves.
  - The small orifices in the desuperheater nozzles.
  - The inner side of the piping downstream of the desuperheater.

  In addition to reducing the TDS levels, all cooling water should be passed through a suitable strainer installed before the water control valve.
- **Pressure and flowrate -** As mentioned in Module 15.2, the pressure of the cooling water, along with the area of the nozzles, determines the flow of cooling water into the desuperheater. Table 15.3.1 shows the typical minimum pressures (above the superheated steam pressure) required for each type of desuperheater. It should be noted that these might vary between manufacturers and for different steam pressures.

  If a booster pump is used, a 'spill back line' will be required to ensure that there is always sufficient flow through the pump at times of low cooling water demand.
- **Control -** A pressure drop will inevitably be required over the water control valve. When using cooling water close to saturation temperature, care is needed to ensure that the pressure drop is not large enough to cause the water to flash into steam.

  An equal percentage characteristic plug in the water control valve may be selected, which will usually complement the pump characteristic.
- **Source -** The availability of water at high pressure and temperature may be difficult. There are a number of possible sources of the cooling water; and options include:
  - Water from the pressure side of the boiler feedpump (providing the boiler uses modulating level control).
  - De-mineralised water.
  - Condensate.
  - Town water. This however may require treatment to improve the quality, otherwise salts may be deposited on the inside of the desuperheater downstream pipework.

Desuperheater installation

The total installed length of a desuperheater station will vary with size and type, but it is typically about 7.5 m.

Most desuperheaters can be installed in any direction (the variable orifice type is a notable exception), but if installed vertically, the flow should be upwards. The venturi type is best installed in a vertical pipe with the flow in the upward direction, as this aids mixing of the water and the steam. However, such installations are not usually possible due to the vertical space required.

Superheated steam pressure control

Although it is possible to design desuperheater installations to operate with varying upstream pressures, it is much simpler if a constant supply pressure is maintained.

The amount of cooling water added is controlled by the temperature of the steam after the desuperheater. The higher the temperature, the more the control valve will open, and the greater the amount of water added. The target is to reduce the steam temperature to within a small margin of the design discharge temperature.

If the superheated steam supply pressure is increased, the saturation temperature will also increase. However, the set value on the coolant controller will not change, and an excessive amount of water will be added, resulting in wet steam.

Pressure sensors used in the control of the superheated steam pressure should ideally be located at the point of use, so that the pressure control valve can compensate for any line loss between the desuperheater and the point of use.

Temperature sensor positioning

The minimum distance from the point of water injection to the temperature sensing point is critical:

- If the sensor is too close to the water injection point, the mixing of the steam and the water will not have been completed, and the temperature sensor will give a false output.
- If the sensor is too far away, it will make the installation unnecessarily long.

The minimum installation distance will vary between different types of desuperheater and with different manufacturers. It is usually specified as a function of the temperature difference between the required outlet temperature and either the inlet temperature or the coolant temperature. Figure 15.4.2 shows a typical manufacturer's sensor positioning chart.

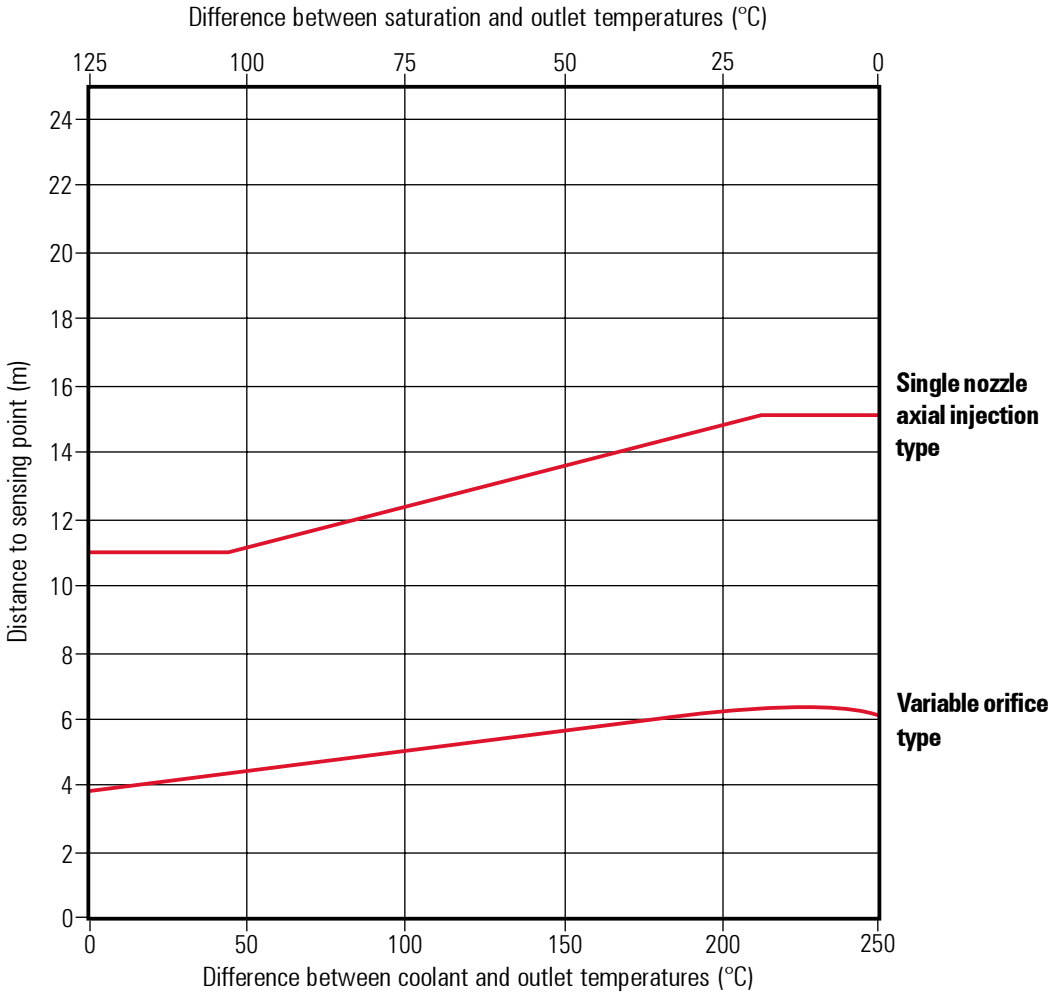


Fig. 15.4.2 Positioning of the temperature sensor

Separator station

Efficient drainage of the pipework following the desuperheater is essential. To ensure that water cannot accumulate at any point, the pipe should be arranged to fall approximately 20 mm per metre in the direction of flow, and should be provided with a separator station.

The steam trap used to drain the separator should be carefully selected to prevent air binding, and the discharge pipe from the steam trap should have ample capacity to deal with the drainage and it should be fixed as near to vertical as possible. In addition, there must be sufficient space in the drainage pipe for the water to flow down and air to pass up the pipe. The steam trap must also be able to withstand superheat conditions.

On critical applications, for example, prior to a turbine, a separator is even more important; the separator station will remove entrained water in the case of control failure, and prevent too much water being added to the steam.

Isolation valves

To allow maintenance to be safely carried out, isolation valves are recommended upstream of:

- The superheated steam pressure control valve.
- The desuperheater.
- The cooling water supply.

Typically, these should be installed approximately, but no less than 10 pipe diameters from the item they are isolating.

Safety valve

A safety valve may be required to protect equipment downstream of the desuperheating station from overpressure, in the event of failure of the pressure control station.

It is necessary to ensure discharge pipework from the safety valve is led away to a safe area. This is of particular importance as high temperature superheated steam may be discharged.

Temperature and pressure ratings

Most equipment to be used on steam systems is designed with saturated steam in mind. It is therefore important that all equipment used in a desuperheater station will tolerate both the maximum temperature and pressure of the superheated steam.

Most equipment will have specified pressure and temperature limitations that are based on the nominal pressure (PN) rating of the material and the specific design of the device. By definition, the PN rating is the maximum pressure that a material can withstand at 120°C. For example, a PN16 rating means that the material will withstand a pressure of 16 bar g at 120°C. At higher temperatures, the maximum pressure will decrease, however, the exact relationship varies and depends on the material.

Figure 15.4.3 depicts typical pressure/temerature gradients for PN16, PN25 and PN40 rated products on a non-specific material. It is important to note that different materials will, by specification, produce variations in the temperature gradient.

In addition, components such as gaskets, fasteners and internal components may have a further limiting effect on the maximum temperature and pressure.

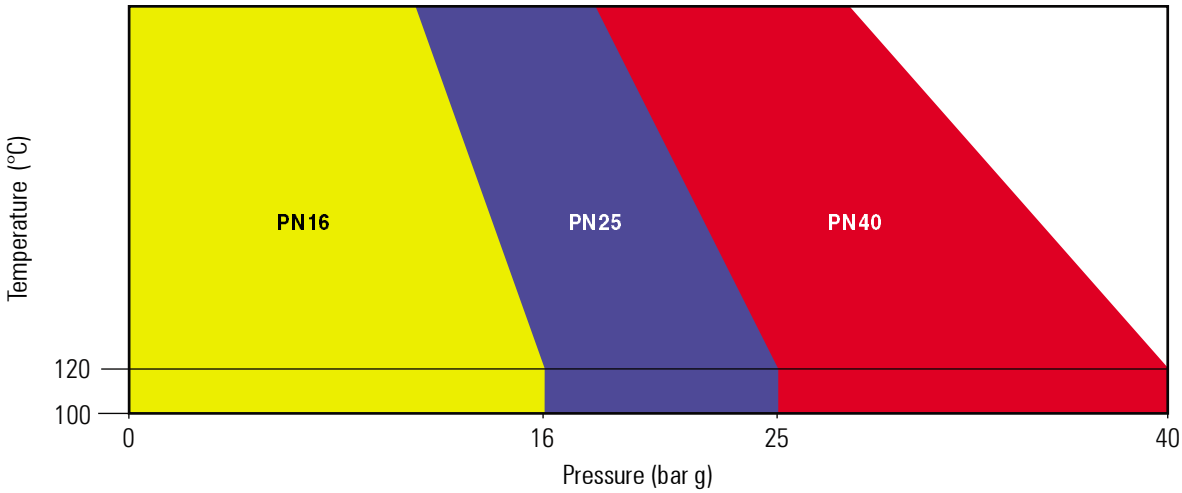


Fig. 15.4.3 Typical PN pressure/temperature gradients

Controls

The selection and installation of the control devices to be used in a desuperheater station are an important consideration, as they can affect the overall turndown of the desuperheater. If the controls installed have a lower turndown ratio than the desuperheater itself, the turndown of the desuperheater station will be reduced (refer to Module 15.2)

Further information on basic control theory and practice can be found in Blocks 5 to 8 inclusive.

