HANDBOOK
ON
THE STEAM ENGINE

HAEDER & POWLES

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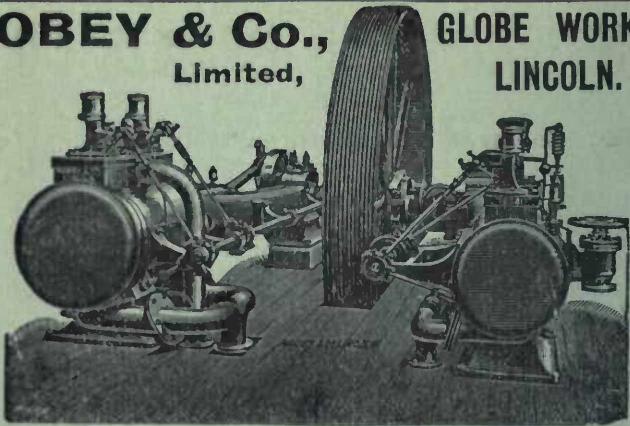

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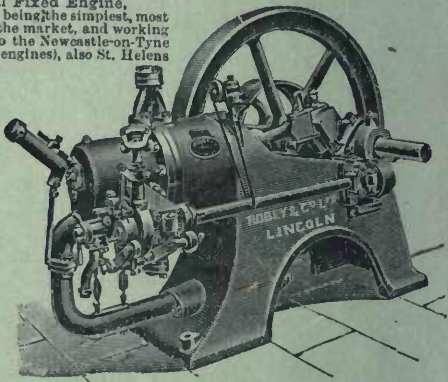

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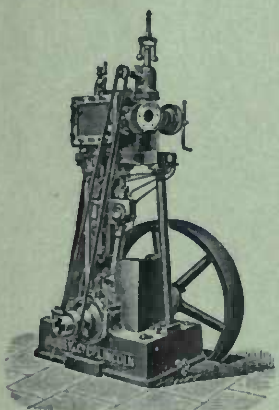

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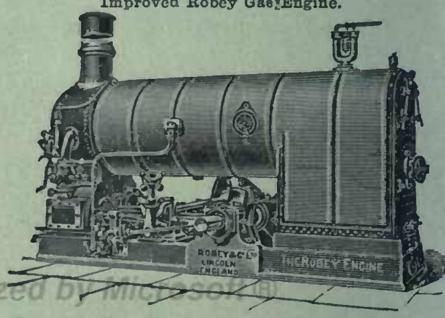

Improved Compound "Robey" Undertype Engine.

A HANDBOOK

ON THE

STEAM ENGINE

WITH ESPECIAL REFERENCE TO

SMALL AND MEDIUM-SIZED ENGINES

FOR THE USE OF ENGINE MAKERS, MECHANICAL DRAUGHTSMEN, ENGINEERING STUDENTS, AND USERS OF STEAM POWER

BY HERMANN HAEDER, CIVIL ENGINEER
DUISBURG-ON-THE RHINE

TRANSLATED FROM THE GERMAN, WITH CONSIDERABLE ADDITIONS AND ALTERATIONS

BY H. H. P. POWLES
ASSOC. MEMB. INST. C.E., M.I.M.E.

Third English Edition, Revised

LONDON
CROSBY LOCKWOOD AND SON
7, STATIONERS'-HALL COURT, LUDGATE HILL
1902

BRADBURY, AGNEW, & CO. LD., PRINTERS,
LONDON AND TONBRIDGE.

AUTHOR'S PREFACE TO FIRST EDITION.

In the present condition of the markets, makers must not occupy too much time in calculations or mathematical research, but collecting results from actual practice, must note them in a convenient form, though not as formulæ, since formulæ are only useful within certain limits, there always being a risk of error in calculation. If anyone compares the results of the formulæ of one writer with those of another, he will often find great discrepancies between them.

The present work aims at showing the results of practical experience. Letterpress has been reduced as much as possible, to allow of the introduction of numerous tables and drawings, these being the language of technical people.

To those firms and others who have supplied me with information, I return my best thanks. May the book and its method gain many friends!

AUTHOR'S PREFACE TO SECOND EDITION.

The first edition of this book had such a favourable reception in technical circles, that within a year a second edition has been found necessary.

Encouraged by this success, I have spared neither trouble nor expense to extend the usefulness of the work, and with that object have introduced a large number of additional figures.

To the numerous technical friends who have favoured me with communications on the subject, I return my grateful thanks, with a request for them to assist me with further exchange of ideas.

TRANSLATOR'S PREFACE.

In preparing this work for an English edition, the plan of the Author has been adhered to as closely as possible, the theoretical part being treated very briefly, and formulæ given only when necessary; whilst numerous tables of dimensions of engine-details, taken from actual examples, and accompanied by illustrations, are appended. These will afford means of comparing the results of Continental with those of English practice.

To the present Edition there have been added examples, with tables and illustrations, of standard types of English Engines by several well-known makers, who have kindly supplied for the purpose dimensions and drawings.

As thus adapted to our own practice, it is hoped that the present edition of the work will be found by many English Engineers, and Mechanical Draughtsmen in particular, to answer a very useful purpose, not as yet fulfilled by any other publication.

To make the tables as clear as possible, the dimensions have been given, in almost every case, uniformly in inches.

London, *May*, 1893.

NOTE TO THIRD ENGLISH EDITION.

The whole work has again been carefully revised and the necessary corrections made. Some of the tables have been extended and some new matter added where necessary.

London, *August*, 1902.

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THE STEAM ENGINE.

INTRODUCTION.

THE knowledge that steam could be made use of as a motive power is very old, but it was not until about the year 1685 that the first model of a steam-engine was made by Dionysius Papin, a native of Blois in France, based on the fact that if cold water was applied or admitted to an enclosed vessel containing steam, the steam was condensed, and a vacuum resulted. In 1699, Captain Savery employed this principle for raising water, and in 1705 Newcomen, of Dartmouth, in Devonshire, constructed the first steam-engine properly so called. This first example was employed for raising water, and was known as "The Atmospheric Engine:" fig. 1 shows the general arrangement.

a. The steam cylinder.
b. The piston.
c. The steam generator or boiler.
d. The cock for admitting steam to the cylinder.
e. The cold water reservoir.
f. The cold water or "injection" cock.
g. A valve opening outwards to allow steam to be blown through, but preventing air from entering the cylinder.
h. The main beam for transmitting the motion of the piston to the pump rod.
i. The pump rod.

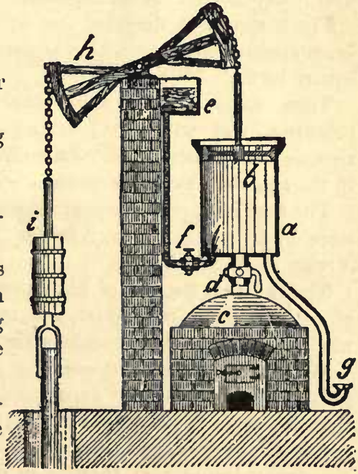

Fig. 1.—Newcomen's Atmospheric Engine.

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The action of the engine is as follows :—In fig. 1 the weight of the pump rod, *i*, has brought the piston, *b*, to the top of the cylinder, the cylinder is then filled with steam from the boiler at atmospheric pressure through the cock, *d*, which is then closed. The cock, *f*, is now opened, and cold water from the tank, *e*, enters the cylinder in a jet ; the steam is condensed, and a vacuum produced below the piston. The atmospheric pressure above the piston forces the latter down to the bottom of the cylinder, at the same time raising the opposite end of the main beam, and with it the pump rod, *i*. By opening and closing the cocks, *d* and *f*, in succession, an alternate up and down movement is given to the piston.

James Watt introduced many improvements, especially the separate condensing chamber, in place of admitting the injection water into the cylinder, which tends to lower the temperature of the cylinder walls, and he remarked in one of his specifications " that the cylinder should be kept as hot as the steam that enters it." James Watt also invented mechanism for automatically opening and closing the valves, introduced the use of steam above atmospheric pressure, and discontinued the use of atmospheric pressure alone, by making use of the expansion of the steam for doing work on the piston. Watt's first engine was built in 1768. A year or two earlier, Smeaton had made many improvements ; but James Watt was amongst the first who really treated the steam-engine on a scientific basis.

Fig. 3 shows in diagram one of Boulton and Watt's condensing beam engines, working with a pressure of from 5 to 15 pounds on the square inch.

These early engines were invariably worked with low boiler pressures, and with condensation. High boiler pressures were regarded with distrust, and James Watt, although he gave attention to the idea, had strong prejudices against using high-pressure steam.

The first high-pressure engines using steam at 3 to 4 atmospheres were by Oliver Evans in America, 1786 to 1801, and Trevithick and Vivian in England, 1802.

The fact that the use of high pressures decreased the consumption of coal was soon recognised, and steam of 4 to 6 atmospheres was used, with and without condensation.

With high-pressure steam the advantage of expansion is very great. By using steam " expansively " is to be generally understood that the steam is admitted at high pressure for a short portion of the piston-stroke only, and then allowed to expand down to the end of the stroke, losing heat thereby in doing work. If the expansion is carried too far, the cooling becomes so great as to condense part of the steam. In order to prevent the waste from this cause, and to prevent the range of temperature between the entering and exhaust steam, the expansion is now carried on in two or more cylinders of different capacities; and engines on this principle are called "compound," if with two cylinders; triple expansion, if with three; quadruple expansion, if with four. Sometimes they are named "continuous" expansion engines, whether with two or more cylinders; also, when with three cylinders, they are occasionally termed "tri-compound."

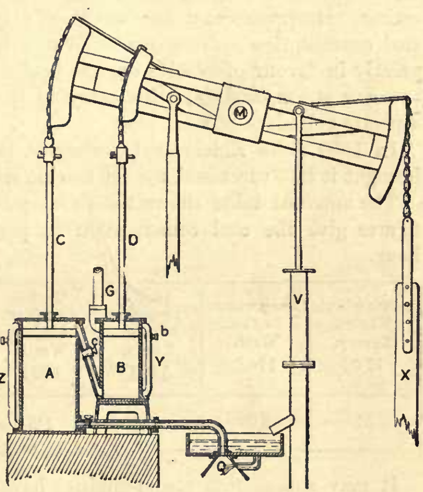


Fig. 2.—Hornblower's Engine.

The earliest attempt at a compound engine seems to have been that of Jonathan Hornblower in 1781, although he does not seem to have had any idea of what is understood now by a "compound" engine. He used two cylinders of unequal size, A and B (fig. 2); the steam was admitted first to the smaller cylinder, B, above the piston. On the piston arriving at the bottom of the stroke, the communication between the cylinder and boiler was cut off, and a communication between top and bottom of the cylinder was opened by means of the pipe, Y, and cock, *b*. The piston then ascended by the weight of the pump-rod, X, and the steam displaced by the rising piston now filled the space below the piston. The cock, *b*, was then closed, and boiler steam admitted above the piston. The downstroke then commencing, the cock, *c*, opened the steam from the cylinder, B, then flowed into cylinder A, where it further expanded. On completion again of the downstroke, communication between upper and lower sides of the piston, A, was established, and the steam, after having done work on the upper side of the piston, passed to the lower side by means of the pipe, Z, and cock, *d*, during the downstroke; the steam eventually passed to the condenser, Q. It will be seen that the engine was really "compound," but it does not seem to have been more economical than the simple engines of James Watt of the same date, and was rather complicated. An outline of this machine is shown in fig. 2. Hornblower's engine was single acting.

In 1804 Arthur Woolf made a two-cylinder high-pressure engine with a condenser, together with a water-tube boiler, and took out an English patent for the same. Woolf's engine was an improvement on Hornblower's, being more simple in construction and double acting. It appears that one of Woolf's best engines was tried for coal consumption against one of Watt's in Cornwall, and resulted greatly in favour of Woolf's engine, probably from his using higher pressure steam, and from his carrying the expansion down in two separate cylinders.

In 1850 John Elder much improved the compound engine, and brought it into successful use for marine engines.

The annexed table shows the development in saving of fuel ; the figures give the coal consumption in pounds per horse-power per hour.

Atmospheric Engine. **Savery.** 1700	Low-pressure Engine. **Watt.** 1768	High-pressure Engine. **Evans.** 1804	Double Cylinder Engine. **Woolf.** 1804	Compound Engine. **Elder.** 1850—1891	Triple Expansion Engine. 1880—1891
31	8·8	6·7	4·5	2·25	1·76 *

It may appear that steam-engines have been brought to a high state of perfection, but it must be remembered that the efficiency of the best engines and boilers taken together is only about 15 per cent. of the theoretical value of the heat energy of the fuel.

* In a trial of a triple expansion marine engine by the Research Committee of the Institution of Mechanical Engineers in July, 1891, the consumption of coal per indicated horse power per hour was as low as 1·46 lbs.

SECTION I.

TYPES OF STEAM ENGINES.

Beam Engine, the original form as constructed by Boulton and Watt. This form owed its existence to the fact that all the earlier steam engines were used for pumping water, the beam forming a convenient means of attachment for the pump rods. Beam engines are still used for pumping, but are gradually giving way to more modern types.

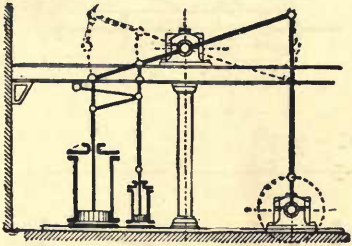
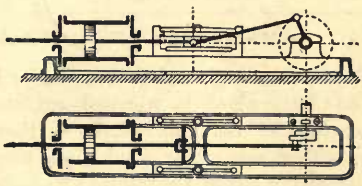


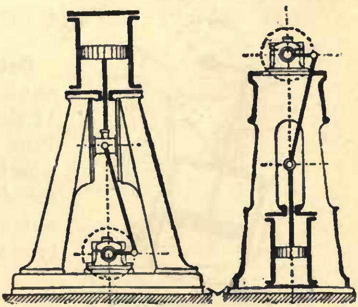
Fig. 3.—Beam Engine.

Horizontal Engine, with open frame cast iron bed-plate, a type much used for all sizes of engines for general purposes. The bed-plate frame is of U section, and is bolted down to a foundation of masonry or brickwork, the cylinder, main bearing and guides being bolted to the bed-plate.

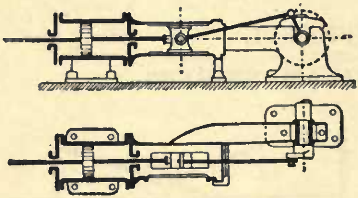


Figs. 4 and 5.—Horizontal Engine with Bed-Plate.

Vertical Engine.—A type now extensively used both for small and large engines; it has the advantage of occupying small floor space, and is not open to the objection so often put forward against horizontal engines,—that of the cylinder wearing oval. An endless number of varieties of this type are now in use, and it is the generally accepted type of marine screw-propeller engines.



Figs. 6 and 7.—Vertical Engines.



Figs. 8 and 9.—Girder Engine.

Corliss Frame or Girder Engines.—A type of horizontal engine now extensively used. Figs. 8 and 9 show an example with bored guides; they are also often made with flat-planed guides. In both cases the guides are formed in the main casting or girder which connects the cylinder to the main bearing. There are many varieties of this type.

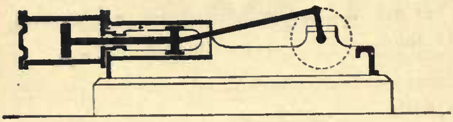
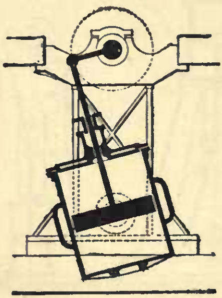


Fig. 10.—Self-contained Horizontal Engine.

Self-contained Horizontal Engines, with bent or slotted out cranks. This type, largely used for small power short-stroke engines, has the cylinder bolted on to the end of an open bed-plate, which is widened out at the other end to take both bearings of the crank-shaft, so that the flywheel may be keyed on either side. The guides are usually formed in the bed-plate, the boring out of the guides and facing of the end flange being done at the same setting.



Oscillating Engines, formerly much used as marine engines. Fig. 11 shows the usual type as made by Penn and others for driving paddle-wheels; this type has also been used for driving screw propellers, but is now almost superseded by the vertical type of marine engine.

Fig. 11.—Oscillating Marine Engine.

Trunk Engines.—Fig. 12 shows an example of this type for driving a screw propeller as made by Penn. It has the advantage of being very compact and simple, but is open to the objection of the very large gland for the trunk, and that the trunk in its outward stroke is exposed to the cooling effect of the air. Very large engines of this type were formerly made. Two varieties are also shown in figs. 13 and 14.

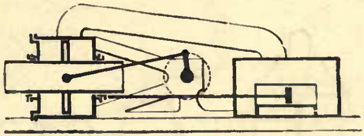
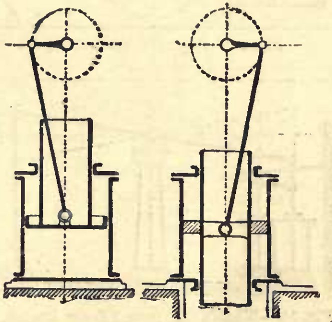
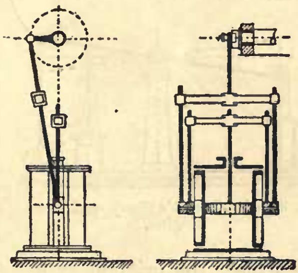


Fig. 12.—Penn's Trunk Engine.



Figs. 13 and 14.—Trunk Engines.

Vertical Marine Engine, old type by Maudslay. Arrangement to reduce height, but at the expense of introducing heavy reciprocating masses.



Figs. 15 and 16.—Maudslay's Marine Engine.

Steeple Engine, formerly, and still occasionally, used for driving paddle-wheels. A variety of this type has been used for small powers, and known as the Table Engine.

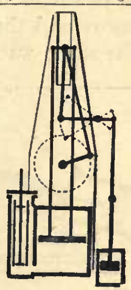


Fig. 17.—Steeple Engine.

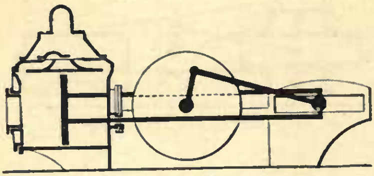


Fig. 18.—Bourne's Marine Engine.

Bourne's "return connecting-rod" engine, derived from the Steeple Engine, and used principally for driving screw propellers. There are two piston-rods in this engine, placed above and below the crank-shaft.

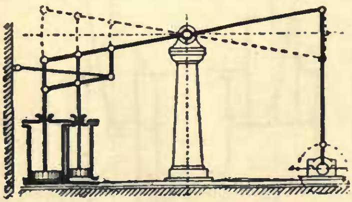


Fig. 19.—Woolf's Compound Beam Engine.

Beam Engine, Woolf's Compound. — Two unequal cylinders, side by side, at one end of the beam. Many pumping engines are of this type.

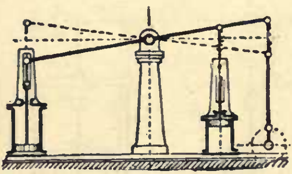


Fig. 20.—M'Naught's Compound Beam Engine.

M'Naught's Compound Beam Engine.—This system, consisting of a small cylinder (high pressure cylinder), placed at the opposite end of the beam to the larger cylinder used as a low-pressure cylinder, was introduced by M'Naught for increasing the power of existing engines. The high-pressure cylinder was the one added, the original cylinder being the low pressure cylinder. The power of the engine was thus increased by increase of boiler pressure and the addition of the new small cylinder, to which the boiler steam was admitted.

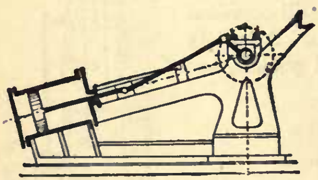


Fig. 21.—Inclined Frame Engine.

Inclined Frame Engines, now used extensively for paddle-steamers in several different varieties, usually compound engines.

A Double-Cylinder Engine variety derived from the above, with the cylinders inclined at an angle of about 45°, is occasionally used for driving rolling-mills in bar iron works.


Fig. 22.—Inclined Frame Engine.

A variety of compound engine, with inclined cylinders, is shown in fig. 23.

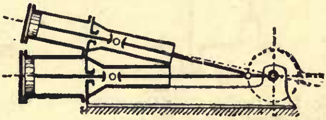


Fig. 23.—Inclined Cylinder Engine.

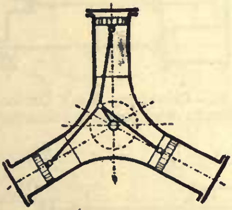


Fig. 24.—Radial Engine.

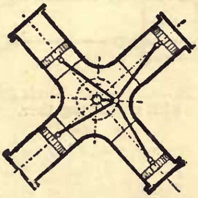


Fig. 25.—Four-Cylinder Radial Engine.

Radial Engines.—A modern type, of which there are many varieties. Fig. 24 shows Brotherhood's three-cylinder single-acting radial engine ; fig. 25, one with four cylinders. These are used for driving fans, steam-launches, and other purposes requiring speed and compactness.

Willans' Central Valve Engine.—One of the most modern types of engine, single-acting, compound or triple-expansion ; a special feature is the hollow piston-rod and central valve. Extensively used for driving dynamo-electric machines coupled direct on to the armature spindle. (See pages 348—352.)


Fig. 26.—Willans' Central Valve Engine.

Various ways of arranging Cylinders and Cranks in Double and Three-cylinder, Compound, and Triple-expansion Engines.—Outline diagrams.

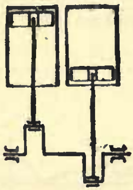


Fig. 27.—Double Cylinder, with cranks opposite or at 180 degrees.

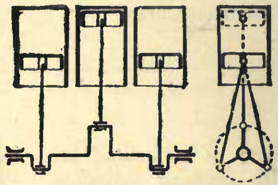


Fig. 28.—Three-cylinder Engine, with cranks at 120 degrees.

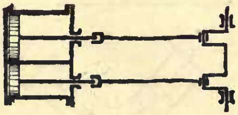


Fig. 29.—Compound Woolf Engine, with cranks together.

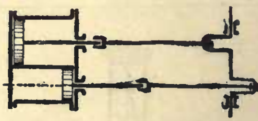


Fig. 30.—Compound Woolf Engine, with cranks opposite or at 180 degrees.

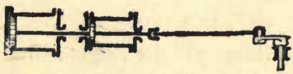


Fig. 31.—Compound Tandem Engine with receiver.

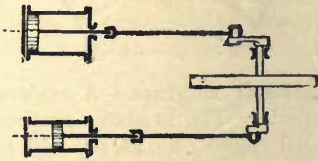


Fig. 32.—Compound Engine, with cylinders side by side with receiver; cranks at 90 degrees.

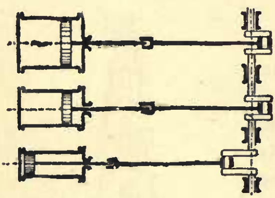


Fig. 33.—Triple Expansion Engine, with cylinders side by side; cranks at 120 degrees.

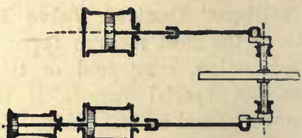


Fig. 34.—Triple Expansion Engine, semi-tandem; two cranks at 90 degrees.

Action of Pistons and Cranks in Woolf, Compound, and Triple-expansion Engines.*—Outline diagrams.

A crank is shown to each piston, to enable the action to be clearly followed; in reality there is only one crank shaft to each engine.

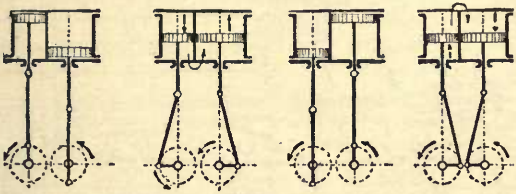


Fig. 35.—Woolf Engine, with cranks opposite (180°), without receiver.

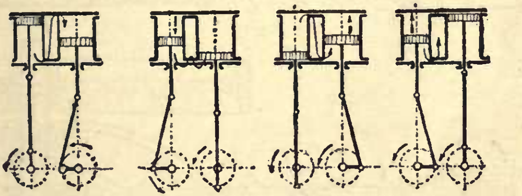


Fig. 36.—Compound Engine, with cranks at right angles, with receiver.

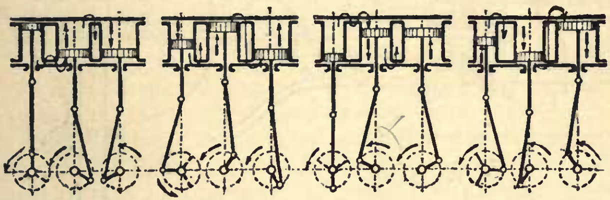


Fig. 37.—Triple Expansion Engine: cranks at 120 degrees, with two receivers.

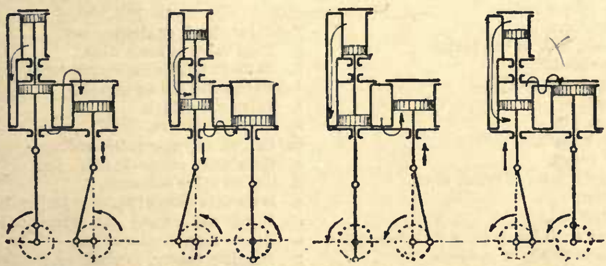


Fig. 38.—Semi-Tandem Triple-Expansion Engine, with two receivers.

* Further details of the action of these compound engines are given in the section on Compound Engines, page 350.

Names of the Parts of a simple Steam Engine.

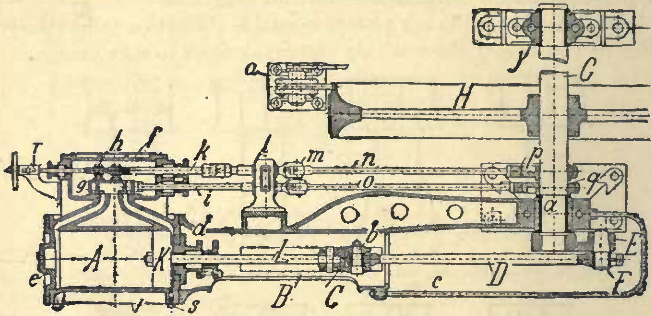


Fig. 39.

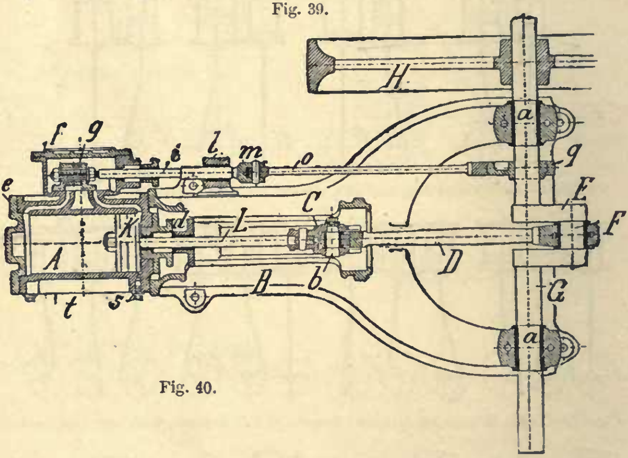


Fig. 40.

A. Cylinder.
B. Frame, bed, or bed plate.
C. Crosshead.
D. Connecting rod.
E. Crank.
F. Crank pin.
G. Crank shaft.
H. Fly wheel.
J. Outer bearing of crank shaft.
K. Piston.
L. Piston rod.
a. Crank shaft bearings, or main bearings.
b. Crosshead pin or gudgeon.
c. Safety rail round engine.
d. Front cylinder cover with stuffing box and gland.
e. Back cylinder cover.
f. Valve chest, or steam chest.
g. Main valve or main slide.
h. Cut off valve or expansion valve.
i. Main valve rod or spindle.
l. Valve rod guides.
m. Valve rod joints.
n. Cut off valve eccentric rod.
o. Main valve eccentric rod.
p. Cut off valve eccentric.
q. Main valve eccentric.
r. Hand adjustment for varying the expansion.
s. Bosses for indicator cocks.
t. Lagging, casing, or clothing.
u. Arrangement for turning the engine slowly round, "barring" arrangement.

Notes and General Remarks.

The size or perhaps better the power of an engine may be estimated by the diameter of the cylinder, the length of the stroke and the boiler pressure. The merits may be judged by the general design, workmanship, materials used, the construction of connecting rod ends, crosshead, crank shaft bearings, and by the nature of the valve gear.

The proportion of stroke to cylinder diameter varies from 1.5 to 2.0 in ordinary engines for driving plants of machinery, and in special high speed engines from .75 to 1.25.

The steam engine maker must consider well before making a series of engines of one type, how best to make the minimum number of patterns serve for different sizes or powers required, and it is important to see that the parts are as far as possible symmetrical about the centre line, in order that engines may be readily built right or left-handed, in short before making either drawings or patterns, all possible dimensions should be tabulated out clearly, a few skeleton outlines being made for the different sizes showing centre lines only, the details such as pistons, crossheads, crank pins, main bearings, etc. can then be designed and a general drawing of each size prepared; from which the larger patterns can be made.

The most convenient starting points are obviously the diameter of the cylinder and the length of the stroke, then follow the speed, the maximum steam pressure and the subordinate details.

In making a set of drawings for a steam engine; for example of the horizontal girder type; the following method will be found both convenient and quick. Draw the centre line for the cylinder in plan and also the centre line for the crank shaft at right angles to the same, then from the table of dimensions draw in the crank pin where the two centre lines cross, on the centre line of crank shaft the width of the crank may then be set out which will give the position of the main bearing, the width of the same being taken from the tables; the same two centre lines are then drawn in elevation, and a circle for the path of the crank pin centre, the length of the connecting rod from the tables, will give the three positions of crosshead pin centre, an outline of crosshead at the two extreme positions will give the necessary length and positions of the guide surfaces for the crosshead slippers, the piston rod gland then drawn in will give the necessary clearance between it and the end of the crosshead boss. The stuffing box and cylinder cover then added will enable the position of the cylinder to be determined ; the remainder of the engine may then be filled in on the same principle, remembering that in all machine designing, whether they be steam engines or other machines of more or less special character, it is always well to have in view the fact that the complete machine to be designed is composed of a number of elementary parts or "units," which have to be brought together in a practical form, and in order to save time and labour all these "units" should be thoroughly worked out before the attempt is made to work out the complete machine.* It is, however, obvious that even elementary parts require modification, but the more thoroughly the smaller details are worked out, the less will be the labour and thought necessary to produce a complete machine for any special purpose. Speaking of the designing of steam engines of small powers, where a large number have to be made of few different sizes, the most successful makers have always been those who keep all the possible elementary parts in stock, this system enabling the cost of production to be greatly decreased, and leaving a margin for better design, material, and workmanship than is possible when each part is only made when actually wanted. Careful proportion of parts is not all that is wanted ; a detail that may work out well in one manufactory will not do so well in another. The designer must know the capabilities of the factory, nay, even the capabilities of the heads of the different workshops, before he can produce designs capable of being carried out so that the works may be run at their maximum efficiency.

* To many, these remarks may seem superfluous, but from neglect of these first principles of machine design much time is often lost in the drawing office.

SECTION II.

DETAILS OF STEAM ENGINES.

THE following leading points in steam engine design and construction are important.

Ample strength—it is not sufficient to trust to calculation alone but to compare calculated results, and to modify such results with judgment based on experience.

The selection of suitable materials for the different parts ;—such as the best possible mixtures of cast iron for the cylinder, the use of phosphor bronze, good gun metal or even white metal in the bearings, and steel for all rods and spindles.

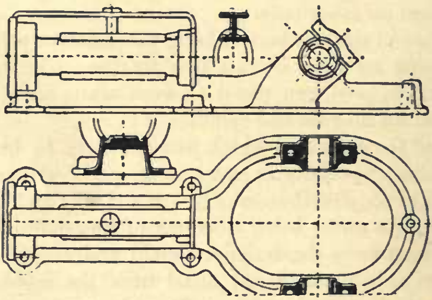
The form of those parts of which patterns have to be made is of importance ; and especially in cast iron or cast steel very great care must be used in the distribution of the metal, any sudden transition from thick to thin metal being a source of unsoundness in the castings, and in most cases the designer should endeavour to study the different parts to be made in cast metal from the moulder's side of the question, the best castings being almost invariably produced from patterns so formed as to leave the sand with ease, without rendering it necessary to "mend up" the mould ;—it being obvious that all mending up of a mould means a source of inaccuracy in the resulting casting, *i.e.* that it will differ from the pattern and no two castings will be obtained alike.

Good workmanship ;—the dearest is not always the best. Special attention should be given to the wearing surfaces, and in order to prevent hot bearings the brasses should always be bored out a fraction larger than the measured diameter of the shaft which runs in them ; it is often difficult to make a workman do this as he is apt to think that such a practice is against the idea of a "good fit ;" hot bearings often cost a factory considerable sums of money and may damage the reputation of an engine maker. Good lubrication is a necessity, and all oil holes in brasses or joints should be supplemented by distributing oil grooves or channels, deep enough to keep clear for a long time, a simple oil hole in a long bearing especially, is totally inadequate.

"An excessive or exaggerated seeking after improvements or originality is very objectionable, one holds to good and well tried schemes. Every innovation should be discussed with a partner or a practical man, four eyes seeing more than two." The seeking after improvements or new appliances should always be encouraged, many useful schemes are lost by diffidence on the part of the schemer, which a little encouragement would allay.

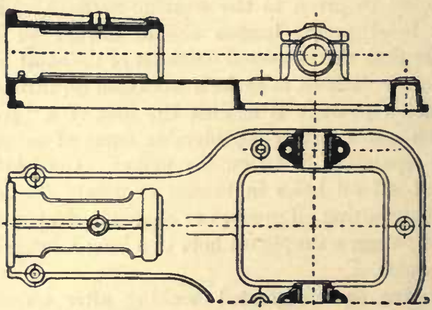
Besides one's own experience we may often make use of that of others, and notes of what we see, systematically arranged, so as to be easy to find, will afford much assistance.

Frames for small Self-contained Horizontal Engines up to 15 inches stroke.

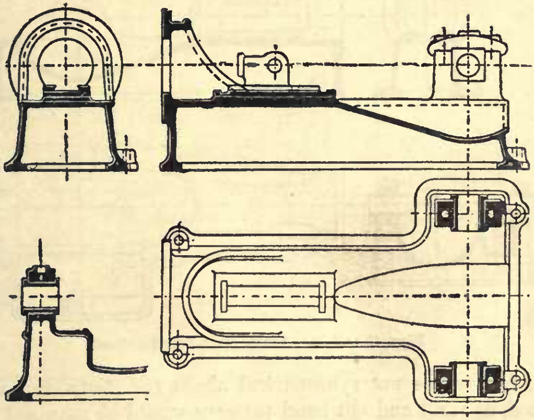


Figs. 41 and 42.—Frame, with double (top and bottom) slipper guides with flat surfaces.

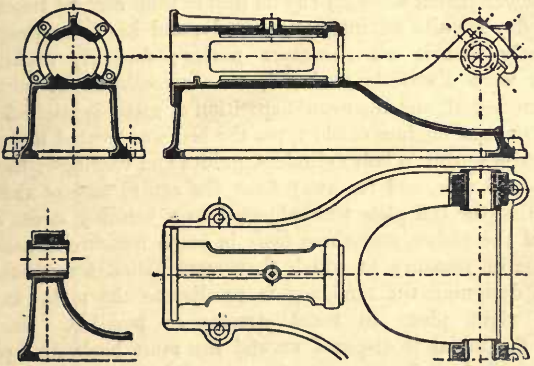
The crank shaft is bent or slotted out and admits of the flywheel being carried at either side of the engine outside the bearings, and if required a driving pulley at the opposite side. The crank shaft bearing brasses are generally in two pieces with a cap and two or four bolts.



Figs. 43 and 44.—Frame with bored Guides.

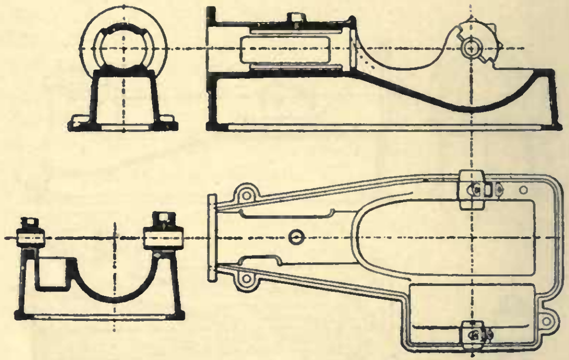


Figs. 45 and 46 – Frame with single slipper guide.



Figs. 47 and 48.—Frames with bored guides.

All the above frames are symmetrical about the centre line in plan



Figs. 49 and 50.—Frame with bored guides.

These frames are not symmetrical about the centre line and but little used, as right and left hand patterns would be required.

These engines can be used as wall engines by being bolted vertically with either the cylinder or crank shaft above; to a substantial wall; they are convenient also for transit as they can be packed complete with everything in its place except the fly-wheel.

The development of steam engine frames from earliest times to the present day is quite an interesting study, and has been progressing so quietly that it often escapes notice; formerly architectural features were always introduced, now smoothness of outline is almost universal, and the even disposition of metal is studied.

In horizontal engines of old types the bed was treated more or less as a plate whereon to bolt cylinders, guides and bearings; the metal was placed below, and far away from the centre line of the piston rod, so that the bed plate was subjected to a bending stress at each stroke of the piston, and these beds in some measure depended on the mass of masonry to which they were bolted for stiffness. In modern designing the tendency is to dispose the metal as nearly in the right place to resist stresses as possible. In girder engines the metal is disposed around the same horizontal plane as the piston rod, and as near to it as possible sideways; in vertical marine engines the frame is a stiff triangle, placed over the crank shaft; in marine paddle engines, &c., of diagonal type the metal is disposed in a most advantageous manner.

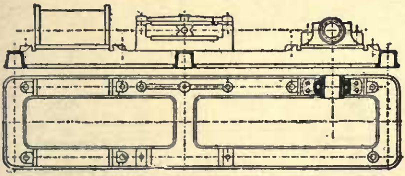


Fig. 51.

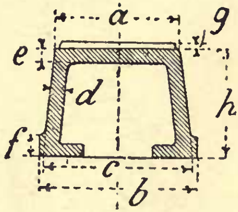


Fig. 52.

Bed-plate Frames of U-form section ; an old type, but still used for large engines.

TABLE 1.—**Dimensions of the above Frames.**

Engine.		All Dimensions are in Inches.							
H*	D	*h*	*b*	*a*	*c*	*d*	*e*	*f*	*g*
12	6	$5\frac{1}{2}$	5	$3\frac{1}{2}$	4	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{4}$
16	8	7	6	$4\frac{1}{2}$	5	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{4}$
24	12	$8\frac{1}{2}$	$7\frac{1}{2}$	$5\frac{1}{2}$	$6\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{3}{8}$
32	16	$10\frac{1}{2}$	9	7	8	$\frac{7}{8}$	$\frac{7}{8}$	1	$\frac{3}{8}$
40	20	12	10	9	$9\frac{1}{2}$	1	1	$1\frac{1}{8}$	$\frac{1}{2}$
48	24	14	12	$9\frac{1}{2}$	11	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$\frac{1}{2}$
56	28	16	$13\frac{1}{2}$	$10\frac{1}{2}$	12	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{8}$
64	32	18	15	$11\frac{1}{2}$	13	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$\frac{5}{8}$
72	36	20	$16\frac{1}{2}$	13	15	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$
80	40	22	18	14	16	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$\frac{3}{4}$

* *Note.*—Throughout the Tables the letter H signifies the stroke of the engine, and D the diameter of the cylinder.

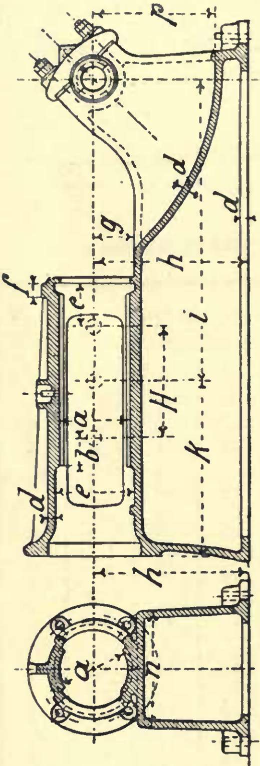


Fig. 53.

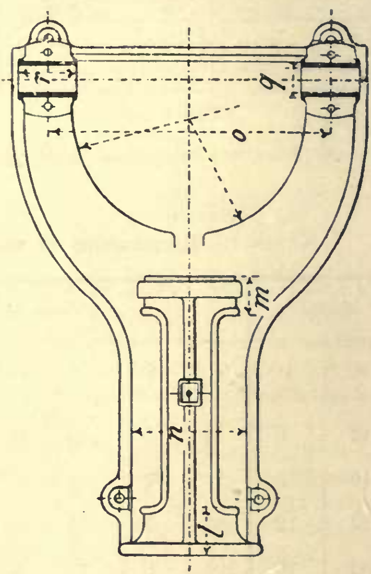


Fig. 54.

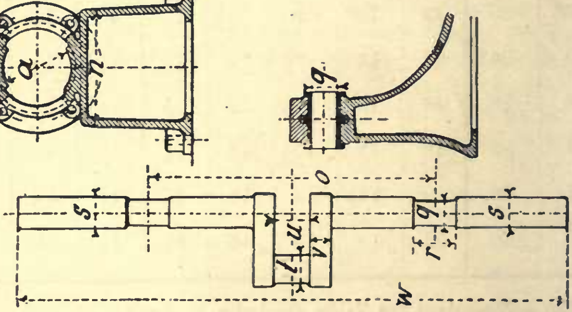


Fig. 55.

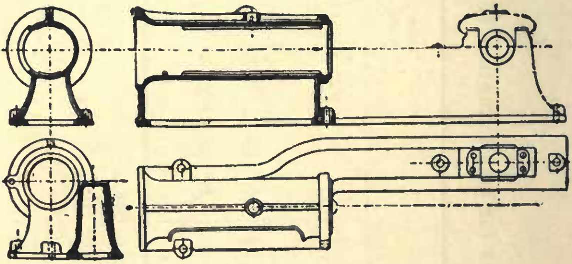
TABLE 2.—**Dimensions of Frames and Crank Shafts for small Engines** (Figs. 53—55).

Engine.		Frame															Main bearing.		Crank Shaft.				
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*
8	4	6	2	$3\frac{1}{4}$	$\frac{1}{2}$	$6\frac{1}{2}$	$1\frac{3}{8}$	$3\frac{1}{2}$	12	22	14	4	$3\frac{1}{2}$	10	22	$9\frac{1}{2}$	$2\frac{1}{2}$	5	3	$2\frac{3}{4}$	$3\frac{1}{4}$	2	48
12	6	$7\frac{1}{2}$	3	$4\frac{1}{4}$	$\frac{5}{8}$	$8\frac{1}{4}$	$1\frac{5}{8}$	$4\frac{1}{2}$	16	32	18	$4\frac{3}{4}$	4	12	32	$10\frac{1}{2}$	3	6	$3\frac{1}{2}$	$3\frac{1}{4}$	$3\frac{1}{2}$	$2\frac{3}{8}$	58

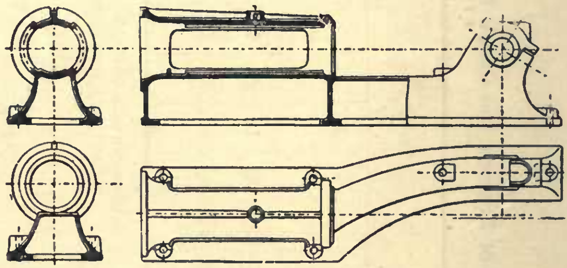
All Dimensions in this Table are given in Inches.

Frames for Medium and Large Engines.

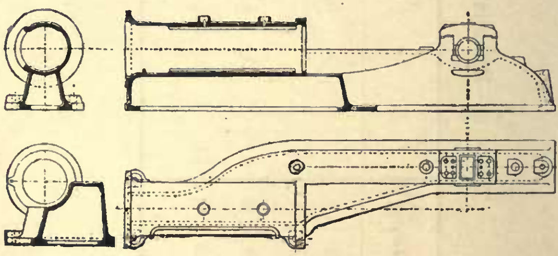
The bed plate frame as in page 19, fig. 51, was formerly largely used but now has given place to the more modern types given below ; it has the advantage of taking firm hold of the foundation and is still much used for winding engines of large size.



Figs. 56 and 57.

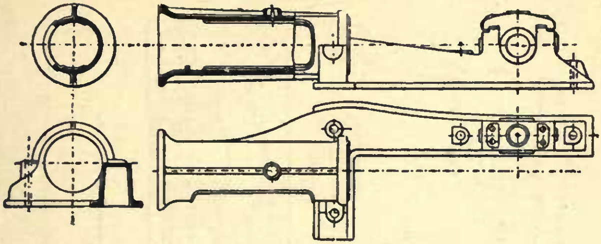


Figs. 58 and 59.

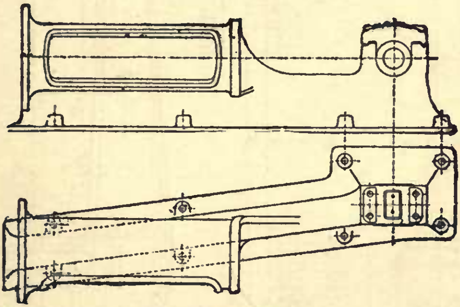


Figs. 60 and 61.

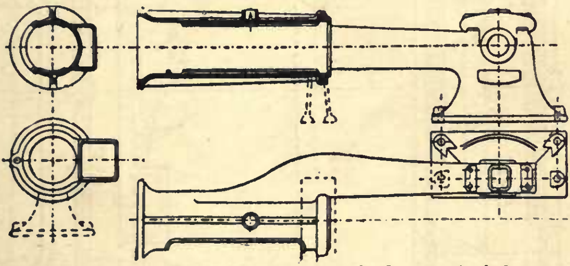
Figs. 61* & 62.



Figs. 63 & 64.



Figs. 65 & 66.



All the above frames Figs. 56—66 are of the semi-girder type having the frame supported throughout its length on the foundation except fig. 65, which is of the true girder type, the girder being free from the foundation, supports being at the ends, one under the cylinder the other at the main bearing. In the semi-girder type the cylinder usually overhangs the end of the frame.

Note.—The frame of a girder-winding engine at Herne, of 5′3″ stroke, cracked at (*a*), fig. 67, and the bending of the girder in the middle could be seen distinctly. A hole was drilled at the end of crack (*b*), and a supporting foot fixed under the girder at C.

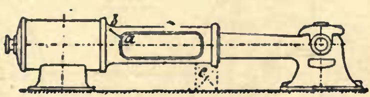
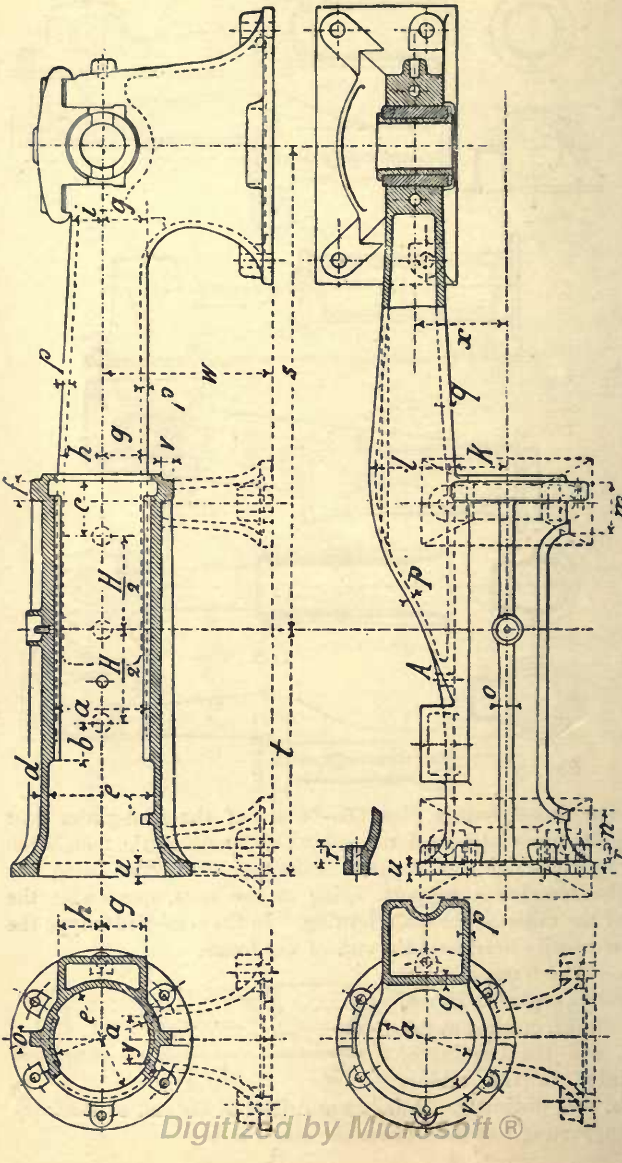


Fig. 67.



Figs. 68—70.

TABLE 3.—**Dimensions of Girder Frames with Bored Guides** (Figs. 68—70).

Engine.		All Dimensions are given in Inches.																						
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*	*x*	*y*
20	10	11	$4\frac{1}{4}$	$5\frac{3}{4}$	$\frac{11}{16}$	$11\frac{7}{8}$	$2\frac{1}{4}$	$4\frac{3}{4}$	$4\frac{1}{2}$	$2\frac{3}{4}$	$5\frac{1}{2}$	9	$5\frac{1}{2}$	$6\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{3}{4}$	50	27	$1\frac{3}{8}$	$1\frac{1}{8}$	18	$9\frac{1}{4}$	5
24	12	12	$4\frac{3}{4}$	$6\frac{1}{2}$	$\frac{3}{4}$	13	$2\frac{3}{8}$	$5\frac{1}{2}$	5	$3\frac{1}{4}$	$6\frac{1}{2}$	10	$6\frac{1}{2}$	$7\frac{1}{2}$	$1\frac{1}{2}$	$\frac{3}{4}$	1	2	60	$30\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	20	$10\frac{3}{4}$	$6\frac{1}{4}$
28	14	$13\frac{1}{2}$	$5\frac{1}{2}$	$7\frac{1}{4}$	$\frac{13}{16}$	$14\frac{5}{8}$	$2\frac{1}{2}$	$6\frac{1}{4}$	6	$3\frac{3}{4}$	$7\frac{1}{2}$	$11\frac{1}{2}$	7	8	$1\frac{3}{4}$	$\frac{7}{8}$	1	$2\frac{1}{4}$	70	34	$1\frac{1}{2}$	$1\frac{1}{2}$	22	$12\frac{1}{4}$	$7\frac{1}{4}$
32	16	15	6	8	$\frac{7}{8}$	$16\frac{1}{2}$	$2\frac{3}{4}$	$7\frac{1}{4}$	$6\frac{3}{4}$	$4\frac{1}{2}$	$8\frac{1}{4}$	13	8	9	2	$\frac{7}{8}$	$1\frac{1}{8}$	$2\frac{1}{2}$	80	$37\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	24	14	8
36	18	17	$6\frac{1}{4}$	$8\frac{3}{4}$	$\frac{7}{8}$	$18\frac{1}{2}$	3	8	$7\frac{1}{2}$	5	$9\frac{1}{2}$	$14\frac{1}{2}$	$8\frac{1}{2}$	$9\frac{1}{2}$	$2\frac{1}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$	$2\frac{7}{8}$	90	41	$1\frac{3}{4}$	2	26	$15\frac{5}{8}$	$8\frac{3}{4}$
40	20	19	$6\frac{3}{4}$	$9\frac{1}{2}$	$\frac{15}{16}$	$20\frac{1}{2}$	$3\frac{1}{4}$	$8\frac{3}{4}$	8	$5\frac{1}{2}$	$10\frac{1}{2}$	$15\frac{1}{2}$	$9\frac{1}{2}$	$10\frac{1}{2}$	$2\frac{1}{2}$	1	$1\frac{1}{4}$	$3\frac{1}{4}$	100	$44\frac{1}{4}$	2	$2\frac{1}{4}$	28	$17\frac{1}{4}$	$9\frac{1}{2}$
44	22	21	7	10	1	$22\frac{1}{2}$	$3\frac{1}{2}$	$9\frac{1}{2}$	$8\frac{3}{4}$	6	$11\frac{1}{2}$	$16\frac{1}{2}$	10	11	$2\frac{3}{4}$	1	$1\frac{1}{4}$	$3\frac{1}{2}$	110	$48\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	30	$18\frac{1}{2}$	10

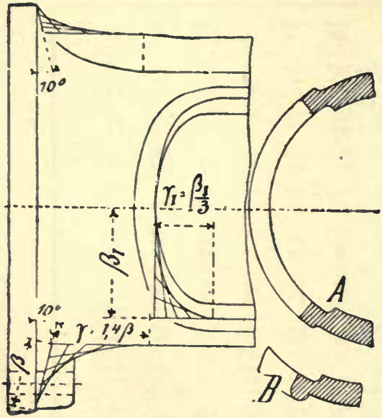
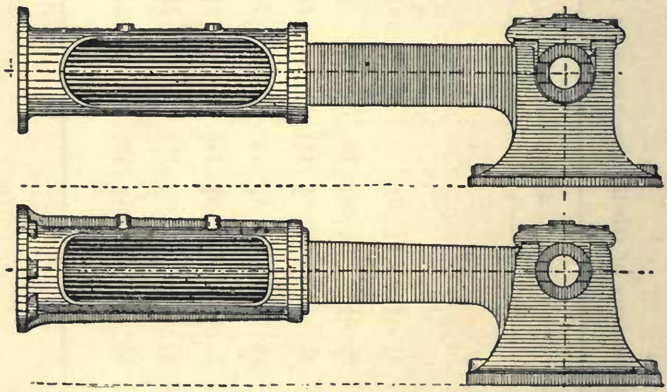


Fig 71

The opening or window for the crosshead has ends in the form of a parabola with axes as 1 to 3 and a fillet is carried round the window either as at A or B. Fig. 71 also shows the blending of the round part of the frame with the flange to which the cylinder is bolted; the guides are sometimes finished as shown in Figs. 72, 73.



Figs. 72 and 73.

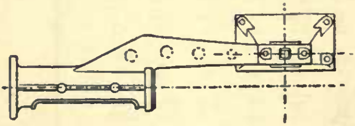


Fig. 74.

Fig. 74 shows the connection between the part of the frame with the bored out guides and the main crank-shaft-bearing, the straight line forms shown in the figure, makes the pattern simple; the dotted circles show the holes for venting the core and for getting it out of the casting.

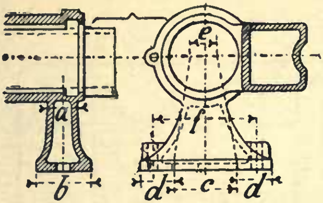


Fig. 75.

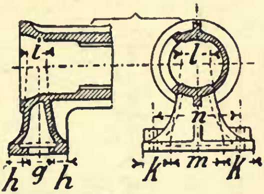


Fig. 76.

TABLE 4.—**Feet for supporting Girder Frames** (Figs. 75—77).

Engines.		Feet supporting middle of Girder.						Feet at Cylinder end of Girder.						
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*
20	10	—	—	—	—	—	—	6	2½	4½	5½	4¾	8	15
24	12	—	—	—	—	—	—	6¾	2½	5	6¼	5¾	9½	17
28	14	—	—	—	—	—	—	7½	2¾	6½	6¾	6¾	11½	19½
32	16	5½	10	12	7¼	6½	21	—	—	—	—	—	—	—
36	18	6¾	10½	12¾	7½	6¾	22	—	—	—	—	—	—	—
40	20	7¼	11	13½	8	7¼	23	—	—	—	—	—	—	—
44	22	7½	11½	14½	8¾	7½	25½	—	—	—	—	—	—	—

All Dimensions are in Inches.

Small engines up to 12″ diameter of cylinder are frequently made with overhanging cylinders and with the foot cast in one with the girder as shown in fig. 76.

For larger engines an intermediate supporting foot is sometimes cast or bolted on to the girder. The curve of the sides of these feet is parabolic, fig. 77

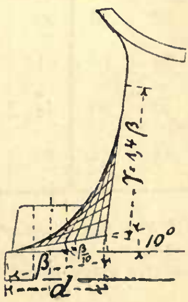


Fig. 77.

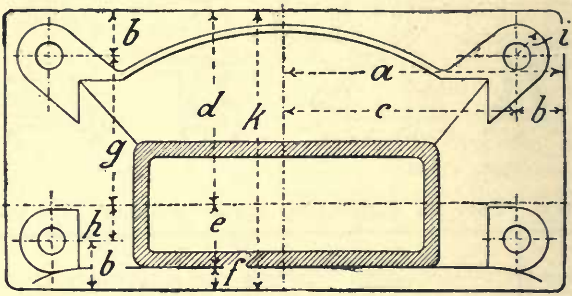


Fig. 78.*

TABLE 5.—**Main Bearing Foot** (Fig. 78).

Engines.		All Dimensions are in Inches.									
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*
20	10	$13\frac{1}{4}$	2	$11\frac{1}{4}$	9	$2\frac{7}{8}$	$1\frac{1}{8}$	7	2	$1\frac{3}{4}$	13
24	12	$14\frac{3}{4}$	$2\frac{1}{2}$	$12\frac{1}{4}$	11	$3\frac{3}{8}$	$1\frac{5}{8}$	$8\frac{1}{2}$	$2\frac{1}{2}$	2	16
28	14	18	$2\frac{1}{2}$	$15\frac{1}{2}$	12	$3\frac{7}{8}$	$2\frac{1}{8}$	$9\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{4}$	18
32	16	20	$2\frac{3}{4}$	$17\frac{1}{4}$	13	$4\frac{3}{8}$	$2\frac{1}{8}$	$10\frac{1}{4}$	$3\frac{3}{4}$	$2\frac{1}{4}$	$19\frac{1}{2}$
36	18	$21\frac{1}{2}$	$2\frac{3}{4}$	$18\frac{3}{4}$	15	$4\frac{7}{8}$	$1\frac{5}{8}$	$12\frac{1}{4}$	$3\frac{3}{4}$	$2\frac{1}{2}$	$21\frac{1}{2}$
40	20	24	$3\frac{1}{4}$	$21\frac{3}{4}$	16	$5\frac{3}{8}$	$2\frac{1}{8}$	$12\frac{3}{4}$	$4\frac{1}{4}$	$2\frac{1}{2}$	$23\frac{1}{2}$
44	22	26	$3\frac{1}{4}$	$22\frac{3}{4}$	18	6	$2\frac{1}{2}$	$14\frac{3}{4}$	$5\frac{1}{4}$	$2\frac{3}{4}$	$26\frac{1}{2}$

* This foot is not symmetrical about the longitudinal centre line, and therefore cannot be used for both right and left-hand engines without alteration to patterns unless bolted to girders by a flange.

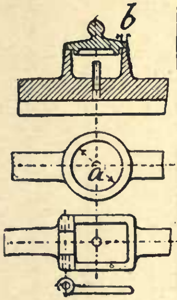


Fig. 79.

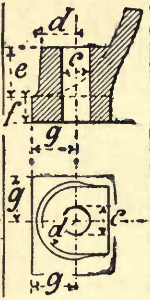


Fig. 80.

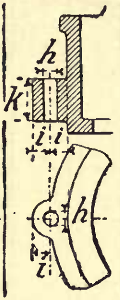


Fig. 81.

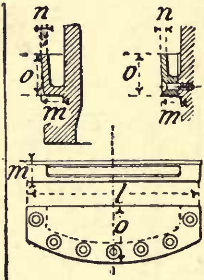
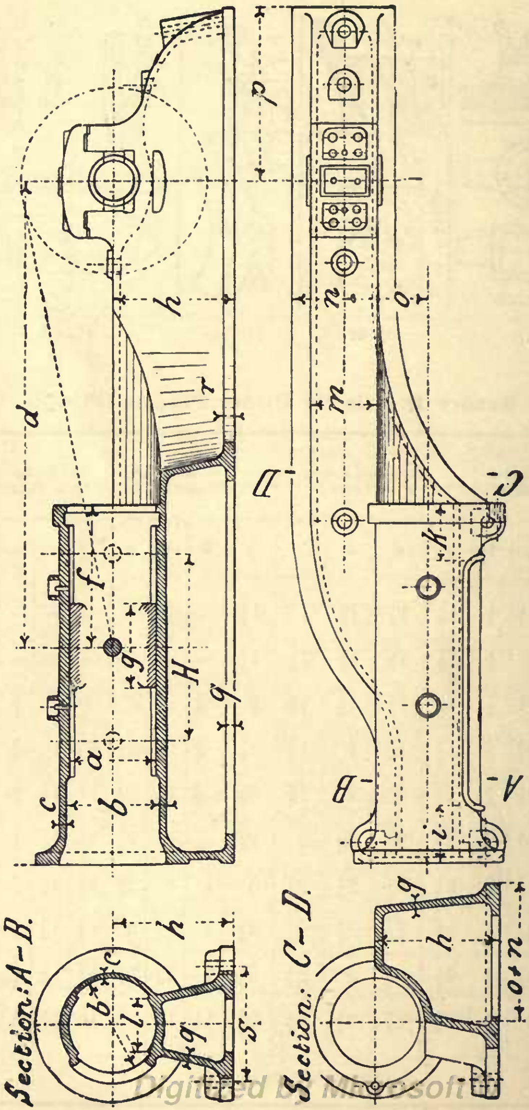


Fig. 82.

Table 6.—**Sundry Details for Girder Frames** (Figs. 79—82).

	Oil Cup for Guides.			Bosses for Foundation Bolts					Boss for Guard Rail.			Oil Catcher for Main Bearing.			
H	No.	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*
8	1	$1\frac{1}{2}$	$\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	1	$1\frac{1}{2}$	—	—	—	—	—	—	—
12	1	2	$\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{3}{4}$	—	—	—	—	—	—	—
16	1	$2\frac{1}{2}$	$\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{8}$	2	$\frac{5}{8}$	$\frac{3}{4}$	2	9	1	$\frac{1}{4}$	2
20	1	$2\frac{3}{4}$	$\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$1\frac{1}{2}$	$2\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$2\frac{1}{4}$	10	1	$\frac{1}{4}$	2
24	1	$3\frac{1}{4}$	$\frac{3}{8}$	$1\frac{3}{4}$	2	2	$1\frac{3}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	1	$2\frac{3}{8}$	11	$1\frac{1}{8}$	$\frac{1}{4}$	$2\frac{3}{8}$
28	1	$3\frac{3}{4}$	$\frac{3}{8}$	2	$2\frac{1}{4}$	$2\frac{3}{8}$	2	$2\frac{3}{4}$	$\frac{7}{8}$	1	$2\frac{1}{2}$	12	$1\frac{1}{8}$	$\frac{1}{4}$	$2\frac{1}{2}$
32	2	$3\frac{1}{4}$	$\frac{3}{8}$	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{4}$	3	1	1	$2\frac{3}{4}$	13	$1\frac{1}{4}$	$\frac{3}{8}$	$2\frac{3}{4}$
36	2	$3\frac{1}{2}$	$\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{3}{8}$	$3\frac{1}{4}$	1	$1\frac{1}{8}$	3	14	$1\frac{1}{4}$	$\frac{3}{8}$	3
40	2	$3\frac{3}{4}$	$\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{1}{2}$	3	$2\frac{1}{2}$	$3\frac{1}{4}$	1	$1\frac{1}{4}$	$3\frac{1}{4}$	15	$1\frac{3}{8}$	$\frac{3}{8}$	$3\frac{1}{4}$
44	2	4	$\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{5}{8}$	$3\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$3\frac{1}{2}$	16	$1\frac{1}{2}$	$\frac{1}{2}$	$3\frac{1}{2}$

All Dimensions are in Inches.

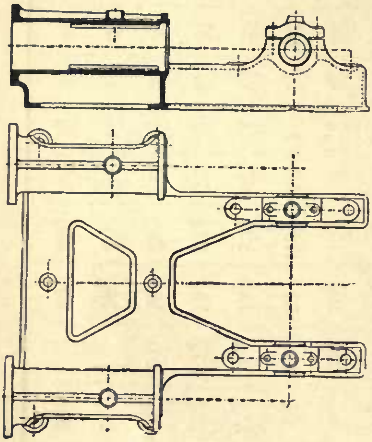


Figs. 83—86.

TABLE 7.—**Dimensions of Frames with bored Guides supported on Foundation from end to end, "semi-girder" type** (Figs. 83—86).

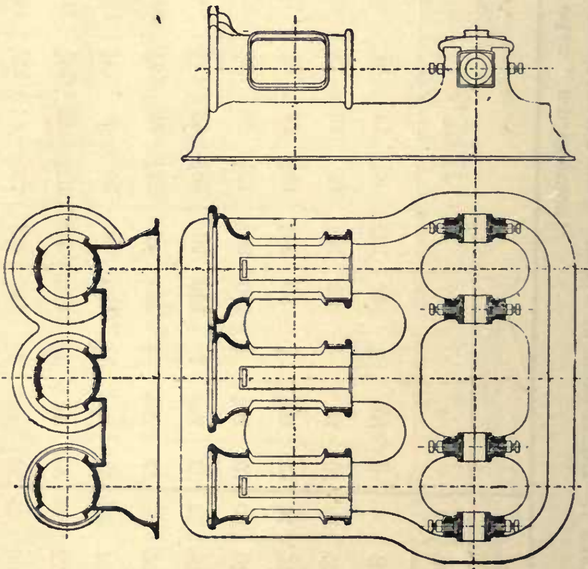
Engine.		All Dimensions are in Inches.																	No. of Bolts
H	D	*a*	*b*	*c*	*d*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	
16	8	9	10	⅝	42	12¾	8¾	14	5½	5	4½	5¾	7¼	9¼	28	¾	¾	17¼	7
20	12	12	13	¾	62	19¼	12	19	7½	6½	6¼	8	8¾	12⅜	38	¾	1	19½	7
32	16	15	16½	¾	80	24¾	15¼	24	8¾	7½	8	9½	10½	15¼	48	⅞	1¼	22	8
40	20	19	20½	⅞	100	30½	17½	28	10½	8½	9½	12	12	19¼	56	1	1⅜	25	8
48	24	23	24½	1	120	36	20½	32	11½	9¼	11¼	15¼	13½	23⅝	67	1⅛	1½	28	9
56	28	27	28½	1⅛	140	42	27¼	36	12½	10	12¾	17¾	15¼	27¼	72	1¼	2	31	10
64	32	31	32½	1¼	160	48	30	40	13½	11	14½	20	16¾	30¾	80	1¼	2½	34	11
72	36	34½	36	1⅜	176	54	32¾	42	14½	11¾	17¼	22	18½	33⅝	84	1⅜	2¾	37	12
80	40	38½	40	1⅜	192	60	36	44	15¼	12¾	20	24¾	20	37⅝	88	1⅜	3¼	40	12

Frames for Horizontal Engines with more than One Cylinder.



Figs. 87 and 88.

Figs. 87, 88 show a frame for a double cylinder engine with fly-wheel between the two cylinders. A very convenient and cheap type for small self-contained engines. The size of the fly-wheel is limited in this arrangement to about three and a half times the crank radius.



Figs. 89—91.

Figs. 89—91 show a frame for a triple expansion horizontal engine of 20 inches stroke, the arrangement for adjusting the crank shaft brasses by set screws is not recommended, although formerly it was a very common practice.*

Frames for Vertical Engines.

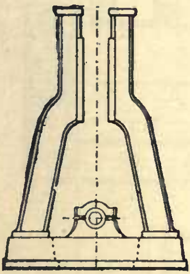


Fig 92.

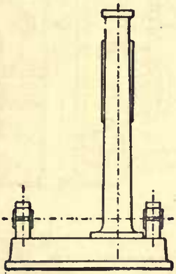


Fig. 93.

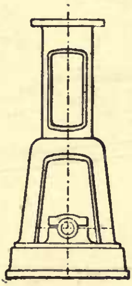


Fig. 94.

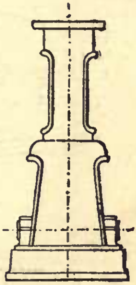


Fig. 95.

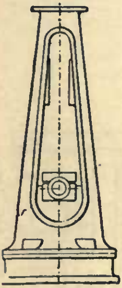


Fig. 96

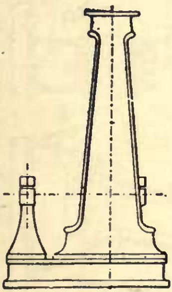


Fig. 97

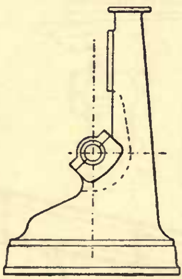


Fig 98

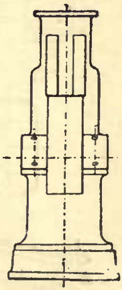


Fig. 99.

The above figs. 92—99 show some early types of vertical engine frames for small engines, a modification of fig. 98 is now used for small quick running engines, and has a wrought-iron stay bar brought from a lug on the cylinder down to the front of the base of the frame.

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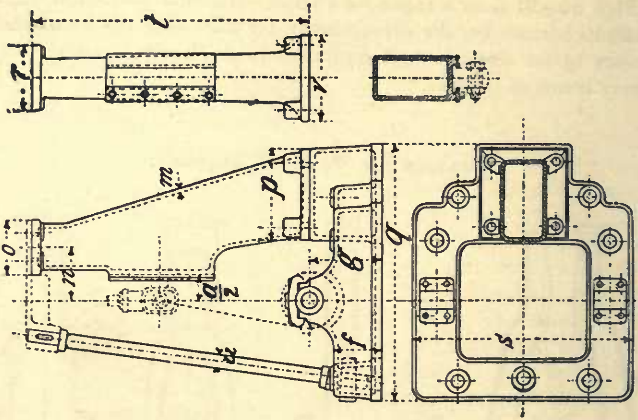
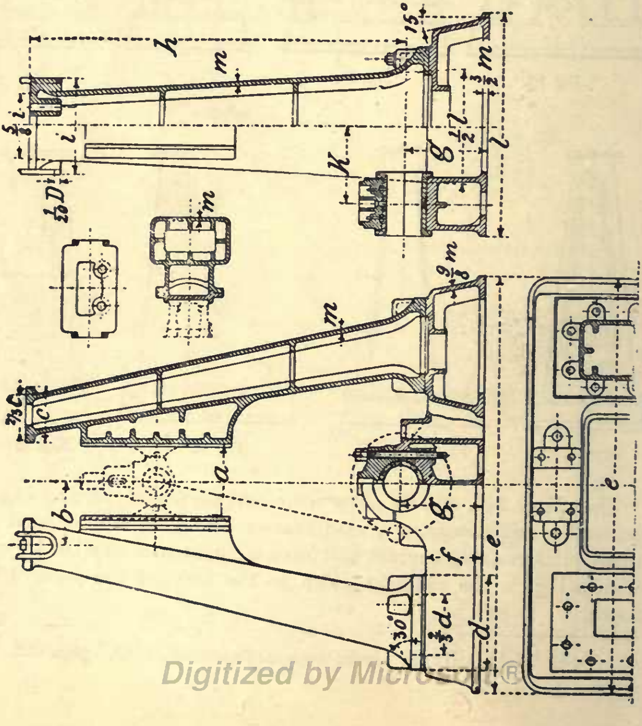
* See "Zeitschrift der Verein deutsch. Ingenieur, 1888," page 226.



Figs. 103—105.



Figs. 100—102.

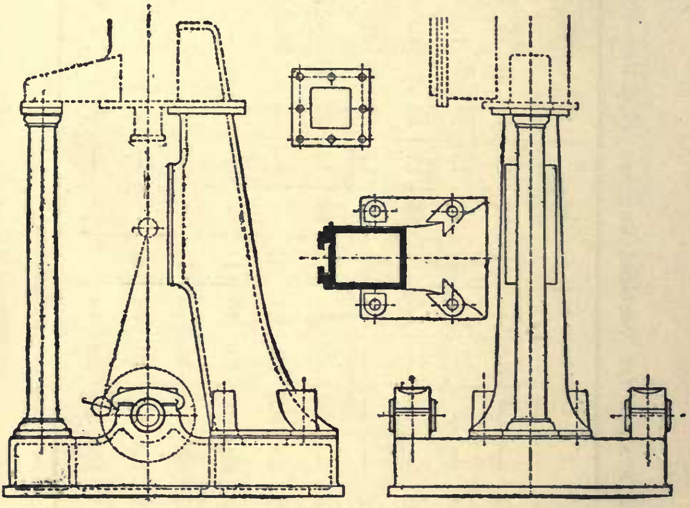
TABLE 8.—**Dimensions of Frames for Vertical Engines** (Figs. 100—105).

H	D	a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	r	s	t	u	v	w
8	6	11	—	—	—	—	11	18	—	—	—	—	$\frac{5}{8}$	11	11	18	48	12	48	42		16	
12	10	13	—	—	—	—	12	19	—	—	—	—	$\frac{11}{16}$	13	13	21	60	15	57	60		2	
16	14	15	—	—	—	—	13	20	—	—	—	—	$\frac{3}{4}$	15	15	25	72	18	66	78		24	
20	18	17	—	—	—	—	14	21	—	—	—	—	$\frac{5}{8}$	17	17	29	84	21	75	96		28	
24	22	19	—	—	—	—	15	22	—	—	—	—	$\frac{7}{8}$	19	19	32	96	24	84	114		32	
28	26	21	17	15	$26\frac{1}{2}$	120	16	23	108	$26\frac{1}{2}$	22	64	$\frac{15}{16}$	—	—	—	—	—	—	—	—	—	—
32	30	23	$18\frac{1}{2}$	16	29	132	$17\frac{1}{2}$	25	120	29	24	70	1	—	—	—	—	—	—	—	—	—	—
36	34	25	20	$17\frac{1}{2}$	$31\frac{1}{2}$	144	19	27	132	$31\frac{1}{2}$	26	76	$1\frac{1}{16}$	—	—	—	—	—	—	—	—	—	—
40	38	27	$21\frac{1}{2}$	19	34	156	$20\frac{1}{2}$	29	144	34	28	82	$1\frac{1}{8}$	—	—	—	—	—	—	—	—	—	—
44	44	29	23	$20\frac{1}{2}$	36	168	22	31	156	$36\frac{1}{2}$	30	88	$1\frac{3}{16}$	—	—	—	—	—	—	—	—	—	—

All Dimensions are given in Inches.

Figs. 100—102 show a vertical engine frame of cast iron for large engines of the marine type, the outer bearing is here on a separate bed outside the fly-wheel when used as a land engine.

Figs. 103—105 show a vertical engine frame of cast iron with wrought-iron stay bar in front, and both crank shaft bearings on one bed thus making the engine self-contained.



Figs. 106—109.

Figs. 106—109 show a frame of the marine type modified for self-contained vertical land engines.

Crank Shaft Bearings.

Fig. 110.

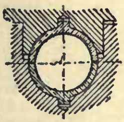


Fig. 111.

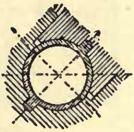


Fig. 112.

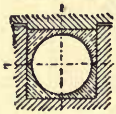
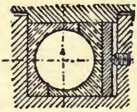


Fig. 113.



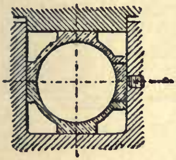


Fig. 114.

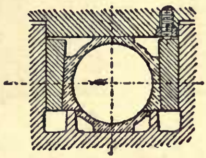


Fig. 115.

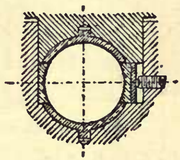


Fig. 116.

Fig. 117.

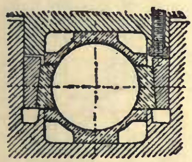


Fig. 118.

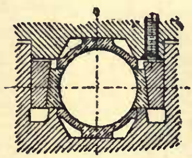
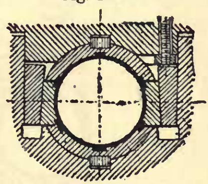


Fig. 119.



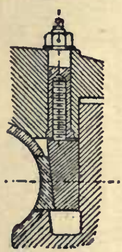


Fig. 120.

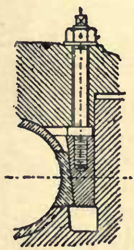


Fig. 121.

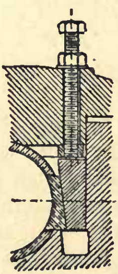


Fig. 122.

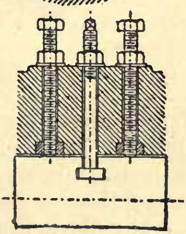
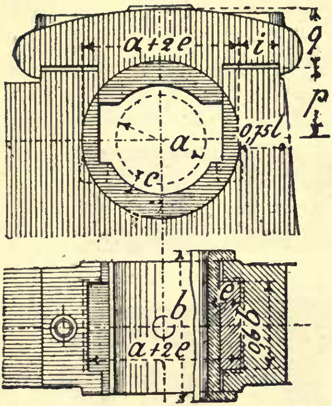


Fig. 123.

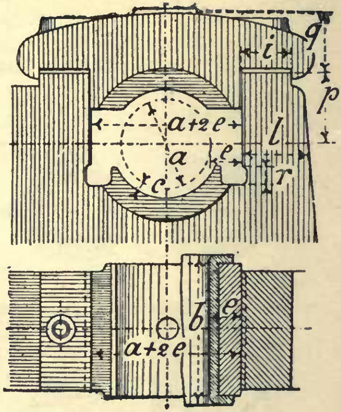
Figs. 110—113.—Bearings with brasses in two parts for small engines.

Figs. 114—116.—Bearings with brasses in three parts for medium-sized engines.

Figs. 117—123.—Bearings with brasses in four parts for large engines.



Figs. 124 and 125.



Figs. 126 and 127.

Figs. 124—127 show two different designs of bearings ; in one, Fig. 124, the adjusting wedge is difficult to fit in, and in fig. 126, if desirable to cover up the wedge at the sides, the flanges of the brasses require to be very deep.

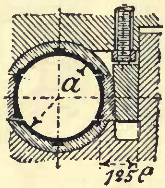
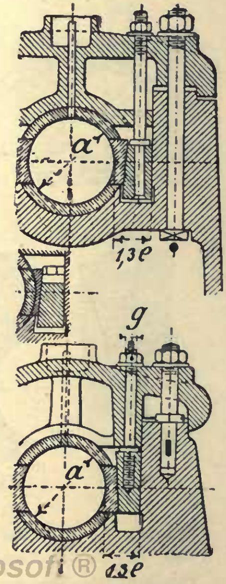


Fig. 128.



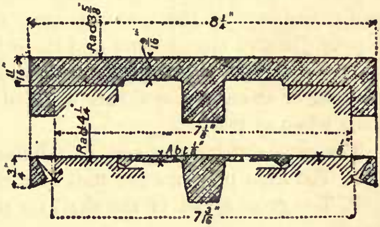
Figs. 129—131.

Fig. 128 shows the bearings of cast iron or cast steel lined with white metal. In large bearings the caps are often cored out to lighten them, as shown in section in figs. 129—131 ; these figures also show two methods of adjusting the brasses endways or horizontally. The outline of the bearing foot is made approximately parabolic, a curve which gives perhaps the neatest appearance.

It will be noted in fig. 128 that the adjusting wedge is shown with the *small end* downwards ; it is generally better to have all adjusting wedges with the *large end* downwards, as then, if the bolt jars loose, the wedge will work down, and not jamb as it might do when put in the other way.

The crank shaft or main bearing, being a very important part of the engine, requires special care both in design and construction. The adjustment of bearings with the brasses in three or more parts by means of wedges placed so as to take up the wear in definite directions, has many advantages if skilfully handled, but may lead to bad results and prove to be far worse, than the ordinary bearing with two-part brasses, if badly handled. On this ground, many firms make even large engines with two-part brasses and no wedge adjustment. It is noticeable that both in marine and locomotive practice the bearings are made as simple as possible without any elaborations.

The part of the brass step which rests on the body casting of the bearing should always be well fitted and have ample surface. Figs. 132, 133, show the section of a crank shaft brass when new and after many years wear, the lower fig. shows how the material of the brass spreads out in the direction of its length by the continual pressure and jarring of ordinary work.*



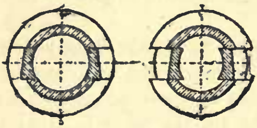
Figs. 132 and 133.

Crank shaft and other similar brasses have a tendency to close in and pinch the shaft as they become warm, to avoid this trouble two-part brasses should be bored out 0·5 per cent. and four-part brasses 0·8 per cent. larger than the diameter of the shaft, attention to this point would often save trouble in ordinary shafting, the brasses of which are often sent out a tight fit on the shaft and have generally to be filed or scraped out until they are a comparatively loose fit.

To bore out and fit up brasses, especially when in many pieces, they are planed up on the flat joint surfaces and soldered together, and can then be bored out, turned or machined where required, and fitted to their places, and afterwards readily separated by heat; sometimes, where possible, they are held together by clamps and steady pins.

* Zeitschrift der Verein deutsches Ingenieur, 1890, page 931.

A suitable mixture for white metal is, according to Kirchweger, composed of 9½ parts copper, 13 parts antimony, 59 parts of pure zinc: the copper should be first melted, then the antimony and zinc added and the mixture well stirred and granulated; 27 parts of this granulated metal should then be melted, and 29½ parts of pure zinc added, the mixture well stirred and run into bars. There are numbers of different metals now used for lining engine and other bearings, one of the first used was Babbit's metal, said to be composed of 1 part copper, 10 parts tin, and 1 part antimony. All these alloys of metals, differing much in their melting points, should be mixed with care, otherwise the easily fusible metals will volatilize, and unsatisfactory results will be obtained. Other standard metals used for engine brasses vary in their composition according to the judgment and experience of the manufacturer, they usually are composed of copper, tin and zinc, copper and tin being the chief ingredients.



Figs. 134 and 135.

The heating of bearings may proceed from the construction, maintenance or attention, or rather want of attention, whilst at work, and even when at rest.

The causes of heating may be tabulated as follows:—

1. Too high pressure per unit of surface P.
2. Too great speed of the shaft for the given pressure, *i.e.*, $p\,v$ too great.
3. Too much pressure from the shaft being not sufficiently stiff for the work, *i.e.*, shaft too elastic.
4. Unsuitable material for the brasses or steps.
5. Insufficient means of lubrication.
6. Blows or jars.

Then let d = diameter of journal in inches.
l = length of journal in inches.
n = speed in revolutions per minute.
P = total pressure in lbs.
p = pressure per square inch in lbs.
v = periphery speed of the journal in feet per second,

then, for ordinary engines:

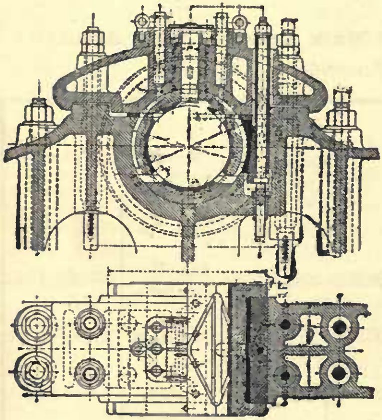
$$p \leqq 270\,;\ p\,v \leqq 840.$$

These figures do not give the limit to which the pressure and speed may be carried, the table gives some examples taken from engines which have been running for some years.

TABLE 9.—**Examples of Main Bearings from Actual Practice.**

Diam. of Steam Cyl. in.	Stroke. in.	Revolution per min.	*d.* in.	*l.* in.	P. lbs.	*p.* lbs. per sq. in.	*p. v.* foot-lbs. p. sec. & sq. in.	Material of Bearings.	Machinery driven by engine.
17·7	27·5	66	6·7	10·8	28200	381	750	Babbited Gun metal	Electric light
23·6	43·3	75	12·6	24·4	37500	122	503	Gun metal	Roller mills
{ 23·6 / 35·4 }	41·3	60	8·6	14·2	46100	378	851	Babbit	Flour mill
{ 23·6 / 39·3 }	39·4	120	9·8	14·5	30800	277	1113	Gun metal	Roller mills
25·6	25·6	150 / 180	9·6	15·7	34800	235	1477 / 1772	Babbit	Roller mills
27·5	39·4	100	10·2	16·5	49500	290	1549	Babbit	Roller mills
31·5	47·2	80	12·2	19·7	52300	217	924	Gun metal	Roller mills
35·4	53·1	80	14·2	22·8	78200	242	1095	Gun metal	Roller mills
49·2	49·2	80 / 90	17·1	23·6	156200	387	2310 / 2599	Babbit	Roller mills

The Lubrication of Bearings is of great importance, and efficient means should always be provided for supply and distribution of the oil or other lubricant. Figs. 136, 137, give a section through the crank-shaft bearing of an engine used for driving a rolling mill. The lubricant is a stiff grease, and is delivered into the bearing by six pistons of gas tube, shown in the fig. 136 in their respective holes in the cap; from these the grease is forced by the weight of the pistons into channels formed at the joints of the different parts of the brass. The engine from which this example is taken has a cylinder of 39 ins. diameter, and the stroke is 55 ins.



Figs. 136 and 137.

The large opening in the centre of the cap can be filled up with lard (*speck*), a material largely used in rolling mills. The adjusting wedge is of the same form as that shown in fig. 127.

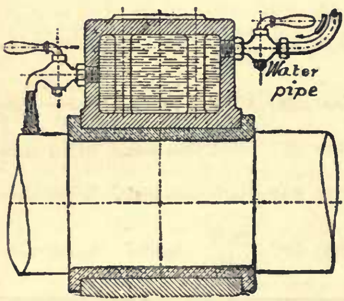


Fig. 138.

Water is occasionally used to cool down hot bearings, applied as shown in fig. 138; with marine engines water is laid on to most of the bearings through small copper pipes from a cistern, each pipe being provided with a tap. Figs. 139—142 give examples of a large crank shaft bearing, with the dimensions given in inches in Table 10 below.

Solidified oil or grease has been successfully introduced for the lubrication of engine and other bearings; it is forced into the oil grooves by a special screw cap to the lubricator, such as those of Stauffer and others.

Fig. 139. Fig. 141.

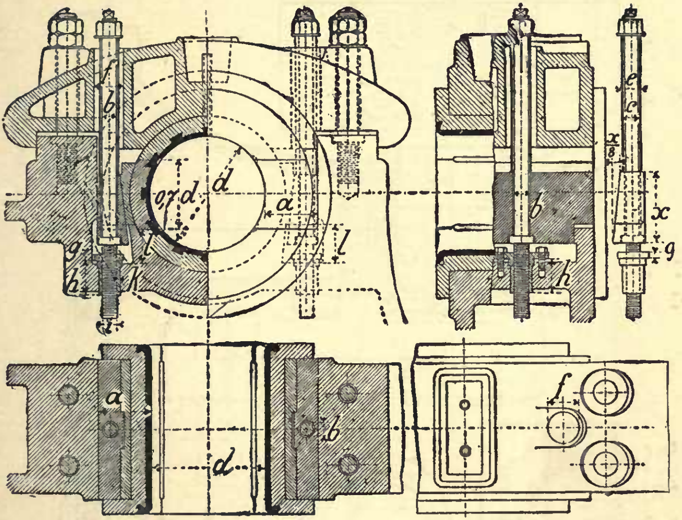


Fig. 140. Fig. 142.