Selection

When selecting a suitable type of desuperheater for a particular application, the following factors need to be considered:

- **Separator station -** This is probably one of the most important considerations, as the different types of desuperheater vary significantly in the range of superheated steam flowrates that can be effectively desuperheated.

  It is important to note here that, although ensuring that the device will have sufficient turndown for the flow likely to be encountered, it is important not to specify more turndown capability than is really needed. This predominantly affects cost, but it can also lead to poor system performance. Poor performance is often aggravated by the fact that most desuperheaters tend to perform better at the higher end of the specified flowrates and a system designer would tend to allow for increases in capacity due to expansion. As an extreme example, if the maximum flow specified was ten times the current requirement (in order to take into account future growth), the desuperheater will operate between 1 and 10% of its full flowrate instead of the 10% to 100% it is designed for.

- **Desuperheated steam temperature -** As seen in the previous Module, different types of desuperheater are capable of reducing the steam temperature to within several degrees of the saturation temperature. For example, if desuperheated steam temperatures within 5°C of the saturation temperature (T_S) were required, a venturi or variable orifice type desuperheater would be selected (see Table 15.3.1).

  Generally, where some degree of residual superheat can be tolerated, the desuperheated steam temperature should be as high above saturation as possible. This is beneficial for several reasons:

  1. Cost – a close approximation to saturation temperature is generally only achievable with the more costly types of desuperheaters.

  2. Controller sensitivity – this may be a problem where the desuperheated steam temperature is required to be close to the saturation temperature. Limited controller sensitivity is one of the reasons why most desuperheaters are limited in their approach to saturation temperature. For example, if a controller had a sensitivity of ±5°C, it would not be able to distinguish between saturation temperature and 5°C above. If such a controller interpreted the steam temperature at 5°C above T_S, and the steam were actually at T_S, it would increase the flow of cooling water. But since the temperature of the saturated steam will not decrease (due to the latent heat of evaporation), the controller will add increasingly more coolant as it would still believe the system to be at 5°C above T_S. This will result in very wet steam flooding the steam main after the desuperheater station.

  3. It becomes increasingly difficult to evaporate the cooling water as the superheated steam temperature drops towards saturation, due to the reduced temperature difference between the two.

  4. The lower temperature difference also reduces the heat transfer rate between the water and the steam, and therefore the water droplets have to stay in suspension for longer to be evaporated. This increases the likelihood that the water particles will fall out of suspension in the pipe. In order to prevent this from occurring, as the temperature approaches T_S, the velocity of the steam needs to be increased in order to create more turbulence.

- **Available coolant supply pressure -** The choice of desuperheater type will also depend on the availability of cooling water at the necessary pressure. It would provide a cost advantage to use cooling water that is already available, for example, from the pressure side of a boiler feedwater pump.

  If the available pressure were not sufficient for a particular type of desuperheater, additional pumping arrangements would have to be made.

A typical manufacturer's selection chart is shown in Figure 15.4.4. It is based on the typical performance and installation characteristics, which can be found in Table 15.3.1. The method used to size a desuperheater will vary depending on the particular manufacturer and the type of desuperheater, and therefore it is outside the scope of this publication.

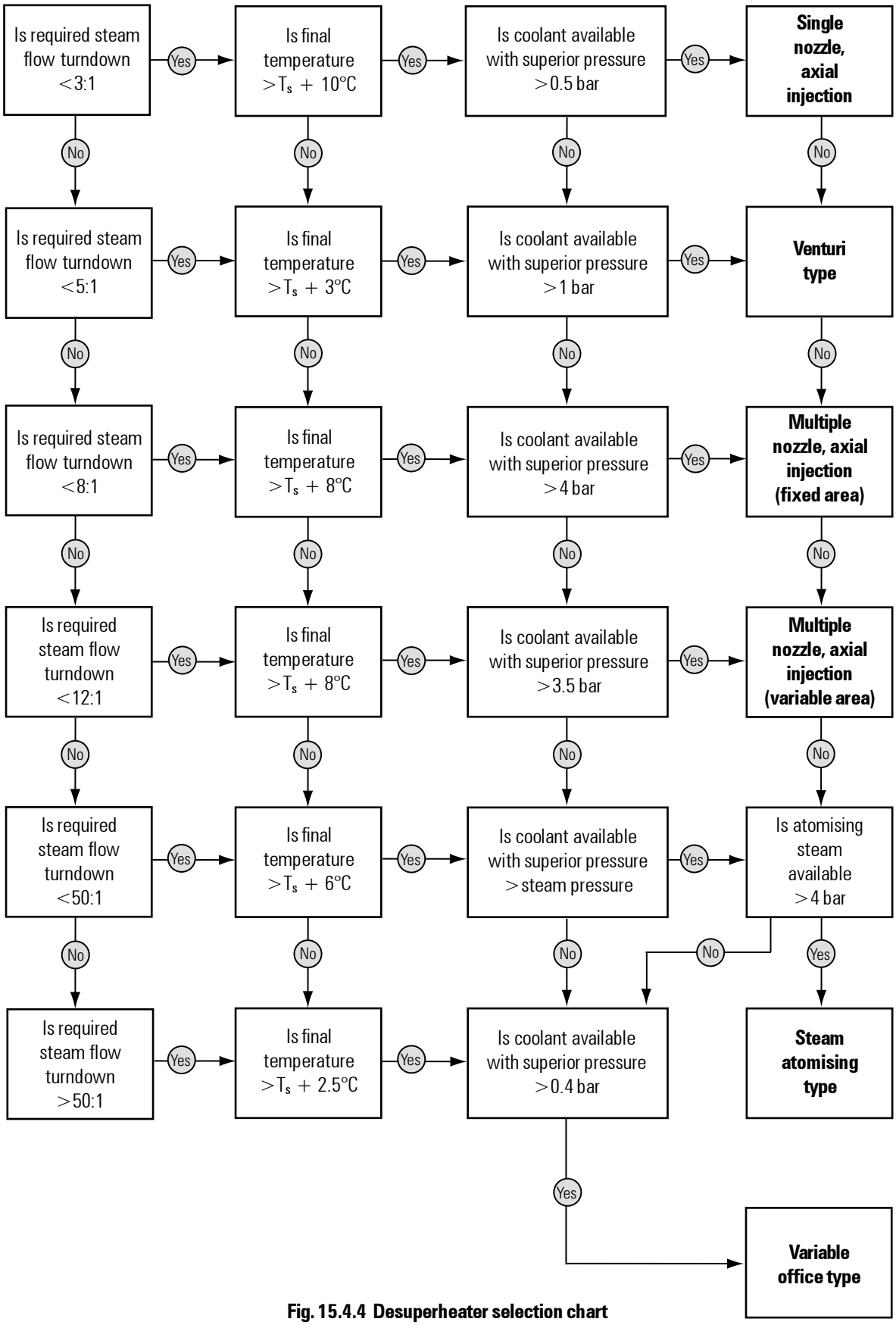


Fig. 15.4.4 Desuperheater selection chart

Typical applications

Desuperheaters are mainly applied in two areas:

1. Power generation - Desuperheaters are mainly used to reduce the temperature of steam emitted from turbine bypass systems to an efficient level for operation on other parts of the plant requiring saturated steam for heat transfer purposes.

2. Process industry - In process industries, desuperheaters are used as part of a system for reducing the temperature and pressure of steam from boilers to economical levels of operation. Table 15.4.1 shows some common application examples in particular industries.

Table 15.4.1 Typical applications for desuperheaters in process industries

Industry	Applications
Paper and board	Paper drying machines.
Food	Steam cooking kettles. Evaporator heat exchangers.
Textiles	Fabric finishing autoclaves.
Tobacco	Tobacco leaf drying plants. Reactor heater jackets and coils. Distillation plants.
Chemical and pharmaceutical	Methanol plants. Sulphur plants. Polymerisation plants. Chemical. Vacuum distillation start-up heaters. Glycol and ethylene plants. Aromatics recovery.
Oil and petrochemical	Vinyl chloride plants. Thermal and catalytic crackers. Vacuum distillation. Polymerisation plants. Sulphur plants. De-salting. Isomerisation processes.
Refineries	Thermal and catalytic crackers. Vacuum distillation. Sulphur plants.
Brewing and distilling	Steam heating systems.
Boiler and turbine installations	Power generation. Shipbuilding.

Questions

1. Which of the following are advantages of using hot, instead of cold water as the cooling fluid in a desuperheater?

i. Water particles spend less time suspended in the steam.

ii. Less water impinges on the inside walls of downstream pipework.

iii. It reduces the absorption length as the water is evaporated more quickly.

a| i only ☐

b| ii only ☐

c| i and ii ☐

d| i, ii and iii ☐

2. Why is tap water not an ideal source for cooling water for most types of desuperheater?

a| It is generally available at low pressures ☐

b| It is at room temperature ☐

c| High TDS levels can result in solids falling out of suspension ☐

d| All of the above ☐

3. Why do most desuperheaters not reduce the temperature of the superheated steam to saturation temperature?

i. The decrease in temperature difference between the cooling water and the steam makes this difficult.

ii. The sensitivity of the cooling water flowrate controller will cause flooding of the pipework at temperatures close to saturation.

iii. Producing steam at saturation temperature is not beneficial.

a| i only ☐

b| ii only ☐

c| i and ii ☐

d| i, ii and iii ☐

4. Why is it important not to over specify the turndown ratio of a desuperheater?

a| It will increase the cost of the desuperheater station ☐

b| It will increase the risk of the downstream pipework flooding ☐

c| It will increase the amount of cooling water required for a given superheated steam flowrate ☐

d| The statement is false and all desuperheaters should be specified with a significantly larger turndown to allow for expansion ☐

5. Using the selection chart (figure 15.4.4), which type of desuperheater would be most suitable for use in a sugar refinery for the following conditions? The site already has a boiler producing 10 bar g saturated steam. Coolant is available at 14 bar g.