Figs. 139—142.—Bearing with Brasses in Four Parts.

TABLE 10.—**Dimensions of Bearings** (Figs. 139—142).

d	*a*	*b*	*c*	*e*	*f*	*g*	*h*	*i*	*k*	*l*
7	$3\frac{5}{8}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{8}$	$2\frac{1}{8}$	$\frac{7}{8}$	$\frac{1}{4}$	$2\frac{1}{2}$
$7\frac{1}{2}$	$3\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$2\frac{1}{2}$	$\frac{3}{8}$	$2\frac{1}{4}$	1	$\frac{1}{4}$	$2\frac{1}{2}$
8	4	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$2\frac{1}{2}$	$\frac{3}{8}$	$2\frac{3}{8}$	1	$\frac{1}{4}$	$2\frac{3}{4}$
$8\frac{1}{2}$	$4\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{1}{8}$	$\frac{5}{16}$	3
9	$4\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$\frac{1}{2}$	$2\frac{3}{4}$	$1\frac{1}{8}$	$\frac{5}{16}$	$3\frac{1}{8}$
10	$4\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$3\frac{1}{8}$	$\frac{1}{2}$	3	$1\frac{1}{4}$	$\frac{3}{8}$	$3\frac{1}{4}$
11	$5\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$2\frac{3}{8}$	$3\frac{1}{8}$	$\frac{1}{2}$	$3\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{8}$	$3\frac{1}{2}$
12	$5\frac{3}{4}$	$1\frac{3}{8}$	2	$2\frac{3}{8}$	$3\frac{1}{2}$	$\frac{1}{2}$	$3\frac{3}{8}$	$1\frac{3}{8}$	$\frac{7}{16}$	$3\frac{3}{4}$
13	$6\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{3}{8}$	$3\frac{3}{4}$	$\frac{5}{8}$	$3\frac{3}{4}$	$1\frac{1}{2}$	$\frac{1}{2}$	4
14	$6\frac{1}{2}$	$1\frac{1}{2}$	2	3	$3\frac{3}{4}$	$\frac{5}{8}$	4	$1\frac{1}{2}$	$\frac{1}{2}$	$4\frac{1}{4}$
15	7	$1\frac{3}{4}$	$2\frac{1}{2}$	3	$4\frac{3}{8}$	$\frac{5}{8}$	$4\frac{1}{4}$	$1\frac{3}{4}$	$\frac{5}{8}$	$4\frac{1}{2}$
16	$7\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{2}$	3	$4\frac{3}{8}$	$\frac{5}{8}$	$4\frac{1}{2}$	$1\frac{3}{4}$	$\frac{5}{8}$	$4\frac{3}{4}$

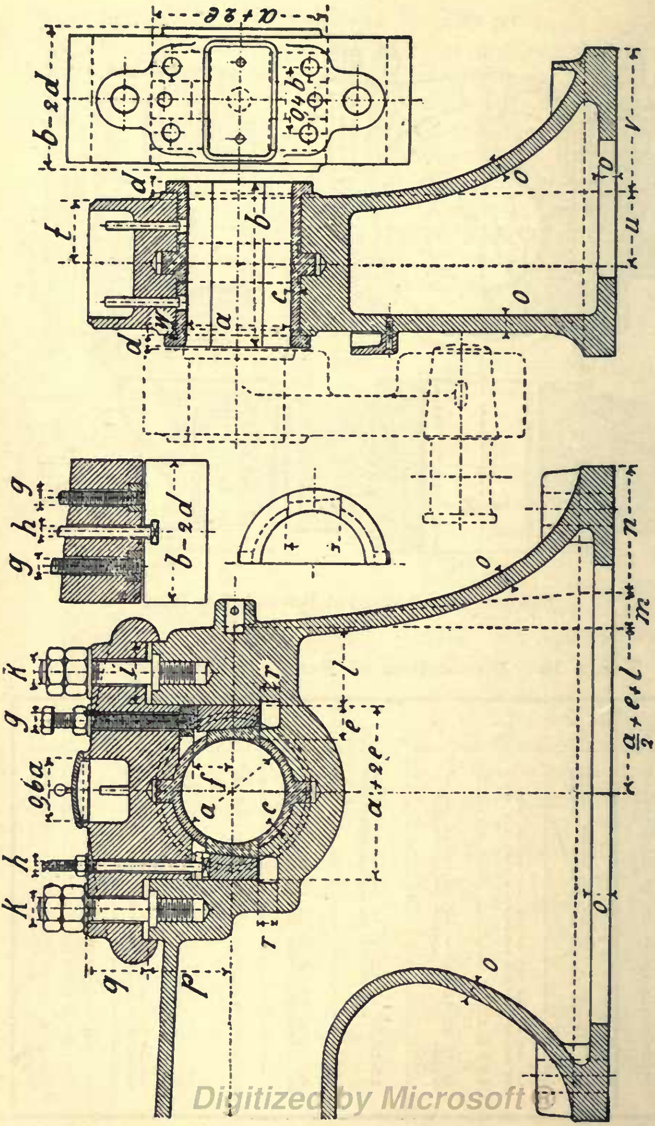


Fig. 143.

Figs. 144 and 145.

TABLE 11.—**Dimensions of Crank Shaft Bearings** (Figs. 143—145).

Engine.		Brasses.									Bolts.												
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	No.	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*t*	*u*	*v*	*w*
16	8	$3\frac{5}{8}$	$5\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$2\frac{1}{4}$	2	$\frac{7}{8}$	$2\frac{5}{8}$	—	$4\frac{1}{4}$	$\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{4}$	$\frac{7}{8}$	2	$2\frac{1}{2}$	—	$\frac{1}{4}$
20	10	$4\frac{1}{2}$	$7\frac{1}{4}$	$\frac{7}{16}$	$\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$2\frac{1}{2}$	2	1	3	$1\frac{5}{8}$	5	$\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{5}{8}$	1	$2\frac{3}{4}$	$3\frac{1}{4}$	$5\frac{3}{4}$	$\frac{1}{4}$
24	12	$5\frac{1}{4}$	$8\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{2}$	2	$\frac{5}{8}$	$\frac{1}{2}$	$2\frac{3}{4}$	2	$1\frac{1}{8}$	$3\frac{5}{8}$	$1\frac{7}{8}$	$6\frac{1}{8}$	$\frac{7}{8}$	4	3	$1\frac{1}{4}$	$3\frac{1}{8}$	$3\frac{3}{4}$	$7\frac{1}{4}$	$\frac{1}{4}$
28	14	$6\frac{3}{8}$	$9\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$1\frac{3}{4}$	$2\frac{3}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$3\frac{1}{4}$	2	$1\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{3}{16}$	$6\frac{5}{8}$	1	$4\frac{3}{4}$	$3\frac{3}{8}$	$1\frac{3}{8}$	$3\frac{5}{8}$	$4\frac{1}{4}$	$7\frac{3}{4}$	$\frac{1}{4}$
32	16	7	$10\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$3\frac{5}{8}$	2	$1\frac{1}{4}$	$4\frac{3}{4}$	$2\frac{3}{8}$	$7\frac{5}{8}$	1	$5\frac{5}{8}$	$3\frac{3}{4}$	$1\frac{1}{2}$	$4\frac{1}{4}$	$4\frac{3}{4}$	$8\frac{1}{4}$	$\frac{1}{4}$
36	18	$7\frac{5}{8}$	12	$\frac{5}{8}$	$\frac{3}{4}$	$2\frac{1}{8}$	$3\frac{1}{4}$	$\frac{7}{8}$	$\frac{5}{8}$	4	4	$1\frac{1}{4}$	$5\frac{3}{8}$	$2\frac{7}{16}$	$7\frac{3}{4}$	$1\frac{1}{8}$	$6\frac{1}{2}$	$4\frac{1}{4}$	$1\frac{3}{4}$	$4\frac{3}{4}$	$5\frac{1}{2}$	$9\frac{1}{2}$	$\frac{3}{8}$
40	20	$8\frac{3}{4}$	$13\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$2\frac{3}{8}$	$3\frac{5}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$4\frac{5}{8}$	4	$1\frac{3}{8}$	6	$2\frac{7}{8}$	$8\frac{3}{8}$	$1\frac{1}{4}$	$7\frac{1}{4}$	$4\frac{5}{8}$	2	$5\frac{1}{4}$	6	10	$\frac{3}{8}$
44	22	10	$14\frac{3}{4}$	$\frac{13}{16}$	1	$2\frac{1}{2}$	4	$\frac{7}{8}$	$\frac{5}{8}$	$5\frac{1}{4}$	4	$1\frac{3}{8}$	$6\frac{3}{4}$	3	$8\frac{3}{4}$	$1\frac{1}{4}$	$7\frac{3}{4}$	5	$2\frac{1}{4}$	$5\frac{3}{4}$	$6\frac{1}{2}$	$11\frac{1}{2}$	$\frac{3}{8}$

All the Dimensions are given in Inches.

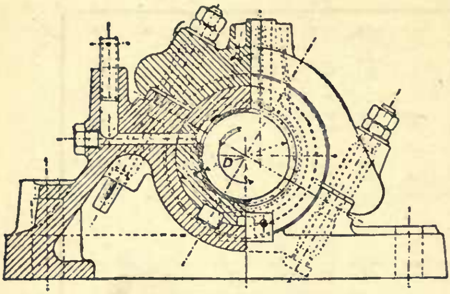


Fig. 146.

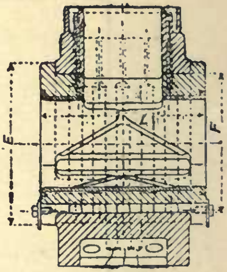


Fig. 147

Figs. 146, 147 show an example of inclined main bearing for high speed rope pulley, and is arranged for water circulation.

Crank Shafts, Cranks, and Crank Discs.

Crank shafts were usually made of wrought iron, but now almost always of mild steel. For very large marine engines they are built up of steel, often Whitworth's compressed steel, and are made hollow to save weight; the diameter of the hollow being about ·6 of the external diameter of the shaft. Fig. 148 shows a built-up hollow crank.

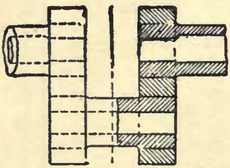


Fig. 148.

In high speed engines the weight of the connecting rod and crank should be balanced.

Let W_1 = the necessary balance weight.
R = radius of the centre of gravity of the balance weight.
W_2 = weight of the crank pin plus half the weight of the connecting rod.
r = the radius of the crank.
W_3 = weight of the piston, piston rod and crosshead.

Then for horizontal engines

$$W_1 = W_2 \frac{r}{R};$$

for vertical engines

$$W_1 = \tfrac{2}{3}(W_2 + W_3)\frac{r}{R};$$

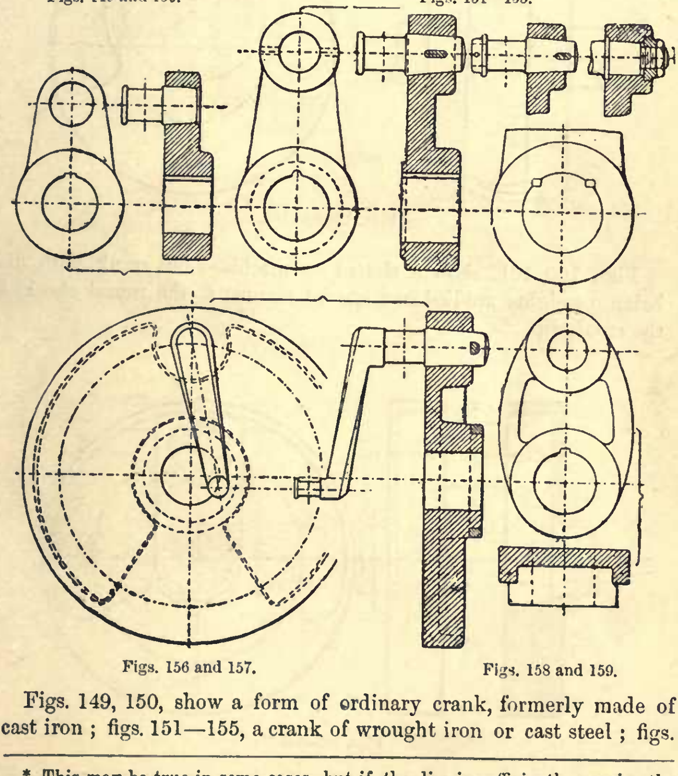
for locomotives it may be taken that

$$W_1 = W_2 \frac{r}{R}.$$

The balance weight should act in the same plane in which the parts to be balanced move ; hence it is wrong to balance the crank, connecting rod, piston, and crosshead by applying the balance weight to the fly-wheel. The employment of crank discs, figs. 160, 163, although somewhat easy to balance, is often objectionable on account of their acting as resonant bodies, and tend to magnify the noise caused by any knock in the engine.* A balanced crank disc is shown in figs. 156, 157.

Cranks, Crank Discs, Slotted out and Bent Crank Shafts.

Figs. 149 and 150. Figs. 151—155.



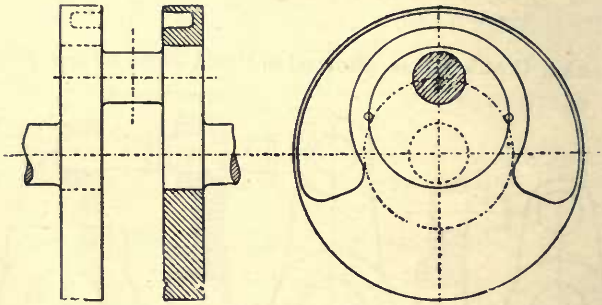
Figs. 156 and 157. Figs. 158 and 159.

Figs. 149, 150, show a form of ordinary crank, formerly made of cast iron ; figs. 151—155, a crank of wrought iron or cast steel ; figs.

* This may be true in some cases, but if the disc is sufficiently massive the effect is not very noticeable, especially if the balance weight is applied as is shown in figs. 182, 184.—Ed.

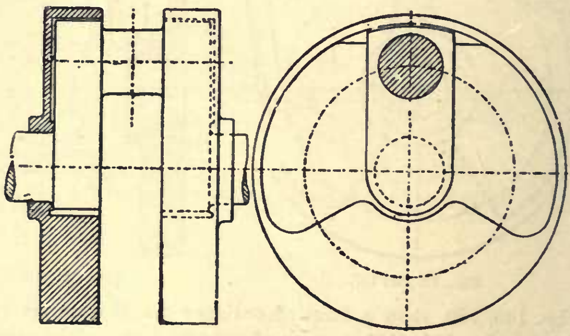
156, 157, a crank disc of cast iron; figs. 158, 159, a crank of either cast iron or cast steel, the former material is not now much used, except in small cheap engines.

Crank Shafts with Balance Weights.



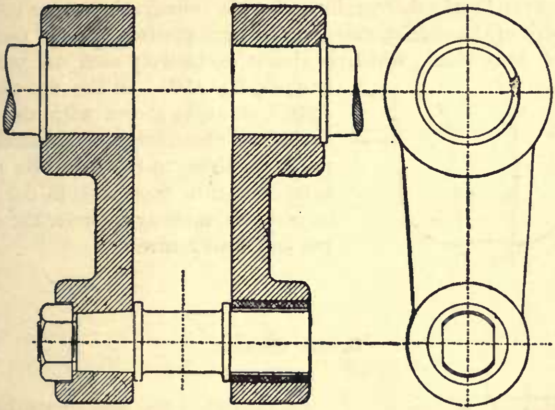
Figs. 160 and 161.

Figs. 160, 161, show a slotted or machined-out crank, with disc balance weights applied in a special manner to the round cheeks of the crank dip.



Figs. 162 and 163.

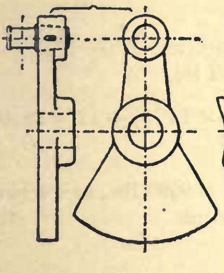
Figs. 162, 163, show an arrangement for applying the balance weights to the cheeks of an ordinary slotted-out crank.



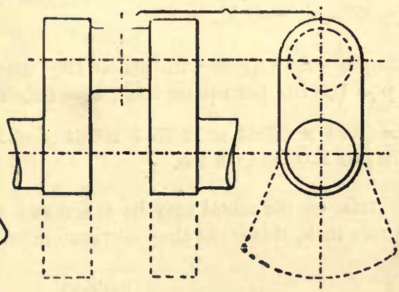
Figs. 164 and 165.

Figs. 164, 165, show a wrought-iron or steel crank, with the crank pin free in one side of the crank; this type is much used for paddle engines, the engine crank shaft being separate from the paddle-wheel shafts, the latter being driven by the crank pin as shown, the left crank being on the engine shaft, the right-hand one on the paddle-wheel shaft.

Cranks, Crank Discs, Slotted-out and Bent Crank Shafts.



Figs. 166 and 167.



Figs. 168 and 169.

Figs. 166, 167 show a form of crank with balance weight in one piece; figs. 168, 169, a slotted-out crank a form much used in "undertype" and other engines where it is desirable to save room in the width of the engine, this form being narrower than the common form of bent crank, which is almost exclusively used on portable engines, fig. 170. In the fig. 169 the slotted crank is shown with the sides shaped off concentric with the shaft and pin respectively at the two ends, but it is usual to turn these ends in the lathe on a centre, midway between the crank pin and shaft centres.

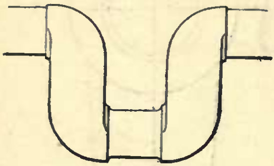


Fig. 170.

Crank Shafts.

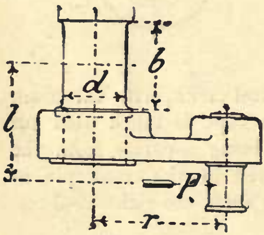


Fig. 171.

Particulars of the neck or journal.

In fig. 171 :—

P = Pressure on piston in lbs. ;
$M_b = Pl$ = the bending moment in inch or foot lbs. ;
$M_d = Pr$ = the turning moment in inch or foot lbs. ;
k = the stress on the material in pounds per square inch ;

so the ideal bending moment (if, as usually the case, $M_b < M_d$) $(M_b)_i = \cdot 625\ M_b + \cdot 6\ M_d = Wk$.

Example.—An engine with piston $15\frac{3}{4}''$ diameter, stroke $27\frac{1}{2}''$, $l = 13\frac{3}{4}''$, $p = 103$ lbs. per square inch, $P = 20064$ lbs.

$(M_b)_i = (\cdot 625 \times 20064 \times 13\cdot 75 + 20064 \times \cdot 6 \times 13\cdot 75) \div 12 = 28400$ foot lbs., or 340600 inch lbs.

The stress on the metal may be taken as $k = 9960$ lbs., or 4·4 tons per square inch, this gives the resistance moment

$$W = \cdot 1\ d^3 = \frac{340600}{9960} = 34\cdot 0$$

$$d = 6\cdot 9 \text{ inches.}$$

Material: Wrought Iron or Mild Steel.

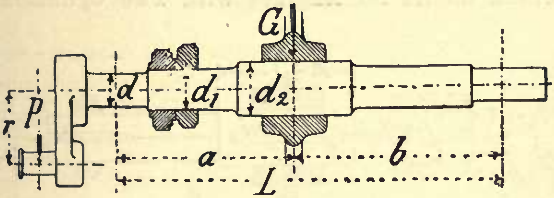


Fig. 172.

The diameter of the neck or journal being d, the diameters at the other parts of the shaft in fig. 172, $d_1 = 1{\cdot}15\, d$, $d_2 = 1{\cdot}4\, d$.

With usual proportions the stress on the metal at d_2 may be taken as from 2,800 to 5,600 lbs., or $1\frac{1}{4}$ to $2\frac{1}{2}$ tons per square inch

If G = the weight of the fly-wheel in pounds, the ideal bending moment for d_2, taking as before, page 50, $M_b < M_d$,

$$(M_b)_i = Wk = 0{\cdot}625\, G \frac{ab}{L} + 0{\cdot}6\, Pr.$$

Cranks and Crank Pins.

Wrought-iron or cast steel cranks (fig. 173).

Length of boss $b = {\cdot}9$ to $1{\cdot}3\, d$;
Thickness of metal $W = {\cdot}4$ to ${\cdot}5\, d$;

The crank web will be exposed to a bending and twisting stress,

$$M_b = Pr\,;\; M_d = Pa.$$

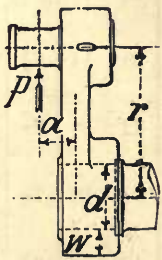


Fig. 173.

The diameter of the boss for the crank pin will be about $\frac{2}{3}$ of the larger boss for the shaft. The bore of the boss is made from $\frac{1}{1000}$ to $\frac{1}{2000}$ smaller than the end of the crank shaft, put on hot, in workshop language "shrunk on," and fitted with one or two keys.

Crank Shafts for Engines with Two Cylinders.

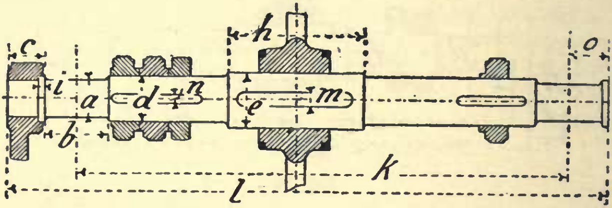


Fig. 174.

TABLE 12.—**Dimensions of Crank Shafts** (Fig. 174).

Diam. of Piston and Stroke.		Journal or Neck.											
H	D	*a*	*b*	*c*	*d*	*e*	*h*	*i*	*k*	*l*	*m*	*n*	*o*
20	10	$4\frac{1}{2}$	$7\frac{1}{4}$	$3\frac{15}{16}$	$5\frac{1}{4}$	$5\frac{1}{4}$	—	$\frac{1}{4}$	4′ 4″	5′ $3\frac{3}{4}$″	$1\frac{3}{8}$	$\frac{5}{8}$	$4\frac{1}{4}$
24	12	$5\frac{1}{4}$	$8\frac{1}{2}$	5	6	6	—	$\frac{1}{4}$	4′ 10″	6′ $0\frac{1}{4}$″	$1\frac{5}{8}$	$\frac{5}{8}$	5
28	14	$6\frac{3}{8}$	$9\frac{1}{2}$	$5\frac{3}{4}$	$7\frac{3}{8}$	$8\frac{3}{8}$	1′ 8″	$\frac{3}{8}$	5′ 4″	6′ $8\frac{1}{8}$″	2	$\frac{3}{4}$	$5\frac{5}{8}$
32	16	7	$10\frac{3}{4}$	$6\frac{5}{8}$	8	$9\frac{3}{8}$	2′ 0″	$\frac{3}{8}$	5′ 10″	7′ $4\frac{3}{4}$″	$2\frac{1}{4}$	$\frac{3}{4}$	$6\frac{3}{4}$
36	18	$7\frac{5}{8}$	12	$7\frac{3}{8}$	$8\frac{5}{8}$	$10\frac{5}{8}$	2′ 4″	$\frac{1}{2}$	6′ 4″	8′ $0\frac{3}{8}$″	$2\frac{5}{8}$	$\frac{3}{4}$	7
40	20	$8\frac{3}{4}$	$13\frac{1}{4}$	$8\frac{1}{8}$	10	12	2′ 8″	$\frac{1}{2}$	7′ 0″	8′ $10\frac{1}{2}$″	$2\frac{3}{4}$	$\frac{3}{4}$	$7\frac{3}{4}$
44	22	10	$14\frac{3}{4}$	$8\frac{3}{8}$	$11\frac{1}{4}$	$13\frac{1}{4}$	3′ 0″	$\frac{1}{2}$	7′ 8″	9′ $8\frac{3}{4}$″	3	$\frac{7}{8}$	$8\frac{5}{8}$

All Dimensions are given in Feet and Inches.

Crank Pins.

These are invariably of mild steel.

d = diameter of pin in inches.
b = length of pin in inches.
P = maximum pressure on piston in lbs.,

then $M_b = P\,\frac{b}{2} = Wk = \cdot 1\, d^3k.$

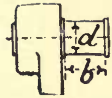


Fig. 175.

The stress k may be taken as from 6,720 to 11,200 lbs. or 3 to 5 tons per square inch. In order to avoid heating, the crank pin is usually made larger than given by the above formula. Let $p = \frac{P}{db}$ = the pressure per square inch in lbs. on the surface of the crank pin and $v = \frac{d\pi n}{12}$ = speed of crank pin surface in feet per ~~minute~~ sec, then for ordinary workmanship $p \leqq 1{,}130$ lbs. per square inch $vp \leqq 845$. With very high class workmanship we may go higher : see table on page 63.

Methods of fixing crank pins into the crank arm.—A very usual method is with cone and key (fig. 177), the pin being shrunk in, the taper of the cone being about 1 in 32. There is always difficulty in boring a tapered hole and making a pin to be an exact fit, an easier and better method of fixing crank pins is by making the pin with a collar and turning the shank parallel to fit a parallel hole in the crank arm, the pin can, when small, be secured by riveting the end over or by a key transversely through crank arm and pin, but in all cases the pin should be shrunk in, or squeezed in with a hydraulic press. A special method of fixing the pin is shown in fig. 176, which also shows an elaborate system of lubrication.

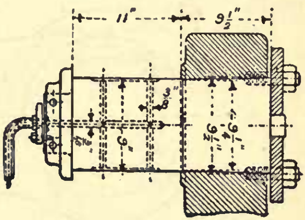
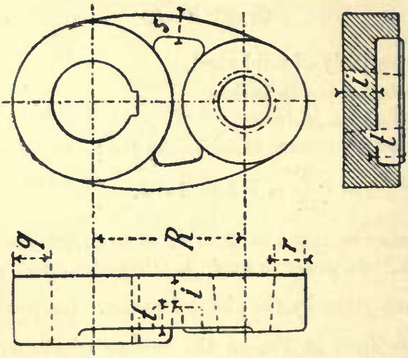
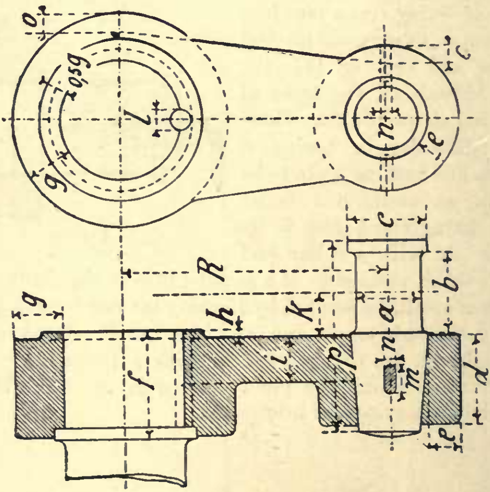


Fig. 176.



Figs. 179—181.



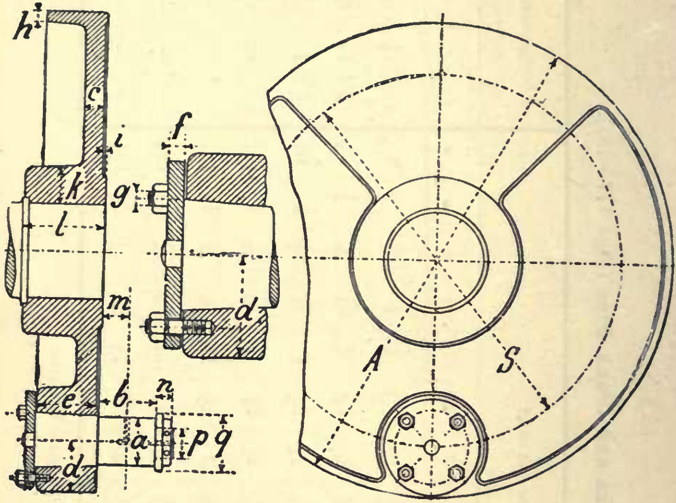
Figs. 177 and 178.

TABLE 13.—**Dimensions of Cranks and Crank Pins** (Figs. 177—181).

Diameter of Piston, Stroke and Radius of Crank.			Wrought iron or cast steel.															Cast iron.			
H	D	R	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*
20	10	10	$2\frac{5}{8}$	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$	$1\frac{1}{4}$	$3\frac{15}{16}$	2	$\frac{5}{16}$	$1\frac{3}{4}$	$1\frac{11}{16}$	$\frac{3}{4}$	1	$\frac{3}{8}$	$\frac{3}{4}$	7	2	$1\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{4}$
24	12	12	$2\frac{3}{4}$	$3\frac{5}{8}$	$3\frac{5}{8}$	4	$1\frac{3}{8}$	5	$2\frac{3}{8}$	$\frac{3}{8}$	2	$1\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$\frac{7}{16}$	$\frac{7}{8}$	$8\frac{1}{4}$	$2\frac{3}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$
28	14	14	$3\frac{1}{4}$	$4\frac{1}{4}$	4	$4\frac{3}{8}$	$1\frac{1}{2}$	$5\frac{3}{4}$	$2\frac{5}{8}$	$\frac{7}{16}$	$2\frac{1}{4}$	$2\frac{3}{16}$	1	$1\frac{3}{8}$	$\frac{1}{2}$	1	$9\frac{1}{2}$	$2\frac{5}{8}$	2	$1\frac{3}{8}$	$1\frac{1}{2}$
32	16	16	$3\frac{3}{4}$	$4\frac{5}{8}$	$4\frac{5}{8}$	$4\frac{3}{4}$	$1\frac{3}{4}$	$6\frac{5}{8}$	$2\frac{7}{8}$	$\frac{7}{16}$	$2\frac{5}{8}$	$2\frac{3}{8}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$\frac{9}{16}$	1	$10\frac{1}{4}$	3	$2\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$
36	18	18	$4\frac{1}{4}$	$5\frac{1}{4}$	5	$5\frac{1}{4}$	2	$7\frac{3}{8}$	$3\frac{1}{4}$	$\frac{1}{2}$	3	$2\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$\frac{5}{8}$	$1\frac{1}{8}$	$11\frac{1}{4}$	—	—	—	—
40	20	20	$4\frac{3}{4}$	$5\frac{3}{4}$	$5\frac{3}{4}$	$5\frac{3}{4}$	$2\frac{1}{4}$	$8\frac{1}{8}$	$3\frac{1}{2}$	$\frac{1}{2}$	$3\frac{3}{8}$	3	$1\frac{1}{4}$	$1\frac{3}{4}$	$\frac{11}{16}$	$1\frac{1}{4}$	$12\frac{1}{2}$	—	—	—	—
44	22	22	$5\frac{1}{4}$	$6\frac{3}{8}$	$6\frac{1}{4}$	$6\frac{3}{8}$	$2\frac{3}{8}$	$8\frac{3}{8}$	$3\frac{3}{4}$	$\frac{9}{16}$	4	$3\frac{5}{16}$	$1\frac{3}{8}$	2	$\frac{3}{4}$	$1\frac{1}{4}$	$13\frac{1}{2}$	—	—	—	—

All Dimensions are given in Inches.

Crank Discs of Cast Iron.



Figs. 182—184.

Table 14.—Dimensions of Crank Discs (Figs. 182—184).

Diameter of piston and stroke.			Crank Pin.						Bolts.									
H	D	A	a	b	c	d	e	f	No.	g	h	i	k	l	m	n	p	q
16	8	22	$2\frac{1}{2}$	$3\frac{1}{2}$	$1\frac{1}{8}$	3	3	$\frac{3}{8}$	3	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{8}$	2	4	$1\frac{3}{4}$	1	$1\frac{3}{4}$	3
20	10	26	$2\frac{3}{4}$	4	$1\frac{1}{4}$	3	$3\frac{1}{2}$	$\frac{7}{16}$	3	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{4}$	$2\frac{1}{4}$	5	$1\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{7}{8}$	$3\frac{1}{2}$
24	12	31	$3\frac{1}{4}$	$4\frac{1}{2}$	$1\frac{3}{8}$	$3\frac{1}{2}$	4	$\frac{1}{2}$	3	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{1}{4}$	$2\frac{1}{2}$	6	2	$1\frac{1}{4}$	$2\frac{1}{4}$	4
28	14	36	$3\frac{3}{4}$	5	$1\frac{5}{8}$	4	$4\frac{1}{2}$	$\frac{9}{16}$	4	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{8}$	$2\frac{3}{4}$	7	$2\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{5}{8}$	$4\frac{1}{2}$
32	16	41	$4\frac{1}{8}$	$5\frac{1}{2}$	$1\frac{3}{4}$	$4\frac{1}{2}$	5	$\frac{5}{8}$	4	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	3	$7\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{1}{2}$	$2\frac{7}{8}$	5
36	18	46	$4\frac{3}{4}$	6	2	5	$5\frac{1}{2}$	$\frac{11}{16}$	4	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{2}$	$3\frac{1}{2}$	8	$2\frac{1}{2}$	$1\frac{3}{4}$	$3\frac{1}{4}$	$5\frac{1}{2}$
40	20	51	5	$6\frac{1}{2}$	$2\frac{1}{8}$	$5\frac{1}{2}$	6	$\frac{3}{4}$	4	$\frac{3}{4}$	1	$\frac{1}{2}$	4	$8\frac{1}{2}$	$2\frac{3}{4}$	2	$3\frac{1}{2}$	6

Preparation of Crank Pins in Special Cases.

Crank pins are sometimes case-hardened and ground up true in a grinding machine. This has been done in America, and also in Germany and other countries. The journals in the Allen engine by Whitworth were ground up perfectly true; no doubt this is conducive to cool running, but with crank pins the heating is not always caused by want of truth in the form of the pin, but probably from the pin not being perfectly in line with the crank shaft.

Lubrication of Crank Pins.

In small engines the necessary lubrication is obtained from an oil cup in the connecting rod end (fig. 185).

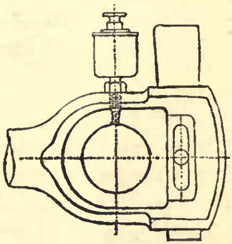


Fig. 185.

An arrangement for oiling the crank pin whilst the engine is running is shown in fig. 186. The oil cup is supported on the guard rail of the engine. The oil from the cup runs down a tube into a hollow ring at the end of a tube fixed into the crank pin. This hollow ring receives the oil which by centrifugal force makes its way through the holes in the crank pin, and thus supplies the pin

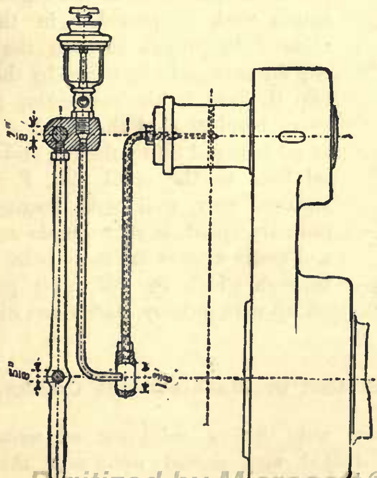


Fig. 186.

and connecting rod end with oil. Two other methods of lubricating with special grease are shown in figs. 187, 188, one with the "Stauffer"

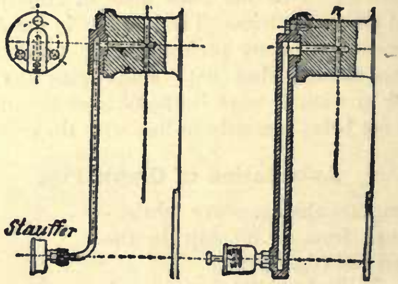


Fig 187. Fig. 188.

grease cup. The cap of this cup is screwed down by hand, and forces the grease along the tube to the pin.

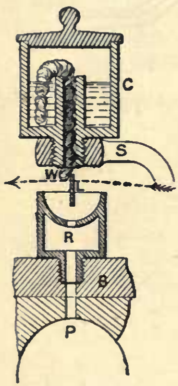


Fig. 189.

Another method much used in large engines is shown in fig. 189. The oil cup, C, is supported on a wrought-iron standard carried up from the main bearing pedestal, so as to stand just over the crank pin when at the highest position. A cotton wick is provided in the cup, and is allowed to project through the bottom of the cup far enough to be wiped by the lip, L, in the cup R, fixed to the connecting rod end, B; at every revolution of the crank a small quantity of oil is wiped off by the lip, and falls down the oil hole to the crank pin, P. This method answers very well with engines running at ordinary speed, is very simple and inexpensive, and easily kept clean, there being no long pipes through which the oil must pass, and which frequently get stopped up with gummy matter and dirt.

Slotted or Machined-out Cranks.

Formerly these were forged solid out of wrought iron, and then the dip drilled and slotted out; now they are usually made of mild steel, and machined out to the required form and dimensions. Small cranks are occasionally cast to form from a pattern in steel.*

Fig. 190. Fig. 192.

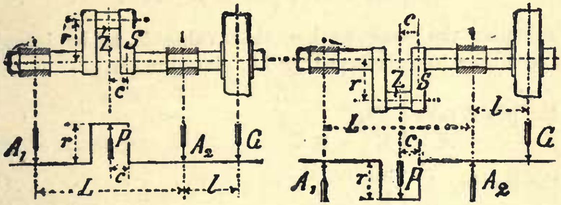


Fig. 191. Fig. 193.

Let P = the pressure on the crank pin in lbs.

G = the weight of the fly-wheel in lbs.

(a.) Single Crank Shaft as in the above figs. 190, 193.

The maximum supported pressure is first to be calculated.

In figs. 190, 191—

$$A_1 = \frac{P \cdot \frac{L}{2} + G \cdot l}{L} = \frac{P}{2} + G\frac{l}{L} \quad . \quad . \quad (3)$$

In figs. 192, 193—

$$A_2 = \frac{P \cdot \frac{L}{2} + G\,(L + l)}{L} = \frac{P}{2} + G + G\frac{l}{L} \quad . \quad (4)$$

For calculating the crank cheeks or dips, S, imagine the right-hand cheek as rigidly fixed; then in figs. 190, 191—

$$\text{I.}\begin{cases} M_b = A_1\left(\frac{L}{2} + c\right) - Pc \quad . \quad . \quad . \quad (5) \\ M_d = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad (6) \end{cases}$$

* There is some risk of confusion in speaking of "mild steel," "cast steel," and "steel castings," or "cast in steel." Generally speaking "mild steel" means such steels as are worked up from open hearth or Bessemer ingots into bars, plates, angles, &c. Such steel is not supposed to harden perceptibly when heated and quenched in cold water. "Cast steel" is a somewhat vague term, but may be generally taken to mean steel worked up from ingots made from melted shear steel, and is usually highly carbonized, and capable of being hardened in the ordinary way. One kind of this steel is known as tool steel. "Cast in steel" or "steel castings" means that the object is actually cast in form from melted steel poured into a mould. Steel castings of the very best quality are often advertised as "crucible steel," meaning obviously that the steel is melted in crucibles, and poured from them into the mould. Other more direct processes than this are used for large steel castings.

When turned 90°—

$$\text{II.}\begin{cases} M_b = Pr \quad . \quad . \quad . \quad . \quad . \quad . \quad . & (7) \\ M_d = A_1\left(\frac{L}{2} + c\right) - Pc \quad . \quad . \quad . & (8) \end{cases}$$

To calculate the cross section, that value is to be introduced which, according to equations I. or II., gives the $(M_b)_i$.

$$\text{For the pin Z take}\begin{cases} M_b = A_1 \frac{L}{2} \quad . \quad . \quad . \quad . \quad . \quad . & (9) \\ M_d = 0. \end{cases}$$

$$\text{At the bearing } A_2 \begin{cases} M_b = Gb \quad . \quad . \quad . \quad . \quad . & (10) \\ M_d = Pr \quad . \quad . \quad . \quad . \quad . \quad . & (11) \end{cases}$$

M_b and M_d may be combined into the ideal bending moment $(M_b)_i$ according to the formula given on page 50.

Example.—Calculation from above formulæ for the shaft of an engine with piston 8″ diameter, and stroke 11¾″, working pressure 75 lbs. per square inch. The engine frame is shown on page 20, fig. 54.

$$P = 3306 \text{ lbs.}, \; G = 880 \text{ lbs.}, \; r = \frac{11{\cdot}75}{2} = 5{\cdot}875 \text{ inches.}$$

$$L = 29{\cdot}5 \text{ inches}, \; l = 8{\cdot}67 \text{ inches}, \; c = 2{\cdot}95 \text{ inches.}$$

$$A_1 = \frac{3306}{2} + 880\,\frac{8{\cdot}67}{29{\cdot}5} = 1912 \text{ lbs.}$$

$$A_2 = \frac{3306}{2} + 880 + 880\,\frac{8{\cdot}67}{29{\cdot}5} = 2792 \text{ lbs.}$$

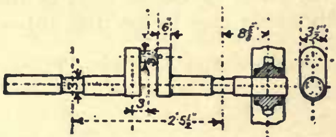


Fig. 194.

For the crank cheeks or dips—

$$\text{II.}\begin{cases} M_b = 3306 \times 5{\cdot}875 = 19423 \text{ inch-lbs.,} \\ M_d = 1913\,(14{\cdot}76 + 2{\cdot}95) - 3306 \times 2{\cdot}95 = 24110 \text{ inch-lbs.;} \end{cases}$$

then $(M_b)_i = {\cdot}625 \times 19423 + {\cdot}6 \times 24126 = 12139 + 14466 =$ 26605 inch-lbs.

The cross-section of the crank cheek $= 3{\cdot}5'' \times 2{\cdot}4'' = 8{\cdot}4$ sq. ins. the value of the smallest cross-section of metal.

$$\frac{26605}{8{\cdot}4} = 3170 \text{ inch-lbs. per square inch nearly.}$$

For the bearing A_2, figs. 190—193,

$$(M_b)_i = \cdot 625 \times 880 \times 8\cdot 625 + \cdot 6 \times 3306 \times 5\cdot 9 =$$
$$4744 + 11706 = 16450 \text{ inch-lbs.}$$

Cross-section = area of 2·95 inches = 6·83 sq. ins.

$$\frac{16450}{6\cdot 83} = 2406 \text{ inch-lbs. per square inch.}$$

For the crank pin, Z, $M_b = 1913 \times \dfrac{29\cdot 5}{2} = 28216$ inch-lbs.

Cross-section = area of 3·14 = 7·74

$$\frac{28216}{7\cdot 74} = 3644 \text{ inch-lbs. per square inch.}$$

(b.) Double Crank Shafts.

Also here the first point to be determined is the pressure on the bearings; for simple calculation the pressure on the two pistons may be taken as equal.

Fig. 195.

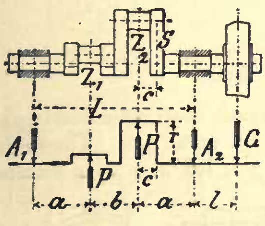


Fig. 196.

Fig. 197.

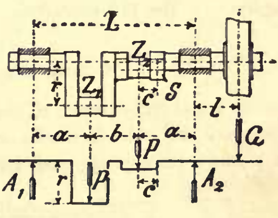


Fig. 198.

In figs. 195, 196,

$$A_1 = \frac{Pa + P(a + b) + Gl}{L} = P + G\frac{l}{L} \quad . \quad . \quad . \quad (12)$$

In figs. 197, 198,

$$A_2 = \frac{Pa + P(a + b) + G(L + l)}{L} = P + G + G\frac{l}{L} \quad . \quad (13)$$

The twisting and bending moments for the cheeks, S, crank pin, Z_2, and bearing, A_2, can be determined by the same method as above.

A useful formula for calculating the diameter of a crank shaft is as follows: $d = \sqrt[3]{\frac{HP}{N}} \times c$. Where N = number of revolutions per minute, c = a constant based on practice. When using the somewhat vague term of "nominal" horse-power, the constant may be taken as 560; $d = \sqrt[3]{\frac{HP}{N}} \times 560$. When using indicated horse-power, the constant may be taken as 190; $d = \sqrt[3]{\frac{IHP}{N}} \times 190$, this will give ample strength for mild steel shafts.

Methods of oiling Crank Pins of Bent and Slotted-out Cranks.—The usual method is based on that shown on page 57 fig. 185, where the crank pin is oiled through the connecting rod end. Another method, but certainly not so good, is shown in fig. 199, where a hole is drilled in the pin shown in dotted lines, and fed with oil by a cup fixed to a ring held concentric with the shaft. No doubt this would freely oil the pin, but the long holes through which the oil has to travel are liable to get choked up, and may do so without being noticed until the pin runs hot.

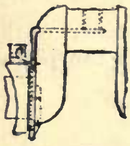


Fig. 199.

Connecting Rods.

Of almost equal importance to the crank shaft main bearing is the connecting rod end which works on the crank pin. The desire to make a perfect connecting rod end has led to a large number of

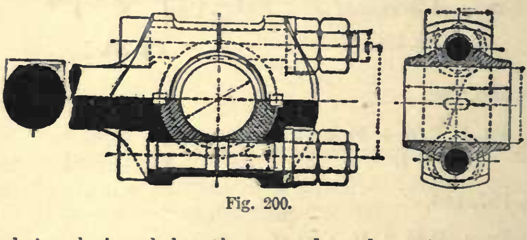


Fig. 200.

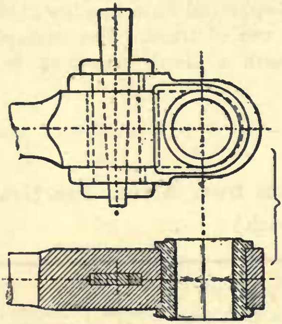
varieties being designed, but the general tendency has been to settle down into two or three types, such as the marine type with bolts, the strap and cotter type, the solid end type, and a few others for special cases.

Fig. 200 shows an example of connecting rod end fitted with bolts, and the seating faces of the brasses of spherical form to allow of slight self-adjustment should any part be out of truth. The example is taken from a pumping engine, with a crank pin 9·84 inches diameter.

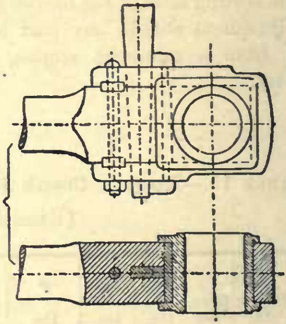
TABLE 15.—**Engine Crank Pins from Actual Practice.**
(Kiesselbach.)

Diam. of Cyl. in.	Stroke in.	Revs. per min.	d ins.	l ins.	P lbs.	p lbs. p. per. sq. in.	$p\ v$ ft. lbs. p. sec. &sq. in.	Material of bearings.	Machinery driven by Engine.
17·7	27·5	66	4·5	4·7	22500	1113	1381	Babbit	Electric light
23·6	43·3	75	8·5	13·0	30900	280	779	Gun metal	Roller mills
23·6 35·4	41·1	60	6·1	6·1	38200	1027	1640	Babbit	Flour mills
23·6 39·3	39·4	120	5·1	5·1	24700	950	1537	Gun metal	Roller mills
25·6	25·6	150 180	5·9	6·3	28200	750	2896 3475	Babbit	Roller mills
27·5	39·4	100	6·3	6·7	40400	957	2631	Babbit	Roller mills
31·5	47·2	80	7·1	8·6	41900	686	1700	Gun metal	Roller mills
35·4	53·1	80	7·9	9·8	65300	861	2374	Gun metal	Roller mills
49·2	49·2	80 90	9·8	11·0	127800	1186	4057 4567	Babbit	Roller mills

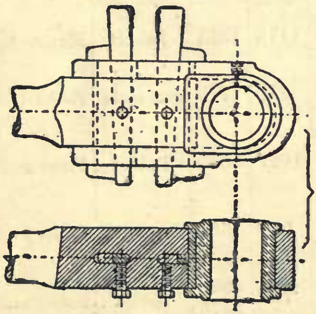
Connecting Rod Ends.



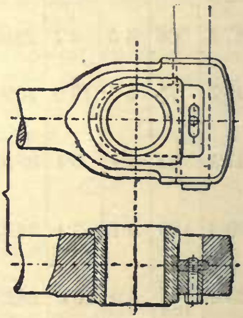
Figs. 201 and 202.



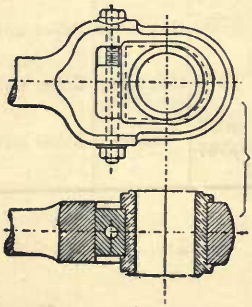
Figs. 203 and 204.



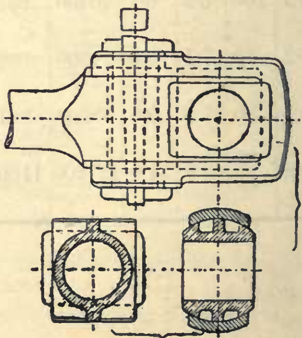
Figs. 205 and 206.



Figs. 207 and 208.



Figs. 209 and 210.

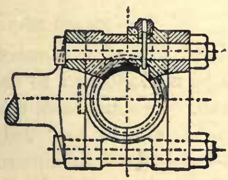


Figs. 211 and 212.

Figs. 201, 203, 205 and 211 show varieties of the connecting rod ends known as the "strap, gib and cottar" or "butt" end, a form much used on locomotives and other engines, it is easy of adjustment, and occupies but little space. Figs. 207 and 209 show two varieties of that kind known as the "solid" end; and this form is probably the lightest in weight of all ordinary ends, but can only be used with crank pins on discs or crank arms. On account of the end being in one piece with the rod, it can only be taken off the pin by pulling outwards. It is a very good form, and very safe from breakdowns and other mishaps.

With fig. 203, when the cottar is driven down to take up wear, the rod is lengthened, that is, the distance from centre to centre of brasses in the two ends is increased; in fig. 201 the reverse is the case; in the solid end, fig. 209, the length is increased, and in fig. 207 diminished. Sometimes connecting rods are made with one end adjusting in the opposite sense to the other end, this tends to keep the rod nearly to the normal length.

Fig. 213.



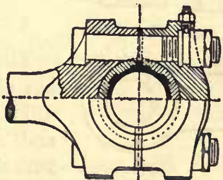
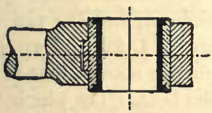


Fig. 215.

Fig. 214.



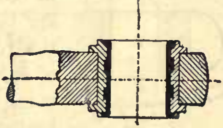
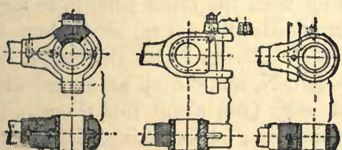
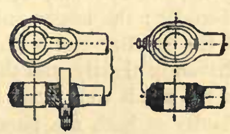


Fig. 216.

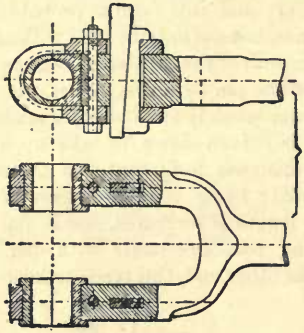
Fig. 213 shows a connecting rod end much used both for small and large engines; it is a good form and cheap to make. There is more brass in the form fig. 213 than in fig. 215, and fig. 215 is more used in large work, such as marine engines. In the above two figs. the "brasses" are lined with Babbit metal, and may sometimes be of cast iron or of cast steel, the latter is used by eminent marine engine builders.



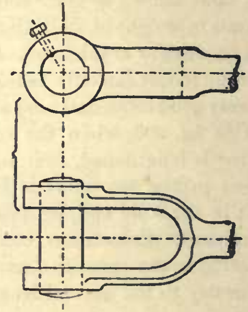


Figs. 217—219. Figs. 220 and 221.

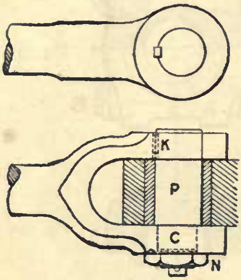
Figs. 217 to 221 show a variety of types used for the crosshead end of rod in small engines. Fig. 218 was formerly used on pump rods, and thus no doubt found its way on to some of the early steam engines. It is now very rarely used, at least for connecting rods.



Figs. 222 and 223.

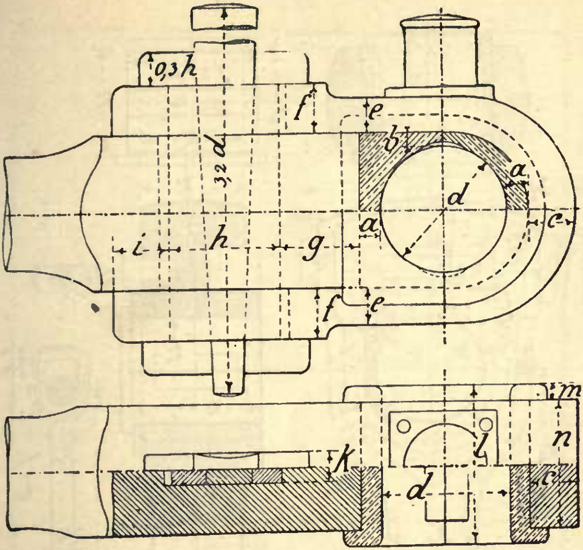


Figs. 224 and 225.



Figs. 226 and 227.

The crosshead end of a connecting rod has often to be made forked to embrace a solid crosshead. In some cases the brasses and adjustment are in the rod, as fig. 222, the pin being fixed in the crosshead; in other cases the brasses and adjustment are in the crosshead, and the pin fixed in the rod, as figs. 224, 226. Two methods of fixing the pin are shown, that in fig. 227 being by far the best. The pin is fixed by boring the two sides of the fork with conical holes of different sizes, so that the similar cones on the pin may be passed in from one side. This method, although good, is difficult to carry out in practice as it is almost impossible to make both sides a perfect fit, and to insure that the sides of the fork are not pulled in when the nut on the small end of the pin is tightened up (see fig. 269). In fig. 227 the pin is parallel, and one end, C, is smaller than the other part, P, to form a shoulder; the larger end is provided with a key, or feather, K, and the smaller end with a nut, N. The pin with the feather, K, is driven in tightly, and then the nut, N, screwed up hard; provided that the pin is well fitted in both sides (and about this there is no difficulty as both are parallel), this forms a good fixing, and experience has shown that it is easily drawn out and does not work loose.



Figs. 228 and 229.

TABLE 16.—Dimensions of Connecting Rod Ends (Figs. 228, 229).

Crank Pin.		NOTE.—All Dimensions are given in Inches.										
d	l	a	b	c	e	f	g	h	i	k	m	n
2	$2\frac{5}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	1	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{5}{8}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{1}{4}$	$1\frac{7}{8}$
$2\frac{1}{2}$	$3\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	1	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{7}{8}$	1	$\frac{1}{2}$	$\frac{5}{16}$	$2\frac{3}{8}$
$2\frac{3}{4}$	$3\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{3}{4}$	1	$1\frac{3}{8}$	$2\frac{1}{8}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$2\frac{3}{4}$
3	$4\frac{1}{4}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$2\frac{3}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$3\frac{1}{2}$
$3\frac{1}{2}$	$4\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{3}{8}$	1	$1\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{2}$	$3\frac{5}{8}$
4	$5\frac{1}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{1}{8}$	$3\frac{1}{4}$	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{5}{8}$	4
$4\frac{1}{2}$	$5\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$2\frac{3}{8}$	$3\frac{5}{8}$	2	1	$\frac{5}{8}$	$4\frac{1}{2}$
5	$6\frac{3}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{3}{4}$	$2\frac{5}{8}$	4	$2\frac{1}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$5\frac{1}{8}$
$5\frac{1}{2}$	7	$\frac{3}{4}$	$\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{1}{2}$	$2\frac{1}{8}$	3	$4\frac{3}{4}$	$2\frac{5}{8}$	$1\frac{1}{8}$	$\frac{5}{8}$	$5\frac{3}{4}$
6	$7\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{1}{2}$	$2\frac{1}{4}$	$3\frac{1}{4}$	$5\frac{1}{4}$	$2\frac{3}{4}$	$1\frac{3}{8}$	$\frac{5}{8}$	$6\frac{1}{2}$
$6\frac{1}{2}$	$8\frac{1}{2}$	$\frac{7}{8}$	$\frac{5}{8}$	$2\frac{5}{8}$	$1\frac{3}{4}$	$2\frac{3}{8}$	$3\frac{3}{8}$	$5\frac{5}{8}$	3	$1\frac{3}{8}$	$\frac{5}{8}$	$7\frac{1}{4}$
7	9	1	$\frac{5}{8}$	$3\frac{1}{8}$	$1\frac{7}{8}$	$2\frac{3}{4}$	$3\frac{3}{4}$	$6\frac{3}{8}$	$3\frac{3}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	$7\frac{1}{2}$
$7\frac{1}{2}$	$9\frac{3}{4}$	1	$\frac{5}{8}$	$3\frac{3}{8}$	2	$2\frac{7}{8}$	4	$6\frac{3}{4}$	$3\frac{5}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	$8\frac{1}{4}$
8	10	$1\frac{1}{8}$	$\frac{5}{8}$	$3\frac{5}{8}$	$2\frac{1}{8}$	3	$4\frac{1}{4}$	$7\frac{1}{4}$	$3\frac{7}{8}$	$1\frac{3}{4}$	$\frac{3}{4}$	$8\frac{1}{2}$

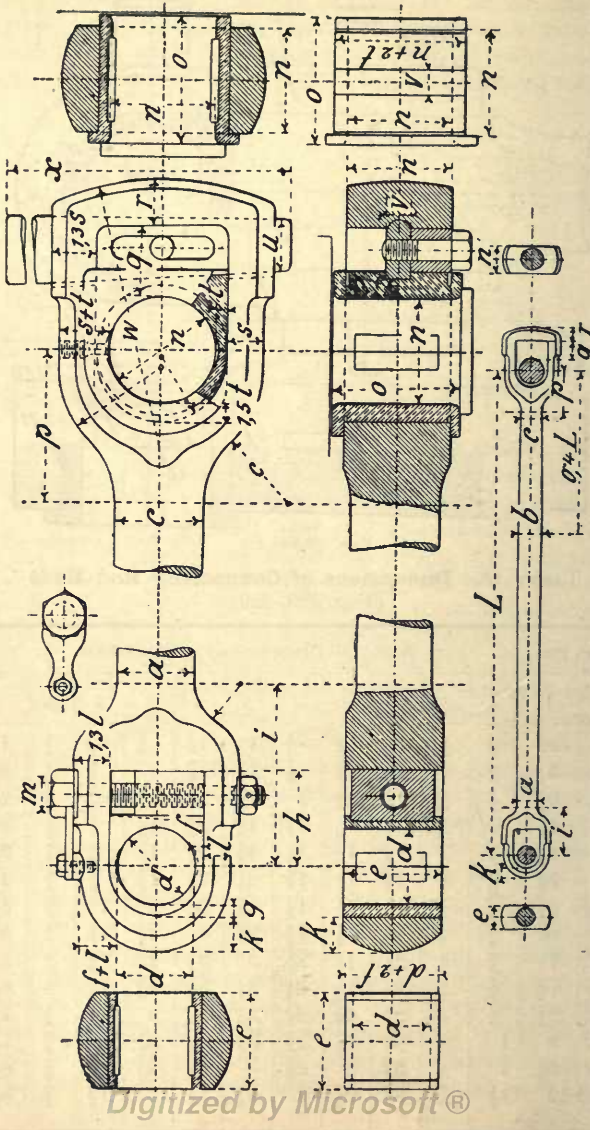


Fig. 230.

Fig. 231

Fig. 232.

Fig. 233.

Fig. 234.

TABLE 17.—**Dimensions of Connecting Rods** (Figs. 230—234).

Engines.		Rod.				Crosshead end.									Crank Pin end.										
H	D	L	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*	*x*
12	6	32	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{4}$	$\frac{3}{16}$	$\frac{1}{4}$	$1\frac{3}{4}$	$3\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	—	—	—	—	—	—	—	—	—	—	—
16	8	42	$1\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$2\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$2\frac{1}{8}$	4	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	—	—	—	—	—	—	—	—	—	—	—
20	10	50	$1\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{1}{8}$	2	$2\frac{3}{4}$	$\frac{3}{16}$	$\frac{5}{16}$	$2\frac{5}{8}$	$5\frac{3}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$2\frac{5}{8}$	$3\frac{1}{4}$	$3\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{4}$	1	$\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	9	8
24	12	60	2	$2\frac{5}{8}$	$2\frac{3}{8}$	$2\frac{1}{4}$	$3\frac{1}{4}$	$\frac{3}{16}$	$\frac{5}{16}$	$2\frac{3}{4}$	6	1	$\frac{7}{8}$	$\frac{5}{8}$	$2\frac{3}{4}$	$3\frac{5}{8}$	$4\frac{3}{8}$	2	$1\frac{1}{4}$	1	$\frac{5}{16}$	$1\frac{3}{8}$	$\frac{7}{8}$	10	$9\frac{1}{2}$
28	14	70	$2\frac{1}{4}$	3	$2\frac{5}{8}$	$2\frac{5}{8}$	$3\frac{5}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$3\frac{1}{4}$	$6\frac{3}{4}$	$1\frac{1}{8}$	1	$\frac{5}{8}$	$3\frac{1}{4}$	$4\frac{1}{4}$	5	$2\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{5}{16}$	$1\frac{3}{8}$	1	12	11
32	16	80	$2\frac{3}{8}$	$3\frac{1}{4}$	$2\frac{3}{4}$	3	4	$\frac{1}{4}$	$\frac{3}{8}$	$3\frac{5}{8}$	$7\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$3\frac{3}{4}$	$4\frac{5}{8}$	$5\frac{5}{8}$	$2\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{8}$	13	12
36	18	90	$2\frac{3}{4}$	$3\frac{5}{8}$	$3\frac{1}{4}$	$3\frac{3}{8}$	$4\frac{3}{8}$	$\frac{5}{16}$	$\frac{1}{2}$	4	$8\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$4\frac{1}{4}$	$5\frac{1}{4}$	$6\frac{1}{4}$	$2\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{3}{8}$	$\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{1}{4}$	14	$13\frac{1}{2}$
40	20	100	3	$3\frac{3}{4}$	$3\frac{3}{8}$	$3\frac{3}{4}$	$4\frac{3}{4}$	$\frac{5}{16}$	$\frac{1}{2}$	$4\frac{3}{8}$	9	$1\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{4}$	$4\frac{3}{4}$	$5\frac{3}{4}$	$6\frac{3}{4}$	$2\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{8}$	16	15
44	22	110	$3\frac{1}{4}$	4	$3\frac{5}{8}$	4	$5\frac{1}{4}$	$\frac{3}{8}$	$\frac{5}{8}$	$4\frac{5}{8}$	$9\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{3}{8}$	$\frac{3}{4}$	$5\frac{1}{4}$	$6\frac{3}{8}$	$7\frac{5}{8}$	3	2	$1\frac{3}{4}$	$\frac{1}{2}$	2	$1\frac{1}{2}$	17	16

All Dimensions are given in Inches.

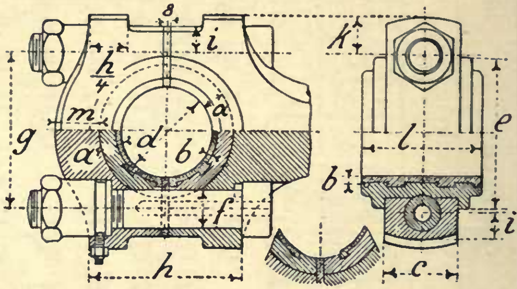
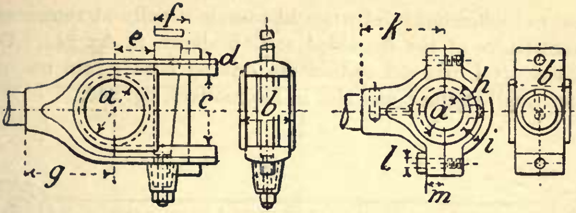


Fig. 235. Fig. 236.

TABLE 18.—Dimensions of Connecting Rod Ends (Marine Type) (Figs. 235, 236).

Crank Pin.		NOTE.—All Dimensions are given in Inches.								
d	*l*	*a*	*b*	*c*	*e*	*f*	*g*=*h*	*i*	*k*	*m*
2	$2\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{16}$	2	$4\frac{3}{8}$	$1\frac{1}{8}$	$4\frac{3}{8}$	$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$
$2\frac{1}{2}$	3	$\frac{5}{8}$	$\frac{3}{16}$	$2\frac{1}{4}$	$4\frac{3}{4}$	$1\frac{1}{4}$	$4\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$
$2\frac{3}{4}$	$3\frac{1}{2}$	$\frac{11}{16}$	$\frac{3}{16}$	$2\frac{3}{8}$	5	$1\frac{1}{4}$	$5\frac{1}{4}$	$\frac{7}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$
$3\frac{1}{4}$	4	$\frac{11}{16}$	$\frac{3}{16}$	$2\frac{3}{4}$	$5\frac{5}{8}$	$1\frac{3}{8}$	$5\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$
$3\frac{1}{2}$	$4\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{16}$	$3\frac{1}{4}$	$6\frac{1}{4}$	$1\frac{1}{2}$	$6\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{3}{4}$
4	$5\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{4}$	$3\frac{3}{8}$	$6\frac{3}{4}$	$1\frac{3}{4}$	$7\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{7}{8}$	2
$4\frac{1}{2}$	$5\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{4}$	$3\frac{3}{4}$	$7\frac{3}{8}$	$1\frac{7}{8}$	$7\frac{3}{4}$	$1\frac{3}{8}$	$2\frac{1}{8}$	$2\frac{1}{4}$
$4\frac{3}{4}$	6	$\frac{7}{8}$	$\frac{1}{4}$	4	$7\frac{3}{4}$	$2\frac{1}{8}$	$8\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$
$5\frac{1}{4}$	$6\frac{5}{8}$	1	$\frac{5}{16}$	$4\frac{3}{8}$	$8\frac{5}{8}$	$2\frac{1}{4}$	$9\frac{1}{4}$	$1\frac{5}{8}$	$2\frac{3}{8}$	$2\frac{5}{8}$
$5\frac{5}{8}$	$7\frac{1}{4}$	1	$\frac{5}{16}$	$4\frac{3}{4}$	$9\frac{1}{4}$	$2\frac{3}{8}$	$9\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{5}{8}$	$2\frac{3}{4}$
6	$7\frac{5}{8}$	$1\frac{1}{8}$	$\frac{5}{16}$	5	$9\frac{5}{8}$	$2\frac{5}{8}$	$10\frac{3}{8}$	2	$2\frac{3}{4}$	3
$6\frac{1}{2}$	$8\frac{1}{4}$	$1\frac{1}{4}$	$\frac{5}{16}$	$5\frac{3}{8}$	$10\frac{3}{8}$	$2\frac{3}{4}$	$11\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{7}{8}$	$3\frac{1}{4}$
$7\frac{1}{4}$	$9\frac{1}{4}$	$1\frac{1}{4}$	$\frac{5}{16}$	6	$11\frac{5}{8}$	$3\frac{1}{4}$	$12\frac{5}{8}$	$2\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{5}{8}$
8	10	$1\frac{3}{8}$	$\frac{5}{16}$	$6\frac{3}{4}$	$12\frac{3}{8}$	$3\frac{3}{8}$	$13\frac{3}{4}$	$2\frac{5}{8}$	$3\frac{5}{8}$	4



Figs. 237 and 238.

TABLE 19.—Dimensions of small Connecting Rod Ends (Figs. 237, 238).

Engines.		NOTE.—All Dimensions are given in Inches.											
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*
12	6	$2\frac{3}{4}$	$3\frac{1}{4}$	$3\frac{3}{8}$	$\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$4\frac{3}{8}$	$\frac{5}{16}$	$1\frac{1}{8}$	$4\frac{3}{4}$	$\frac{5}{8}$	$1\frac{1}{4}$
16	8	$3\frac{1}{4}$	$3\frac{5}{8}$	$3\frac{7}{8}$	$\frac{7}{8}$	2	2	$4\frac{3}{4}$	$\frac{5}{16}$	$1\frac{1}{4}$	$5\frac{1}{4}$	$\frac{3}{4}$	$1\frac{3}{8}$

These types, especially fig. 237, are not much used now, having been superseded by those which are capable of being turned out by machine work, with as little hand work as possible.

Connecting Rods.

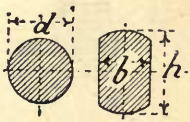
Wrought-iron rods of round or flattened cross section. Figs. 239, 240.

P = pressure on piston in lbs.

l = length of rod in inches.

d = diameter of round rod in inches at centre of length.

m = constant for different speeds, see table 20, then



Figs. 239 and 240.

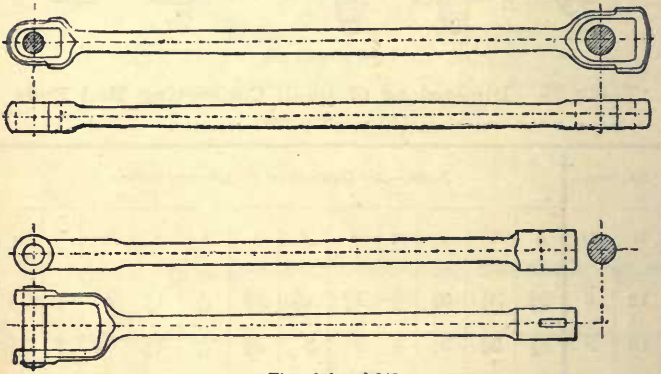
$$d = \cdot 0164 \sqrt[4]{m \,.\, P \,.\, l^2}.$$

TABLE 20.

Piston speed in feet per minute.	200	400	600	800
m	30	20	15	10

With locomotives, m may be taken from 10 to 6. The diameter of the rod at crosshead end is $\cdot 8d$, at the crank end $\cdot 9d$.

The rod when made of wrought iron is usually of round section, but sometimes of the flat-sided section shown in fig. 244. Of late years cast-steel rods of **I** section are coming largely into use, especially for small engines, and also in locomotives, fig. 245.



Figs. 241 and 242.

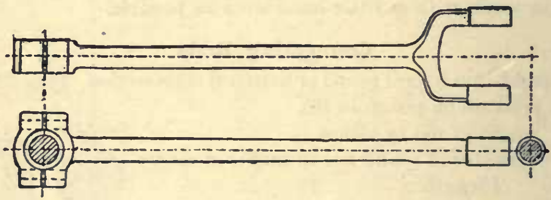
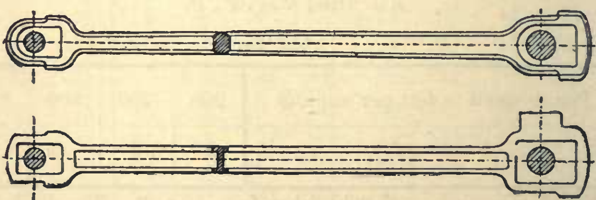


Fig. 243.



Figs. 244 and 245.

Conversion of round into rectangular cross section.

TABLE 21.

$h : b$	1·5	1·75	2	2·25	2·5
$b : d$	·79	·76	·74	·72	·7
$h : d$	1·19	1·33	1·48	1·62	1·75

Example.—In an engine P = 15432 lbs., the length of the rod $l = 71$ inches, the piston speed = 400 feet per minute; for the round rod $d = \cdot 0164 \sqrt[4]{20 \times 15432 \times 71^2} = 3\cdot 23$ inches, for a rectangular rod with the proportion $h : b = 1\cdot 75$, from the table $b : d = \cdot 76$,

$$\text{then } b = \cdot 76 \times 3\cdot 23 = 2\cdot 45 \text{ inches,}$$
$$\text{and } h = 1\cdot 33 \times 3\cdot 23 = 4\cdot 29 \text{ inches.}$$

Crossheads.

These are made in endless variety of form and construction and the materials used are cast iron, wrought iron and cast steel, the wearing surface of the guide blocks attached to the crossheads are almost invariably of cast iron. Formerly guide bars were often made circular, turned steel bars being much used on small engines, but now the bars are either flat or bored out, both systems having merits of their own. Possibly flat surfaces work with the least friction, other things being equal, but bored-out guides and turned crossheads have the merit of being made perfectly true with less difficulty than flat surfaces. The figures below show a variety of crossheads. The pressure on the guide surfaces may be from 30 to 45 lbs. per square inch.

Figs. 246, 247 are examples of American practice.

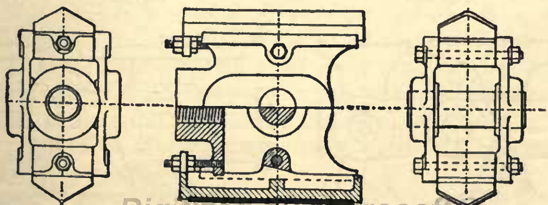


Fig. 246.

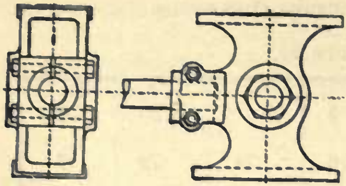


Fig. 247.

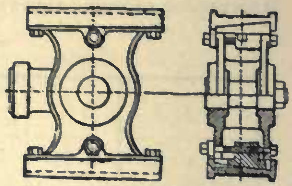


Fig. 248.

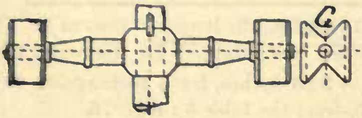


Fig. 249.

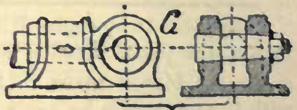


Fig. 250.

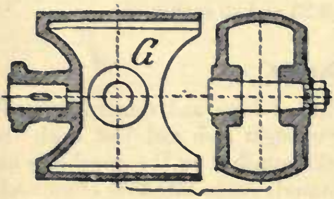


Fig. 251.

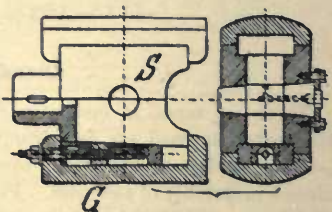


Fig. 252.

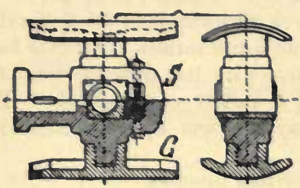


Fig. 253.

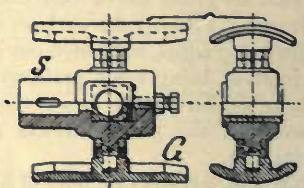


Fig. 254.

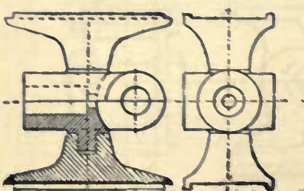


Fig. 255.

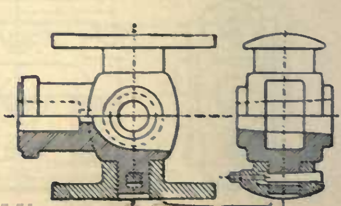


Fig. 256.

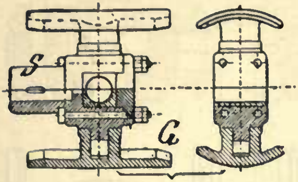


Fig. 257.

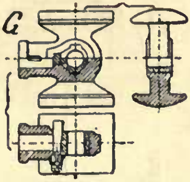


Fig. 258.

G. Cast iron. S. Wrought iron or cast steel.

Crosshead Pin.

The maximum pressure on the surface of the pin is up to 1400 lbs. per square inch.* The length of the pin varies from two diameters down to 1·3 diameters. The crosshead pins are lubricated by automatic means for engines which have to make long runs, with an arrangement like that on page 58, fig. 189, for oiling the crank pin. See also fig. 259.

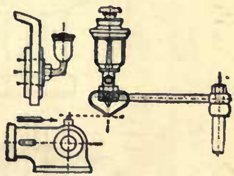


Fig. 259.

There are advantages and disadvantages in some of these types of crossheads with respect on the one hand to the guide blocks or slippers ; in the old types with two blocks and four bars, such as those shown in fig. 267, page 80, there are a number of parts and a great deal of labour in fitting up ; however, for a long time such crossheads, blocks, and bars held their own and were universally used ; now following the same lines of development as in engine frames, that is of making the structure in as few pieces as possible ; two bars, top and bottom, are almost universal, the guiding surfaces or bars being in one with the engine frame, and this lessens the labour and decreases the liability of error in fitting up.

In some cases, however, there is only one bar with the guide block embracing it all round ; this is common in locomotives, and has the merit of being easy of access. In marine engines and in some large land engines the crosshead is simple, as in fig. 250, and two strips bolted to the frame form the upper guide surfaces ; this is only used in land engines when they run one way continually, so that the pressure is always on the bottom bar, and when the engine is horizontal the guide block can be made to run in a bath of oil, a very effective way of lubricating.

* This is very high, a good English example giving only 850 lbs., with maximum boiler pressure on the piston.

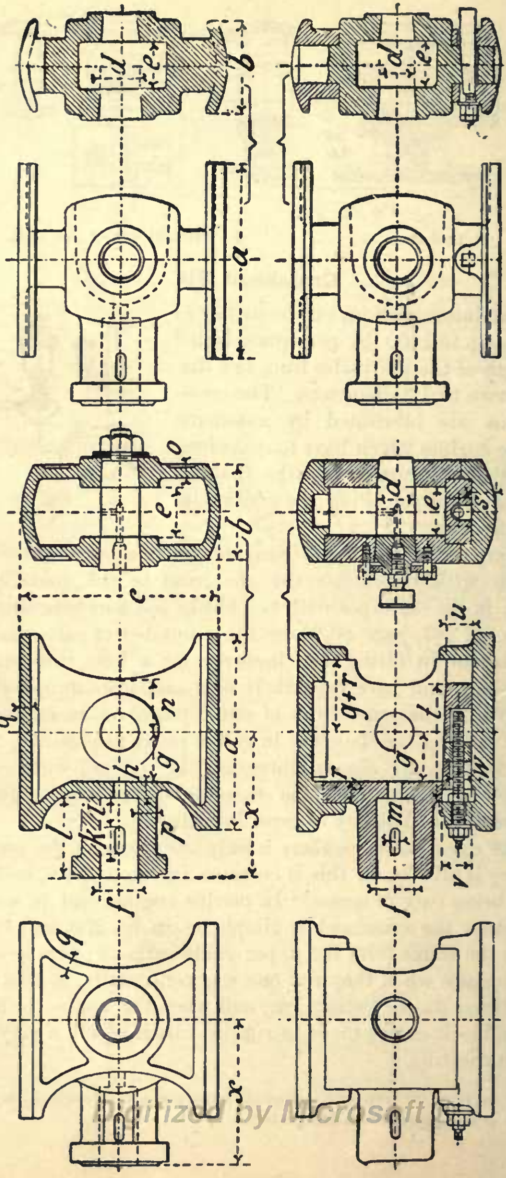


Fig. 260

Fig. 261

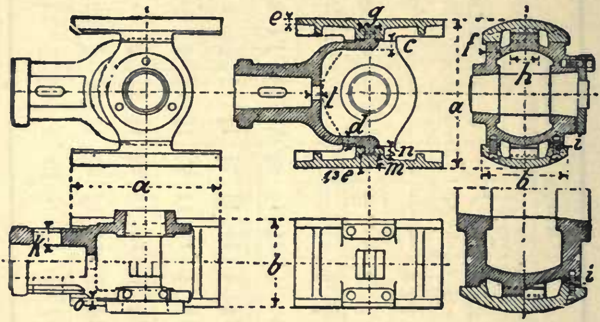
Fig. 262

Fig. 263

TABLE 22.—**Dimensions of Crossheads** (Figs. 260—263) **for Engine Frame**, page 24.

Engines.		Cast Iron.																Wrought Iron with Cast-iron Slippers.						
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*	*x*
12	6	6	$3\frac{5}{8}$	7	$1\frac{3}{8}$	$1\frac{3}{4}$	1	2	$\frac{1}{2}$	$\frac{3}{4}$	1	$2\frac{3}{4}$	$\frac{3}{8}$	$1\frac{3}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	—	—	—	—	—	—	$5\frac{1}{4}$
16	8	$7\frac{5}{8}$	$4\frac{1}{2}$	9	$1\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{1}{4}$	$2\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$3\frac{5}{8}$	$\frac{3}{8}$	$2\frac{1}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{8}$	—	—	—	—	—	—	$6\frac{1}{2}$
20	10	$8\frac{3}{4}$	5	11	2	$2\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{3}{4}$	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$4\frac{1}{8}$	$\frac{7}{16}$	$2\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{8}$	—	—	—	—	—	—	$7\frac{1}{2}$
24	12	$10\frac{3}{8}$	$6\frac{1}{4}$	12	$2\frac{1}{4}$	$3\frac{1}{4}$	$1\frac{7}{8}$	$3\frac{1}{8}$	$\frac{3}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$4\frac{5}{8}$	$\frac{1}{2}$	$3\frac{1}{8}$	$\frac{5}{8}$	1	$\frac{1}{2}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$6\frac{3}{8}$	1	$1\frac{3}{8}$	$\frac{5}{8}$	$8\frac{1}{2}$
28	14	12	$7\frac{1}{4}$	$13\frac{1}{2}$	$2\frac{5}{8}$	$3\frac{5}{8}$	$2\frac{1}{8}$	$3\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	5	$\frac{9}{16}$	$3\frac{3}{8}$	$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$7\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$\frac{5}{8}$	$9\frac{1}{4}$
32	16	$13\frac{5}{8}$	8	15	3	4	$2\frac{1}{4}$	$3\frac{7}{8}$	$\frac{7}{8}$	$1\frac{5}{8}$	2	$5\frac{3}{4}$	$\frac{5}{8}$	$3\frac{3}{4}$	$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{8}$	8	$1\frac{3}{8}$	$1\frac{7}{8}$	$\frac{3}{4}$	$10\frac{1}{2}$
36	18	$14\frac{3}{4}$	$8\frac{3}{4}$	17	$3\frac{3}{8}$	$4\frac{3}{8}$	$2\frac{1}{2}$	$4\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$6\frac{1}{8}$	$\frac{11}{16}$	$4\frac{1}{8}$	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$8\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{1}{8}$	$\frac{3}{4}$	$11\frac{1}{2}$
40	20	$16\frac{3}{8}$	$9\frac{1}{2}$	19	$3\frac{3}{4}$	$4\frac{3}{4}$	$2\frac{5}{8}$	$4\frac{5}{8}$	$1\frac{1}{4}$	2	$2\frac{1}{4}$	$6\frac{5}{8}$	$\frac{3}{4}$	$4\frac{3}{8}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$9\frac{3}{8}$	$1\frac{3}{4}$	$2\frac{3}{8}$	$\frac{7}{8}$	$12\frac{1}{2}$
44	22	$21\frac{5}{8}$	10	21	4	$5\frac{1}{4}$	$3\frac{1}{8}$	5	$1\frac{5}{8}$	$2\frac{1}{4}$	$2\frac{5}{8}$	$7\frac{3}{8}$	$\frac{13}{16}$	$4\frac{3}{4}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{7}{8}$	10	2	$2\frac{5}{8}$	$\frac{7}{8}$	14

All Dimensions are given in Inches.



Figs. 264 and 265.

TABLE 23.—**Dimensions of Cast Steel Crossheads with Cast-iron Slippers** (Figs. 264, 265).

Engines.															
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*
24	12	12	$6\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{3}{4}$	2	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{1}{16}$
32	16	15	8	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$2\frac{3}{8}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{1}{16}$
40	20	19	$9\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$1\frac{3}{8}$	$1\frac{1}{4}$	3	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{1}{8}$
48	24	23	$11\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$3\frac{5}{8}$	$\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{1}{8}$
56	28	27	$12\frac{3}{4}$	$1\frac{1}{8}$	1	$\frac{7}{8}$	2	$1\frac{5}{8}$	4	$\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{1}{8}$
64	32	31	$14\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{8}$	1	$2\frac{1}{4}$	2	$4\frac{3}{4}$	$\frac{3}{4}$	2	$1\frac{7}{8}$	1	$1\frac{7}{8}$	$\frac{1}{8}$
72	36	$34\frac{1}{2}$	$17\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{5}{8}$	$2\frac{1}{4}$	$5\frac{3}{8}$	$\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{1}{8}$	$\frac{3}{16}$
80	40	$38\frac{1}{2}$	20	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{3}{8}$	6	$\frac{7}{8}$	$2\frac{3}{8}$	$2\frac{3}{8}$	$1\frac{1}{4}$	$2\frac{3}{8}$	$\frac{3}{16}$

All Dimensions are given in Inches.

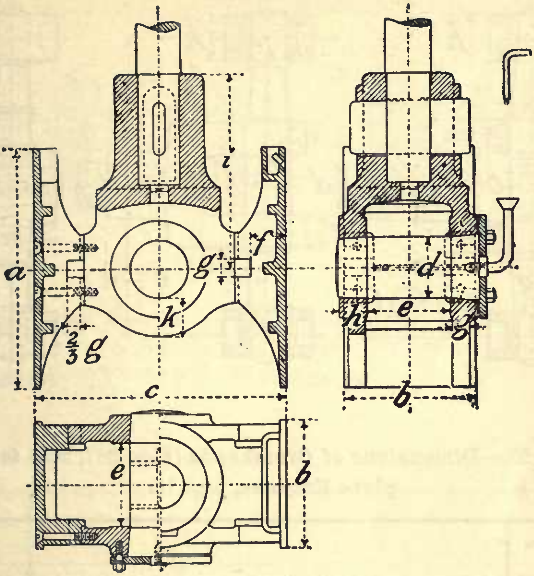


Fig. 266.

TABLE 24.—**Dimensions of Cast-steel Crossheads with Cast-iron Slippers** (Fig. 266).

Engine.											
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*
28	26	18	11	21	$4\frac{1}{2}$	$6\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{3}{8}$	2	15	$2\frac{1}{2}$
30	28	20	$11\frac{1}{2}$	22	5	7	$2\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{1}{4}$	16	$2\frac{3}{4}$
32	30	22	12	23	$5\frac{1}{2}$	$7\frac{1}{2}$	3	$1\frac{5}{8}$	$2\frac{1}{2}$	17	3
34	32	23	$12\frac{1}{2}$	24	$5\frac{3}{4}$	8	$3\frac{1}{4}$	$1\frac{3}{4}$	$2\frac{3}{4}$	18	$3\frac{1}{4}$
36	34	24	13	25	6	$8\frac{1}{2}$	$3\frac{1}{2}$	$1\frac{7}{8}$	3	19	$3\frac{1}{2}$
38	36	25	$13\frac{1}{2}$	26	$6\frac{1}{4}$	9	$3\frac{3}{4}$	2	$3\frac{1}{4}$	20	$3\frac{3}{4}$
40	38	26	14	27	$6\frac{1}{2}$	$9\frac{1}{2}$	4	$2\frac{1}{4}$	$3\frac{1}{2}$	21	4
42	40	27	15	28	$6\frac{3}{4}$	10	$4\frac{1}{4}$	$2\frac{1}{2}$	$3\frac{3}{4}$	22	$4\frac{1}{4}$
44	44	28	16	29	7	$10\frac{1}{2}$	$4\frac{1}{2}$	$2\frac{3}{4}$	4	23	$4\frac{1}{2}$

All Dimensions are given in Inches.