Maximum superheated steam flow 16000 kg/h

Minimum superheated steam flow 2000 kg/h

Final temperature T_S + 7°C

a | Tube bundle type ☐

b | Multiple nozzle, axial injection type ☐

c | Steam atomising type ☐

d | Variable orifice type ☐

6. Which of the following considerations have to be made when selecting a type of desuperheater?

a | Turndown ratio ☐

b | The required final temperature ☐

c | The availability of cooling water ☐

d | All of the above ☐

Answers

1: d, 2: d, 3: c, 4: a, 5: c, 6: d

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Module 16.1

Equations

Equations

Block 1: Introduction

Equation number	Equation
	There are no equations in Block 1

Block 2: Steam engineering principles and heat transfer

Equation number	Equation
2.1.1	**Thermodynamic temperature** T (K) = Temperature (°C) + 273.15
2.1.2	**Density of a material** $\rho = \frac{m}{V} = \frac{1}{v_g}$ Where: ρ = Density (kg/m³) m = Mass (kg) V = Volume (m³) v_g = Specific volume (m³/kg)
2.1.3	**Specific gravity of a material** $\text{Specific gravity} = \frac{\text{Density of substance } \rho_s}{\text{Density of water } \rho_w}$
2.1.4	**Energy transfer equation** $Q = m\, c_p\, \Delta T$ Where: Q = Quantity of energy (kJ) m = Mass of the substance (kg) c_p = Specific heat capacity of the substance (kJ/kg °C) ΔT = Temperature rise of the substance (°C)
2.1.5	**Change in entropy** $\text{Change in entropy } (\Delta S) = \frac{\text{Change in enthalpy } (\Delta H)}{\text{Average absolute temperature } (\Delta T)}$
2.1.6	**Change in specific entropy** $\text{Change in specific entropy } (\Delta s) = \frac{\text{Change in specific enthalpy } (\Delta h)}{\text{Average absolute temperature } (\Delta T)}$
2.2.1	**Total enthalpy of saturated steam** $h_g = h_f + h_{fg}$ Where: h_g = Total enthalpy of saturated steam (Total heat) (kJ/kg) h_f = Liquid enthalpy (Sensible heat) (kJ/kg) h_{fg} = Enthalpy of evaporation (Latent heat) (kJ/kg)

Equation number	Equation
2.2.2	**Enthalpy of evaporation of wet steam** $\text{Actual enthalpy of evaporation} = h_{fg}\chi$ Where: h_{fg} = Enthalpy of evaporation (Latent heat) (kJ/kg) χ = Dryness fraction
2.2.3	**Total enthalpy of wet steam** $\text{Actual total enthalpy} = h_f + h_{fg}\chi$ Where: h_f = Liquid enthalpy (Sensible heat) (kJ/kg) h_{fg} = Enthalpy of evaporation (Latent heat) (kJ/kg) χ = Dryness fraction
2.2.4	**Specific volume of wet steam** $\text{Actual specific volume} = v_g\chi$ Where: v_g = Specific volume of dry steam at same pressure χ = Dryness fraction
2.2.5	**Flash steam produced from hot water and condensate** $\text{Proportion of flash steam} = \frac{(h_f \text{ at } P_1) - (h_f \text{ at } P_2)}{h_{fg} \text{ at } P_2}$ Where: P_1 = Initial pressure P_2 = Final pressure h_f = Liquid enthalpy (kJ/kg) h_{fg} = Enthalpy of evaporation (kJ/kg)
2.3.1	**Carnot efficiency** $\text{Carnot efficiency } \eta_C = \frac{T_i - T_e}{T_i}$ Where: T_i = Temperature at turbine inlet (K) T_e = Temperature at turbine exhaust (K)
2.3.2	**Rankine efficiency** $\text{Rankine efficiency } \eta_R = \frac{H_i - H_e}{H_i - h_e}$ Where: H_i = Heat at turbine inlet (kJ/kg) H_e = Heat at turbine exhaust (kJ/kg) h_e = Sensible heat in condensate (kJ/kg)

Equation number	Equation
2.4.1	**Dalton's law of partial pressures** $\begin{pmatrix}\text{Effective steam}\\ \text{pressure (bar a)}\end{pmatrix} = \begin{pmatrix}\text{Amount of steam as a proportion}\\ \text{of total (by volume)}\end{pmatrix} \times \begin{pmatrix}\text{Indicated pressure}\\ \text{(bar a)}\end{pmatrix}$
2.5.1	**Heat transfer by conduction through a layer (Fourier's law)** $\dot{Q} = k A \frac{\Delta T}{\varkappa}$ Where: $\dot{Q}$ = Heat transferred per unit time (W) k = Thermal conductivity of the material (W/m K or W/m°C) A = Heat transfer area (m²) ΔT = Temperature difference across the material (K or °C) $\varkappa$ = Material thickness (m)
2.5.2	**Heat transfer by convection (Newton's law of cooling)** $\dot{Q} = h A \Delta T$ Where: $\dot{Q}$ = Heat transferred per unit time (W) h = Convective heat transfer coefficient of the process (W/m² °C) A = Heat transfer area of the surface (m²) ΔT = Temperature difference between the surface and the bulk fluid (K or °C)
2.5.3	**General heat transfer** $\dot{Q} = U A \Delta T$ Where: $\dot{Q}$ = Heat transferred per unit time (W) U = Overall heat transfer coefficient (W/m² °C) A = Heat transfer area (m²) ΔT = Temperature difference between the primary and secondary fluid (K or °C) **Note:** $\dot{Q}$ will be a mean heat transfer rate ($\dot{Q}_M$) if ΔT is a mean temperature difference (ΔT_{LM} or ΔT_{AM}).
2.5.4	**Arithmetic mean temperature difference (AMTD or ΔT_{AM})** $\Delta T_{AM} = T_s - \left(\frac{T_1 + T_2}{2}\right)$ Where: T_s = Steam temperature (°C) T_1 = Secondary fluid in temperature (°C) T_2 = Secondary fluid out temperature (°C)

Equation number	Equation
2.5.5	**Log mean temperature difference (LMTD or ΔT_{LM})** $\Delta T_{LM} = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)}$ Where: T_s = Steam temperature (°C) T_1 = Secondary fluid in temperature (°C) T_2 = Secondary fluid out temperature (°C) ln = A mathematical function known as 'natural logarithm'
2.5.6	**Rate of heat transfer across a barrier knowing the thickness and conductivity** $\dot{Q} = A \frac{\Delta T}{\varkappa/k}$ Where: $\dot{Q}$ = Heat transferred per unit time (W) A = Heat transfer area (m²) ΔT = Temperature difference across the barrier (°C) $\varkappa/k$ = Barrier thickness/material thermal conductivity $\left(\frac{m^2 °C}{W}\right)$
2.5.7	**Rate of heat transfer across a barrier knowing thermal resistance** $\dot{Q} = A \frac{\Delta T}{R}$ Where: $\dot{Q}$ = Heat transferred per unit time (W) A = Heat transfer area (m²) ΔT = Temperature difference across the barrier (°C) R = Thermal resistance of the barrier (m² °C/W)
2.5.8	**Resistivity from conductivity** $r = \frac{1}{k}$ Where: r = Thermal resistivity (m°C/W) k = Thermal conductivity (W/m°C)
2.5.9	**Thermal transmittance (heat transfer coefficient) from thermal resistance** $U = \frac{1}{R}$ Where: U = Thermal transmittance of the barrier (W/m² °C) R = Thermal resistance of the barrier (m² °C/W)

Equation number	Equation
2.5.10	**Thermal transmittance (U) from the individual thermal resistances** $U = \frac{1}{R_1 + R_2 + R_3 + R_4 + R_5 + R_6}$ Where: R_1 = Resistance of the air film R_2 = Resistance of the condensate film R_3 = Resistance of the scale film on the steam side R_4 = Resistance of the of the metal wall R_5 = Resistance of the scale film on the water side R_6 = Resistance of the product film
2.5.11	**Thermal transmittance (U) from the individual thicknesses and conductivities** $U = \frac{1}{\frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_4}{k_4} + \frac{x_5}{k_5} + \frac{x_6}{k_6}}$ Where: $\frac{x_1}{k_1} = \frac{\text{Thickness of air film}}{\text{Thermal conductivity of air}}$ $\frac{x_2}{k_2} = \frac{\text{Thickness of condensate film}}{\text{Thermal conductivity of condensate}}$ $\frac{x_3}{k_3} = \frac{\text{Thickness of scale film on steam side}}{\text{Thermal conductivity of scale}}$ $\frac{x_4}{k_4} = \frac{\text{Thickness of metal wall}}{\text{Thermal conductivity of metal}}$ $\frac{x_5}{k_5} = \frac{\text{Thickness of scale film on water side}}{\text{Thermal conductivity of scale}}$ $\frac{x_6}{k_6} = \frac{\text{Thickness of water film}}{\text{Thermal conductivity of water}}$
2.6.1	**Energy requirement for a non-flow application (e.g. batch or tank)** $\dot{Q} = \frac{m\, c_p\, \Delta T}{t}$ Where: $\dot{Q}$ = Mean heat transfer rate (kW (kJ/s)) m = Mass of the fluid (kg) c_p = Specific heat capacity of the fluid (kJ/kg °C) ΔT = Increase in fluid temperature (°C) t = Time for the heating process (seconds)
2.6.2	**Quantity of heat transferred by condensing steam** $Q = m_s\, h_{fg}$ Where: Q = Quantity of heat (kJ) m_s = Mass of steam (kg) h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)

Equation number	Equation
2.6.3	**Heat transfer of condensing steam** $\dot{Q} = \dot{m}_s \, h_{fg}$ Where: $\dot{Q}$ = Mean heat transfer rate (kW or kJ/s) $\dot{m}_s$ = Mean steam consumption (kg/s) h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)
2.6.4	**Energy balance between steam and secondary fluid of a non-flow process** $\dot{m}_s \, h_{fg} = \dot{Q} = \frac{m \, c_p \, \Delta T}{t}$ Where: $\dot{m}_s$ = Mean steam consumption rate (kg/s) h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg) $\dot{Q}$ = Mean heat transfer rate (kW (kJ/s)) m = Mass of the secondary fluid (kg) c_p = Specific heat capacity of the secondary fluid (kJ/kg °C) ΔT = Temperature rise of the secondary fluid (°C) t = Time for the heating process (seconds)
2.6.5	**Energy requirement for a flow-type application (e.g heat exchanger)** $\dot{Q} = \dot{m} \, c_p \, \Delta T$ Where: $\dot{Q}$ = Mean heat transfer rate (kW) $\dot{m}$ = Mean secondary fluid flowrate (kg/s) c_p = Specific heat capacity of the secondary fluid (kJ/kg K) or (kJ/kg°C) ΔT = Temperature rise of the secondary fluid (K or °C)
2.6.6	**Energy balance between steam and fluid of a flow-type application** $\dot{m}_s \, h_{fg} = \dot{Q} = \dot{m} \, c_p \, \Delta T$ Where: $\dot{m}_s$ = Mean steam consumption rate (kg/s) h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg) $\dot{Q}$ = Mean heat transfer rate (kW (kJ/s)) $\dot{m}$ = Mass flowrate of the secondary fluid (kg/s) c_p = Specific heat capacity of the secondary fluid (kJ/kg °C) ΔT = Temperature rise of the secondary fluid (°C)
2.6.7	**Mean steam consumption of a flow type application** $\dot{m}_s = \frac{\dot{m} \, c_p \, \Delta T}{h_{fg}}$ Where: $\dot{m}_s$ = Mean steam consumption rate (kg/s) $\dot{m}$ = Mass flowrate of the secondary fluid (kg/s) c_p = Specific heat capacity of the secondary fluid (kJ/kg °C) ΔT = Temperature rise of the secondary fluid (°C) h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)