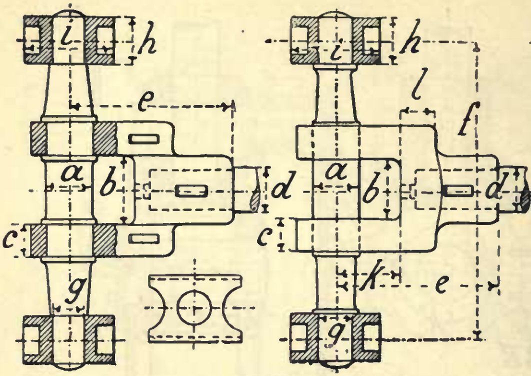


Fig. 267.

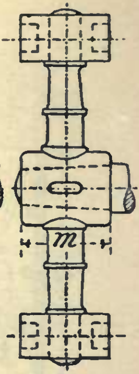


Fig. 268.

TABLE 25.—**Dimensions of Crossheads** (Figs. 267, 268) **for Bed-plate Engines,** page 19.

Engines.													
H	D	*d*	*a*	*b*	*c*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*
12	6	$1\frac{3}{4}$	$2\frac{1}{8}$	3	$1\frac{1}{2}$	8	14	$1\frac{1}{4}$	2	$3\frac{1}{2}$	3	$1\frac{1}{4}$	$4\frac{1}{4}$
16	8	2	$2\frac{3}{8}$	$3\frac{1}{4}$	$1\frac{3}{4}$	9	16	$1\frac{1}{2}$	$2\frac{3}{8}$	$4\frac{3}{4}$	$3\frac{3}{8}$	$1\frac{1}{2}$	$4\frac{3}{4}$
20	10	$2\frac{1}{4}$	$2\frac{3}{4}$	$3\frac{5}{8}$	2	10	$19\frac{1}{2}$	$1\frac{3}{4}$	3	6	$3\frac{3}{4}$	$1\frac{3}{4}$	$5\frac{1}{4}$
24	12	$2\frac{5}{8}$	3	4	$2\frac{1}{4}$	11	$21\frac{1}{2}$	2	$3\frac{5}{8}$	$7\frac{1}{4}$	4	$2\frac{1}{8}$	$5\frac{5}{8}$
28	14	$2\frac{3}{4}$	$3\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{3}{8}$	12	24	$2\frac{1}{4}$	$4\frac{5}{8}$	$8\frac{3}{8}$	$4\frac{1}{4}$	$2\frac{1}{4}$	$6\frac{5}{8}$
32	16	3	$3\frac{5}{8}$	$4\frac{5}{8}$	$2\frac{5}{8}$	13	26	$2\frac{3}{8}$	$5\frac{1}{4}$	$9\frac{1}{2}$	$4\frac{5}{8}$	$2\frac{3}{8}$	7
36	18	$3\frac{1}{4}$	4	5	$2\frac{3}{4}$	14	28	$2\frac{5}{8}$	6	$10\frac{3}{4}$	5	$2\frac{5}{8}$	$7\frac{5}{8}$
40	20	$3\frac{5}{8}$	$4\frac{3}{8}$	$5\frac{1}{4}$	3	15	30	$2\frac{3}{4}$	$6\frac{3}{4}$	12	$5\frac{3}{8}$	$2\frac{7}{8}$	$8\frac{3}{8}$
48	24	$4\frac{3}{8}$	$5\frac{3}{8}$	$6\frac{3}{8}$	$3\frac{3}{8}$	19	32	$3\frac{3}{8}$	$7\frac{5}{8}$	$13\frac{5}{8}$	$6\frac{3}{8}$	$3\frac{1}{2}$	10
56	28	$5\frac{1}{4}$	$6\frac{3}{8}$	$7\frac{5}{8}$	$3\frac{3}{4}$	23	36	4	$8\frac{3}{8}$	$15\frac{1}{4}$	$7\frac{3}{8}$	$4\frac{1}{8}$	$12\frac{3}{8}$
64	32	6	$7\frac{3}{8}$	$8\frac{3}{4}$	$4\frac{3}{8}$	26	42	$4\frac{5}{8}$	$9\frac{5}{8}$	$16\frac{3}{4}$	$8\frac{3}{8}$	$4\frac{3}{4}$	$14\frac{3}{4}$
72	36	7	$8\frac{3}{8}$	10	5	30	52	$5\frac{1}{4}$	$10\frac{3}{4}$	$18\frac{3}{8}$	$9\frac{3}{8}$	$5\frac{1}{2}$	$17\frac{1}{4}$
80	40	8	$9\frac{5}{8}$	$11\frac{1}{4}$	$5\frac{3}{8}$	34	60	6	12	20	$10\frac{1}{2}$	$6\frac{3}{8}$	20

All Dimensions are given in Inches.

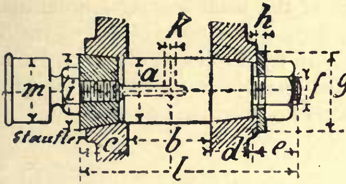


Fig. 269.

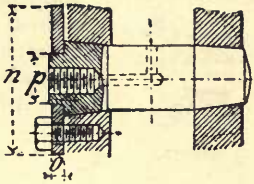


Fig. 270.

The above figs. show two methods of fixing crosshead pins, another method is shown on page 66, fig. 226, under connecting rods with remarks on different ways of fixing pins.

Table 26.—**Dimensions of Crosshead Pins** (Figs. 269, 270).

Engines.																
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*
12	6	$1\frac{3}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	1	$1\frac{1}{4}$	$\frac{5}{8}$	2	$\frac{3}{16}$	$1\frac{5}{8}$	$\frac{1}{4}$	$5\frac{1}{4}$	$\frac{3}{4}$	—	–	—
16	8	$1\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{3}{4}$	$2\frac{3}{8}$	$\frac{3}{16}$	$1\frac{7}{8}$	$\frac{1}{4}$	$6\frac{1}{8}$	$1\frac{1}{4}$	—	–	—
20	10	2	$2\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$2\frac{5}{8}$	$\frac{3}{16}$	$2\frac{3}{8}$	$\frac{1}{4}$	$7\frac{3}{8}$	$1\frac{1}{4}$	—	–	—
24	12	$2\frac{1}{4}$	$3\frac{1}{4}$	2	$1\frac{3}{4}$	$1\frac{1}{2}$	$\frac{7}{8}$	3	$\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{16}$	$8\frac{1}{2}$	$1\frac{1}{2}$	4	$\frac{3}{8}$	$1\frac{1}{2}$
28	14	$2\frac{5}{8}$	$3\frac{5}{8}$	$2\frac{1}{4}$	2	$1\frac{1}{2}$	$\frac{7}{8}$	$3\frac{3}{8}$	$\frac{1}{4}$	3	$\frac{5}{16}$	$9\frac{3}{8}$	$1\frac{1}{2}$	$4\frac{5}{8}$	$\frac{1}{2}$	$1\frac{3}{4}$
32	16	3	4	$2\frac{3}{8}$	$2\frac{1}{4}$	$1\frac{3}{4}$	1	$3\frac{3}{4}$	$\frac{5}{16}$	$3\frac{1}{2}$	$\frac{5}{16}$	$10\frac{3}{8}$	2	$5\frac{3}{8}$	$\frac{5}{8}$	2
36	18	$3\frac{3}{8}$	$4\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{5}{8}$	$1\frac{3}{4}$	1	$4\frac{1}{4}$	$\frac{3}{8}$	4	$\frac{3}{8}$	$11\frac{1}{2}$	2	6	$\frac{5}{8}$	$2\frac{1}{4}$
40	20	$3\frac{3}{4}$	$4\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{3}{4}$	2	$1\frac{1}{8}$	$4\frac{5}{8}$	$\frac{3}{8}$	$4\frac{1}{4}$	$\frac{3}{8}$	$12\frac{3}{4}$	$2\frac{3}{8}$	$6\frac{3}{8}$	$\frac{5}{8}$	$2\frac{3}{8}$
44	22	4	$5\frac{1}{4}$	$3\frac{5}{8}$	$3\frac{1}{4}$	$2\frac{3}{8}$	$1\frac{1}{4}$	$4\frac{3}{4}$	$\frac{3}{8}$	$4\frac{5}{8}$	$\frac{3}{8}$	$14\frac{1}{2}$	$2\frac{3}{8}$	$6\frac{3}{4}$	$\frac{3}{4}$	$2\frac{3}{4}$

All Dimensions are given in Inches.

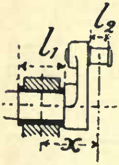


Fig. 271

The length of the main-bearing journal and the dimension x, depend in some measure on the design of the engine. In the Table d = diameter, l = length of journals and the upper figures are for 110, and the lower for 90 lbs. working pressure.

TABLE 27.—**Dimensions of Main Bearings, Crank Pins, and Crosshead Pins.**

Diam. of Cylinder.	Main Bearing.		Crank Pin.		Crosshead Pin.		Dimension of x.
D	d_1	l_1	d_2	l_2	d_3	l_3	$x=0{\cdot}9\,(l_1+l_2)$
10	$4\frac{1}{2}$	$7\frac{1}{4}$	$2\frac{5}{8}$	$3\frac{1}{4}$	2	$2\frac{3}{4}$	$9\frac{1}{4}$
	4	$7\frac{1}{4}$	$2\frac{1}{4}$	$3\frac{1}{4}$	2	$2\frac{3}{4}$	
12	$5\frac{1}{4}$	$8\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{5}{8}$	$2\frac{1}{4}$	$3\frac{1}{4}$	$10\frac{3}{4}$
	$4\frac{5}{8}$	$8\frac{1}{2}$	$2\frac{5}{8}$	$3\frac{5}{8}$	$2\frac{1}{4}$	$3\frac{1}{4}$	
14	$6\frac{3}{8}$	$9\frac{1}{2}$	$3\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{5}{8}$	$3\frac{5}{8}$	$12\frac{1}{4}$
	$5\frac{1}{4}$	$9\frac{1}{2}$	3	$4\frac{1}{4}$	$2\frac{5}{8}$	$3\frac{5}{8}$	
16	7	$10\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{5}{8}$	3	4	14
	6	$10\frac{3}{4}$	$3\frac{3}{8}$	$4\frac{5}{8}$	3	4	
18	$7\frac{5}{8}$	12	$4\frac{1}{4}$	$5\frac{1}{4}$	$3\frac{3}{8}$	$4\frac{3}{8}$	$15\frac{5}{8}$
	$6\frac{3}{4}$	12	$3\frac{3}{4}$	$5\frac{1}{4}$	$3\frac{3}{8}$	$4\frac{3}{8}$	
20	$8\frac{3}{4}$	$13\frac{1}{4}$	$4\frac{3}{4}$	$5\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{3}{4}$	$17\frac{1}{4}$
	$7\frac{5}{8}$	$13\frac{1}{4}$	$4\frac{1}{4}$	$5\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{3}{4}$	
22	10	$14\frac{3}{4}$	$5\frac{1}{4}$	$6\frac{3}{8}$	4	$5\frac{1}{4}$	$18\frac{1}{2}$
	$8\frac{3}{8}$	$14\frac{3}{4}$	$4\frac{5}{8}$	$6\frac{3}{8}$	4	$5\frac{1}{4}$	

All Dimensions are given in Inches.

TABLE 28.—**Dimensions of Piston Rods, Main Bearings, Crank Pins and Crosshead Pins for Engines with 90 to 110 lbs. Working Pressure.** (For Engine Frame, see page 30.)

Engines.		Revs. per Min.	Piston speed ft. per min.	Piston Rods.	Main Bearings.		Crank Pins.		Crosshead Pins.	
H	D	n	c	d	d	l	d	l	d	l
12	6	125	250	$1\frac{1}{8}$	—	—	—	—	$1\frac{3}{8}$	$1\frac{3}{4}$
16	8	110	283	$1\frac{3}{8}$	—	—	—	—	$1\frac{1}{2}$	$2\frac{1}{8}$
20	10	100	334	$1\frac{5}{8}$	$4\frac{1}{2}$	$7\frac{1}{4}$	$2\frac{5}{8}$	$3\frac{1}{4}$	2	$2\frac{3}{4}$
24	12	90	360	2	$5\frac{1}{4}$	$8\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{5}{8}$	$2\frac{1}{4}$	$3\frac{1}{4}$
28	14	85	396	$2\frac{1}{4}$	$6\frac{3}{8}$	$9\frac{1}{2}$	$3\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{5}{8}$	$3\frac{5}{8}$
32	16	80	426	$2\frac{3}{8}$	7	$10\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{5}{8}$	3	4
36	18	78	468	$2\frac{5}{8}$	$7\frac{5}{8}$	12	$4\frac{1}{4}$	$5\frac{1}{4}$	$3\frac{3}{8}$	$4\frac{3}{8}$
40	20	75	500	$2\frac{3}{4}$	$8\frac{3}{4}$	$13\frac{1}{4}$	$4\frac{3}{4}$	$5\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{3}{4}$
44	22	72	528	$3\frac{1}{4}$	10	$14\frac{3}{4}$	$5\frac{1}{4}$	$6\frac{3}{8}$	4	$5\frac{1}{4}$
48	24	70	560	$3\frac{3}{4}$	$11\frac{1}{4}$	$16\frac{3}{4}$	$5\frac{3}{4}$	$7\frac{1}{4}$	5	$5\frac{3}{4}$
56	28	61	570	$4\frac{3}{8}$	$12\frac{1}{2}$	$18\frac{3}{4}$	$6\frac{1}{2}$	8	$5\frac{3}{4}$	$6\frac{3}{8}$
64	32	$54\frac{1}{2}$	580	$5\frac{1}{4}$	14	$21\frac{1}{4}$	$7\frac{3}{4}$	$9\frac{1}{4}$	$6\frac{1}{2}$	$7\frac{1}{4}$
72	36	49	590	6	$15\frac{3}{4}$	24	$8\frac{3}{4}$	$10\frac{3}{8}$	$7\frac{1}{4}$	8
80	40	45	600	$6\frac{3}{4}$	$17\frac{1}{4}$	26	$9\frac{1}{2}$	$11\frac{5}{8}$	8	$9\frac{1}{4}$
88	44	41	600	$7\frac{5}{8}$	$18\frac{3}{4}$	$28\frac{3}{4}$	$10\frac{1}{2}$	$13\frac{1}{4}$	$8\frac{3}{4}$	$10\frac{3}{8}$

All Dimensions are given in Inches, except those in Column c.

Piston Rods.

Piston rods are now almost always made of either Bessemer or open-hearth steel

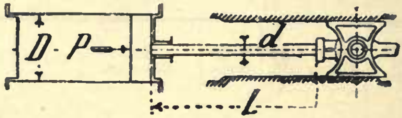


Fig. 272.

Both ends of the piston rod can be taken as fixed, then the diameter to resist crushing may be taken from the formula
$d = \cdot0324 \sqrt[4]{P\ L^2}$ where d = diameter of rod in inches, P the maximum pressure on the piston in lbs., and L the length of the rod. A more direct method of determining the diameter of the piston rod is in proportion to the area of the piston and the maximum pressure thereon.

Thus if A = area of piston and a = area of piston rod, then for a maximum boiler pressure of 80 lbs. $\frac{A}{41} = a$ or $A \times \cdot0244 = a$ gives a very safe strength for the smallest part of the rod, usually that part of the rod around the cottar hole at crosshead end, and as the rod has to be of a somewhat larger diameter in that part which passes through the cylinder cover, there is ample strength to resist crushing or bending.

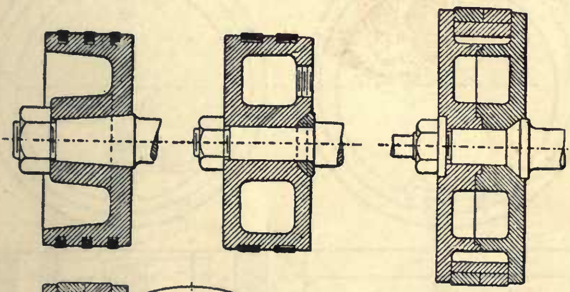
Another way is to give sufficient metal in the smallest area of the rod to give a maximum stress of $1\frac{3}{4}$ tons per square inch, in some cases the stress may be as high as 2 tons per square inch. The staff or main part of the rod is often made about $\frac{1}{16}''$ to $\frac{1}{8}''$ larger than the largest diameter where it enters the crosshead, thus giving a small shoulder, this is for allowing the rod to be trued up after wear, without disturbing the part which fits into the crosshead.

Pistons.

Of pistons for steam engines there are endless varieties, high speeds and high pressures rendering it difficult to construct a perfect piston, and in fact this difficulty is well shown by the number of new pistons that are constantly being advertised in the technical journals. A perfect piston should work silently, and keep steam tight for a reasonable length of time, and should be as near frictionless as possible; the number of parts should be few, and all bolts and nuts secured from working loose. The attachment of the piston to the rod is a matter of great importance, the cone and nut shown in figs. 273, 276 has been successfully used on small pistons, but it is questionable whether a parallel rod with collar and nut is not better for very large pistons, accidents having happened from the piston body being burst by gradually mounting up the cone, especially when the taper of the latter has been too slight; a good proportion for the cone is a taper of $1\frac{1}{2}$ inches of diameter to one foot of length.

Figs. 273, 274 show a form of plain piston fitted with Ramsbottom rings, figs. 275, 276, 277, a good and cheap form for small pistons with one inside and two outside rings, all of cast iron, fig. 284, a spiral coil-spring piston of cast iron, with cast-iron outside rings.

Fig. 273. Fig. 274. Fig. 275.



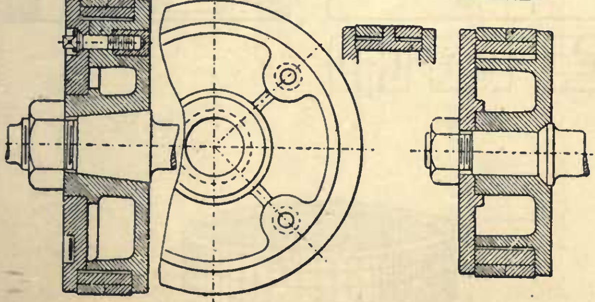


Fig. 276. Fig. 277.

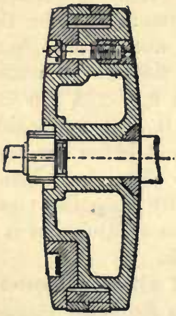


Fig. 278.

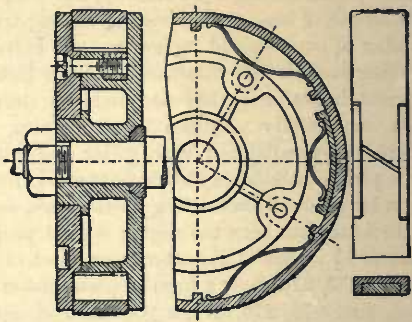


Fig. 279.

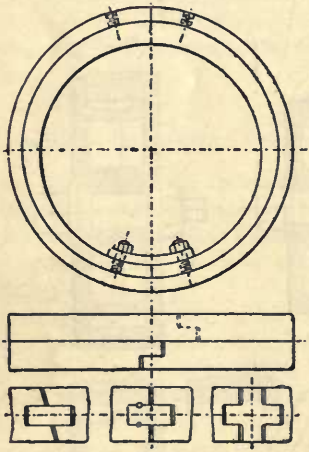


Fig. 280.

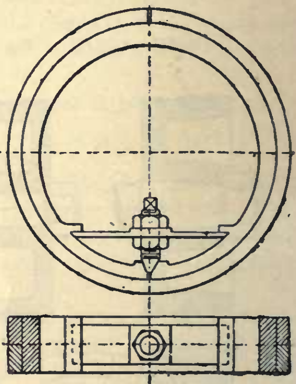


Fig. 281.

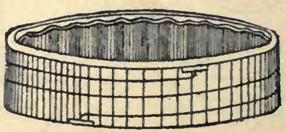


Fig. 282.—Cremer's Piston with spiral ring.

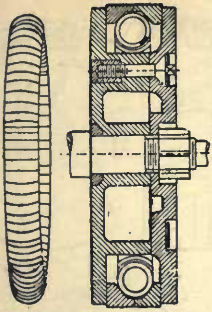


Fig. 283.

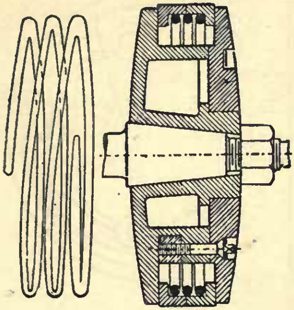


Fig. 284.

Fig. 284.—A Piston with helical spring bearing on outside cast-iron rings

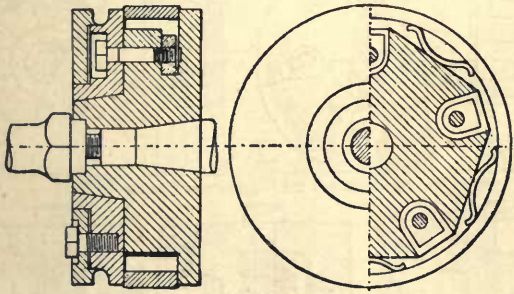


Fig. 285.

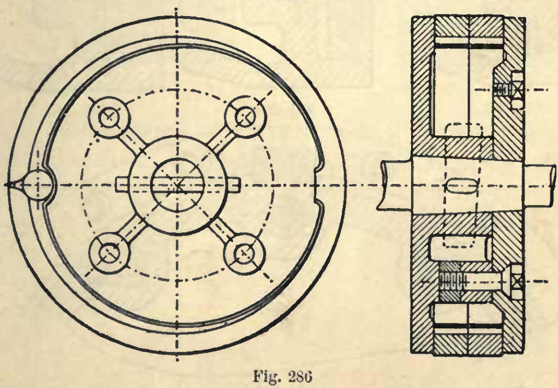


Fig. 286

Fig. 286.—Piston with two outside cast-iron rings and steel spring ring, with block at joint. This is a cheap form derived from an older form of locomotive piston, and has been much used for portable engines.

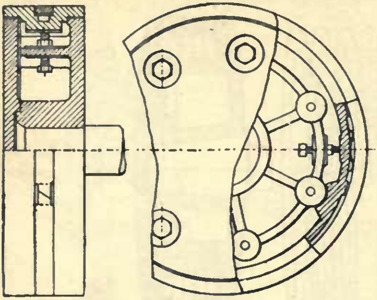


Fig. 287.

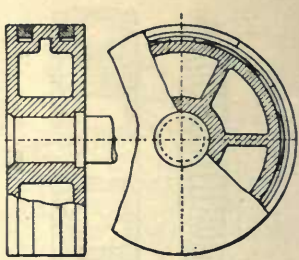
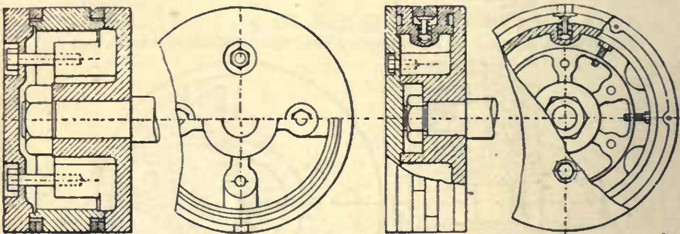


Fig. 288.


Figs. 289 and 290.—American Pistons.

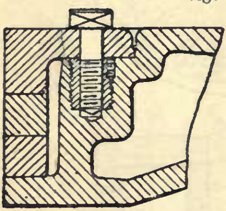


Fig. 291.

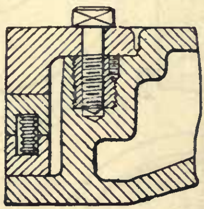


Fig. 292.

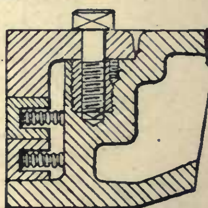


Fig. 293.

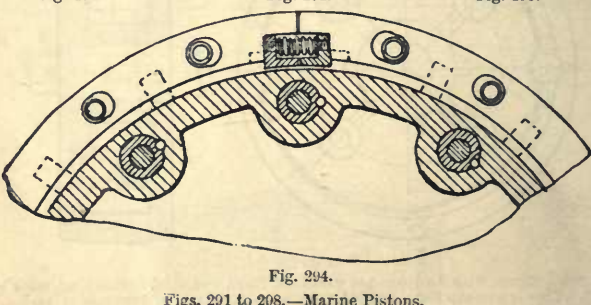


Fig. 294.

Figs. 291 to 208.—Marine Pistons.

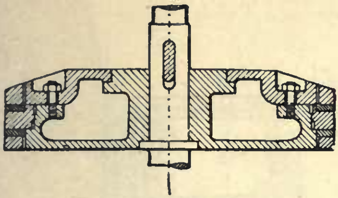


Fig. 295.

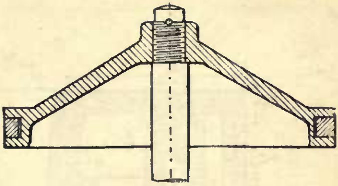


Fig. 296.

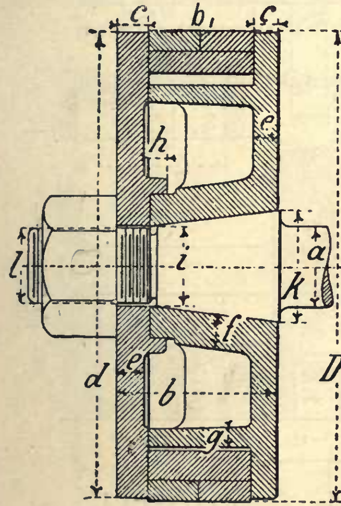


Fig. 297.

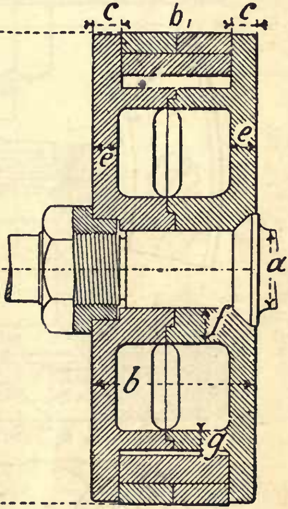


Fig. 298.

TABLE 29.—**Dimensions of Pistons from 6″ to 16″ Diameter, with Cover held on by Piston-rod Nut** (Figs. 297, 298.)

D	a	b	b_1	c	e	f	g	h	i	k	l
6	$1\frac{1}{8}$	3	2	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{3}{4}$	1
8	$1\frac{3}{8}$	$3\frac{1}{4}$	$2\frac{1}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{5}{16}$	$1\frac{3}{8}$	2	$1\frac{1}{4}$
10	$1\frac{5}{8}$	$3\frac{5}{8}$	$2\frac{3}{8}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{3}{8}$	$1\frac{1}{2}$
12	2	4	$2\frac{5}{8}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{7}{8}$
14	$2\frac{1}{4}$	$4\frac{1}{2}$	3	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{1}{2}$	$\frac{3}{8}$	$2\frac{1}{8}$	$3\frac{1}{4}$	2
16	$2\frac{3}{8}$	$4\frac{3}{4}$	$3\frac{1}{4}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$2\frac{5}{16}$	$3\frac{5}{8}$	$2\frac{1}{4}$

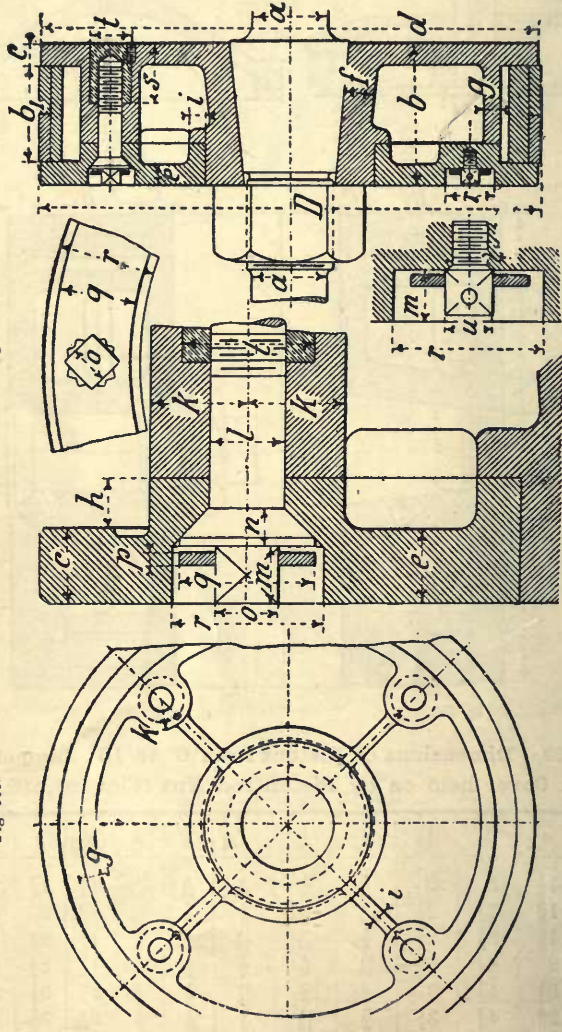


Fig. 299.

Fig. 300.

Fig. 301.

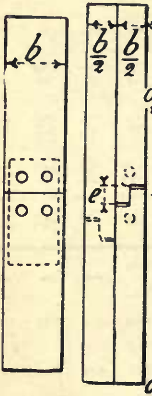
Fig. 302.

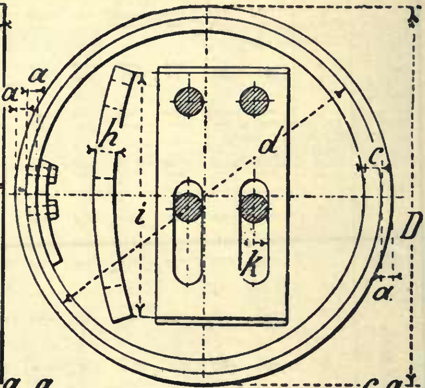
Fig 303.

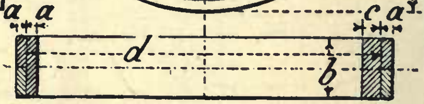
TABLE 30.—**Dimensions of Pistons with Cover held on by Bolts. From 14″ up to 40″ Diameter** (Fig. 299).

	NOTE.—All Dimensions are in Inches.										Cover bolts.										
D	*a*	*b*	b_1	*c*	*e*	*f*	*g*	*h*	*i*	*k*	No.	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*
14	$2\frac{1}{4}$	$4\frac{1}{2}$	3	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$	1	4	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	2	$1\frac{3}{8}$	$\frac{1}{2}$
16	$2\frac{3}{8}$	$4\frac{3}{4}$	$3\frac{1}{4}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	1	5	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	2	$1\frac{3}{8}$	$\frac{1}{2}$
18	$2\frac{5}{8}$	5	$3\frac{3}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	1	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	5	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{2}$	$\frac{1}{2}$
20	$2\frac{3}{4}$	$5\frac{1}{4}$	$3\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{4}$	1	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{9}{16}$	$1\frac{1}{4}$	5	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{2}$	$\frac{1}{2}$
22	$3\frac{1}{4}$	$5\frac{1}{2}$	$3\frac{5}{8}$	$\frac{15}{16}$	$\frac{3}{4}$	1	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	$1\frac{1}{4}$	6	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{2}$	$\frac{1}{2}$
24	$3\frac{3}{4}$	6	4	1	$\frac{13}{16}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{5}{8}$	$1\frac{3}{8}$	6	1	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{16}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$	$1\frac{3}{4}$	$\frac{5}{8}$
28	$4\frac{3}{8}$	$6\frac{3}{4}$	$4\frac{3}{4}$	1	$\frac{7}{8}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{3}{8}$	7	1	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{16}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$	$1\frac{3}{4}$	$\frac{5}{8}$
32	$5\frac{1}{4}$	$7\frac{3}{4}$	$5\frac{1}{4}$	$1\frac{1}{4}$	1	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{2}$	7	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	1	$\frac{3}{16}$	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{3}{4}$	2	$\frac{3}{4}$
36	6	$8\frac{3}{8}$	$5\frac{7}{8}$	$1\frac{1}{4}$	1	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{2}$	8	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	1	$\frac{3}{16}$	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{3}{4}$	2	$\frac{3}{4}$
40	$6\frac{3}{4}$	$8\frac{3}{4}$	6	$1\frac{3}{8}$	$1\frac{1}{16}$	2	1	1	1	$1\frac{5}{8}$	8	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{5}{8}$	1	$\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	3	$2\frac{1}{8}$	$\frac{3}{4}$

Fig. 306.







Figs. 308 and 307. Fig. 305.

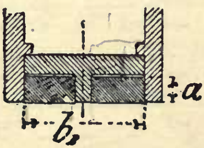


Fig. 304.

Figures 304 to 308 show an ordinary set of piston rings with tongue piece on inside ring, and the outside ring cut so as to prevent marking the cylinder and also to prevent the leakage of steam. This way of cutting the rings is not usually considered necessary, a diagonal cut being generally sufficient with or without the tongue pieces. Table 31 gives the dimensions of these rings.

TABLE 31.—**Dimensions of Piston Rings** (Figs. 304 to 308).

D	b	a	c	D_1	f	e	D	d	d_1	g	d	h	i	k
4	$1\frac{3}{4}$	$\frac{5}{16}$	$\frac{1}{2}$	$4\frac{1}{4}$	—	$\frac{3}{8}$	4	$3\frac{3}{8}$	$3\frac{7}{16}$	—	$3\frac{3}{8}$	$\frac{1}{4}$	2	$\frac{5}{16}$
6	2	$\frac{5}{16}$	$\frac{9}{16}$	$6\frac{1}{4}$	—	$\frac{1}{2}$	6	$5\frac{3}{8}$	$5\frac{7}{16}$	$\frac{1}{16}$	$5\frac{3}{8}$	$\frac{1}{4}$	$2\frac{3}{8}$	$\frac{5}{16}$
8	$2\frac{1}{8}$	$\frac{3}{8}$	$\frac{5}{8}$	$8\frac{5}{16}$	—	$\frac{5}{8}$	8	$7\frac{1}{4}$	$7\frac{5}{16}$	$\frac{1}{8}$	$7\frac{1}{4}$	$\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{16}$
10	$2\frac{3}{8}$	$\frac{3}{8}$	$\frac{5}{8}$	$10\frac{3}{8}$	$\frac{3}{16}$	$\frac{5}{8}$	10	$9\frac{1}{4}$	$9\frac{3}{8}$	$\frac{3}{16}$	$9\frac{1}{4}$	$\frac{5}{16}$	$3\frac{1}{4}$	$\frac{3}{8}$
12	$2\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{4}$	$12\frac{7}{16}$	$\frac{5}{16}$	$\frac{5}{8}$	12	$11\frac{1}{8}$	$11\frac{1}{4}$	$\frac{3}{16}$	$11\frac{1}{8}$	$\frac{5}{16}$	$3\frac{5}{8}$	$\frac{3}{8}$
14	3	$\frac{1}{2}$	$\frac{3}{4}$	$14\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{4}$	14	13	$13\frac{5}{16}$	$\frac{1}{4}$	13	$\frac{3}{8}$	4	$\frac{3}{8}$
16	$3\frac{1}{4}$	$\frac{1}{2}$	$\frac{13}{16}$	$16\frac{9}{16}$	$\frac{1}{2}$	$\frac{3}{4}$	16	15	$15\frac{1}{4}$	$\frac{3}{8}$	15	$\frac{3}{8}$	$4\frac{3}{8}$	$\frac{1}{2}$
18	$3\frac{3}{8}$	$\frac{5}{8}$	$\frac{7}{8}$	$18\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	18	$16\frac{3}{4}$	16	$\frac{1}{2}$	$16\frac{3}{4}$	$\frac{1}{2}$	$4\frac{3}{4}$	$\frac{1}{2}$
20	$3\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{8}$	$20\frac{5}{8}$	$\frac{11}{16}$	$\frac{7}{8}$	20	$18\frac{3}{4}$	$19\frac{1}{4}$	$\frac{5}{8}$	$18\frac{3}{4}$	$\frac{1}{2}$	$5\frac{1}{4}$	$\frac{1}{2}$
22	$3\frac{5}{8}$	$\frac{11}{16}$	1	$22\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	22	$20\frac{5}{8}$	21	$\frac{3}{4}$	$20\frac{5}{8}$	$\frac{1}{2}$	$5\frac{5}{8}$	$\frac{1}{2}$
24	4	$\frac{11}{16}$	1	$24\frac{3}{4}$	$\frac{7}{8}$	1	24	$22\frac{5}{8}$	23	1	$22\frac{5}{8}$	$\frac{1}{2}$	6	$\frac{5}{8}$
28	$4\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{8}$	$28\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{1}{16}$	28	$26\frac{1}{2}$	27	$1\frac{1}{4}$	$26\frac{1}{2}$	$\frac{1}{2}$	$6\frac{3}{8}$	$\frac{5}{8}$
32	$5\frac{1}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	33	$1\frac{1}{4}$	$1\frac{1}{4}$	32	$30\frac{1}{2}$	31	$1\frac{3}{8}$	$30\frac{1}{2}$	$\frac{9}{16}$	$6\frac{3}{4}$	$\frac{5}{8}$
36	$5\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	37	$1\frac{3}{8}$	$1\frac{1}{2}$	36	$34\frac{1}{4}$	$34\frac{7}{8}$	$1\frac{1}{2}$	$34\frac{1}{4}$	$\frac{5}{8}$	$7\frac{1}{4}$	$\frac{5}{8}$
40	6	$\frac{7}{8}$	$1\frac{3}{8}$	$41\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	40	$38\frac{1}{4}$	39	$1\frac{7}{8}$	$38\frac{1}{4}$	$\frac{5}{8}$	$7\frac{5}{8}$	$\frac{5}{8}$

All Dimensions are given in Inches.

Piston Rods.

Fig. 309.

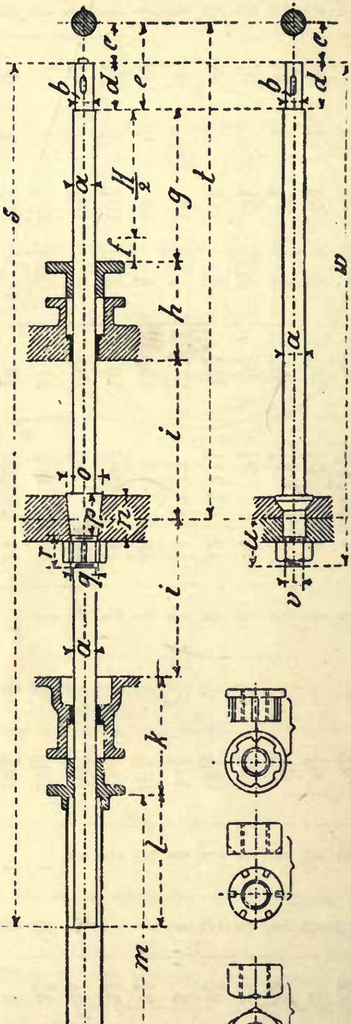


Fig. 313.

Figs. 310—312

TABLE 32.—**Dimensions of Piston Rods** (Figs. 309, 313).

NOTE.—All Dimensions are given in Inches.

Engine. H	Engine. D	a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	r	s	t	u	v	w
12	6	1 1/8	1	2 1/2	2 3/4	5 1/4	1 1/16	7 1/16	7	7 11/16	—	—	—	3	—	2 3/4	—	—	—	27	2 3/4	1 1/8	27 1/4
16	8	1 3/8	1 1/4	2 7/8	3 5/8	6 1/2	1 1/16	9 1/16	8 1/4	9 13/16	—	—	—	3 1/4	—	3	—	—	—	33 5/8	3 1/4	1 3/8	34
20	10	1 5/8	1 1/2	3 3/8	4 1/8	7 1/2	1 1/16	11 1/8	9 3/8	12	—	—	—	3 5/8	—	3 1/2	—	—	—	40	3 3/4	1 3/4	40 3/8
24	12	2	1 7/8	3 7/8	4 5/8	8 1/2	1 1/16	13 1/16	10 1/2	14 3/16	—	12	28	4	3 1/4	3 3/4	2 3/8	2 3/8	—	46 1/4	—	—	—
28	14	2 1/4	2 1/8	4 1/4	5	9 1/4	1 1/8	15 1/8	11 1/8	16 1/2	—	14	32	4 1/2	3 1/2	4	2 1/2	2 1/2	—	52	—	—	—
32	16	2 3/8	2 1/4	4 3/4	5 3/4	10 1/2	1 3/8	17 3/8	11 1/2	18 5/8	—	16	36	4 3/4	3 3/4	4 1/2	2 3/4	2 3/4	—	58	—	—	—
36	18	2 5/8	2 1/2	5 3/8	6 1/8	11 1/2	1 9/16	19 9/16	12 1/8	20 13/16	—	18	40	5	4	4 5/8	3	3	—	64	—	—	—
40	20	2 3/4	2 5/8	5 7/8	6 5/8	12 1/2	1 9/16	21 9/16	12 3/4	22 15/16	—	20	44	5 1/4	4 1/2	4 3/4	3 1/4	3 1/4	—	69 3/4	—	—	—
44	22	3 1/4	3 1/8	6 5/8	7 3/8	14	1 9/16	23 9/16	14 1/8	25 1/16	—	22	48	5 1/2	4 3/4	5	3 5/8	3 5/8	—	76 3/4	—	—	—

Steam Engine Cylinders.

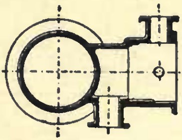


Fig. 314.

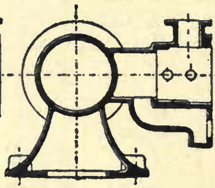


Fig. 315.

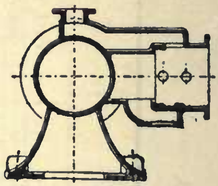


Fig. 316.

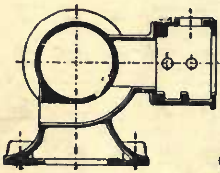


Fig. 317.

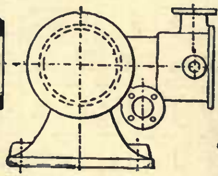


Fig. 318.

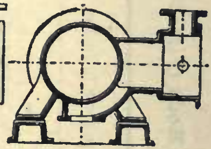


Fig. 319.

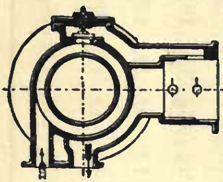


Fig. 320.

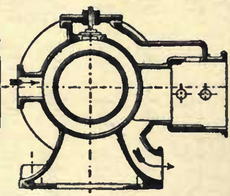


Fig. 321.

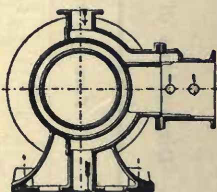


Fig. 322.

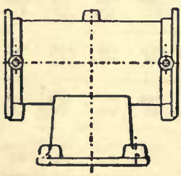


Fig. 323.

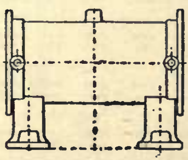


Fig. 324.

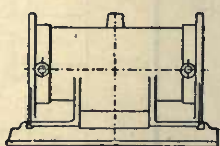
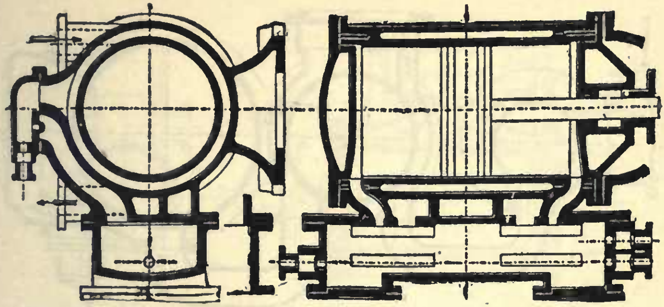
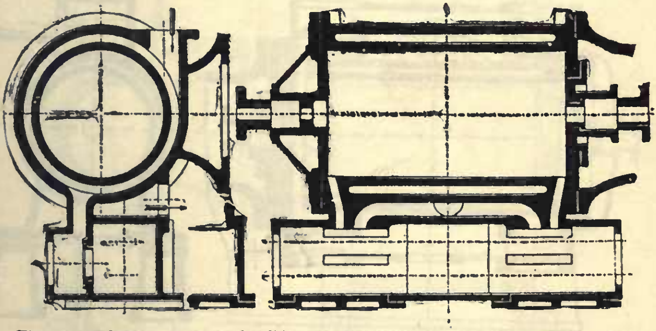


Fig. 325.

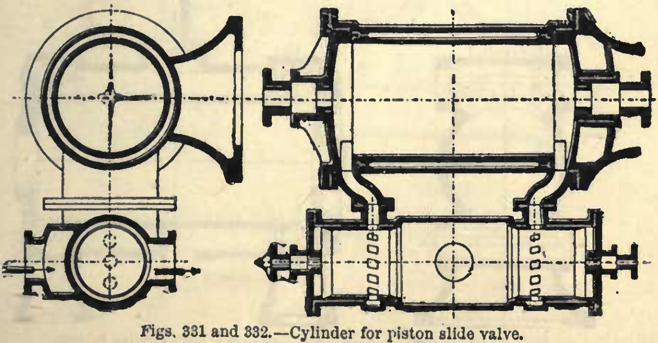
The above figures show different designs for cylinders of horizontal engines with and without steam jackets.



Figs. 327 and 328.—Cylinder for double slide valve gear, with the valves in two parts; an arrangement which reduces the length of the steam ports to the shortest length possible with slide valves



Figs. 329 and 330.—Cylinder for Rider's valve gear, with the valves in two parts to reduce length of steam ports.



Figs. 331 and 332.—Cylinder for piston slide valve.

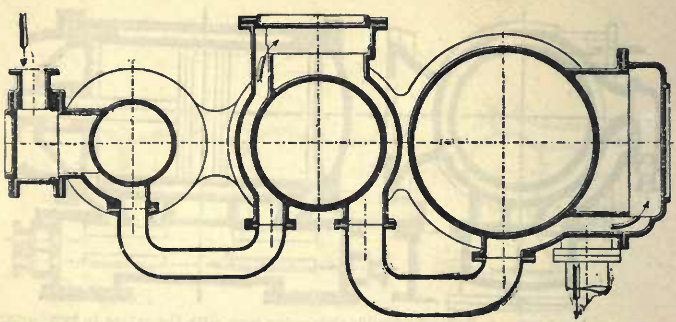
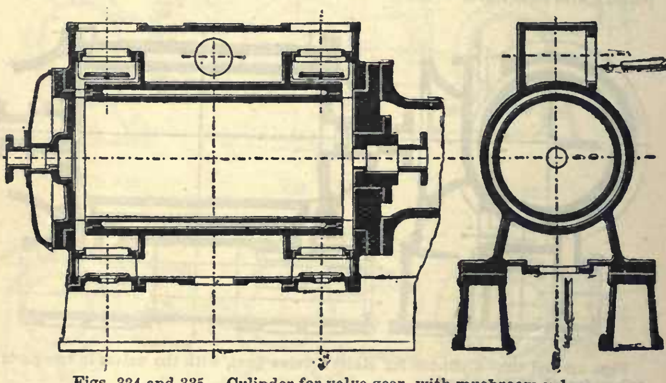
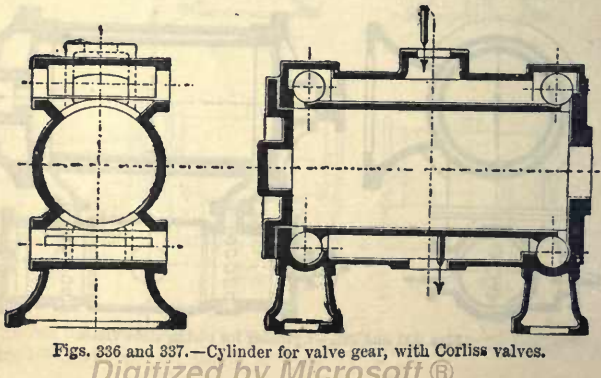


Fig. 333.—Cylinders for triple-expansion engine, designed for use with frame shown fig. 89, page 32. In this arrangement all the three slide valves are easy of access.



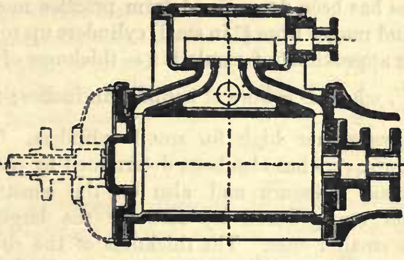
Figs. 334 and 335.—Cylinder for valve gear, with mushroom valves.



Figs. 336 and 337.—Cylinder for valve gear, with Corliss valves.

Fig. 338.

Fig. 339.



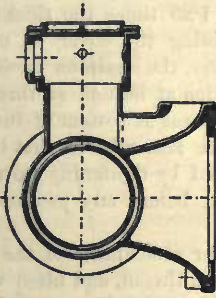


Fig. 340.

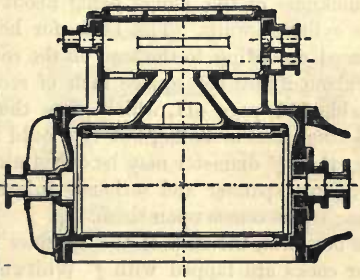


Fig. 341.

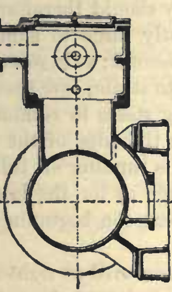


Fig. 342.

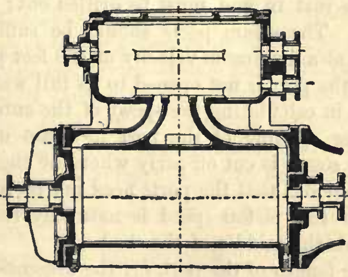


Fig. 343.

Figs. 338 and 339.—Cylinder for ordinary simple slide valve gear.
Figs. 340 and 341.—Cylinder for Meyer's slide valve gear.
Figs. 342 and 343.—Cylinder for Rider's valve gear.

In the above examples the slide valve chest is shown shorter than the cylinder ; this arrangement saves weight, but it is often more convenient to make it the same length as the cylinder and to bolt the stuffing boxes on.

The thickness of the cylinder walls in small or medium sized engines has been determined from practice more than from calculation, and ranges from $\frac{5}{8}''$ in small cylinders up to 9" bore; beyond that size an approximate formula is t = thickness of walls in inches = $\cdot75 + \frac{D}{100}$, where D = bore of cylinder in inches; this formula gives the thickness rather high for small cylinders. The thickness of the cylinder covers has also been determined by practice and depends on the steam pressure and also on the construction of the cover, whether strengthened by ribs for the larger cylinders or plain for the smaller ones. The thickness of the cover at the bolt circle is often made a little less than that of the cylinder flange; the thickness of the flange being about 1·25 times the thickness of the cylinder walls. The bolts for holding the cover on must be spaced according to the load on the cover, the stress ~~strain~~ on the bolts being about 2 tons per square inch of section at bottom of thread; (see table 151, page 411, which gives the areas at bottom of thread of the usual sizes of bolts), but it should be remembered that bolts smaller than $\frac{5}{8}''$ diameter may be overstrained by tightening up with an ordinary spanner and seriously injured before any pressure or working stress comes upon them.

The bosses on the ends of the cylinder for attachment of the Indicator cocks are tapped with $\frac{3}{4}''$ Whitworth thread, and fitted with screw plugs often made of iron, but if not occasionally removed are apt to rust in and must be drilled out; they should therefore be of brass. The steam ports should be sufficiently large to admit the steam at a maximum velocity of 100 feet per second; as in some valve gears the port is not opened to its full width to steam, care should be taken in calculating the speed of the entering steam to reckon only for the opening of the port to steam up to the edge of the valve. If the steam is cut off early when the engine is running on full load it is obvious that the ports need not be so large as for the later cut off, as the piston speed is naturally less near the beginning than towards the middle of the stroke.

The length of the port, *i.e.* the dimension measured at right angles to the length of the cylinder, depends on the design of the cylinder, and this being determined, the other dimension is readily calculated from the area.

In cylinder construction it is important to arrange bosses for connection of all pipes required, and specially to arrange for efficient means of draining and lubricating the cylinder barrel, for admitting steam to the jacket and draining the same in an efficient manner. Other important points are the means for neatly clothing the cylinder with some non-conducting medium.

It may be again mentioned here that in order to save multiplications of patterns, the cylinder as well as other parts of engines should be designed symmetrically about their centre lines, so that a cylinder by being turned end for end may be used for either a right or a left hand engine.

Clearance space in Steam Engine Cylinders.—A small distance between the piston at each end of the stroke and the cylinder covers is necessary in order to prevent the piston from coming into contact with the covers. This space, called "clearance," should be as small as possible, but cannot be reduced beyond a certain limit, as the length of the connecting rod alters by wear and adjustment; therefore the clearance is seldom less than $\frac{5}{16}''$ in small engines and somewhat more in larger engines. In calculating the amount of steam used in an engine this space has to be added to the volume swept by the piston in its stroke; in addition to the clearance space at the end of the stroke, there are the steam passages, and if a double slide valve gear, the hollow part of the main valve to be taken in. Fig. 344 shows the clearance space and steam passage shaded at the left hand end of the cylinder. In this case it amounts to 450 cubic inches, the volume swept by the piston equals 7320 cubic inches $\frac{450 \times 100}{7320} = 6{\cdot}1$ per cent., so the clearance space amounts to 6·1 per cent. of the volume of the stroke. Table 33 shows the clearance spaces for different varieties of ports.

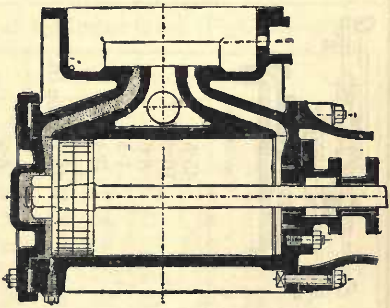
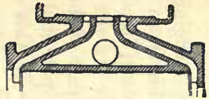
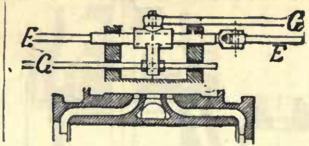
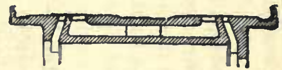


Fig. 344.

TABLE 33.—**Percentage of Clearance Space with Steam Ports of different Design.**

Engine.	For the Normal Velocity of the Steam.	 Fig. 345.	 Fig. 346.	 Fig. 347.
D=10	Steam port .	52·5	45·5	23·5
	Clearance . .	16·2	16·2	16·2
H=16	Recess in cover .	8·6	8·6	8·6
	Recess in cover .	1·5	1·5	1·5
n=120	Ring in cylinder	3·4	3·4	3·4
	Total .	82·2 cube in.	75·2 cube in.	53·2 cube in.
	Per cent. of cylinder capacity	7	6·5	4·5
D=22	Steam port .	687·3	633·3	244·3
	Clearance . .	114·7	114·7	114·7
H=40	Recess in cover .	24·6	24·6	24·6
	Recess in cover .	21·5	21·5	21·5
n=65	Ring in cylinder	20·0	20·0	20·0
	Total .	868·1 cube in.	814·1 cube in.	425·1 cube in.
	Per cent. of cylinder capacity	6	5·7	3

Feet for Cylinders.

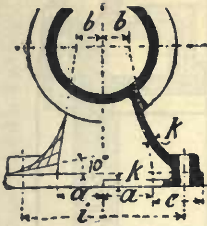


Fig. 348.

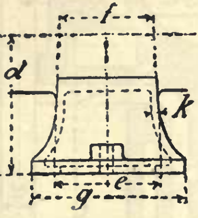


Fig. 349.

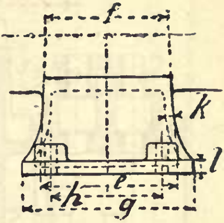


Fig. 350.

TABLE 34.—**Dimensions of Cylinder Feet** (Figs. 348—350).

Engines.		Note.—All Dimensions are in Inches.										
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*
20	10	5	3	5	18	12	$10\frac{1}{2}$	17	—	16	$\frac{3}{4}$	$1\frac{1}{2}$
24	12	$5\frac{1}{2}$	$3\frac{1}{2}$	6	20	14	12	20	14	18	$\frac{3}{4}$	$1\frac{3}{4}$
28	14	$6\frac{1}{2}$	4	$6\frac{1}{2}$	22	16	14	22	16	21	$\frac{7}{8}$	2
32	16	$7\frac{1}{2}$	$4\frac{1}{2}$	$7\frac{1}{4}$	24	18	16	25	18	23	1	$2\frac{1}{4}$
36	18	8	$5\frac{1}{2}$	8	26	20	$18\frac{1}{2}$	$27\frac{1}{2}$	19	26	$1\frac{1}{8}$	$2\frac{3}{8}$
40	20	9	6	$8\frac{1}{2}$	28	22	19	30	20	28	$1\frac{1}{4}$	$2\frac{1}{2}$
44	22	$9\frac{1}{2}$	7	9	30	24	21	32	22	30	$1\frac{1}{4}$	$2\frac{5}{8}$

The expansion of the metal of the cylinder due to the difference in temperature between the cylinder at atmospheric and working heat has occasionally been allowed for by arranging the cylinder foot on a sliding bed, as in fig. 351. The difference in length is given in Table 35. It will be seen that the amount is small, and usually no allowance need be made for it.

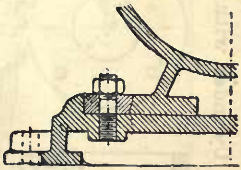


Fig. 351.

TABLE 35.

Stroke of Engine in Inches.	16	24	32	39	78
Amount of expansion, or increase of length —inches . . .	·0196	·0315	·0432	·0629	·1377

Cylinder with Steam Jacket (see TABLE 36).

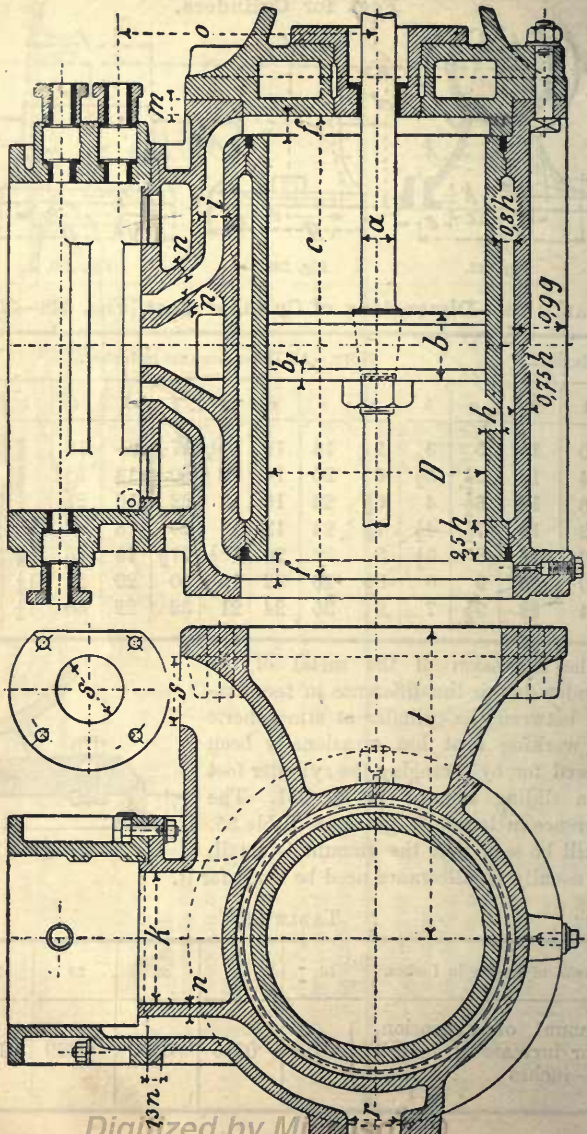


Fig. 353.

Fig. 352.

TABLE 36.—**Dimensions of Cylinders from 10″ up to 22″ diameter for Double Slide Valves**

(Figs. 352, 353, 357, 358.)

Engine.		Piston.									Steam Ports.						Bolts.		Bolts.					
H	D	*a*	*b*	b_1	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	No.	*p*	No.	*q*	*r*	*s*	*t*	*u*
16	8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
20	10	$1\frac{5}{8}$	$3\frac{5}{8}$	$\frac{5}{8}$	24	1	13	$\frac{13}{16}$	$4\frac{1}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$5\frac{3}{8}$	$18\frac{1}{4}$	1	$\frac{5}{8}$	$14\frac{1}{4}$	4	$\frac{7}{8}$	8	$\frac{3}{4}$	$2\frac{3}{8}$	$2\frac{3}{4}$	18	$5\frac{1}{4}$
24	12	2	4	$\frac{11}{16}$	$28\frac{3}{8}$	$1\frac{5}{16}$	$15\frac{1}{2}$	$\frac{7}{8}$	$4\frac{1}{4}$	$\frac{7}{8}$	1	$6\frac{3}{4}$	$20\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$	$16\frac{1}{2}$	5	1	8	$\frac{7}{8}$	3	$3\frac{5}{8}$	20	$6\frac{3}{8}$
28	14	$2\frac{1}{4}$	$4\frac{1}{2}$	$\frac{3}{4}$	33	$1\frac{1}{2}$	18	1	$4\frac{5}{8}$	$\frac{15}{16}$	$1\frac{1}{8}$	$8\frac{3}{8}$	$23\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$18\frac{3}{4}$	5	$1\frac{1}{8}$	10	$\frac{7}{8}$	$3\frac{5}{8}$	$4\frac{3}{8}$	22	$7\frac{5}{8}$
32	16	$2\frac{3}{8}$	$4\frac{3}{4}$	$\frac{3}{4}$	$37\frac{1}{4}$	$1\frac{7}{8}$	$20\frac{1}{2}$	$1\frac{1}{16}$	$4\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{4}$	10	$25\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{8}$	$21\frac{1}{4}$	6	$1\frac{1}{8}$	10	$\frac{7}{8}$	$4\frac{1}{4}$	5	24	$8\frac{3}{4}$
36	18	$2\frac{5}{8}$	5	$\frac{13}{16}$	$41\frac{5}{8}$	$2\frac{3}{16}$	23	$1\frac{3}{16}$	$5\frac{1}{4}$	1	$1\frac{3}{8}$	$11\frac{5}{8}$	$28\frac{1}{2}$	$1\frac{3}{8}$	$\frac{7}{8}$	$22\frac{1}{2}$	6	$1\frac{1}{4}$	10	1	$4\frac{5}{8}$	$5\frac{5}{8}$	26	$9\frac{5}{8}$
40	20	$2\frac{3}{4}$	$5\frac{1}{4}$	$\frac{7}{8}$	$45\frac{7}{8}$	$2\frac{9}{16}$	$25\frac{1}{2}$	$1\frac{1}{4}$	$5\frac{1}{2}$	1	$1\frac{5}{8}$	$13\frac{1}{4}$	31	$1\frac{1}{2}$	1	$25\frac{5}{8}$	8	$1\frac{1}{4}$	12	1	$5\frac{1}{4}$	$6\frac{3}{8}$	28	10
44	22	$3\frac{1}{4}$	$5\frac{1}{2}$	$\frac{15}{16}$	$50\frac{1}{8}$	$2\frac{3}{4}$	28	$1\frac{5}{16}$	$5\frac{1}{2}$	$1\frac{1}{16}$	$1\frac{3}{4}$	15	33	$1\frac{1}{2}$	1	$28\frac{1}{4}$	8	$1\frac{3}{8}$	14	1	$5\frac{3}{4}$	$7\frac{1}{4}$	30	$10\frac{3}{8}$

All Dimensions are given in Inches.

Steam Jackets.

The use of steam jackets is now almost universally adopted in all but very small cheap engines. A well arranged steam jacket tends to keep the steam dry in the cylinder, and slightly reduces the

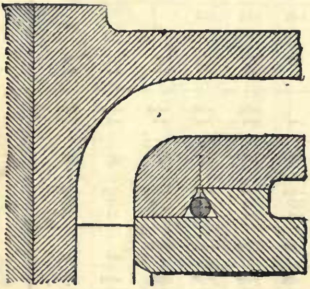


Fig. 354.

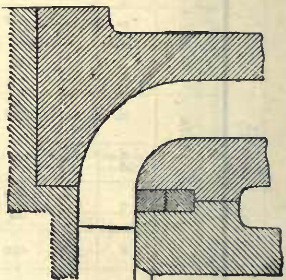


Fig. 355.

amount of steam used per horse power. In some cases the cylinder and jacket are in one casting, but usually the working barrel or liner is separate from the cylinder body, and forced steam tight into it by a hydraulic press. The liner can be made of harder iron than would be safe, if the whole were in one casting. Figs. 352, 353 show methods of rendering the joint between liner and body tight by means of packing. In fig. 353, the packing consists of a copper ring caulked in, but for small work this is quite unnecessary. A good fit is not difficult to obtain, and when the liner is either shrunk in by warming the body, or forced in cold by the hydraulic press, a leak is of very rare occurrence.

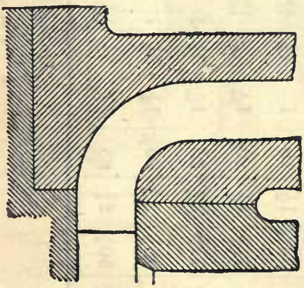


Fig. 356.

Cylinder for Double Slide Valves.

For dimensions see Table 36, p. 104.

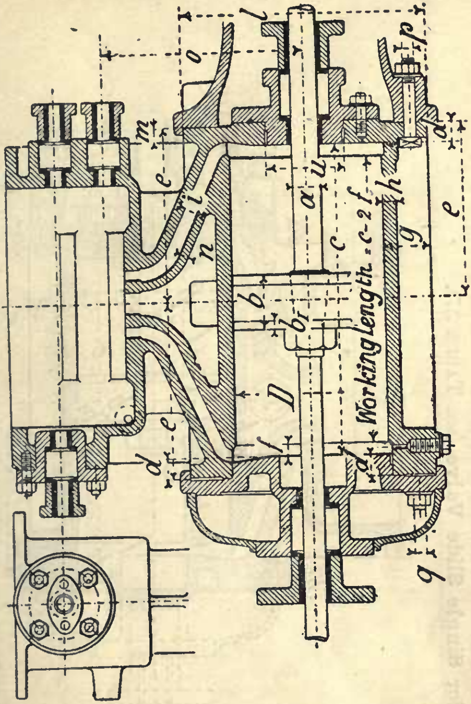


Fig. 358.

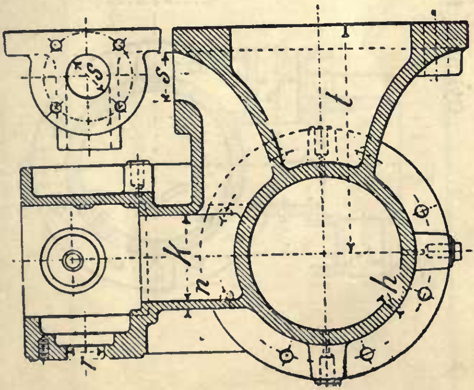


Fig. 357.

Cylinder for Single Slide Valve (see Table 37).

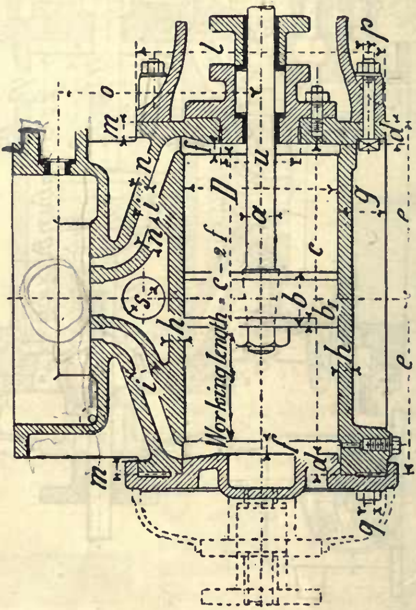


Fig. 360.

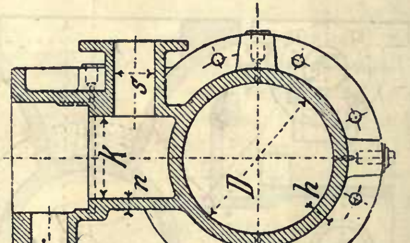


Fig. 359

TABLE 37.—Dimensions of Cylinders from 6″ up to 14″ diameter for Single Slide Valve (Figs. 359, 360).

Engine.		Piston.									Steam Ports.						Bolts.		Bolts.				
H	D	a	b	b_1	c	d	e	f	g	h	i	k	l	m	n	o	No.	p	No.	q	r	s	u
12	6	$1\frac{1}{8}$	3	$\frac{1}{2}$	$15\frac{3}{8}$	$\frac{9}{16}$	$8\frac{1}{4}$	$\frac{3}{4}$	$3\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$3\frac{1}{4}$	$13\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{2}$	—	4	$\frac{3}{4}$	6	$\frac{3}{4}$	$1\frac{3}{8}$	$1\frac{3}{4}$	$3\frac{1}{4}$
16	8	$1\frac{3}{8}$	$3\frac{1}{4}$	$\frac{9}{16}$	$19\frac{5}{8}$	$\frac{11}{16}$	$10\frac{1}{2}$	$\frac{3}{4}$	4	$\frac{3}{4}$	$\frac{5}{8}$	$4\frac{3}{8}$	16	$\frac{7}{8}$	$\frac{5}{8}$	—	4	$\frac{3}{4}$	6	$\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$4\frac{1}{4}$
20	10	$1\frac{5}{8}$	$3\frac{5}{8}$	$\frac{5}{8}$	24	1	13	$\frac{13}{16}$	$4\frac{1}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$5\frac{5}{8}$	$18\frac{1}{4}$	1	$\frac{5}{8}$	$14\frac{1}{4}$	4	$\frac{7}{8}$	8	$\frac{3}{4}$	$2\frac{3}{8}$	$2\frac{3}{4}$	$5\frac{1}{4}$
24	12	2	4	$\frac{11}{16}$	$28\frac{5}{8}$	$1\frac{5}{16}$	$15\frac{1}{2}$	$\frac{7}{8}$	$4\frac{1}{4}$	$\frac{7}{8}$	1	$6\frac{3}{4}$	$20\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$	$16\frac{1}{2}$	5	1	8	$\frac{7}{8}$	3	$3\frac{5}{8}$	$6\frac{3}{8}$
28	14	$2\frac{1}{4}$	$4\frac{1}{2}$	$\frac{3}{4}$	33	$1\frac{1}{2}$	18	1	$4\frac{5}{8}$	$\frac{15}{16}$	$1\frac{1}{8}$	$8\frac{3}{8}$	$23\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$18\frac{3}{4}$	5	$1\frac{1}{4}$	10	$\frac{7}{8}$	$3\frac{5}{8}$	$4\frac{3}{8}$	$7\frac{5}{8}$

All Dimensions are given in Inches.

Attachment of Cylinders to Girder Frames.

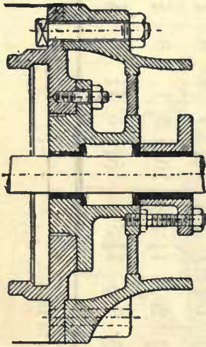


Fig. 361.

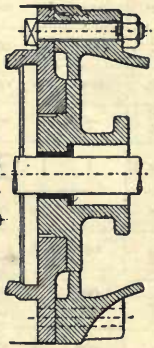


Fig. 362.

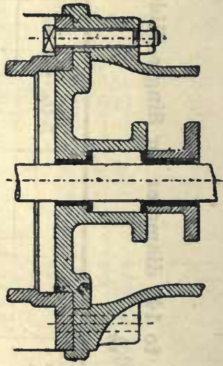


Fig. 363.

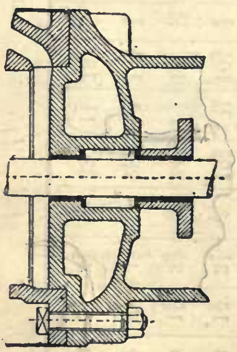


Fig. 364.

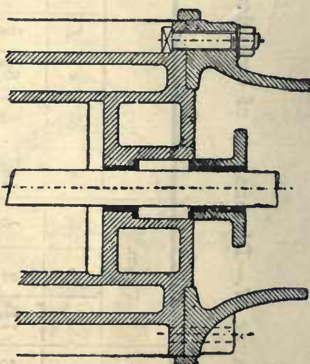


Fig 365.

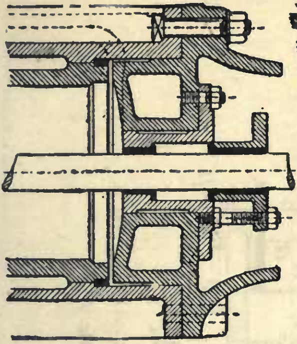


Fig. 366.

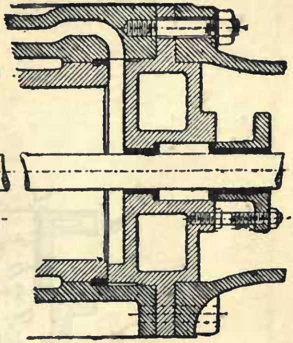


Fig. 367.

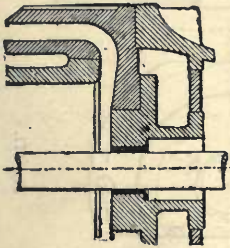


Fig. 368.

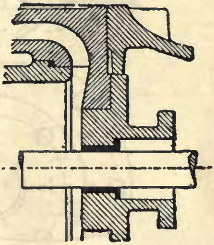


Fig. 369.

Figs. 361 to 369 show different ways of bolting cylinders on to girder frames. Figs. 362, 363, 369 show the simplest and perhaps best way; in fig. 362 the small cover can be removed without disturbing any other part, but in fig. 363 the girder and cylinder must be parted in order to take the cover off, and is therefore not so convenient as figs. 362 and 369. In figs. 364, 366, the girder makes a steam joint with the cylinder; not a good arrangement.

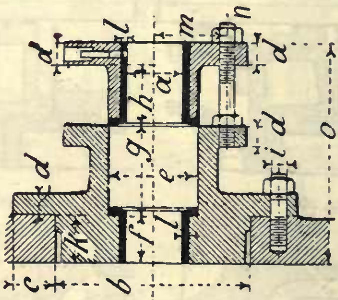


Fig. 372.

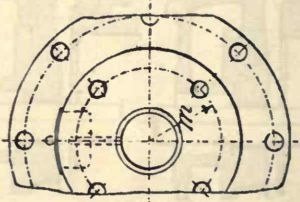


Fig. 371.

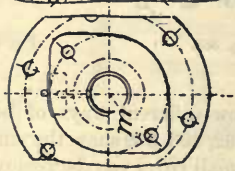


Fig. 370.

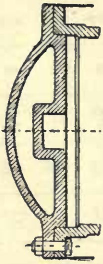
TABLE 38.—Dimensions of Front Cylinder Covers and Glands (Figs. 370—372).

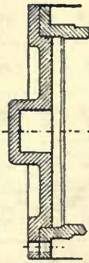
Engine.										Bolts.					Bolts.		
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	No.	*i*	*k*	*l*	*m*	No.	*n*	*o*
12	6	1⅛	3¼	1½	¾	2⅜	¾	2⅝	1¾	4	⅝	1⅜	$\frac{3}{16}$	2⅜	2	⅝	7
16	8	1⅜	4¼	2	⅞	2¾	1¼	3	1⅞	4	⅝	1½	$\frac{3}{16}$	2½	2	⅝	8¼
20	10	1⅝	5¼	2¼	1	3	1⅜	3⅜	2⅛	6	⅝	1½	$\frac{3}{16}$	2⅞	2	¾	9⅜
24	12	2	6⅜	2¼	1	3⅝	1⅝	4	2⅜	6	¾	1¾	¼	$3\frac{3}{16}$	3	¾	10½
28	14	2¼	7⅝	2¼	1	3¾	1¾	4¼	2½	6	¾	2	¼	3¼	3	¾	11⅛
32	16	2⅜	8¾	2⅜	1⅛	4	2	4¼	2⅝	6	¾	2½	$\frac{5}{16}$	3⅜	3	¾	11½
36	18	2⅝	9⅝	2⅜	1⅛	4¼	2¼	4⅜	2¾	6	¾	2¾	$\frac{5}{16}$	3½	3	¾	12⅛
40	20	2¾	10	2⅜	1¼	4⅜	2⅜	4⅝	3	8	¾	3	$\frac{5}{16}$	$3\frac{9}{16}$	3	¾	12¾
44	22	3¼	10⅜	2⅜	1¼	5	2¾	5	3⅜	8	¾	3¼	$\frac{5}{16}$	4	4	⅞	14⅛

All Dimensions are given in Inches.

Back Cylinder Covers, with and without Tail Rod Guides.

Figs. 373 & 374.





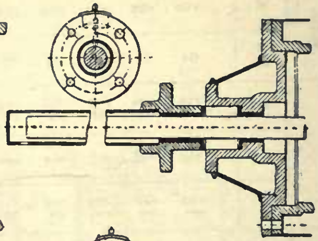


Fig. 375

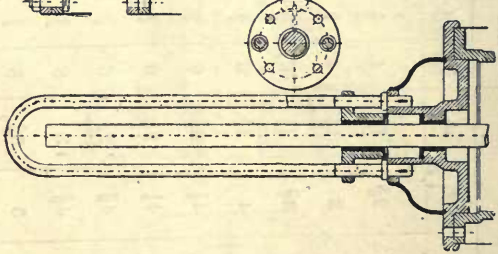


Fig 376.

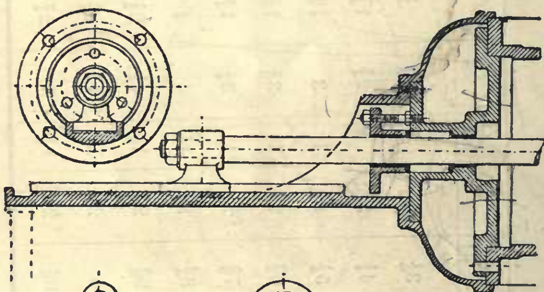


Fig. 377.

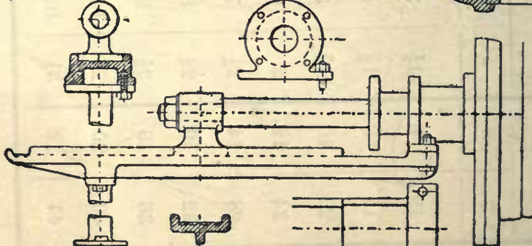


Fig. 378

Fig. 373 shows a chambered cover cast in one piece; fig 374 a plain back cover. Fig. 375 a simple tail rod guide formed by a stuffing box and gland in back cover and protected by a covering of iron pipe, a very neat arrangement and one that is much used; fig. 376, the guide for the tail rod, is the same as in the last figure; the rod is protected by a pair of guard rails, an arrangement which is neither so neat nor good as the tube in fig. 375. In figs. 377, 378 the tail rod guide is more elaborate, being guided by a planed bar and slipper, a more expensive arrangement than those of figs. 375, 376, but more suitable for large engines. Figs. 379, 380 show further methods of guiding the tail rod.

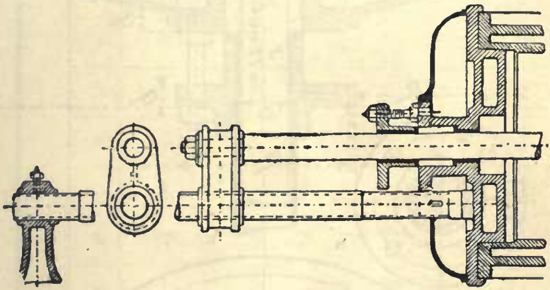


Fig. 379.

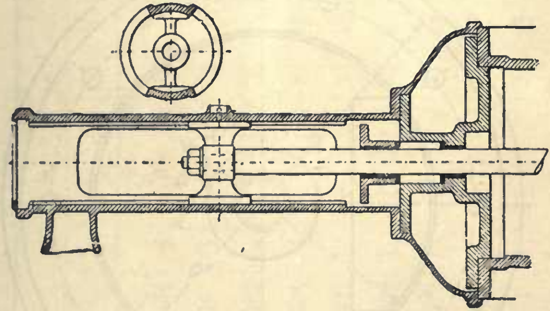


Fig 380

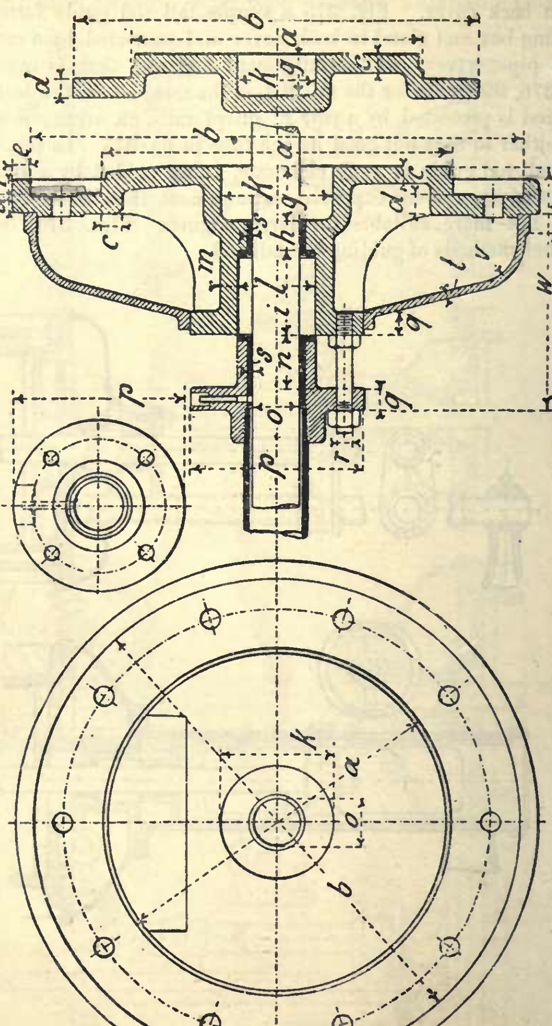


Fig. 383.

Fig. 382.

Fig. 384.

Fig. 381.

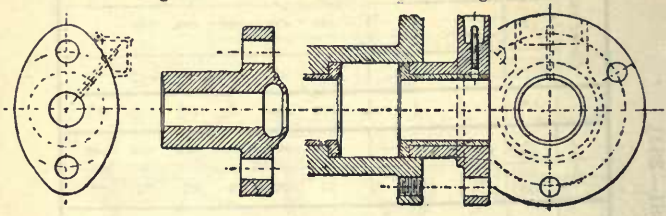
TABLE 39.—**Dimensions of Back Cylinder Covers** (Figs. 381—384).

Engines.																		Bolts.			Cover casing.		
H.	D.	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	No.	*r*	*s*	*t*	*u*	*v*
12	6	6½	13⅞	¾	1	—	—	1⅜	—	—	2¾	—	—	—	—	—	—	—	—	—	—	—	—
16	8	8½	16	¾	1	—	—	1¾	—	—	3½	—	—	—	—	—	—	—	—	—	—	—	—
20	10	10½	18¼	⅞	1	—	—	2¼	—	—	4½	—	—	—	—	—	—	—	—	—	—	—	—
24	12	12½	20½	⅞	1	⅝	⅝	2⅜	1⅝	4	4⅝	3⅝	¾	2⅜	2	8¼	1	3	¾	¼	5/16	⅝	
28	14	14¾	23¼	1	1¼	⅝	⅝	2⅝	1¾	4¼	4¾	3¾	¾	2½	2¼	8½	1	3	¾	¼	5/16	⅝	2½
32	16	16¾	25¾	1	1¼	¾	⅝	2¾	2	4¼	5¼	4	⅞	2⅝	2⅜	8⅝	1⅛	3	¾	5/16	⅜	¾	2½
36	18	18¾	28½	1⅛	1⅜	⅞	¾	3	2¼	4⅜	6	4¼	⅞	2¾	2⅝	9¼	1⅛	3	¾	5/16	⅜	¾	2¾
40	20	20¾	31	1¼	1½	1	¾	3¼	2⅜	4⅝	6¼	4⅜	⅞	3	2¾	9⅜	1¼	3	¾	5/16	7/16	⅞	3
44	22	22¾	33	1¼	1½	1	1	3½	2¾	5	6½	5	1	3⅜	3¼	10⅛	1¼	4	⅞	5/16	½	1	3¼

All Dimensions are given in Inches.

Stuffing Boxes and Glands for Piston Rods.

Fig. 385. Fig. 386.



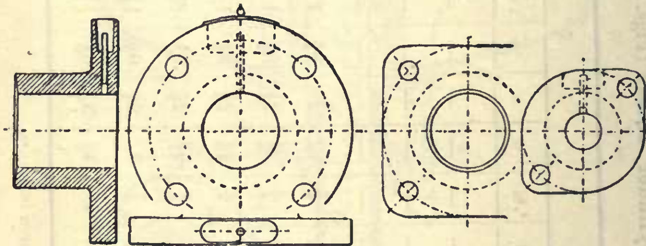


Fig. 387. Fig. 388.

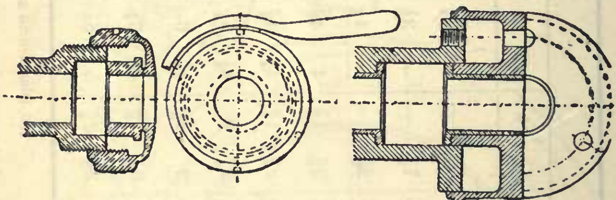


Fig. 389. Fig. 390.

Figs. 385, 386 show ordinary simple stuffing boxes and glands; figs. 389, 390 glands of old fashioned type having the disadvantage of often corroding up round the outside of the stuffing box where the cap goes over. Fig. 391 shows the very ingenious device of Mr.

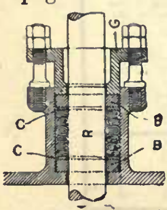


Fig. 391.

Yarrow especially adapted for high speed engines. The piston rod has often given trouble in high speed engines by getting very hot, the reason being that the crosshead guides and cylinder bore are seldom in perfect truth with one another, so that a severe side strain is exerted by the rod against the gland. In Mr. Yarrow's gland this is obviated by introducing loose collars B, B, between the packing and the gland G; these collars are a good fit on the rod but free from touching the sides of the stuffing box; the gland is bored larger than the rod R, but is turned a good fit in the stuffing box. It will be seen in this arrangement if the rod R is guided by the crosshead slightly out of truth with the cylinder bore, the loose collars will allow it to move in its own path freely, the elasticity of the packing allowing of this, and that no severe strain can come upon the gland causing the rod to heat.