Equation number	Equation
2.6.8	**Mean steam consumption of a flow type application** $\dot{m}_s = \frac{\dot{Q}}{h_{fg}}$ Where: $\dot{m}_s$ = Mean steam consumption rate (kg/s) $\dot{Q}$ = Mean heat transfer rate (kW) h_{fg} = Specific enthalpy of evaporation of steam (kJ/kg)
2.8.1	**To determine the required steam flowrate from a kW rating** $\text{Steam flowrate (kg/h)} = \frac{\text{Load in kW x 3 600}}{h_{fg} \text{ at operating pressure}}$
2.11.1	**To determine the steam flowrate for a steam injection process** $\dot{m}_s = \frac{\dot{Q}}{h_g - (T\ c_p)}$ Where: $\dot{m}_s$ = Mean steam flowrate (kg/s) $\dot{Q}$ = Mean heat transfer rate (kW) h_g = Specific total enthalpy of the steam upstream of the control valve (kJ/kg) T = Final temperature of the water c_p = Specific heat capacity of the water (kJ/kg °C)
2.11.2	**Steam consumption to provide tank heat losses** $\dot{m}_s = \frac{\dot{Q}}{2\,256.7}$ Where: $\dot{m}_s$ = Mean steam flowrate to provide the heat losses from the tank (kg/s) $\dot{Q} = \dot{Q}_{(sides)} + \dot{Q}_{(surface)}$ (kW) 2256.7 = Enthalpy of evaporation at atmospheric pressure (kJ/kg)
2.11.3	**Mass and heat balance for steam injection into a tank** $m\,h_1 + m_s\,h_g = (m + m_s)\,h_2$ Where: m = Initial mass of water in the tank (kg) m_s = The mass of steam to be injected (kg) h_1 = The heat in the water at the initial temperature (kJ/kg) h_2 = The heat in the water at the final temperature (kJ/kg) h_g = The total enthalpy of the steam upstream of the control valve (kJ/kg)

Equation number	Equation
2.11.4	**Steam consumption by injection into a tank** $m_s = \frac{m\,(h_2 - h_1)}{h_g - h_2}$ Where: m_s = The mass of steam to be injected (kg) m = Initial mass of water in the tank (kg) h_1 = The heat in the water at the initial temperature (kJ/kg) h_2 = The heat in the water at the final temperature (kJ/kg) h_g = The total enthalpy of the steam upstream of the control valve (kJ/kg)
2.12.1	**Steam start-up load to bring steam pipework to operating temperature** $\dot{m}_s = \frac{60\,W\,(T_s - T_{amb})\,c_p}{h_{fg}\,t}$ kg/h Where: $\dot{m}_s$ = Mean rate of condensation of steam (kg/h) W = Total weight of pipe plus flanges and fittings (kg) T_s = Steam temperature (°C) T_{amb} = Ambient temperature (°C) c_p = Specific heat of pipe material (kJ/kg°C) h_{fg} = Enthalpy of evaporation at operating pressure (kJ/kg) t = Time for warming up (minutes)
2.12.2	**Steam running load to keep a steam main at operating temperature** $\dot{m}_s = \frac{3.6\,\dot{Q}\,L\,f}{h_{fg}}$ kg/h Where: $\dot{m}_s$ = Rate of condensation (kg/h) $\dot{Q}$ = Heat emission rate (W/m) L = Effective length of pipe allowing for flanges and fittings (m) f = Insulation factor (dimensionless) h_{fg} = Enthalpy of evaporation at operating pressure (kJ/kg) **Note:** The constant 3.6 gives the answer in kg/h
2.12.3	**Steam condensing rate for air heating equipment** $\dot{m}_s = \frac{3\,600\,\dot{V}\,\Delta T\,c_p}{h_{fg}}$ kg/h Where: $\dot{m}_s$ = Rate of steam condensation (kg/h) $\dot{V}$ = Volumetric flowrate of air being heated (m^3/s) ΔT = Air temperature rise (°C) c_p = Specific heat of air at constant pressure (kJ/m^3°C) h_{fg} = Enthalpy of evaporation of steam in the coils (kJ/kg) **Note:** The constant 3 600 gives the solution in kg/h

Equation number	Equation
2.12.4	**Steam condensing rate for horizontal pipes in still air** $\dot{m}_s = \frac{3.6\,\dot{Q}\,L}{h_{fg}}$ kg/h Where: $\dot{m}_s$ = Rate of steam condensation (kg/h) $\dot{Q}$ = Heat emission from pipe (W/m) L = Effective length of pipes (m) h_{fg} = Enthalpy of evaporation at the working pressure (kJ/kg) **Note:** The constant 3.6 gives the answer in kg/h
2.13.1	**Mean steam flowrate to a storage calorifier** $\dot{m}_s = \frac{m\,c_p\,\Delta T}{h_{fg}\,t}$ Where: $\dot{m}_s$ = Mean rate of condensation (kg/h) m = Mass of water heated (kg) c_p = Specific heat of water (kJ/kg°C) ΔT = Change in temperature of water (°C) h_{fg} = Enthalpy of evaporation of steam (kJ/kg) t = Recovery time to heat the water (hours)
2.14.1	**Steam consumption of drying cylinders** $\dot{m}_s = \frac{1.5\left([W_w - W_d]\,2550 + 1.26\,W_d\,[T_2 - T_1]\right)}{h_{fg}}$ Where: $\dot{m}_s$ = Mass flowrate of steam (kg/h) W_w = Throughput of wet material (kg/h) W_d = Throughput of dry material (kg/h) T_2 = Temperature of material leaving the machine (°C) T_1 = Temperature of material entering the machine (°C) h_{fg} = Enthalpy of evaporation of steam in cylinders (kJ/kg)
2.16.1	**The kinetic energy in steam** Kinetic energy (E) $= \frac{m\,u^2}{2\,g\,J}$ Where: E = Kinetic energy (kJ) m = Mass of the fluid (kg) u = Velocity of the fluid (m/s) g = Acceleration due to gravity (9.806 65 m/s²) J = Joule's mechanical equivalent of heat (101.972 m kg/kJ)

Equation number	Equation
2.16.2	**Velocity of steam passing through an orifice in terms of kinetic energy** $u^2 = \frac{2 E g J}{m}$ Where: u = Velocity of the fluid (m/s) E = Kinetic energy (kJ) g = Acceleration due to gravity (9.806 65 m/s²) J = Joule's mechanical equivalent of heat (101.972 m kg/kJ) m = Mass of the fluid (kg)
2.16.3	**Velocity of steam passing through an orifice in terms of heat drop** $u = 44.72 \sqrt{h}$ Where: u = Veolocity of the fluid (m/s) h = Heat drop per unit mass (kJ/kg)
2.16.4	**Mass flow of steam through an orifice** $\text{The mass flow} = \frac{\text{Velocity m/s x Orifice area m}^2}{\text{Specific volume m}^3/\text{kg}} \text{ kg/s}$
2.16.5	**Velocity of steam passing through an orifice in terms of heat drop** $u = \sqrt{2 h}$ Where: u = Velocity of the fluid in m/s h = Heat drop in J/kg 2 = Constant of proportionality incorporating the gravitational constant 'g'.

Block 3: The boiler house

Equation number	Equation
3.2.1	**Stress in a boiler shell resulting from boiler pressure** $\sigma = \frac{P\,D}{2\,\varkappa}$ Where: σ = Hoop stress (N/m²) P = Boiler pressure (N/m² = bar x 10^5) D = Diameter of cylinder (m) $\varkappa$ = Plate thickness (m)
3.2.2	**Relating boiler pressure to heat transfer rate** $P\ \alpha\ \frac{1}{\dot{Q}}$ Where: P = Boiler pressure (N/m² = bar x 10^5) $\dot{Q}$ = Heat transfer rate (kW)
3.5.1	**To determine the evaporation factor of a boiler from its 'From & At' rating** $\text{Evaporation factor} = \frac{A}{B - C}$ Where: A = Specific enthalpy of evaporation at atmospheric pressure. B = Specific enthalpy of steam at operating pressure. C = Specific enthalpy of water at feedwater temperature.
3.5.2	**To determine the actual evaporation rate of a boiler from its kW rating and the energy required to be added to the feedwater to make steam** $\dot{m} = \dot{Q} \times \frac{3\,600\ \text{s/h}}{\text{Energy to be added (kJ/kg)}}$ Where: $\dot{m}$ = Steam output (kg/h) $\dot{Q}$ = Boiler rating (kW)
3.5.3	**To determine boiler horse power from heat transfer area** $\text{Heat transfer area (ft}^2\text{)} \times \frac{1}{17} = \text{BoHP}$ Where: BoHP = Boiler horsepower
3.6.1	**Calculating boiler efficiency** $\text{Boiler efficiency (\%)} = \frac{\text{Heat exported in steam}}{\text{Heat provided by the fuel}} \times \frac{100}{1}$

Equation number	Equation
3.12.1	**To determine the TDS of a sample by the density method** TDS (ppm) = (relative density at 15.5°C - 1) x 1.1 x 10^6
3.12.2	**To determine the TDS of a sample by the conductivity method** TDS (ppm) = (conductivity in µS/cm) x 0.7
3.12.3	**To correct the conductivity of a sample at a temperature from 25°C** $\sigma_T = \sigma_{25} [1 + \alpha (T - 25)]$ Where: σ_T = Conductivity at temperature T (µS/cm) σ_{25} = Conductivity at 25°C (µS/cm) α = Temperature coefficient, per °C (Typically 0.02 / °C or 2%°C) T = Temperature (°C)
3.12.4	**The electrical resistance of a conductivity probe** $R = \frac{K}{\sigma}$ Where: R = Resistance (Ohm) K = Cell constant (cm^{-1}) σ = Conductivity (S/cm)
3.12.5	**To determine the blowdown rate of a boiler** Blowdown rate = $\frac{F\ S}{B - F}$ Where: F = Feedwater TDS (ppm). S = Steam generation rate (kg/h). B = Required boiler water TDS (ppm).
3.16.1	**Ohm's Law** $I = \frac{V}{R}$ Where: I = Current (amperes) V = Voltage (volts) R = Resistance (ohms)
3.16.2	**Capacitance Law** $C = K \frac{A}{D}$ Where: C = Capacitance (farad) K = Dielectric constant (non-dimensional) A = Area (m^2) D = Distance between plates (m)

Equation number	Equation
3.21.1	**Steam injection required to power a steam deaerator** $\dot{m}_s = \dot{m} \frac{(h_2 - h_1)}{(h_g - h_2)}$ Where: $\dot{m}_s$ = Mass of steam to be injected (kg/h) $\dot{m}$ = Maximum boiler output at the initial feedwater temperature (kg/h) h_1 = Enthalpy of water at the initial temperature (kJ/kg) h_2 = Enthalpy of water at the required temperature (kJ/kg) h_g = Enthalpy of steam supplying the control valve (kJ/kg)
3.21.2	**Sizing a control valve for saturated steam** $\dot{m}_s = 12 K_v P_1 \sqrt{1 - 5.67 (0.42 - x)^2}$ Where: $\dot{m}_s$ = Steam mass flowrate (kg/h) K_v = Valve coefficient required P_1 = Pressure upstream of the control valve (bar a) P_2 = Pressure downstream of the control valve (bar a) x = Pressure drop ratio $\left(\frac{P_1 - P_2}{P_1}\right)$
3.21.3	**Sizing a control valve for liquid** $\dot{V} = K_v \sqrt{\frac{\Delta P}{G}}$ Where: $\dot{V}$ = Volumetric flowrate (m^3/h) K_v = Valve coefficient required ΔP = Pressure drop across the valve (bar) G = Relative density of fluid (water = 1)
3.22.1	**Steam storage capacity of an accumulator** Steam storage capacity (kg) = $\frac{\text{Difference in enthalpy of water (kJ/kg) x mass of water (kg)}}{\text{Enthalpy of evaporation at the lower pressure (kJ/kg)}}$

Block 4: Flowmetering

Equation number	Equation
4.1.1	**To determine the Absolute or Dynamic viscosity of a fluid by dropping a sphere through a fluid** $\text{Dynamic viscosity}\,(\mu) = \frac{2\,\Delta\rho\,g\,r^2}{9\,u}$ Where: μ = Absolute (or dynamic) viscosity (Pa s) $\Delta\rho$ = Difference in density between the sphere and the liquid (kg/m^3) g = Acceleration due to gravity (9.81 m/s^2) r = Radius of sphere (m) $u = \text{Velocity}\left(\frac{\text{d - Distance sphere falls (m)}}{\text{t - Time taken to fall (seconds)}}\right)$ (m/s)
4.1.2	**To determine the Kinematic viscosity of a fluid** $\text{Kinematic viscosity}\,(\nu) = \frac{\text{Dynamic viscosity}\,(\mu) \times 10^3}{\text{Density}\,(\rho)}$ Where: ν = Kinematic viscosity (centistokes) μ = Dynamic viscosity (Pa s) ρ = Density (kg/m^3)
4.1.3	**To determine the Reynolds number of a fluid in a circular pipe** $\text{Reynolds number}\,(R_e) = \frac{\rho\,u\,D}{\mu}$ Where: R_e = Reynolds number (dimensionless) ρ = Density (kg/m^3) u = Mean velocity in the pipe (m/s) D = Internal pipe diameter (m) μ = Dynamic viscosity (Pa s)
4.1.4	**To determine volumetric flowrate from velocity** $q_v = A\,u$ Where: q_v = Volume flow (m^3/s) A = Cross sectional area of the pipe (m^2) u = Velocity (m/s)
4.1.5	**To determine mass flowrate from volumetric flowrate** $q_m = \frac{q_v}{v_g}$ Where: q_m = Mass flow (kg/s) q_v = Volume flow (m^3/s) v_g = Specific volume (m^3/kg)

Equation number	Equation
4.1.6	**To determine mass flowrate from velocity** $q_m = \frac{A\,u}{v_g}$ Where: q_m = Mass flow (kg/s) A = Cross sectional area of the pipe (m^2) u = Velocity (m/s) v_g = Specific volume (m^3/kg)
4.2.1	**To determine the turndown ratio of a steam flowmeter** $\text{Turndown} = \frac{\text{Maximum flow}}{\text{Minimum flow}}$
4.2.2	**Bernoulli's Equation for a liquid** $\frac{P_1}{\rho g} + \frac{u_1^2}{2g} + h_1 = \frac{P_2}{\rho g} + \frac{u_2^2}{2g} + h_2$ Where: P_1 and P_2 = Pressure at points within a system (Pa) u_1 and u_2 = Velocities at corresponding points within a system (m/s) h_1 and h_2 = Relative vertical heights within a system (m) ρ = Density (kg/m^3) g = Gravitational constant (9.81 m/s^2)
4.2.3	**Bernoulli's Equation multiplied throughout by 'ρg'** $P_1 + \rho g h_1 + \frac{1}{2}\rho u_1^2 = P_2 + \rho g h_2 + \frac{1}{2}\rho u_2^2$ Where: P_1 and P_2 = Pressure at points within a system (Pa) u_1 and u_2 = Velocities at corresponding points within a system (m/s) h_1 and h_2 = Relative vertical heights within a system (m) ρ = Density (kg/m^3) g = Gravitational constant (9.81 m/s^2)
4.2.4	**Bernoulli's Equation with constant potential energy terms** $P_1 + \frac{1}{2}\rho u_1^2 = P_2 + \frac{1}{2}\rho u_2^2$ Where: P_1 and P_2 = Pressure at points within a system (Pa) u_1 and u_2 = Velocities at corresponding points within a system (m/s) ρ = Density (kg/m^3)
4.2.5	**Bernoulli's Equation with constant potential energy terms and frictional losses** $P_1 + \frac{1}{2}\rho u_1^2 = P_2 + \frac{1}{2}\rho u_2^2 + h_f$ Where: P_1 and P_2 = Pressure at points within a system (Pa) u_1 and u_2 = Velocities at corresponding points within a system (m/s) ρ = Density (kg/m^3) h_f = Friction loss (Pa)

Equation number	Equation
4.2.6	**The pressure drop for a liquid equals the friction loss** $P_1 - P_2 = h_f$ Where: P_1 = Upstream pressure (Pa) P_2 = Downstream pressure (Pa) h_f = Friction loss (Pa)
4.2.7	**Potential energy** $\text{Potential energy} = m\ g\ h$ Where: m = Mass of all the molecules directly between and including molecule 1 and molecule 2. g = Gravitational constant (9.81 m/s²) h = Cumulative height of molecules above the hole
4.2.8	**Kinetic energy** $\text{Kinetic energy} = \frac{1}{2} m u^2$ Where: m = Mass of the object (kg) u = Velocity of the object at any point (m/s)
4.2.9	**Potential energy at the start of process = Kinetic energy at the end of process** $m\ g\ h = \frac{1}{2} m u^2$ Where: m = Mass of the object (kg) g = Gravitational constant (9.81 m/s²) h = Height of the object above a reference point (m)
4.2.10	**Velocity of liquid through an orifice** $u = \sqrt{2\ g\ h}$ Where: u = Velocity (m/s) g = Gravitational constant (9.81 m/s²) h = Pressure head (m)
4.2.11	**Volumetric flowrate of liquid through an orifice** $q_v = C\ A \sqrt{2\ g\ h}$ Where: q_v = Volumetric flowrate (m³/s) C = Coefficient of discharge (dimensionless) A = Area of orifice (m²) g = Gravitational constant (9.8 m/s²) h = Differential pressure (m)

Equation number	Equation
4.2.12	**Volumetric flowrate of liquid is proportional to the square root of pressure drop** $q_v \propto \sqrt{\Delta p}$ Where: q_v = Volumetric flowrate (m^3/s) Δp = Pressure drop (m)
4.2.13	**The liquid velocity measured by a Pitot tube** $u_1 = \sqrt{\frac{2\,(\Delta P)}{\rho}}$ Where: u_1 = The fluid velocity in the pipe ΔP = Static pressure - dynamic pressure ρ = The fluid density
4.3.1	**To determine the β ratio for an orifice plate** $\beta = \frac{d\ (\text{orifice diameter})}{D\ (\text{pipe diameter})}$ **Note:** Diameters must be in the same unit of measurement
4.3.2	**To determine the vortex shedding frequency around a bluff body** $f \propto \frac{Sr\ u}{d}$ Where: f = Shedding frequency (Hz) Sr = Strouhal number (dimensionless) u = Mean pipe flow velocity (m/s) d = Bluff body diameter (m)
4.3.3	**The volumetric flowrate from the shedding frequency** $q_v = A\,\frac{f}{k}$ Where: q_v = Volumetric flowrate (m^3/s) A = Cross sectional area of the orifice (m^2) f = Shedding frequency (Hz) k = A constant for all fluids for a given design of meter

Equation number	Equation
4.4.1	**Percentage error when using a velocity sensing meter which is not pressure compensated** $\varepsilon = \left[\left(\frac{\text{Specified } \rho}{\text{Actual } \rho}\right) - 1\right] \times 100\%$ Where: ε = Flow error expressed as a percentage of the actual flow Specified ρ = Density of steam at the specified steam line pressure Actual ρ = Density of steam at the actual line pressure
4.4.2	**Percentage error when using a pressure difference meter which is not pressure compensate** $\%\ \text{error}\ (\varepsilon) = \left(\frac{\sqrt{\text{Specified } \rho}}{\sqrt{\text{Actual } \rho}} - 1\right) \times 100$ Where: ε = Percentage flow error Actual ρ = Density of steam at actual pressure (kg/m³) Specified ρ = Density of steam at specified pressure (kg/m³)
4.4.3	**To determine the density of steam with known dryness fraction** $\text{Density of steam } (\rho) = \frac{1}{\nu_g \chi}$ Where: ρ = Density of steam with dryness fraction χ ν_g = Specific volume of dry steam χ = Dryness fraction
4.4.4	**Approximation of relationship between indicated and actual flowrate with a deviation in dryness fraction** $\frac{\text{Indicated mass flowrate}}{\text{Actual flowrate}} = \frac{\sqrt{\text{density at calibrated dryness fraction}}}{\sqrt{\text{density at actual dryness fraction}}}$
4.4.5	**Actual value of superheated steam flowing through a flowmeter calibrated for saturated steam** $\text{Actual value} = \frac{\text{Displayed value}}{\left[1 + \left(\frac{\%\ \text{error}}{100}\right)\right]}$