Fig. 392 shows a gland and packing made by the United States Metallic Packing Co., Ltd., of Bradford.

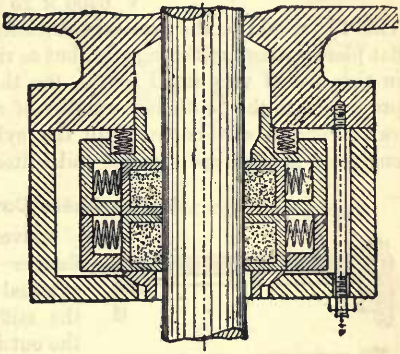
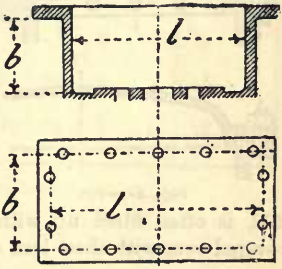


Fig. 392.

Slide Valve Chests and Covers.

Slide valve chests and their covers are made of cast iron; for calculating the strength of the cover and chest walls, they are considered as plates held fast round the edges (figs. 393, 394).



Figs. 393 and 394.

Thickness of walls, $\delta = \cdot 5l.\ \sqrt{\frac{Ph}{kl}}$

Stress $k = 0 \cdot 25\ \frac{Pbl}{\delta^2} =$ for cast iron 6400 lbs. per square inch.

$P =$ press in lbs. per square inch.
$h =$ breadth of walls.
$l =$ length.

Example.—An engine with $15\frac{3}{4}''$ diameter of piston $27\frac{1}{2}''$ stroke, 88·2 lbs. pressure of steam.

(1.) For the valve chest walls,

$l = 22''$; $b = 11''$; $k = 6400$ lbs. per square inch.

$$\delta = \cdot 5 \times 22\ \sqrt{\frac{88 \cdot 2 \times 11}{6400 \times 22}} = \cdot 91'' \text{ about } \tfrac{7}{8}''.$$

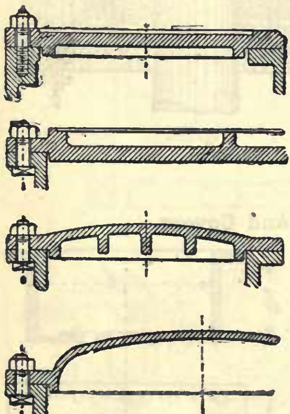
(2.) For the valve chest cover,

$l = 25''$; $b = 15''$; $k = 6400$ lbs. per square inch

$$\delta = \cdot 5 \times 25 \sqrt{\frac{88 \cdot 2 \times 15}{6400 \times 25}} = 1\tfrac{1}{16}'' \text{ inches.}$$

This $1\frac{1}{16}''$ would be the necessary thickness if the cover were a plain flat plate without stiffening ribs, but as ribs are generally used, except in the case of very small covers, the thickness may be reduced in proportion to the depth and number of ribs. In small engines the valve chest is cast in one with the cylinder, but with very large engines it is often cast separate and bolted on.

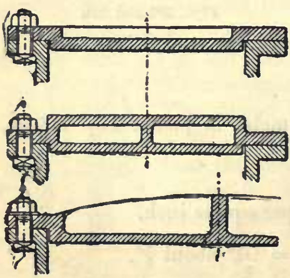
Valve Chest Covers.



Figs. 395—398.

Valve chests have either internal flanges as figs. 395 and 402, or external flanges as figs. 396 to 401; the stiffening ribs are usually on the outside as figs. 396, 402.

The design shown in fig. 402, where the valve chest is finished off with a bold curve, is one that has been largely used on portable engines, and gives an exceedingly neat appearance; but it is not easy to clothe a valve chest with lagging when the corners are round, as when the corners are square and a small external flange cast on the chest to receive the lagging screws. The cover when ribbed as in fig. 402, is often filled up with non-conducting material and neatly covered over with sheet iron or steel.



Figs. 399—401.

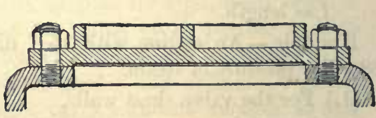


Fig. 402.

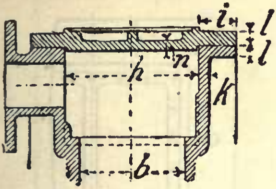


Fig. 403.

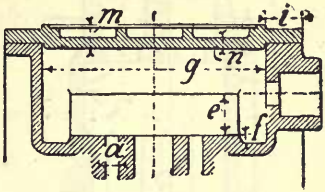


Fig. 404.

TABLE 40.—**Dimensions of Valve Chests of Single Slide Valves** (Figs. 403, 404).

Engines.												
H	D	*a*	*b*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*
12	6	$\frac{3}{8}$	$3\frac{1}{4}$	$1\frac{1}{2}$	$\frac{1}{2}$	6	$4\frac{3}{4}$	$1\frac{7}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	1	$\frac{3}{8}$
16	8	$\frac{1}{2}$	$4\frac{3}{8}$	2	$\frac{1}{2}$	$7\frac{5}{8}$	6	2	$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{1}{2}$
20	10	$\frac{5}{8}$	$5\frac{5}{8}$	$2\frac{1}{4}$	$\frac{1}{2}$	$9\frac{1}{4}$	$7\frac{1}{2}$	$2\frac{1}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{1}{2}$
24	12	$\frac{3}{4}$	$6\frac{3}{4}$	$2\frac{3}{8}$	$\frac{1}{2}$	12	9	$2\frac{3}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{5}{8}$
28	14	1	$8\frac{3}{8}$	$2\frac{5}{8}$	$\frac{1}{2}$	$14\frac{1}{2}$	$10\frac{3}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$

All Dimensions are given in Inches.

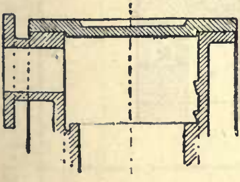


Fig. 405.

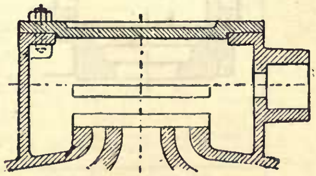


Fig. 406.

Figs. 405, 406 show two sections of a slide valve chest of ordinary construction with raised port face.

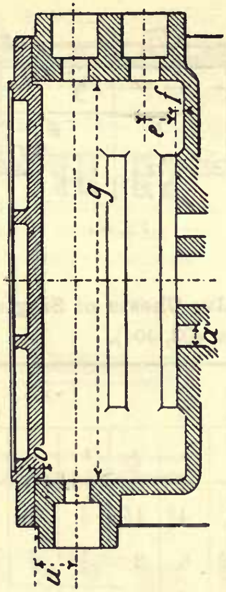


Fig. 407.

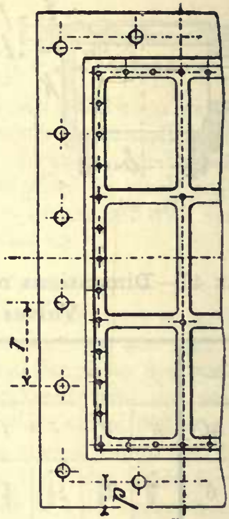


Fig. 408.

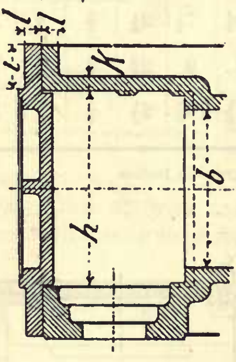


Fig. 409.

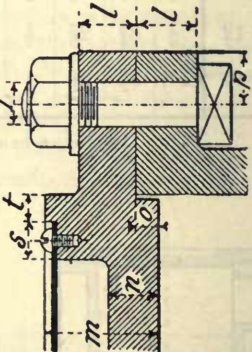


Fig. 410.

TABLE 41.—**Dimensions of Valve Chests with Cover for Meyer's Slide Valves** (Figs. 407—410).

Engines.														Bolts.						
H	D	*a*	*b*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	No.	*q*	*r*	*s*	*t*	*u*
16	8	1/2	4 3/8	1	1/2	11 1/4	6	2	1/2	5/8	1 3/8	5/8	1/4	5/8	10	1/2	4	5/8	3/8	1 3/4
20	10	5/8	5 5/8	1 1/4	1/2	13 1/4	7 1/2	2 1/8	5/8	3/4	1 1/2	3/4	1/4	3/4	12	5/8	4 3/8	5/8	3/8	2
24	12	3/4	6 3/4	1 3/8	1/2	16	9	2 3/8	3/4	7/8	1 3/4	7/8	1/4	3/4	14	5/8	4 5/8	5/8	3/8	2 1/4
28	14	1	8 3/8	1 1/2	1/2	18 1/2	10 3/4	2 1/2	3/4	1	2	1	3/8	1	16	3/4	4 3/4	5/8	3/8	2 3/8
32	16	1 1/8	10	1 3/4	5/8	21 1/4	12 1/2	2 3/4	7/8	1 1/8	2 1/4	1 1/8	3/8	1	18	3/4	4 3/4	5/8	1/2	2 5/8
36	18	1 1/4	11 5/8	2	5/8	24	14 3/4	3	1	1 1/4	2 3/8	1 1/4	3/8	1 1/8	20	7/8	4 3/4	5/8	1/2	2 3/4
40	20	1 1/2	13 1/4	2	3/4	27 1/4	16 1/2	3 1/4	1 1/8	1 1/4	2 5/8	1 1/4	1/2	1 1/8	20	7/8	5	5/8	1/2	3
44	22	1 5/8	15	2 1/4	3/4	30	18	3 1/2	1 3/8	1 1/2	2 3/4	1 3/8	1/2	1 3/8	22	1	5 1/4	5/8	1/2	3 1/4

All Dimensions are given in Inches.

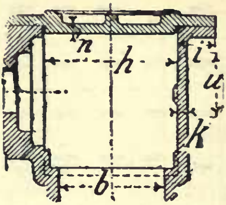


Fig. 411.

TABLE 42.—**Dimensions of Valve Chests for Rider's Slide Valves** (Figs. 411, 412).

H	20	24	28	32	36	40	44
D	10	12	14	16	18	20	22
u	$3\frac{1}{2}$	$4\frac{1}{2}$	5	6	7	8	9
y	15	$17\frac{1}{2}$	$20\frac{1}{2}$	$23\frac{1}{2}$	$26\frac{1}{2}$	30	33

All Dimensions are given in Inches.

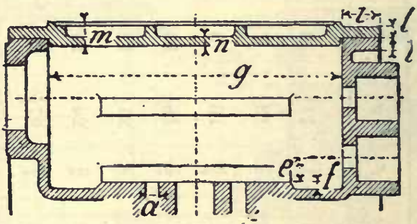


Fig. 412.

Fly-wheels.

The fly-wheel of an engine may be looked upon as a speed regulator, especially for reducing those irregularities which occur in the engine itself when passing the centres ; the fly-wheel also serves to soften down the irregularities proceeding from external causes, such as sudden variations in the load, and thus assists the governor, whose special duty is to control the speed of the engine when under a varying load.

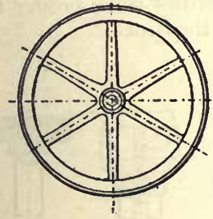


Fig. 413.

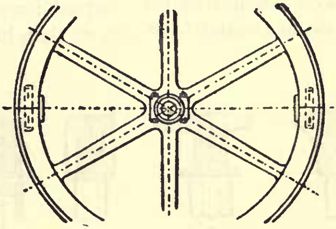


Fig. 414.

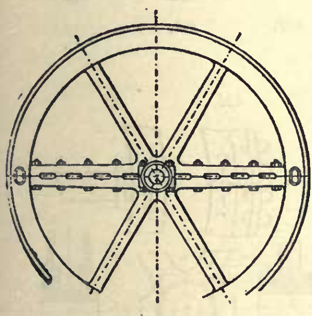


Fig. 415.

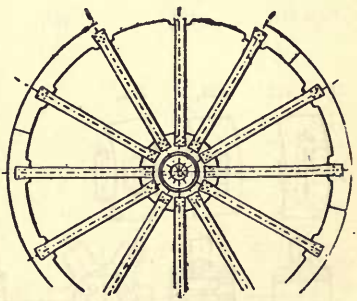
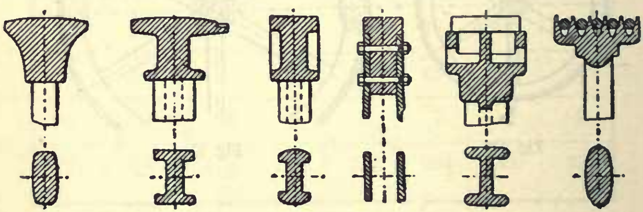


Fig 416.

The above figs. show examples of fly-wheels : fig. 414 is the most usual method of making a fly-wheel in halves; fig. 416 shows a built-up wheel with wrought-iron arms, figs. 420 and 431, on page 126 show respectively the section of rim and boss of such a wheel.

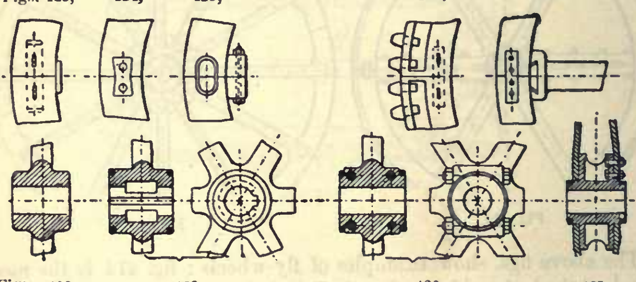
Various rules can be given for the weight of fly-wheels, but it should always be remembered that where very steady running is required the fly-wheel should be heavier than most formulæ give, especially if the engine is used, as is often the case, with electric lighting to drive one machine direct ; this is perhaps the severest test of steady running that can be given. Where a number of machines are driven from an engine by means of shafting, steady running is not nearly so difficult of attainment, on account of the steadying power of the shafting pulleys, belting, &c.

Engines working with a high rate of expansion require heavy fly-wheels on account of the internal irregularities being greater than when steam is admitted for more than half the stroke.



Figs. 417, 418, 419, 420, 421, 422.

Figs. 423, 424, 425, 426, 427



Figs. 428, 429, 430, 431.

Sections of rims, arms, and bosses of fly-wheels, figs. 417, 418, 419, are ordinary sections of rims and arms ; fig. 430, an ordinary fly-wheel boss in halves ; other figs. show special examples.

Let N = the effective horse power of the engine.
n = the number of revs. per minute.
r = the crank radius in inches.
l = the length of the connecting rod in inches.
$\frac{r}{l}$ = the ratio of crank radius to length of connecting rod.
G = the weight of the fly-wheel rim in lbs.
R = the mean radius of the fly-wheel rim in feet.
v = the mean speed of the rim in feet per second.
$\frac{1}{\delta} = \frac{v \text{ max.} - v \text{ min.}}{v}$ = the extent of irregularity.
δ = regularity.

Values of δ—

For ordinary steam engines $\delta = 40$ to 60.

For engines driving spinning machinery or for electric lighting, $\delta = 60$ to 100

For a single cylinder engine,

$$G = 100 i \frac{\delta N}{v^2 n} \text{ and } \delta = \frac{G v^2 n}{100 i N}$$

The coefficient i is dependent on the rate of expansion and may be taken from the table where the values of i are given for $\frac{r}{l} = \frac{1}{4}$ to $\frac{1}{6}$ and the clearance space s = from 2 to 7 per cent. of the stroke, and p = absolute initial pressure, w = absolute pressure of release, then $\frac{P}{w}$ = total expansion, and h = the cut off.

TABLE 43.—**Values of i.**

$\frac{P}{w}$	1	2	3	4	5	6	7	8	9
h	1	·5	·33	·25	·20	·15	·10	·08	·06
i	1265	1610	1840	2070	2185	2300	2415	2530	2645

Example.—For an engine with piston $15\frac{3}{4}''$ diameter, $27\frac{1}{2}''$ stroke, 85 revolutions per minute, 55 effective HP, $\frac{r}{l} = \frac{1}{5}$, R = 5′ 3″

$$v = \frac{85 \times 2\pi \times 5{\cdot}25}{60} = 46{\cdot}5 \text{ feet per second}; \; h = {\cdot}17, \; \delta = 50.$$

$$G = 100 \times 2185 \frac{50 \times 55}{(46.5)^2 \times 85} = 3270 \text{ lbs.}$$

The total weight of the fly-wheel, including the arms and boss, is about 1·35 times that of the rim.

For double-cylinder or compound engines, the weight of the wheel may be somewhat less if the cranks are at right angles. $G = 30\, i \frac{\delta N}{v^2\, n}$ where N = the total effective HP of the engine and the value of δ for such engines being rather greater than 70.

Let Gs = the weight in lbs. of a segment whose length is l, then $C = \frac{Gsv^2}{gR}$ = the centrifugal force.

The piece l, fig. 432, may be taken as a beam with an equally distributed load and fixed at the two ends; the centrifugal force acting outwards will then be the sum of these equal loads. In order to

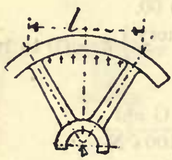


Fig. 432.

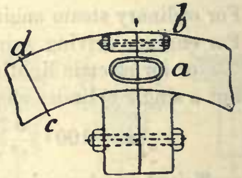


Fig. 433.

increase the strength of the joint in the rim when the fly-wheels are made in halves a wrought iron ring, a, fig. 433, is shrunk on over lugs cast on the rim; (the joint in fig. 433 is of unusual construction, see those shown in figs. 438, 440). The stress S_1 on the rim in lbs. per square inch of section may be calculated by the formula, $S_1 = \cdot 144\, v^2$, where v is in feet per second; this includes the bending strain on the arms. The total stress on the cross section, $c - d$, fig. 433, is $f \times S_1$, where f is the cross section in square inches. If q is the cross section of the two rings taken together and q_1 the cross section of the bolts or dowells according to which plan of construction is adopted, then the stress in rings and bolts is $S = \frac{f S_1}{q + q_1}$.

The arms are exposed to a tearing stress from the centrifugal force, and to a bending stress from the variations of the force communicated to the wheel; the stress may be for cast iron, 1300 lbs. per square inch, for wrought iron, 5000 lbs. per square inch.

The maximum safe velocity for the periphery of a cast iron fly-wheel is usually taken as 80 feet per second, and care must be taken that when the fly-wheel is in halves, that the joints are sufficiently strong to resist bursting, and if not made in halves the boss should be split to allow of contraction in cooling, a hoop being shrunk on after the wheel is cool.*

Fly-wheels are often used as pulleys for driving machinery by means of belts; the rim is then turned slightly arched, or if ropes are used, the rim is grooved, as shown in fig. 434.

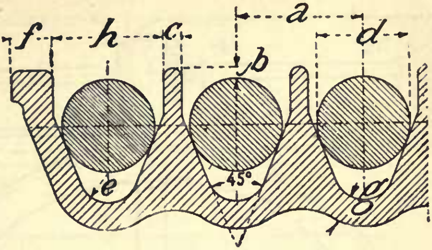


Fig. 434.

Table 44.—Dimensions of Grooves for Rope Pulleys (Fig. 434).

d	*a*	*b*	*c*	*e*	*f*	*g*	*h*
$1\frac{9}{16}$	$2\frac{1}{8}$	$\frac{3}{16}$	$\frac{5}{16}$	about $\frac{7}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{13}{16}$
$1\frac{3}{4}$	$2\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{2}$	2
2	$2\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$2\frac{1}{4}$

All Dimensions are given in Inches.

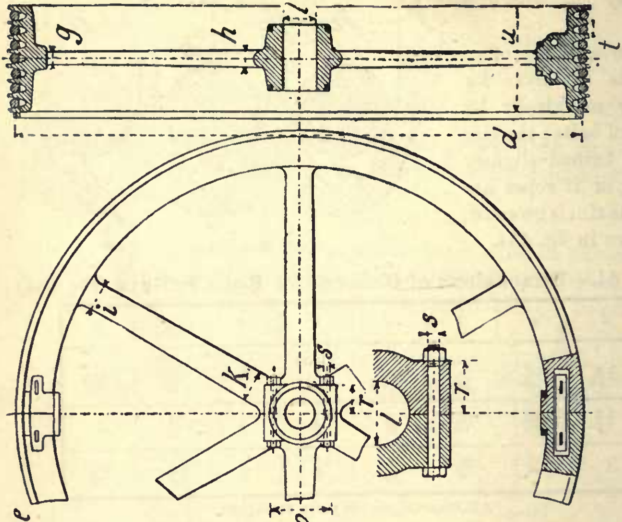
Table 45.—Dimensions and Particulars of Rope Drives.

Circumference of ropes . .	$2\frac{3}{4}$	3	$3\frac{1}{2}$	$4\frac{1}{4}$	$5\frac{1}{4}$	6	$6\frac{1}{2}$
Diameter of ropes . . .	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2
Weight in lbs. per foot, Hemp	·37	·46	·56	·66	1·00	1·37	1·62
„ „ „ Cotton	·27	·30	·39	·59	·87	1·17	1·50
†Diameter of smallest pulley .	18	24	30	36	46	60	66
Pitch of grooves . . .	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	$2\frac{3}{4}$	3
Indicated HP transmitted by one rope for the above pulleys at 100 revolutions per minute	For ropes running horizonal or nearly so						
	2	3	5	12	20	28	36
	For ropes running vertical or nearly so.						
	1	2	4	8	14	20	26

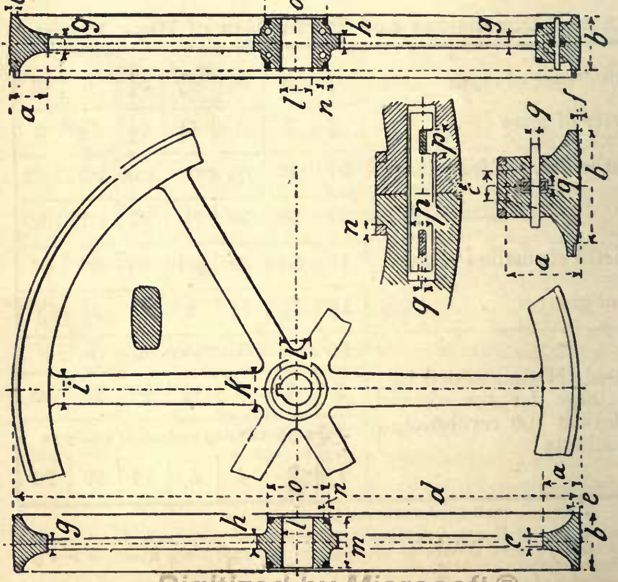
All Dimensions are given in Inches.

* A fly-wheel, 20 feet diameter, running at 70 revolutions per minute, burst, and the pieces were carried to great distances—the speed in this case being about 70 feet per second.—*Engineer*, Oct. 7th, 1892.

† By "diameter of smallest pulley," it is to be understood that it is best not to use smaller pulleys, except in cases of emergency.



Figs. 433—440.



Figs. 435—437.

TABLE 46.—**Dimensions of Fly-Wheels** (Figs. 435—440).

Engines.																				Centre of gravity of Rim.	Weight of Rim.	Rope Pulleys.		
H.	D.	d	b	a	c	e	f	g	h	i	k	l	m	n	o	p	q	r	s	d_1	lbs.	Ropes No.	t	u
12	6	42	6	3	2	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	2	$2\frac{1}{2}$	3	5	—	6	—	—	—	—	40	300	—	—	—
16	8	60	7	$3\frac{1}{2}$	$2\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$2\frac{1}{2}$	$3\frac{1}{4}$	$3\frac{1}{2}$	6	—	7	—	—	—	—	58	800	—	—	—
20	10	78	8	4	$2\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$1\frac{1}{2}$	2	$3\frac{1}{2}$	$4\frac{1}{2}$	$5\frac{1}{4}$	$7\frac{1}{2}$	1	10	—	—	—	—	75	1,250	3	$1\frac{1}{2}$	7
24	12	96	10	5	$2\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{16}$	2	$2\frac{1}{2}$	$4\frac{1}{2}$	$5\frac{1}{2}$	6	9	$1\frac{1}{4}$	12	—	—	—	—	93	1,900	4	$1\frac{1}{2}$	$8\frac{1}{2}$
28	14	114	12	$5\frac{1}{2}$	$3\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{7}{8}$	$5\frac{1}{2}$	$6\frac{1}{2}$	$8\frac{3}{8}$	$9\frac{1}{2}$	$1\frac{3}{8}$	14	$2\frac{1}{2}$	1	5	$1\frac{3}{8}$	$110\frac{1}{2}$	2,700	4	$1\frac{3}{4}$	10
32	16	132	15	6	$3\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{4}$	$2\frac{3}{4}$	$3\frac{1}{4}$	$6\frac{1}{2}$	$7\frac{5}{8}$	$9\frac{3}{8}$	11	$1\frac{3}{8}$	16	3	$1\frac{1}{8}$	$5\frac{3}{4}$	$1\frac{1}{2}$	128	3,800	5	2	$13\frac{1}{2}$
36	18	156	18	$6\frac{1}{2}$	4	$\frac{3}{8}$	$\frac{5}{16}$	$3\frac{1}{4}$	$3\frac{3}{4}$	7	$8\frac{3}{8}$	$10\frac{5}{8}$	$12\frac{1}{2}$	$1\frac{1}{2}$	17	$3\frac{1}{4}$	$1\frac{1}{4}$	$6\frac{1}{2}$	$1\frac{5}{8}$	152	5,100	6	2	$16\frac{1}{2}$
40	20	168	20	$7\frac{1}{2}$	$4\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{8}$	$3\frac{1}{2}$	$4\frac{1}{2}$	$7\frac{1}{2}$	9	12	14	$1\frac{5}{8}$	20	$3\frac{1}{2}$	$1\frac{3}{8}$	$7\frac{1}{2}$	$1\frac{3}{4}$	163	6,850	7	2	19
44	22	192	22	8	5	$\frac{1}{2}$	$\frac{3}{8}$	4	5	8	$9\frac{1}{2}$	$13\frac{1}{4}$	16	$1\frac{3}{4}$	22	4	$1\frac{1}{2}$	8	2	187	10,000	8	2	22

All Dimensions are given in Inches.

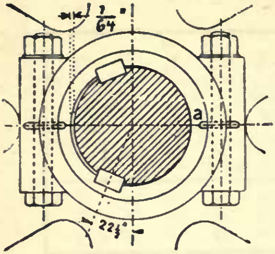


Fig. 441.

An American practice for fly-wheels is to bore out the boss exactly to the size of the shaft, and then machine or slot out a clearance, as shown to the left of the figure 441, two keys being fitted and driven home. The wheel is thus pulled hard over on to the true part of the original bore.

Throttle Valves.

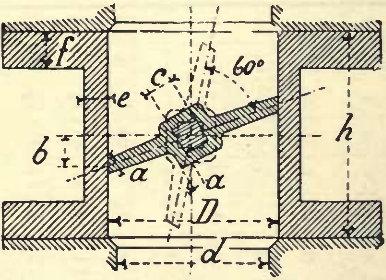


Fig. 442.

TABLE 47.—**Dimensions of Throttle Valves.**

d	D.	*a*	*b*	*c*	*e*	*f*	*h*
¾	1¼	⅛	¼	5/16	⅜	⅝	2
1¼	1½	⅛	¼	⅜	⅜	⅝	2½
1½	2	⅛	⅜	⅜	⅜	⅝	3
2	2½	⅛	⅜	⅜	½	¾	3½
2½	2¾	3/16	½	½	½	¾	4
2¾	3¼	3/16	½	½	½	¾	4½
3¼	3½	3/16	⅝	½	½	¾	5
3½	4	3/16	¾	⅝	½	¾	5½
4	4¼	¼	¾	⅝	½	¾	6
4½	5	¼	⅞	⅝	½	⅞	6½
4¾	5½	¼	1	¾	⅝	⅞	7
5¼	6	¼	1	¾	⅝	⅞	7½
5½	6½	5/16	1⅛	⅞	⅝	1	8
6	7	6/16	1⅛	1	⅝	1	8½

SECTION III.

GOVERNORS.

STEAM-ENGINE GOVERNORS have been greatly elaborated since the time of James Watt's simple conical pendulum governor, and new varieties have been introduced. Governors may be roughly divided into two main types, those in which some form of revolving pendulum is the basis, and those in which the centrifugal force of weights carried on levers is balanced against a spring; the former with a few exceptions revolve on vertical spindles, the latter are usually keyed direct on the engine crank shaft, and are sometimes called "crank shaft" governors. These latter also act directly, in some cases, on the valve eccentric and govern the engine by altering the degree of expansion; the former at the present time are seldom used to control the engine by means of a throttle valve,* as in the earliest engines, but are connected indirectly to the valve gear and control the engine by altering the degree of expansion. There are further sub-divisions into which governors may be divided, such as static, pseudo-astatic, and astatic.

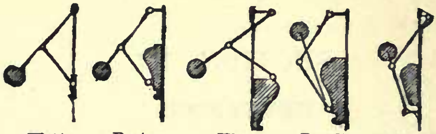
The simple ball governor of Watt is an example of the static governor, and each different speed of revolution corresponds to a definite position of the balls. Astatic governors are too sensitive to be of any use unless indirectly connected with the controlling mechanism, the slightest increase of speed of revolution causing the balls to fly out to their fullest limit; they have, however, been used, when connected to the controlling mechanism, through the medium of three bevel wheels and a clutch (fig. 465).

Pseudo-astatic governors approach nearer to the static governors and are much used; the cross-armed governor of Mr. J. Head (fig. 445) is an example in this kind, and any degree of sensitiveness can be attained by suitable proportions of the arms. When these pendulum governors are combined with a sliding weight as in Porter's governor (fig. 452), they are then called loaded governors, and are better adapted to control an engine than the simple governor of Watt.

* With very quick running engines, the difference whether they are controlled by a throttle valve or by varying the expansion is very small.

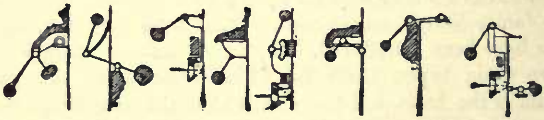
Outlines of Different Types of Governors.

Fig. 443.



Watt. Porter. Kley. Proell. Hartung.

Fig. 444.



Cosinus. Buss. Ericsson. Göpel. Ottos. Gasmot. Girard.

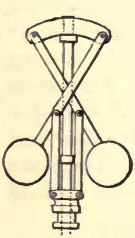


Fig. 445.
Cross-armed.

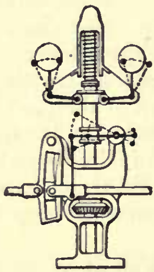


Fig. 446.
Hartnell.

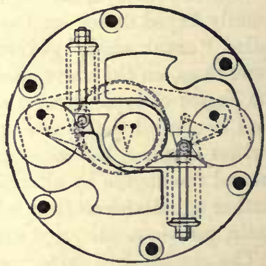


Fig. 447.
Turner—Hartnell.

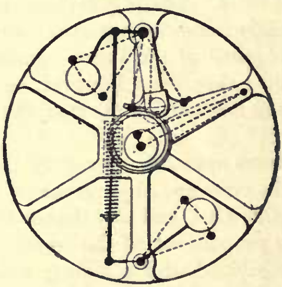
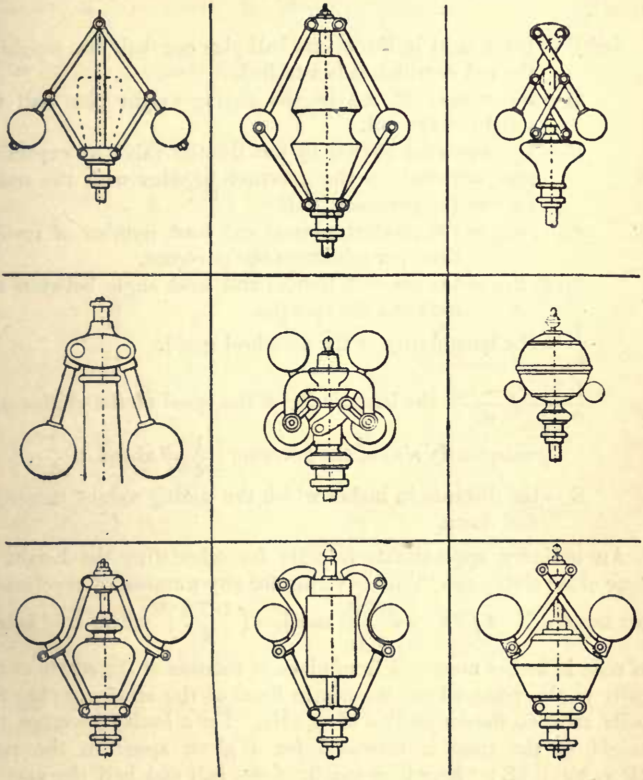


Fig. 448.
Hartnel

Figs. 443—445 show varieties of loaded and unloaded ball governors without springs; fig. 446, one of Hartnell's loaded centrifugal governors (with spring) on a vertical spindle; figs. 447, 448 two examples of crank-shaft governors.



Figs. 449—451. Figs. 452, 453, 454. Figs. 455—457.

Varieties of Governors of the Pendulum and Loaded Pendulum Types.

Fig. 449.—Watt.
Fig. 450.—Tangye.
Fig. 451.—Proell.

Fig. 452.—Porter.
Fig. 453.—Buss.
Fig. 454.—Proell.

Fig. 455.—Crossarm. loaded.
Fig. 456.—Cosine.

The formulæ given below serve to show the action of pendulum governors and to calculate the proportions for any given normal speed.

Let P = the weight in lbs. of one ball plus one half the weight of the rod to which it is attached.

Q = the weight in lbs. of the sliding weight plus half the weight of the rod.

R = the resistance offered by the throttle valve or expansion gear actuated by the governor together with the resistance of the governor itself.

$n_1, n, n_{11},$ = the greatest, normal and least number of revolutions per minute of the governor.

$\beta_1, \beta, \beta_{11},$ = the greatest, normal and least angle between the rods and the spindle.

$\frac{1}{\delta_0}$ = the irregularity of the fly-wheel speed.

$\frac{1}{\delta} = \frac{n_1 - n_{11}}{n}$ the irregularity of the speed of the engine and consequently also of the governor; $\frac{1}{\delta}$ = about $\cdot 9 \frac{1}{\delta_0}$.

S = the distance in inches which the sliding weight moves up and down.

An ordinary approximate formula for calculating the height of cone of revolving pendulum governor for any number of revolutions per minute N, 's; $N = \frac{187 \cdot 5}{\sqrt{L}}$ and $L = \left(\frac{187 \cdot 5}{N}\right)^2$ where L = height of cone in inches measured from plane of rotation of the centre of the balls to the point where the centre lines of the arms carrying the balls cross on the centre line of spindle. For a loaded governor the height of the cone is increased for a given speed in the ratio $2P + 2Q : 2P$; where P = weight of one ball and half the arm in lbs., and Q the load or sliding weight and half arm in lbs.

d = diameter in inches of the balls. For the normal speed n, R is taken to equal 0 in the equations.

Watt's governor, fig. 458.

$$n^2 = \frac{2936}{a \cos. \beta}\left(1 + \frac{Qb}{Pa}\right); \ \frac{1}{\delta} = \frac{Rb}{Pa + Qb}.$$

Dimensions in feet; the constant 2936, in from $\frac{3600g}{4\pi^2}$ where $g = 32 \cdot 2$.

The following proportions will be found to give good results—

$$\frac{Q}{P} = 0 \text{ to } 4,\ \frac{b}{a} = \cdot 6,\ S = \cdot 12b,\ \beta = 25^{\circ},\ \frac{1}{\delta} = \cdot 03.$$

For Porter's governor, fig. 459.

$$n^2 = 2936 \frac{1 + \frac{Q \pm R}{P}}{a \cos. \beta + e \cot. \beta}$$

$$\frac{1}{\delta} = \frac{R}{P + Q}, \text{ dimensions in feet.}$$

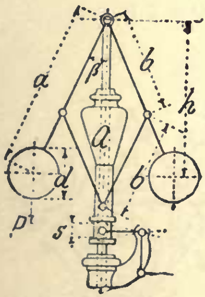


Fig. 458.

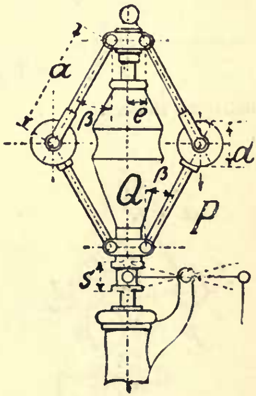


Fig. 459.

Table 48 gives the leading dimensions for a few examples of Porter's governor.

TABLE 48.

$n =$	150	140	130	120	110	100	90
$a =$	$8\frac{3}{4}$	$10\frac{3}{8}$	12	$14\frac{3}{8}$	$18\frac{3}{4}$	20	24
$e =$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{5}{8}$	2	$2\frac{3}{8}$
$s =$	$1\frac{1}{8}$	$1\frac{1}{2}$	2	$2\frac{3}{8}$	$2\frac{3}{4}$	$3\frac{1}{4}$	$3\frac{5}{8}$
$d =$	$3\frac{1}{4}$	$4\frac{1}{4}$	$5\frac{1}{4}$	6	$6\frac{3}{4}$	$7\frac{5}{8}$	8

All Dimensions are given in Inches.

Kley's governor (astatic), fig. 460.

$$n^2 = \frac{2936}{a \cos. \beta - e \cot. \beta}\left(1 + \frac{Qb}{Pa}\right)$$

$$\frac{1}{\delta} = \frac{Rb}{Pa + Qb}$$

Proell's governor (pseudo-astatic), fig. 461.

$$n^2 = 2936 \frac{-1 + \frac{Q + 2P \pm R}{P} \times \frac{b}{a} \times \frac{\sin. \gamma}{\sin. \beta}}{a \cos. \beta + e \cot. \beta}$$

$$\frac{1}{\delta} = \frac{R}{-P\frac{a}{b} \times \frac{\sin. \gamma}{\sin. \beta} + Q + 2P}$$

dimensions in feet.

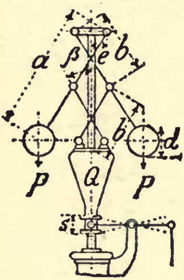


Fig. 460

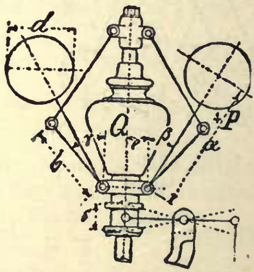


Fig. 461

TABLE 49.—**Leading Dimensions of Kley's Governor (Fig. 460).**

$\frac{a}{b}$ = ·7 to ·5.

$\frac{Q}{P}$ = about 4.

β = 30°.

n =	110	100	90	85	80	75	70
a =	$13\frac{3}{8}$	$15\frac{5}{8}$	18	$21\frac{3}{4}$	$25\frac{3}{8}$	30	36
e =	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{5}{8}$	2	$2\frac{3}{8}$
s =	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{3}{4}$	$3\frac{3}{8}$	4	$4\frac{3}{4}$	$5\frac{5}{8}$
d =	$4\frac{5}{8}$	$5\frac{3}{8}$	6	$7\frac{1}{4}$	$8\frac{3}{8}$	10	$11\frac{1}{4}$

All Dimensions are given in Inches.

TABLE 50.—**Leading Dimensions of Proell's Governor (Fig. 461).**

$\frac{Q}{P}$ = about 4.

$\beta = 20°$.

$\gamma = 30°$.

$n =$	135	125	115	110	105	100	95
$a =$	8	$9\frac{5}{8}$	$11\frac{1}{4}$	$12\frac{3}{4}$	$14\frac{3}{8}$	$16\frac{3}{8}$	$18\frac{3}{8}$
$b =$	$5\frac{3}{8}$	$6\frac{3}{4}$	$7\frac{3}{4}$	9	10	$11\frac{3}{8}$	$12\frac{3}{4}$
$e =$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{5}{8}$	2	$2\frac{3}{8}$
$s =$	2	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$
$d =$	4	$4\frac{3}{4}$	$5\frac{5}{8}$	$6\frac{3}{8}$	$7\frac{1}{4}$	$8\frac{1}{4}$	$9\frac{1}{4}$

All Dimensions are given in Inches.

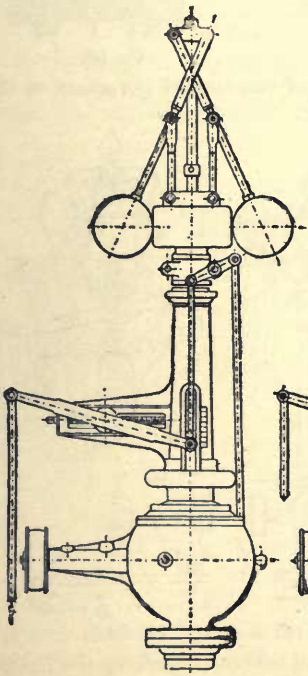


Fig. 462.

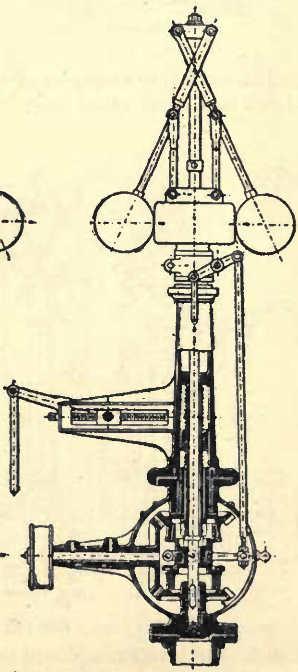


Fig. 463.

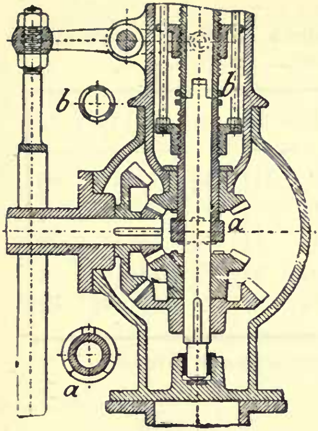


Fig. 464.

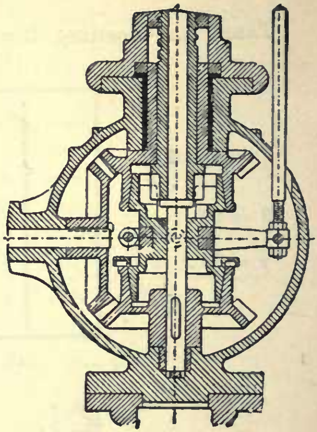


Fig. 465.

Figs. 462—465 give examples of cross-armed governors on stands with clutch and bevel-wheel gear.

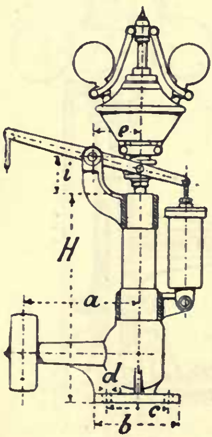


Fig. 466.

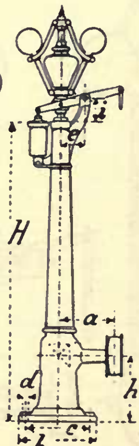


Fig. 467.

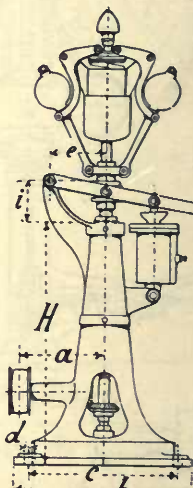


Fig. 468.