Block 5: Basic control theory

Equation number	Equation
	There are no equations in Block 5

Block 6: Control hardware: Electric/pneumatic actuation

Equation number	Equation
6.1.1	**Closing force to close a valve** $(A \times \Delta P) + \text{Friction allowance} = F$ Where: A = Valve seating area (m^2) ΔP = Differential pressure (kPa) F = Closing force required (kN)
6.3.1	**Calculate valve K_v for liquids** $K_v = \dot{V}\sqrt{\frac{G}{\Delta P}}$ Where: K_v = Flow of liquid that will create a pressure drop of 1 bar (m^3/ h bar) $\dot{V}$ = Flowrate (m^3/h) G = Relative density/specific gravity of the liquid (dimensionless). ΔP = Pressure drop across the valve (bar)
6.3.2	**Volumetric flow of water through a valve** $\dot{V} = K_v\sqrt{\Delta P}$ Where: $\dot{V}$ = Flowrate (m^3/h) K_v = Flow of liquid to create a pressure drop of 1 bar (m^3/h bar) ΔP = Pressure drop across a valve (bar)
6.3.3	**The flow of liquid through a constant bore pipe relative to pressure loss** $\frac{\dot{V}_1^2}{\dot{V}_2^2} = \frac{P_1}{P_2}$ Where: $\dot{V}_1$ = Flowrate at pressure loss P_1 $\dot{V}_2$ = Flowrate at pressure loss P_2
6.3.4	**Valve authority** $N = \frac{\Delta P_1}{\Delta P_1 + \Delta P_2}$ Where: N = Valve authority ΔP_1 = Pressure drop across a fully open control valve ΔP_2 = Pressure drop across the remainder of the circuit $\Delta P_1 + \Delta P_2$ = Pressure drop across the whole circuit

Equation number	Equation
6.4.1	**Critical pressure ratio for dry steam and gases** $\text{Critical pressure ratio} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}}$ Where: γ = Isentropic exponent of the steam or gas
6.4.2	**Speed of sound in steam** $C = 31.6 \sqrt{\gamma R T}$ Where: C = Speed of sound in steam (m/s) 31.6 = Constant of proportionality γ = Steam isentropic exponent (1.135 : saturated, 1.3 : superheated) R = 0.4615 the gas constant for steam (kJ/kg) T = Absolute steam temperature (K)
6.4.3	**Steam flow through a valve under critical flow conditions** $\dot{m}_s = 12 K_v P_1$ Where: $\dot{m}_s$ = Mass flow through a valve (kg/h) K_v = Valve capacity (m³/h bar) P_1 = Upstream pressure (bar a)
6.5.1	**Volumetric flow through an equal pecentage valve** $\dot{V} = \frac{e^x}{\tau} \dot{V}_{max}$ Where: $\dot{V}$ = Volumetric flow through the valve at lift H. x = (ln τ) H Note: 'In' is a mathematical function known as 'natural logarithm'. τ = Valve rangeability (ratio of the maximum to minimum controllable flowrate, typically 50 for a globe type control valve) H = Valve lift (0 = closed, 1 = fully open) $\dot{V}_{max}$ = Maximum volumetric flow through the valve
6.5.2	**Required capacity of a water control valve** $K_{vr} = \frac{\dot{V}}{\sqrt{\Delta P}}$ Where: K_{vr} = The actual valve capacity required by the installation (m³/h bar) $\dot{V}$ = Flowrate through the valve (m³/h) ΔP = The differential pressure across the valve (bar)

Equation number	Equation
6.5.3	**Percentage lift of an equal percentage valve in terms of relative flow** $H\% = \frac{\ln\left[\frac{\dot{V}\tau}{\dot{V}_{max}}\right]}{\ln\tau} \times 100$ Where: H% = Percentage lift In = Natural logarithm $\dot{V}$ = Flow through the valve at lift 'H' (m³/h) τ = Valve rangeability $\dot{V}_{max}$ = Maximum flow through the valve at full lift (m³/h)
6.5.4	**Percentage lift of an equal percentage valve in terms of relative K_V** $H\% = \frac{\ln\left[\frac{K_{vr}\,\tau}{K_{vs}}\right]}{\ln\tau} \times 100$ Where: H% = Percentage lift In = Natural logarithm K_{vr} = Required capacity at lift 'H' (m³/h bar) τ = Valve rangeability K_{vs} = Valve capacity full open (m³/h bar)
6.5.5	**The required capacity for a steam valve under sub-sonic flow** $K_{vr} = \frac{\dot{m}_s}{12\,P_1\sqrt{1 - 5.67\,(0.42 - x)^2}}$ Where: K_{vr} = Required capacity at lift 'H' (m³/h bar) $\dot{m}_s$ = Steam mass flowrate (kg/h) P_1 = Upstream pressure (bar a) P_2 = Downstream pressure (bar a) x = $(P_1 - P_2)/P_1$

Block 7: Control hardware: Self-acting actuation

Equation number	Equation
7.1.1	**Stem force required to close a control valve** Force on valve stem (newton) = $\frac{\pi\,d^2}{40} \times \Delta P$ Where: d = Diameter of valve orifice (mm) ΔP = Differential pressure (bar)

Block 8: Control applications

Equation number	Equation
	There are no equations in Block 8

Block 9: Safety valves

Equation number	Equation
9.1.1	**Flow area of a safety valve** $\text{Flow area} = \frac{\pi d^2}{4}$ Where: d = The area of the inlet port at its narrowest point
9.1.2	**Curtain area of a safety valve** $\text{Curtain area} = \pi d_1 L$ Where: d_1 = Minimum area of opening between the valve and seat L = Maximum lift from seat to valve
9.2.1	**Required opening force for a safety valve with the spring housing vented via the discharge vent pipe** $P_V A_N = F_S + P_B A_N$ Where: P_V = Fluid inlet pressure A_N = Nozzle area F_S = Spring force P_B = Backpressure
9.2.2	**Required opening force for a safety valve with the spring housing vented to atmosphere** $P_V A_N = F_S - P_B (A_D - A_N)$ Where: P_V = Fluid inlet pressure A_N = Nozzle area F_S = Spring force P_B = Backpressure A_D = Disc area
9.2.3	**Required opening force for a safety valve with the spring housing vented via the discharge vent pipe and taking into effect the build-up backpressure** $P_S A_N = F_S + A_N (P_B - P_O)$ Where: P_S = Set pressure of safety valves A_N = Nozzle area F_S = Spring force P_B = Backpressure P_O = Overpressure

Equation number	Equation
9.2.4	**Required opening force for a balanced safety valve** $P_V A_N = F_S$ Where: P_V = Fluid inlet pressure A_N = Nozzle area F_S = Spring force
9.3.1	**Cold differential pressure** $CDSP = RISP - CBP$ Where: CDSP = Cold differential set pressure RISP = Required installed set pressure CBP = Constant backpressure
9.4.1	**Coefficient of discharge** $K_d = \frac{\text{Actual flowing capacity}}{\text{Theoretical flowing capacity}}$ Where: K_d = Coefficient of discharge
9.4.2	**Critical pressure ratio** $\frac{P_B}{P_1} = \left(\frac{2}{k+1}\right)^{\left(\frac{k}{k-1}\right)}$ Where: P_B = Critical backpressure (bar a) P_1 = Actual relieving pressure (bar a) k = Isentropic coefficient of the gas or vapour at the relieving conditions
9.4.3	**AD-Merkblatt valves - Minimum flow area for steam** $A_O = \frac{\chi \dot{m}}{\alpha_w P_R}$ Where: A_O = Minimum cross sectional flow area (mm²) χ = Pressure medium coefficient $\dot{m}$ = Mass flow to be discharged (kg/h) α_W = Outflow coefficient P_R = Absolute relieving pressure (bar a)
9.4.4	**AD-Merkblatt valves - Minimum flow area for dry gases and air** $A_O = \frac{0.1791 \dot{m}}{\Psi \alpha_w P_R} \sqrt{\frac{T Z}{M}}$ Where: A_O = Minimum cross sectional flow area (mm²) $\dot{m}$ = Mass flow to be discharged (kg/h) Ψ = Outflow function α_W = Outflow coefficient P_R = Absolute relieving pressure (bar a) T = Inlet temperature (K) M = Molar mass (kg/kmol) Z = Compressibility factor

Equation number	Equation
9.4.5	**AD-Merkblatt valves - Minimum flow area for liquids** $A_O = \frac{0.6211\ \dot{m}}{\alpha_w \sqrt{\rho\ \Delta P}}$ Where: A_O = Minimum cross sectional flow area (mm²) $\dot{m}$ = Mass flow to be discharged (kg/h) α_W = Outflow coefficient ρ = Density (kg/m³) ΔP = P_R - P_B P_R = Absolute relieving pressure (bar a) P_B = Absolute backpressure (bar a)
9.4.6	**Compressibility factor for compressible steam and dry gases** $Z = \frac{10^5\ P_R\ M\ \nu}{R_u\ T}$ Where: Z = Compressibility factor P_R = Safety valve relieving pressure (bar a) ν = Specific volume of the gas at the actual relieving pressure and temperature (m³/kg) M = Molar mass (kg/kmol) R_u = Universal gas constant (8 314 Nm/kmol K) T = Actual relieving temperature (K)
9.4.7	**Proportion of vapour in two phase discharge** $n = \frac{h_{f1} - h_{f2}}{h_{fg2}}$ Where: n = The proportion of discharge fluid which is vapour h_{f1} = Enthalpy of liquid before the valve (kJ/kg) h_{f2} = Enthalpy of liquid after the valve (kJ/kg) h_{fg2} = Enthalpy of evaporation after the valve (kJ/kg)
9.4.8	**ASME (API RP 520) valves - Minimum flow area for steam** $A_O = \frac{\dot{m}}{51.5\ P_R\ K_d\ K_{SH}}$ Where: A_O = Required effective discharge area (in²) $\dot{m}$ = Required mass flow through the valve (lb/h) P_R = Upstream relieving pressure (psi a) K_d = Effective coefficient of discharge K_{SH} = Superheat correction factor
9.4.9	**ASME (API RP 520) valves - Minimum flow area for dry gases and air** $A_O = \frac{\dot{V}\ \sqrt{T\ Z\ G}}{1.175\ C_g\ K_d\ P_R\ K_B}$ Where: A_O = Required effective discharge area (in²) $\dot{V}$ = Required volume flow through the valve (ft³/min) T = Relieving temperature (°R) Z = Compressibility factor G = Specific gravity of the air or gas C_g = Nozzle gas constant K_d = Effective coefficient of discharge P_R = Upstream relieving pressure (bar a) K_B = Backpressure correction factor for gas and vapour

Equation number	Equation
9.4.10	**ASME (API RP 520) valves - Minimum flow area for liquids** $A_O = \frac{\dot{V}^1}{38\ K_d\ K_\mu\ K_w}\ \sqrt{\frac{G}{P_R - P_B}}$ Where: A_O = Required effective discharge area (in²) $\dot{V}^1$ = Required volume flow through the valve (U.S. gal/min) K_d = Effective coefficient of discharge (specified by the manufacturer) K_μ = Viscosity factor K_W = Backpressure correction factor for liquids (bellows balanced valves only) G = Specific gravity (ratio of molar mass of the fluid to the molar mass of air P_R = Upstream relieving pressure (psi a) P_B = Absolute backpressure (psi a)
9.4.11	**ASME (API RP 520) valves - Nozzle gas constant** $C_g = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\left(\frac{k+1}{k-1}\right)}}$ for $k > 1$ $C_g = 315$ for $k = 1$ Where: C_g = Nozzle gas constant k = Isentropic coefficient of the gas or vapour at the relieving conditions
9.4.12	**ASME (API RP 520) valves - Backpressure correction factor** $K_B = \frac{C_1}{C_2}$ Where: K_B = Backpressure correction factor C_1 = Capacity of valve with backpressure applied C_2 = Capacity of valve when discharging to atmosphere
9.4.13	**ASME (API RP 520) valves - Bellows balanced valves** $\%\ \text{of gauge backpressure} = \frac{P_B}{P_S} \times 100$ Where: P_B = Backpressure (psi g) P_S = Set pressure (psi g)
9.4.14	**ASME (API RP 520) valves - Conventional valves** $\%\ \text{of gauge backpressure} = \frac{P_B}{P_R} \times 100$ Where: P_B = Backpressure (psi g) P_R = Relieving pressure (psi g)
9.4.15	**ASME (API RP 520) valves - Reynolds number: Metric units** $R_e = 0.3414\ \frac{\dot{m}}{\mu\sqrt{A_O}}$ Where: R_e = Reynolds number $\dot{m}$ = Mass flow to be discharged (kg/h) μ = Dynamic viscosity (Imperial – cP, Metric – Pa s) A_O = Discharge area (Imperial – in², Metric – mm²)

Equation number	Equation
9.4.16	**ASME (API RP 520) valves - Reynolds number: Imperial units** $R_e = \frac{2800\, G\, \dot{V}}{\mu \sqrt{A_O}}$ Where: R_e = Reynolds number $\dot{V}$ = Volume flow to be discharged (U.S. gal/min) $\dot{m}$ = Mass flow to be discharged (kg/h) μ = Dynamic viscosity (Imperial – cP, Metric – Pa s) A_O = Discharge area (Imperial – in², Metric – mm²)
9.4.17	**BS 6759 valves - Minimum orifice area for steam** $A_O = \frac{\dot{m}}{0.525\, P_R\, K_{dr}\, K_{SH}}$ Where: A_O = Flow area (mm²) $\dot{m}$ = Mass flow to be discharged (kg/h) P_R = Absolute relieving pressure (bar a) K_{dr} = Derated coefficient of discharge K_{SH} = Superheat correction factor
9.4.18	**BS 6759 valves - Minimum orifice area for air** $A_O = \frac{\dot{V}}{0.193\, P_R\, K_{dr}} \sqrt{\frac{T}{288}}$ Where: A_O = Flow area (mm²) $\dot{V}$ = Volumetric flow to be discharged (l/s) P_R = Absolute relieving pressure (bar a) K_{dr} = Derated coefficient of discharge T = Inlet temperature (K)
9.4.19	**BS 6759 valves - Minimum orifice area for dry gases** $A_O = \frac{\dot{m}}{P_R\, C_g\, K_{dr}} \sqrt{\frac{Z\, T}{M}}$ Where: A_O = Flow area (mm²) $\dot{m}$ = Mass flow to be discharged (kg/h) P_R = Absolute relieving pressure (bar a) C_g = Nozzle gas constant K_{dr} = Derated coefficient of discharge Z = Compressibility factor T = Inlet temperature (K) M = Molar mass (molecular weight) (kg/kmol)

Equation number	Equation
9.4.20	**BS 6759 valves - Minimum orifice area for liquids** $A_O = \frac{\dot{m}}{1.61\, K_{dr}\, K_\mu \sqrt{\rho\ \Delta P}}$ Where: A_O = Flow area (mm²) $\dot{m}$ = Mass flow to be discharged (kg/h) K_{dr} = Derated coefficient of discharge K_μ = Viscosity correction factor ρ = Density (kg/m³) $\Delta P = P_R - P_B$ P_R = Absolute relieving pressure (bar a) P_B = Absolute backpressure (bar a)
9.4.21	**BS 6759 valves - Minimum orifice area for hot water** $A_O = \frac{\dot{Q}}{0.329\, P_R\, K_{dr}}$ Where: A_O = Flow area (mm²) $\dot{Q}$ = Hot water heating capacity (kW) P_R = Absolute relieving pressure (bar a) K_{dr} = Derated coefficient of discharge
9.4.22	**BS 6759 valves - Nozzle gas constant** $C_g = 3.948 \sqrt{k \left(\frac{2}{k+1}\right)^{\left(\frac{k+1}{k-1}\right)}}$ Where: k = Isentropic coefficient of gas or vapour
9.4.23	**EN ISO 4126 valves - Minimum orifice area for steam, air and dry gas at critical flow** $A = \frac{\dot{m}}{0.288\,3\, C\, K_{dr} \sqrt{\frac{P_o}{\nu_g}}}$ Where: A = Flow area (not curtain area) (mm²) $\dot{m}$ = Mass flowrate (kg/h) C = Function of the isentropic exponent K_{dr} = Certified derated coefficient of discharge P_o = Relieving pressure (bar a) ν_g = Specific volume at relieving pressure and temperature (m³/kg)
9.4.24	**EN ISO 4126 valves - Minimum orifice area for wet steam at critical flow** $A = \frac{\dot{m}}{0.288\,3\, C\, K_{dr} \sqrt{\frac{P_o}{\nu_g\, x}}}$ Where: A = Flow area (not curtain area) (mm²) $\dot{m}$ = Mass flowrate (kg/h) C = Function of the isentropic exponent K_{dr} = Certified derated coefficient of discharge P_o = Relieving pressure (bar a) ν_g = Specific volume at relieving pressure and temperature (m³/kg) x = Dryness fraction of wet steam

Equation number	Equation
9.4.25	**EN ISO 4126 valves - Minimum orifice area for air and dry gas at sub-critical flow** $A = \frac{\dot{m}}{0.2883\, C\, K_{dr} \sqrt{\frac{P_o}{\nu_g}}}$ Where: A = Flow area (not curtain area) (mm^2) $\dot{m}$ = Mass flowrate (kg/h) C = Function of the isentropic exponent K_{dr} = Certified derated coefficient of discharge K_b = Theoretical correction factor for sub-critical flow P_o = Relieving pressure (bar a) ν_g = Specific volume at relieving pressure and temperature (m^3/kg)
9.4.26	**EN ISO 4126 valves - Minimum orifice area for liquids** $A = \frac{\dot{m}}{1.61\, K_{dr}\, K_v \sqrt{\frac{P_o - P_b}{\nu_g}}}$ Where: A = Flow area (not curtain area) (mm^2) $\dot{m}$ = Mass flowrate (kg/h) K_{dr} = Certified derated coefficient of discharge K_v = Viscosity correction factor P_o = Relieving pressure (bar a) P_b = Backpressure (bar a) ν_g = Specific volume at relieving pressure and temperature (m^3/kg)
9.5.1	**Safety valve vent pipe diameter** $d = \sqrt[5]{\frac{L_e\, \dot{m}^2 \left(\frac{\nu_g + 2}{2}\right)}{0.08\, P}}$ Where: d = Pipe diameter (mm) L_e = Equivalent length of pipe (m) $\dot{m}$ = Discharge capacity (kg/h) P = Safety valve set pressure (bar g) x 0.1 v_g = Specific volume of steam at the pressure (P) (m^3/kg)
9.5.2	**Reaction force at the end of a safety valve vent pipe** $F = 129\, \dot{m} \sqrt{\frac{k\, T}{(k+1)\, M}} + 0.1\, A\, P$ Where: F = Reaction force at the point of discharge to atmosphere (newtons) $\dot{m}$ = Discharge mass flowrate (kg/s) k = Isentropic coefficient of the fluid T = Fluid temperature (K) M = Molar mass of the fluid (kg/kmol) A = Area of the outlet at the point of discharge (mm^2) P = Static pressure at the outlet at the point of discharge (bar g)

Equation number	Equation
9.5.3	**Sound power level at the safety valve outlet** $L_P = 60 \log_{10} u + 10 \log_{10} \dot{m} - 52$ Where: L_P = Sound power level in dB (A) $\dot{m}$ = Mass flow (kg/h) u = Speed of sound in an ideal gas (m/s), $u = \sqrt{\frac{k R_u T}{M}}$ k = Isentropic coefficient of the gas R_u = Universal gas constant (8 314 J/kmol K) T = Absolute gas temperature at the safety valve outlet (K) M = Molar mass (kg/kmol)
9.5.4	**Sound pressure level at the safety valve outlet** $L = L_P - 10 \log_{10} (2 \pi R^2)$ Where: L = Sound pressure level in dB (A) L_P = Sound power level in dB (A) R = Distance from the source (m)
9.5.5	**Sound power level from EN ISO 4126** $L_P = 20 \log (10^{-3} D_A) - 10 \log v_g + 80 \log u - 53$ Where: L_P = Sound power level in dB (A) D_A = Discharge pipe bore (mm) v_g = Specific volume at relieving pressure and temperature (m^3/kg) u = Velocity of fluid in the outlet pipe (m/s)

Block 10: Steam distribution

Equation number	Equation
10.2.1	**The 'SI based' D'Arcy equation for determining pressure drop due to frictional resistance** $h_f = \frac{4 f L u^2}{2 g D}$ Where: h_f = Head loss to friction (m) f = Friction factor (dimensionless) L = Length (m) u = Flow velocity (m/s) g = Gravitational constant (9.81 m/s^2) D = Pipe diameter (m)

Equation number	Equation
10.2.2	**The 'Imperial based' D'Arcy equation for determining pressure drop due to frictional resistance** $h_f = \frac{f L u^2}{2 g D}$ Where: h_f = Head loss to friction (m) f = Friction factor (dimensionless) L = Length (m) u = Flow velocity (m/s) g = Gravitational constant (9.81 m/s²) D = Pipe diameter (m)
10.2.3	**Friction factor for turbulent fluids (Colebrook-White formula)** $\frac{1}{\sqrt{f}} = 3.48 - 1.74 \ln\left(\frac{2 k_s}{D} + \frac{9.3}{R_e \sqrt{f}}\right)$ Where: f = Friction factor (Relates to the SI Moody chart) k_s = Absolute pipe roughness (m) D = Pipe bore (m) R_e = Reynolds number (dimensionless)
10.2.4	**'SI based' friction factors - f** $f \approx \frac{0.08}{\sqrt[4]{R_e}}$ Where: f = Friction factor R_e = Reynolds number
10.2.5	**'Imperial based' friction factors - f** $f \approx \frac{0.03}{\sqrt[4]{R_e}}$ Where: f = Friction factor R_e = Reynolds number
10.2.6	**Reynolds number** $R_e = \frac{\rho u D}{\mu}$ Where: R_e = Reynolds number ρ = Density of water (kg/m³) u = Velocity of water (m/s) D = Pipe diameter (m) μ = Dynamic viscosity of water (kg/m s)
10.2.7	**Relative pipe roughness** Relative pipe roughness $\left(\frac{k_s}{D}\right)$ = Pipe roughness / Pipe bore Where: k_s = Pipe roughness (m) D = Pipe bore (m)

Equation number	Equation
10.2.8	**Pressure factor** $F = \frac{P_1 - P_2}{L}$ Where: F = Pressure factor P_1 = Factor based on the inlet pressure P_2 = Factor based on the pressure at a distance of L metres of pipe L = Equivalent length of pipe (m)
10.2.9	**Pressure drop formula 1** $\frac{(P_1)^{1.9375} - (P_2)^{1.9375}}{L} = \frac{\dot{m}^{1.853}}{0.011 D^{4.987}}$ Where: P_1 = Upstream pressure (bar a) P_2 = Downstream pressure (bar a) L = Length of pipe (m) $\dot{m}$ = Mass flowrate (kg/h) D = Pipe diameter (mm)
10.2.10	**Pressure drop formula 2 (Maximum pipe length: 200 metres)** $\Delta P = \frac{L v_g \dot{m}^2}{0.08 D^5}$ Where: ΔP = Pressure drop (bar) L = Length of pipe (m) v_g = Specific volume of steam (m^3/kg) $\dot{m}$ = Mass flowrate (kg/h) D = Pipe diameter (mm)
10.4.1	**Thermal expansion of pipe** Expansion (mm) = $L \Delta T \alpha$ Where: L = Length of pipe between anchors (m) ΔT = Temperature difference between ambient temperature and operating temperatures (°C) α = Expansion coefficient (mm/m °C x 10^{-3})

Block 11: Steam trapping

Equation number	Equation
	There are no equations in Block 11

Block 12: Pipeline ancillaries

Equation number	Equation
12.2.1	**Pressure drop across a valve in a liquid system** $\Delta P = G\left(\frac{\dot{V}}{K_v}\right)^2$ Where: ΔP = Pressure drop across the valve (bar) G = Specific gravity of the liquid (non-dimensional) $\dot{V}$ = Flowrate of liquid (m^3/h) K_v = Valve flow coefficient (m^3/h bar)
12.3.1	**Equivalent water flowrate through a check valve** $\dot{V}w = \sqrt{\frac{\rho}{1000}}\,\dot{V}$ Where: $\dot{V}w$ = Equivalent water volume flowrate (m^3/h) ρ = Density of the liquid (kg/m^3) $\dot{V}$ = Volume flowrate of liquid (m^3/h)
12.3.2	**Converting water mass flow to volumetric flow** $\dot{V} = \dot{m}\ v_g$ Where: $\dot{V}$ = Volume flowrate (m^3/h) $\dot{m}$ = Mass flowrate (kg/h) v_g = Specific volume (m^3/kg)
12.4.1	**Largest particle size through a strainer screen** $c^2 = a^2 + b^2$ Where: a = Mesh hole length b = Mesh hole width c = Particle size
12.4.2	**Pressure drop across a steam valve** $\Delta P = P_1\left[0.42 - \sqrt{\frac{1}{5.67}\left(1-\left(\frac{\dot{m}_s}{12\,K_v\,P_1}\right)^2\right)}\right]$ Where: ΔP = Pressure drop across the valve (bar) $\dot{m}_s$ = Steam flowrate (kg/h) K_v = Valve flow coefficient (m^3/h bar) P_1 = Upstream pressure (bar a)

Block 13: Condensate removal

Equation number	Equation
13.2.1	**Calculating the heating area of a heat exchanger** $A = \frac{\dot{Q}}{U \Delta T_M}$ Where: A = Area of heating surface (m^2) $\dot{Q}$ = Mean heat transfer rate (W) U = Heat transfer coefficient ($W/m^2\,°C$) ΔT_M = Mean temperature difference.
13.2.2	**The heat exchanger temperature design constant** $TDC = \frac{T_s - T_1}{T_s - T_2}$ Where: TDC = Temperature design constant of the heat exchanger T_s = Steam temperature (°C) T_1 = Secondary fluid inlet temperature (°C) T_2 = Secondary fluid outlet temperature (°C)
13.2.3	**The steam temperature at any load** $T_s = \frac{(T_2 \times TDC) - T_1}{TDC - 1}$ Where: TDC = Temperature design constant of the heat exchanger T_s = Steam temperature (°C) T_1 = Secondary fluid inlet temperature (°C) T_2 = Secondary fluid outlet temperature (°C)
13.2.4	**The secondary fluid inlet temperature at any load** $T_1 = T_s - [\, TDC\,(T_s - T_2)\,]$ Where: TDC = Temperature design constant of the heat exchanger T_s = Steam temperature (°C) T_1 = Secondary fluid inlet temperature (°C) T_2 = Secondary fluid outlet temperature (°C)
13.2.5	**The secondary fluid outlet temperature at any load** $T_2 = T_S - \left[\frac{T_S - T_1}{TDC}\right]$ Where: TDC = Temperature design constant of the heat exchanger T_s = Steam temperature (°C) T_1 = Secondary fluid inlet temperature (°C) T_2 = Secondary fluid outlet temperature (°C)

Equation number	Equation
13.3.1	**Mean differential temperature between the primary and secondary fluids** $\Delta T_M = \frac{\dot{Q}}{U A}$ Where: ΔT_M = Mean temperature difference. Note: ΔT_M may be either ΔT_{LM} (LMTD) or ΔT_{AM} (AMTD) $\dot{Q}$ = Mean heat transfer rate (W) U = Heat transfer coefficient (W/m² °C) A = Heating area (m²)
13.4.1	**The secondary inlet temperature at any load** $T_X = [(T_2 - T_1)\ (1- x)] + T_1$ Where: T_x = The secondary inlet temperature at any load factor 'x' (°C) T_1 = The secondary inlet temperature at full-load (°C) T_2 = The secondary outlet temperature at full-load (°C) x = The load factor
13.5.1	**The stall load for a constant flow secondary** $\% \text{ Stall load} = \frac{D - B}{A - B} \times 100$ Where: A = The steam temperature in the steam space at full-load (°C) B = The secondary fluid outlet temperature (°C) D = The backpressure equivalent saturated steam temperature (°C)
13.6.1	**Calculating the stall load with a variable flow secondary** $\% \text{ Stall load} = \frac{D - \left(\frac{B + C}{2}\right)}{A - \left(\frac{B + C}{2}\right)} \times 100$ Where: A = Steam temperature at full-load (°C) B = Secondary fluid outlet temperature at full-load (°C) C = Secondary fluid inlet temperature at full-load (°C) D = Equivalent backpressure steam temperature (°C)

Block 14: Condensate recovery

Equation number	Equation
14.1.1	**Cost of fuel saved by returning condensate** $\text{Fuel savings/year} = \frac{X\,A\,B\,C\,D}{E\,10^6}$ Where: X = Expected improvement in condensate return expressed as a percentage between 1 and 100 A = Cost of fuel to provide 1 GJ of energy B = Energy required per kilogram of make-up water to reach condensate temperature (kJ/kg). C = Average boiler evaporation rate (kg/h) D = Operational hours per year (h/year) E = Boiler efficiency (%)
14.1.2	**Cost of water saved by returning condensate** $\text{Savings in water costs/year} = \left(\frac{X\,C\,D}{10^5}\right) \times \text{Cost of water/m}^3$
14.1.3	**Cost of effluent saved by returning condensate** $\text{Savings in effluent costs/year} = \left(\frac{X\,C\,D}{10^5}\right) \times \text{Cost of effluent/m}^3$

Equation number	Equation
14.4.1	**Calculating pump delivery head** $\text{Total delivery head } (h_d) = h_s + h_f + h_p$ Where: h_d = Total delivery head h_s = Pressure required to raise the water to the desired level (static head) h_f = Pressure required to move the water through the pipes (friction head) h_p = Pressure in the condensate system
14.4.2	**Calculate condensate velocity in a pipe** $\text{Condensate velocity (m/s)} = \frac{\text{Condensate flowrate (kg/h)} \times \text{Condensate specific volume (m}^3\text{/kg)}}{(\text{pipe bore[mm]})^2 \times 0.002827}$

Block 15: Desuperheating

Equation number	Equation
15.1.1	**Calculate cooling water flowrate for a desuperheater** $\dot{m}_{cw} = \frac{\dot{m}_s\,(h_s - h_d)}{(h_d - h_{cw})}$ Where: $\dot{m}_{cw}$ = Mass flowrate of cooling water (kg/h) $\dot{m}_s$ = Mass flowrate of superheated steam (kg/h) h_s = Enthalpy at superheat condition (kJ/kg) h_d = Enthalpy at desuperheated condition (kJ/kg) h_{cw} = Enthalpy of cooling water at inlet connection (kJ/kg)