Figs. 466—468 give examples of governors made by the Lauchammer Iron Works fitted with an oil-brake or dash-pot, with a table of dimensions

TABLE 51.—**Leading Dimensions of the Lauchammer Governors** (Figs. 466—468).

Distinguishing No.	I.	II.	III.	IV.
Height of stand, H	$21\frac{5}{8}$	61	$37\frac{3}{8}$	$42\frac{1}{2}$
Dimension a	$9\frac{7}{8}$	$11\frac{1}{2}$	13	$14\frac{3}{4}$
" b	$8\frac{5}{8}$	$15\frac{3}{4}$	$27\frac{5}{8}$	$31\frac{1}{2}$
" c	$6\frac{1}{4}$	13	$23\frac{5}{8}$	$27\frac{1}{8}$
" d	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$
" e	$4\frac{3}{8}$	$4\frac{7}{8}$	$8\frac{5}{8}$	$9\frac{7}{8}$
" i	$4\frac{7}{8}$	$4\frac{1}{2}$	$6\frac{3}{8}$	$7\frac{1}{8}$
Weight in lbs. complete as shown in figures	660	704	750	840

All Dimensions are given in Inches.

Fig. 466 is especially adapted for horizontal engines; fig. 467 for vertical engines; fig. 468 a heavy type with a spring enclosed in a case in place of the sliding weight.

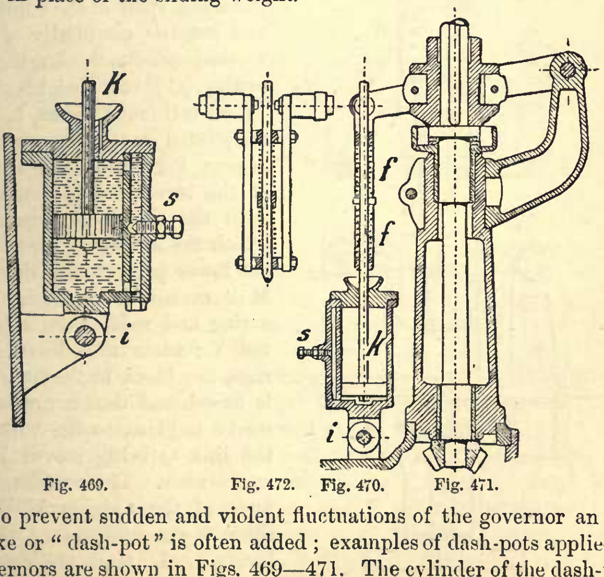


Fig. 469. Fig. 472. Fig. 470. Fig. 471.

To prevent sudden and violent fluctuations of the governor an oil-brake or "dash-pot" is often added; examples of dash-pots applied to governors are shown in Figs. 469—471. The cylinder of the dash-pot, fig. 469, is jointed to the frame or stand of the governor, by the pin i.

The cylinder is filled with oil or glycerine, and as the piston is moved up or down, the oil passes from the upper to the lower side of the piston by a passage shown at the side of the cylinder ; this passage can be more or less throttled by the screw S, so as to give more or less resistance to the movement of the piston, and thus "damp" the tendency to violent fluctuations in the governor. The screw S can be adjusted whilst the engine is running. Fig. 471 shows a combination of dash-pot with an elastic connection between the governor lever and piston rod of dash-pot, a spiral spring being inserted as shown at *f f*, fig. 471. This elastic connection assists the damping action of the dash-pot and should be conducive to very steady running. In some cases the dash-pot is incorporated with the governor, and enclosed in a neat case.

Automatic Expansion Governor by Wilson Hartnell (Fig. 473).

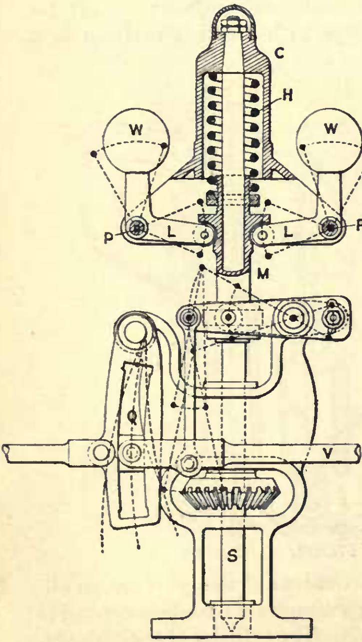


Fig. 473.

This governor is driven by bevel wheels and pulleys from the crank shaft of the engine, and consists essentially of a vertical spindle, S, carrying a casting, C ; two weights, W W, on bell-crank levers, L L, are jointed on to C, by means of pins, P P ; the inner ends of the levers are in contact with the sleeve M, against which the spring H presses ; the lower part of the sleeve M is connected by means of a ring and rods, to the valve rod V ; when the sleeve M rises, the block in the link Q is raised, and thus a varying stroke is given to the valve, the link Q being moved by an eccentric. The centrifugal force of the weights W W, through the medium of the levers L L, tends to compress the spring H. The motion of the different parts is clearly shown in the fig. by the dotted lines.

This governor is much used for controlling the speed of engines, by varying the cut-off, and is usually applied to valve gears with two valves, the main valve driven by an eccentric in the usual way, the cut-off by a separate eccentric through the medium of the link on the governor by which the cut-off is varied.

Crank-Shaft Governors.

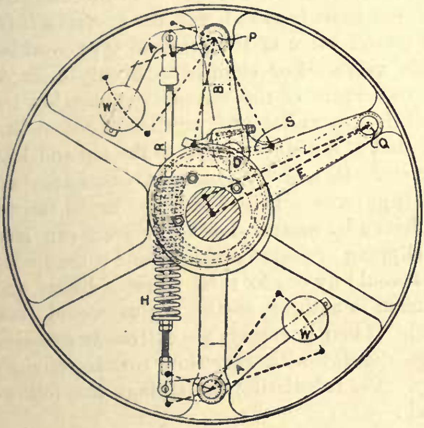


Fig. 474.

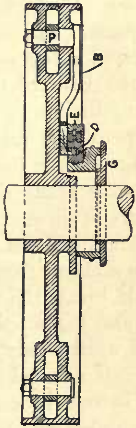


Fig. 475.

Figs. 474, 475, show two views of a Hartnell crank-shaft governor for controlling the speed of an engine by varying the cut-off. In the above figures the governor is shown in a six-armed wheel or pulley ; in some cases the governor is carried in the fly-wheel itself. Two weights, W W, are carried on two levers, A, turning on pins in the arms of the wheel ; the two levers are connected together by the rod R by means of joints and pins, and when the weights move outwards by centrifugal force they tend to compress the spring H. The upper lever, fig. 474, is rigidly fixed to its centre pin, and this pin carries the arm B, terminating at the lower end in the sector S, so that when the weight W moves outwards or inwards the sector S moves to the right or left. The sector is formed as shown in the cross section, fig. 475, and engages in the swivel block D. The block D is free to turn in a recess formed to receive it, in the arm of the eccentric E, this arm being centred on the pin Q. On referring to fig. 474 it will be seen that the sector S is not concentric with the centre P, upon which it swings, but its centre lies to the right hand of the centre P, so that the sector is inclined, the right hand end being the lower. Now as the sector swings on centre P towards the left, and in sliding through the block D pushes the eccentric arm E downwards, and as the eccentric itself is attached to the arm E, its centre will approach towards the centre of the crank-shaft, and thus the stroke of the eccentric will be reduced and the steam cut off earlier. In the figure the line in which the centre of the eccentric approaches the centre of the shaft is radial, and the governor is for application to a separate cut-off valve of the gridiron type working on the balk of the main valve. For engines with only a single valve, the line in which the centre of the eccentric approaches the centre of the crank-shaft is not radial but parallel to radial line such as that in the figure, but at a distance equal to the lap and lead from the crank-shaft centre. It will be seen that one important feature in these crank-shaft governors, is that they are keyed fast on the shaft, and not being driven by means of belts or gear, can lead to no accident from belt slipping or gears breaking ; and if the spring should break, the weights would fly out to their extreme limits and diminish the supply of steam to nothing, so the engine would slow down until nearly stopped. These governors are extremely sensitive and well adapted for high speeds : with very slow running-engines they become rather heavy. For calculating the springs the following simple formula is used :—

W = weight in lbs. of one of the weights.

N = normal speed in revolutions per minute.

N_1 = minimum speed in revolutions per minute.

N_2 = maximum speed in revolutions per minute.

$N_2 - N_1$ = allowed variation, upon the limit of this variation the sensitiveness of the governor largely depends ; about 3 per cent. under and 4 per cent. over the normal speed gives good results, and the usual variations in load of an engine in ordinary working conditions will not give nearly as much variation in speed.

R_1 = the radius in inches from centre of shaft to centre of weight when the latter is nearest to the centre of the shaft and corresponds to the speed of N_1.

R_2 = the corresponding radius to the speed of N_2.

L = the length ir inches of the lever carrying the weight.

l = the length in inches of the lever to which the spring rod R is attached.

r = the distance through which the spring is compressed by the action of the governor.

P_1 = the load on the spring in lbs. due to the centrifugal force at the minimum speed.

P_2 = the corresponding load in lbs. due to the maximum speed.

The $P_2 - P_1$ will equal the difference of maximum and minimum loads and $\left(\frac{P_2 - P_1}{r}\right) 2$ will equal the load in lbs. per inch of compression of the spring as the two weights act together on the spring.

$$P_1 = \frac{W \times R_1 \times N_1^2 \times L \times \cdot 000028}{l},$$

$$P_2 = \frac{W \times R_2 \times N_2^2 \times L \times \cdot 000028}{l}.$$

Spring-makers seem to be very successful in making springs accurate enough for such purposes as centrifugal governors from such data as stiffness per inch required, maximum load to which the spring will be subjected, external diameter and length when uncompressed.

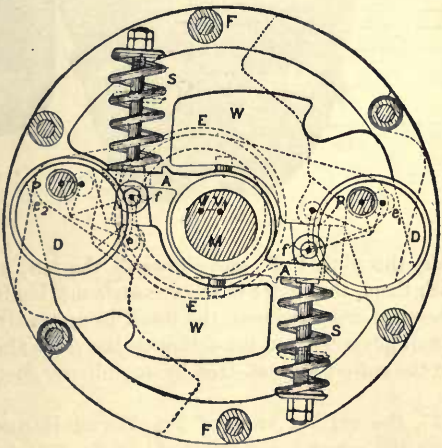


Fig. 476.

A modification of Hartnell's crank shaft governor, has been introduced by Messrs. Turner, of Ipswich, shown in diagram in fig. 476, and in perspective in figs. 477, 478.

Referring to fig. 476, two weights, W W, are formed with turned bosses, D D, fitted to and free to turn in holes in the plates or discs which form the governor body, and which are keyed on the crank shaft, M, the weights thus turning with the shaft and discs, and having their centrifugal force balanced by the springs, S S; the weights are connected by the eccentric, E (shown in dotted lines), by means of the pins, P P; the weights are shown in fig. 476, closed in full lines, and open in dotted lines. It will be seen that by the weights open-

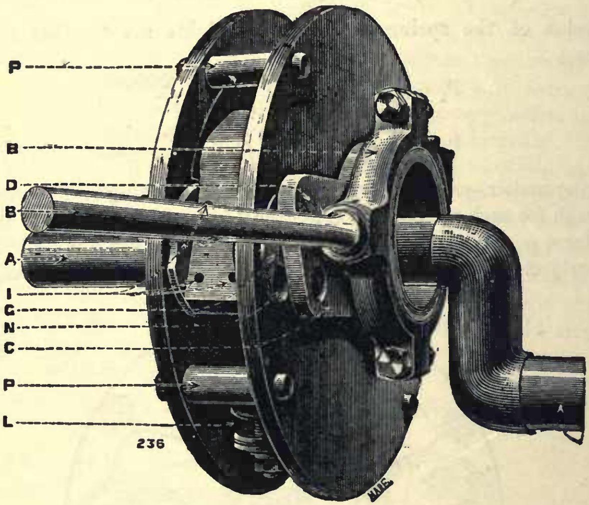


Fig. 477.

ing outwards, the pins, P P, move towards the left, and carry the eccentric with them, the centre of the eccentric moving from V to V_1, the stroke being thereby reduced, the result being nearly the same as in the Hartnell governor, and the action on the valve almost as if the variation in the cut-off were effected by an ordinary Stephenson link motion.

In fig. 477, the outside view of the Turner-Hartnell governor is shown with the eccentric rod attached; in fig. 478, one side disc is removed, and the weights shown open.

The reference letters are, A, the engine crank shaft; B, the valve eccentric rod; C, the eccentric; D D, pins carrying the eccentric, and passing through the bosses of the weights; H H, the weights;

P, bolts with collars and nuts, forming stanchions or pillars for holding the discs together.

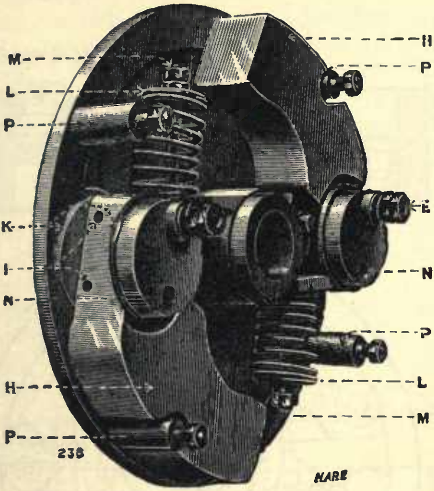


Fig. 478.

Another variety of crank shaft governor, is that of Mr. Moore, made by Messrs. Marshall, of Gainsborough. Figs. 479—482, show the different parts, and the governor consists of a heavy loose rim, A, attached to the shaft through an elastic arrangement of links carrying the weights; C C, are the weights turning on centres attached to the rim; D D, the springs; B B, links connected at one end to the weights and at the other to the points, H H, on the tripod G, fig. 482; the arm of the eccentric, E, fig. 481, is centred on the single end of G, fig. 482, at I; the link F connects the eccentric E to tne rim A.

The rim A revolves with the shaft, but is capable of running round the shaft to a small extent, defined by the limit of extension and contraction of the link centres, that is, by the extent to which the weights will open out, which can be regulated by stops on the back of the weights.

In connection with the loose rim is a variable expansion eccentric, shifting with the rim and increasing or decreasing the stroke of the eccentric and valve, and thus varying the cut-off. The action of this governor is peculiar, and may be explained as follows:—The inertia of the rim assists the weights to rise; the momentum of the rim, by

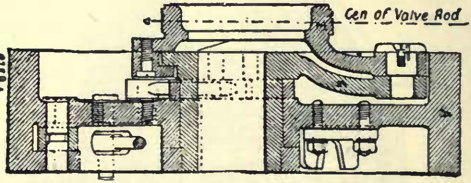


Fig. 479.

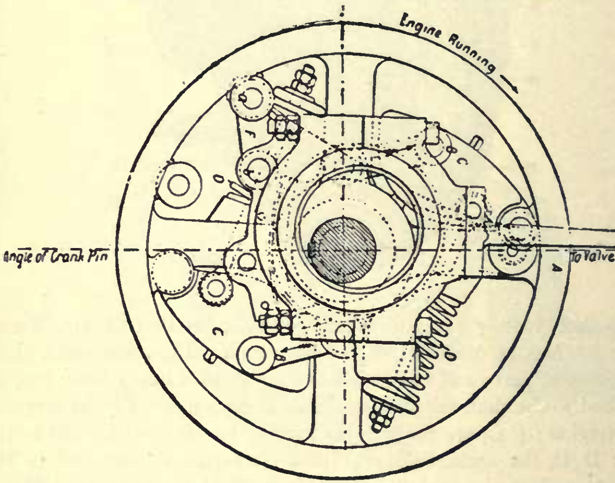
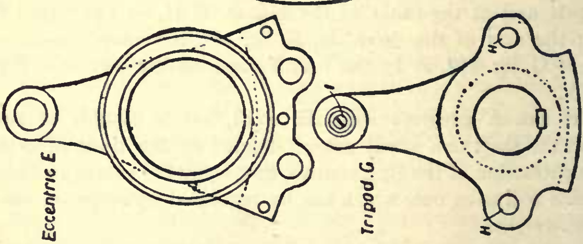


Fig. 480.



Figs. 481, 482.

straightening out the link and weight when the speed of the shaft diminishes, assists the weights to fall.

When the governor is expanding (rising), with an increasing speed, the shaft acts against the rim, through the link and weight, as shown by the arrow. When the governor is closing (falling) with a diminishing speed, the rim acts against it, tending to pull it forward, straightening out the link and weight. The weights are maintained in a larger plane of revolution, so to speak, than that due to centrifugal force, by the governor acting against the inertia of the rim when rising, *i.e.*, speed increasing. The weights are maintained in a smaller plane of revolution than that due to centrifugal force by the momentum of the rim acting against the governor, tending to pull it forward when closing, *i.e.*, speed decreasing.

The governor has a tendency to go too far both ways, either up or down, whenever there is the slightest variation above or below the normal speed, this tendency is checked by the work done by the governor in shifting the eccentric and valve. From the assistance given to the action of the governor weights by the rim, this governor is extremely sensitive.

SECTION IV.

VALVE GEARS.

Indicator Diagrams.

In order to follow the effect of the different distribution of steam effected by different valve gears, it is necessary to understand the indicator diagram. By the term "indicator diagram" is to be understood the graphic representation of the varying pressure of the steam on the engine piston at all positions of the crank. The steam engine "Indicator" invented by James Watt is the instrument by which the diagrams are automatically drawn. A brief description of indicators will be found on page 373. Figs. 483—488 show a few ideal forms of diagrams, with the following letters of reference :—

H = the stroke of the piston.

h = length of the period during which steam is admitted. If H be taken to equal 1, then $\frac{H}{h}$ = the ratio of expansion.

p = the initial absolute pressure measured from the line of perfect vacuum. p is usually taken in lbs. per square inch, but may sometimes be conveniently taken in atmospheres.

p_m = the average or mean effective pressure on the piston.

w = the terminal pressure.

p_o = the back pressure.

s = the clearance and steam passage for one end of the cylinder, usually taken in cubic inches or feet, and in diagrams figs. 483—488 it is expressed in terms of the piston area, so that the line s represents the volume of the clearance space.

va = the line of perfect vacuum.

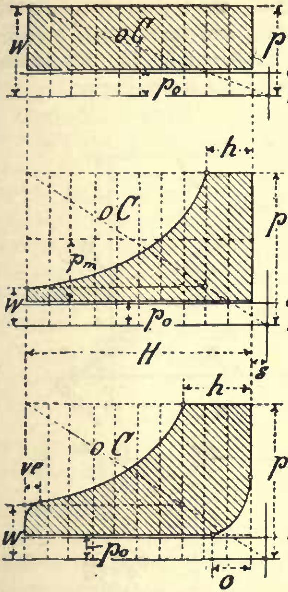
at = the atmospheric line.

ve = the portion of the stroke from the point where the exhaust opens to the end of the stroke; in the diagrams the line ve represents the volume of that portion of the stroke alluded to.

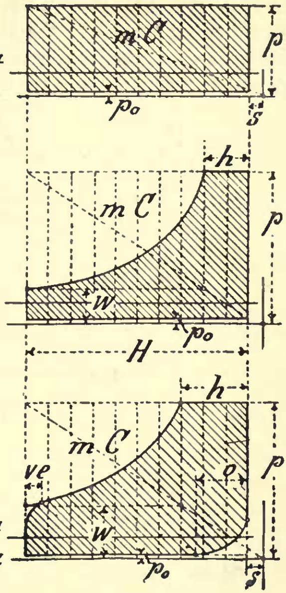
oC, diagrams for non-condensing engines.

mC, diagrams for condensing engines.

Figs. 483, 484, show the diagram for an engine with steam admitted for the whole stroke and in which the average pressure throughout the whole stroke is equal to the initial pressure—the back pressure



Figs. 483, 485, 487.



Figs. 484, 486, 488.

and the diagram is a rectangle; such a case seldom if ever occurs in practice. The figures 485, 486 show imaginary diagrams with steam cut off at ·2 of the stroke; figures 487, 488, show diagrams of a more or less perfect form, with steam cut off at ·3 of the stroke and the corners rounded off as they always come in practice from compression as at *o* and early opening of the exhaust as at *ve*.

The curve formed in these diagrams by the expansion of steam is approximately a rectangular hyperbola, and knowing the initial absolute pressure and the point of cut off, the mean pressure can be calculated by the formula :—

$$p_m = p\left(\frac{1 + \text{hyp. log. R}}{\text{R}}\right)$$

from the value of p_m from this formula the deductions for back pressure, &c., must be made. In the formula R = the ratio of expansion and in the diagrams is the ratio $\frac{\text{H}}{h}$. The approximate expansion curve may be graphically laid down in the following manner, see figs. 489, 490, which is very convenient for comparing the actual curve formed by the indicator and that which would be formed were the action of the steam unaltered by the condensation which always takes place in the cylinder of a steam engine.

Fig. 489.

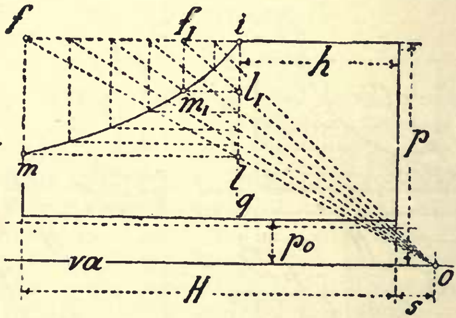
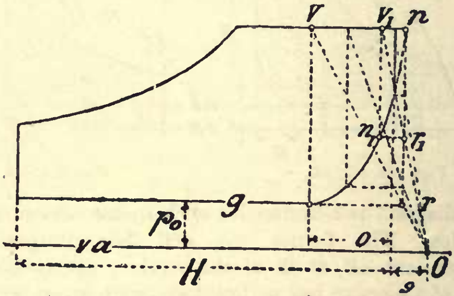


Fig. 490.



H = the length of the diagram and represents the stroke of the piston, and as the cylinder is parallel throughout its length, the line H also represents the volume swept by the piston in making one complete stroke.

h = the period of admission.

s = the clearance expressed in terms of the piston area, so that s in line H represents the volume of the clearance.

p = the initial absolute pressure in lbs. per square inch laid off to any convenient scale.

va = the line of perfect vacuum.

p_o = the back pressure of the exhaust steam.

o = the length of the compression period or the remainder of the stroke left to be completed after the exhaust has closed.

Complete the rectangle on the lines pH, then from point O draw the line $o\,f$, and the line $i\,q$ parallel to the line p; from the point where $o\,f$ cuts $i\,q$ draw $l\,m$ parallel to H, then m is the terminal point of the curve; the remaining points are found by drawing lines from O to points on the upper side of the rectangle, and the horizontal lines from where they cut the line $i\,q$, fig. 489.

Fig. 490 shows the same process for drawing the compression curve.

For obtaining the mean pressure on the piston from a diagram thus drawn, it may be either measured by means of a planimeter, or by dividing the figure into equal parts by vertical lines and measuring the sum of the vertical heights of these parts and dividing by their number for the average. Further explanation of these diagrams and their uses will be found at page 373. The above curve does not actually represent the expansion curve, only an approximation; a truer curve may be obtained by Rankine's formula:

$$\text{Pressure} \times (\text{volume})^{\frac{17}{16}} = \text{constant}.$$

If the pressure be expressed in lbs. per square inch, and the volumes in cubic feet, the constant has a value of 475, when adopting Zeuner's modification of Rankine's formula $\text{Pressure} \times (\text{volume})^{1\cdot0646} = 475$ The formula is referring to one pound of steam. The volume occupied by one pound of steam at different pressures is called the specific volume for that pressure.*

The back pressure (p_o) of exhaust in lbs. per square inch absolute for a speed of about 80 feet per second is given in the table.

TABLE 52.

Terminal pressure, w =	8·8	11·7	14·7	17·6	22·0	29·4	44·0	58·8
For non-condensing engine, p_o =	—	—	14·7	15·4	16·2	17·1	18·0	18·5
For condensing engine, p_o =	3·23	3·7	4·1	4·4	4·7	4·9	5·3	5·6

All pressures are given in lbs. per square inch absolute.

For ordinary work p_o may be taken as 17 lbs. absolute for non-condensing engines, and 3·7 for condensing engines.

* See further remarks on p. 446.

Classification of Valves and Valve Gears.

Valves and valve gears may be divided under the following heads:

A. Gears of simple kind, with eccentrics working either ordinary slide valves or piston valves.

B. Trip valve gears with Corliss valves.

C. Trip or positive motion gears with mushroom valves.

A. Simple Slide or Piston Valve Gears.

(*a*) With one slide or piston valve, rate of expansion usually fixed; or when combined with a Hartnell governor or link motion, the rate of expansion is variable.

(*b*) Double slide valve gears, with two separate slide valves; the expansion or cut-off valve working on the back of the main or exhaust valve, including Meyer's valve gear, Rider's, and others also, in which the expansion valve is controlled by the governor. In these gears both valves are worked by separate eccentrics, with the exception of one or two special examples

B. Trip Valve Gears with Corliss Valves.

(*a*) Gears in which the valves, such as Corliss valves, are opened positively, and let go at a period determined by the governor, by a so-called "trip" gear, the closing of the valves being effected by springs or dash-pots. The valves are parts of cylinders, and work to and fro in bored-out seatings.

(*b*) Gears in which the Corliss valves revolve continuously, such as those of Siegel and Ehrhardt.

In Corliss valve gears there are usually two steam valves and two exhaust valves. The advantages of this system will be referred to when treating of Corliss engines, pages 252 *et seq.*

C. Gears with Mushroom Valves.

(*a*) Gears with positively driven valves of mushroom form.

(*b*) Gears with valves driven through the medium of trip gear, the valves being of mushroom form. Examples of these are given on pages 234 *et seq.*

Slide Valves.

The figures 491—500 show longtitudinal and cross sections of simple slide valves, such as are used in small steam engines.

Fig. 491.

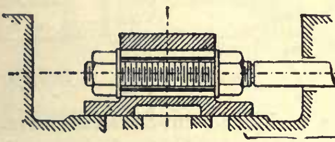
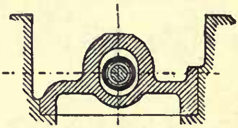


Fig. 492



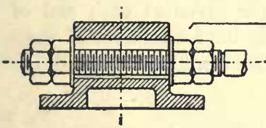


Fig. 493.

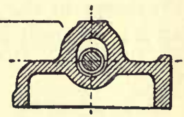


Fig. 494.

Fig. 495.

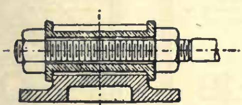


Fig. 496.

Fig. 497.

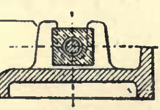


Fig. 498.

Fig. 499.

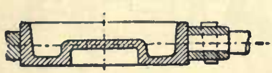
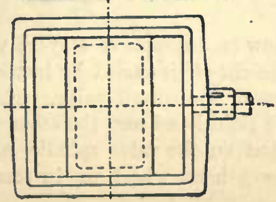


Fig. 500.



Figs. 501—503 show sections of the "Trick" valve, with the double opening for steam shown by the arrows; this is a good form of slide valve for quick running engines. The width of the port, $a = 2c + d$, and the eccentric radius or half stroke, $r = e + 2c$.

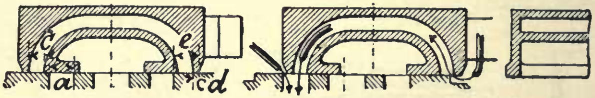
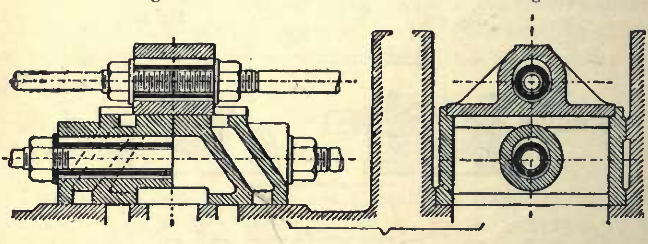


Fig. 501. Fig. 502. Fig. 503.

Figs. 504 and 505, show a main and cut-off slide valve, each worked by a separate eccentric, and for a fixed rate of expansion. Only one port is shown in the example given at each end of the main valve, but it is generally better to have a number of ports, 4 to 6, or even more. They are then termed multiple ported, or "gridiron slide valves.

Fig. 504. Fig. 505.



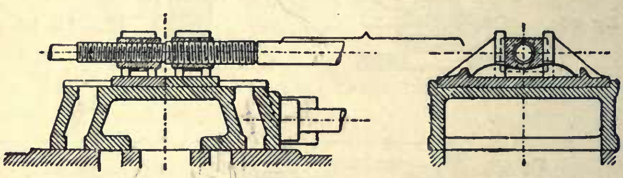


Fig. 506. Fig. 507.

Figs. 506 and 507 show an example of Meyer's valves for variable expansion, in which the cut-off is varied by increasing the distance between the outer edges of the cut-off valve. The valve is in two parts, one part having a right hand nut, the other a left hand, and a corresponding thread cut on the valve spindle, which is capable of being turned round by a hand wheel. or, in some cases, by being actuated by the governor.

In Rider's valve gear, of which the valves are shown in figs. 508—511, the cut-off valve itself corresponds to the right and left hand screw of Meyer's. The rate of expansion is varied by turning the cut-off valve on its seat, which is formed by boring out a recess on

Fig. 508.

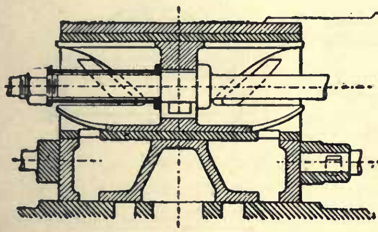
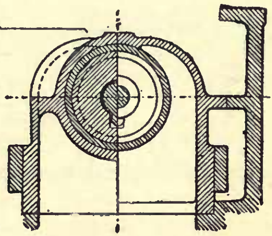


Fig. 509.



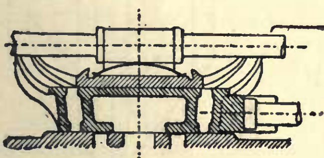


Fig. 510.

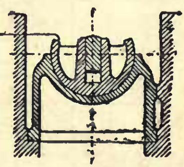


Fig. 511.

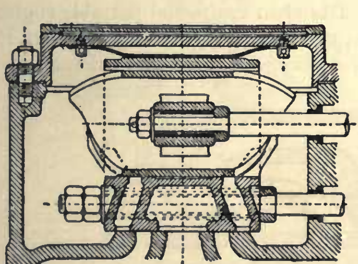


Fig. 512.

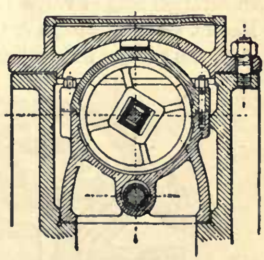


Fig. 513.

the back of the main valve. Fig. 508 gives an example of a closed Rider valve, and fig. 510, an open one; the cut-off valve is usually connected with the governor.

Figs. 512 and 513 give another example of closed Rider valves.

A modification of Rider's valves is shown in figs. 514—516, where in place of the cut-off valve being cylindrical it is flat with inclined edges, and is moved across the back of the main valve by means of a rack and pinion. The figs. are taken from valves by Heinrich Lanz of Mannheim.

Fig. 514.

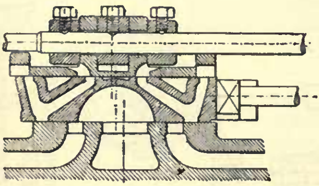
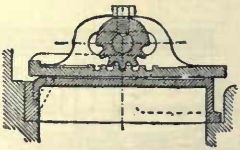


Fig. 515.



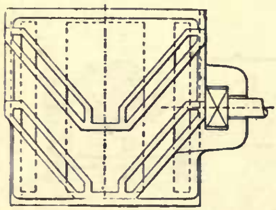


Fig. 516.

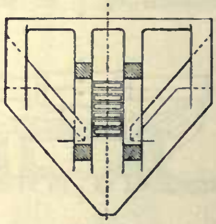


Fig. 517.

An arrangement of valve for a tandem compound portable engine by Messrs. Garrett, Smith & Co., of Buckau-Magdeburg, is shown in fig. 518, the arrows indicating the passage of steam and exhaust.

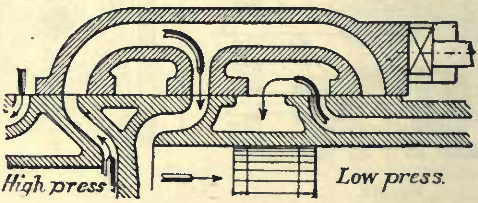


Fig. 518.

The double ported valve, fig. 519, is a form where the two end ports of the main valve are separate. This is necessary where a variation in the cut-off is effected by turning the cut-off valve

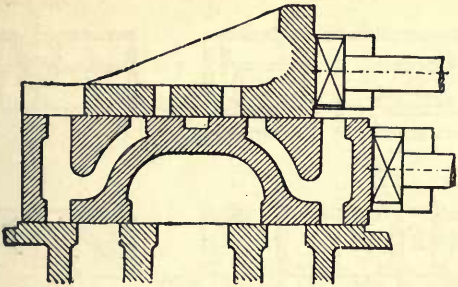


Fig. 519.

eccentric round on the shaft; in other words, by increasing the angle of advance of the cut-off valve eccentric. When the variation is effected by altering the stroke of the cut-off valve eccentric, the main valve may be a hollow case, with a number of ports at the back, all of which ports will be in use, admitting steam to whichever end of the cylinder the main valve allows it to enter. The advantages of having a number of ports are, that the cut-off is much more rapid than with only one or two ports.

A double slide valve arrangement, fig. 520, has a separate chamber for the expansion valve. There are some advantages in this, and it

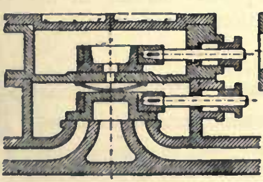


Fig. 520.

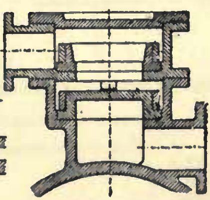


Fig. 521.

has been very successfully used by Messrs. Davey, Paxman & Co., of Colchester.

In order to reduce the stroke of a slide valve, the double ported single valve, figs. 522, 523, was introduced by Messrs. Penn for marine engines: a equalling the total width of the steam passage, then the radius of the eccentric, or half stroke, $r = \cdot 5a + e$. The lead and lap are half the amount which would be required for a simple valve for the same width of port. The steam enters by the passages E E, as well as over the back of the valve.

Fig. 522.

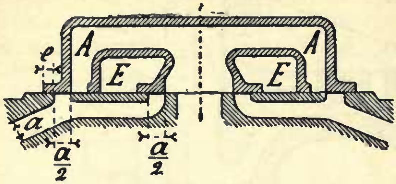
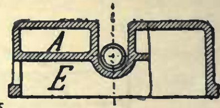


Fig. 523.



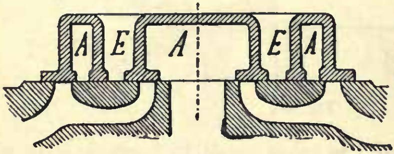


Fig. 524.

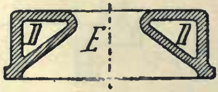


Fig. 525.

A variety of Penn's valve, by Borsig, is shown in figs. 524, 525. The passages, A A, are in connection with the side passages, D D, fig. 525. This arrangement allows of an expansion valve to be used on the back of the main valve in the usual way.

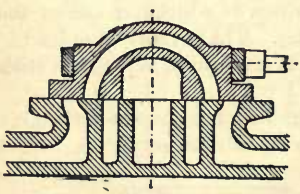
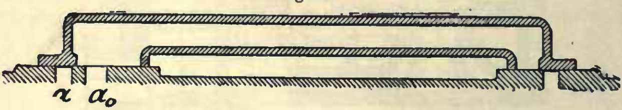


Fig. 526.

Fig. 526 shows the valve of Hick for a compound engine, with both pistons moving in the same direction.

Fig. 527.



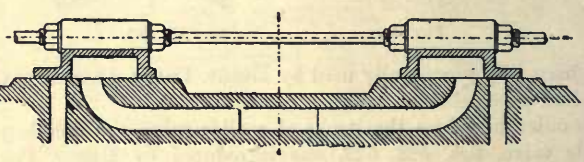
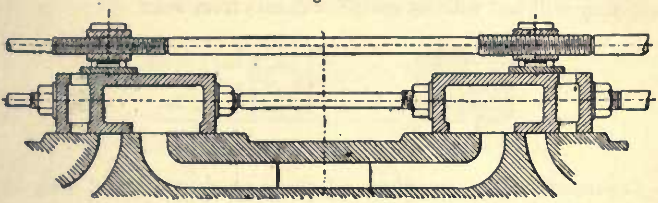


Fig. 528.

In order to reduce the length of the ports to a minimum, valves are often made in two parts, fig. 528. The same effect is produced by the long valve, fig. 527, but is not so good as separating the two ends as in fig. 528. In Meyer's and Rider's valve gears they can be arranged as in figs. 529, 530, where the two ends are separate, and the ports thus made very short.

Fig. 529.



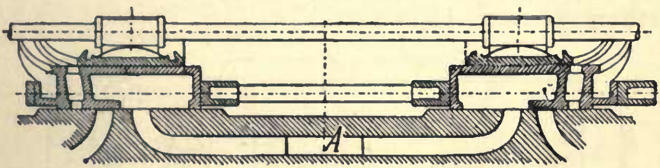
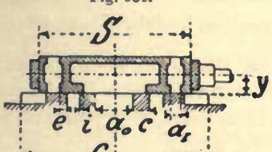


Fig. 530.

The attachment of the valve to the valve rod or spindle, is effected in various ways, and it should always be remembered that the centre line of the rod should be as near the port face as possible, so that the push and pull may be near to the line of greatest resistance, and thus to move the valve as quietly and steadily as possible.

In order to place the rod close to the port face, the rod is often made with a T head, fitting into a recess in the end of valve,

Fig. 531.



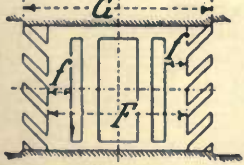
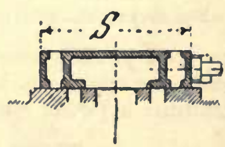


Fig. 532.

$f = a_1 + e + \frac{1}{4}''$.
$G = s + 1{\cdot}5\,r$.

Fig. 533.



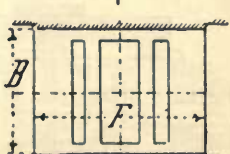


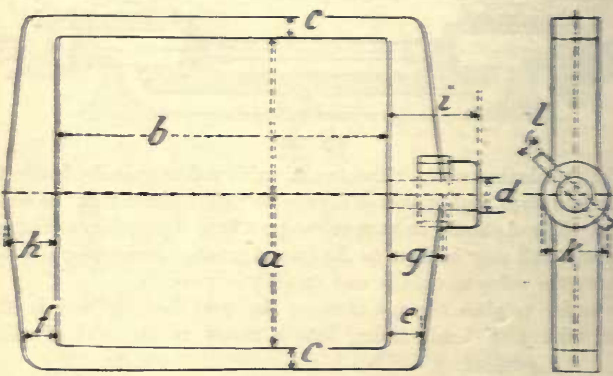
Fig. 534.

$F = s + r$.

fig. 533, or a bridle, figs. 536, 531, is used. Considering the small surface given by these T headed rods, it is somewhat surprising how long they will last without undue slackness from wear.

Fig. 535.

Expansion valves are often attached, as shown in fig. 535, by means of a block through which the rod passes, and in the case of Meyer's valves, is screwed into the block, or, with other valves for ordinary work, secured by a cottar.



Figs. 536 and 537.—Valve bridle for valves of the kind shown in fig. 531.

The great pressure exerted by the steam on slide valves has induced many schemes to be brought out to relieve this pressure. To construct a perfect "balanced" slide valve, is a point of great

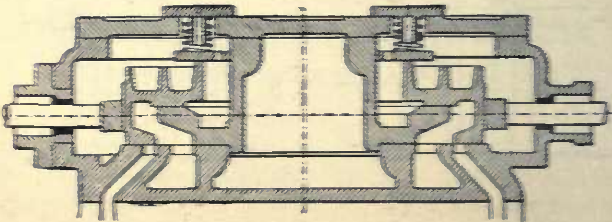


Fig. 538.

difficulty; many so-called balanced valves remaining so for a very short time after the engine has been working. A few examples are shown in figs. 538—541. In fig. 539, the valve is balanced by a ring

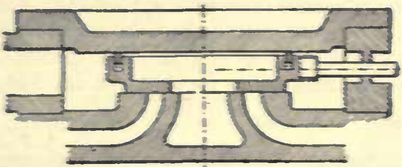


Fig. 539.

on the back, held up by light springs against the cover, so as to prevent the pressure of steam from acting on the back of the valve.

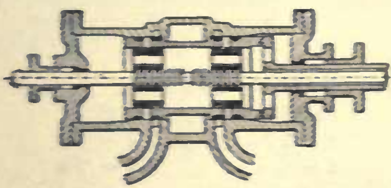


Fig. 540.

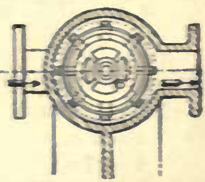


Fig. 541.

Perhaps the best form of balanced valve is the piston valve, which has been successfully used in marine engines with very high pressures. Examples of these will be found on pages 222, 223.

Simple Valve Gears.

The following reference letters are used throughout the investigations of valve gear diagrams and valves in the next pages 164—204.

a. The admission or steam port.

a_x. The exhaust port.

c. The breadth of the bars between the ports.

e. The outside lap.

i. The inside lap.

r. The radius of the eccentric, or half the stroke of valve.

v. The lead or length of steam port open to steam at beginning of stroke.

v_x. The inner lead on exhaust side.

The simple valve, worked by an eccentric, is chiefly used in small engines with cut off at from ·5 to ·8 of stroke, much better results being obtained from valve gears with two valves, worked by separate eccentrics. A single valve, however, attached to a link motion, has been used as a variable expansion apparatus, as well as for reversing.

Single valves in combination with a Hartnell governor, figs. 474, 476, page 145, have given good results,* the action on the valve being nearly the same in this case as with the link motion. Simple slide valves with the reference letters are shown in figs. 542 and 543.

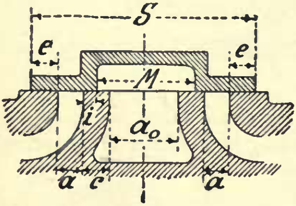


Fig. 542.

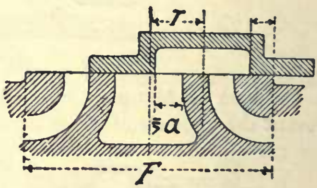


Fig. 543.

The proportions for given rates of expansion are :—

For cut off at ·5	at ·6	at ·7
$e = 2\,a$	$e = 1{\cdot}3\,a$	$e = {\cdot}8\,a$
$i = {\cdot}7\,a$	$i = {\cdot}5\,a$	$i = {\cdot}3\,a$
$r = {\cdot}8\,a + e$	$r = a + e$	$r = a + e$
$v = {\cdot}25\,a$ to $0{\cdot}5\,a$	$v = {\cdot}2\,a$ to ${\cdot}4\,a$	$v = {\cdot}2\,a$ to ${\cdot}3\,a$.

The greater values of v are for quick running.

Figs. 544—547 show in diagram the relative positions of the crank and valve in a simple valve gear, at four different points in the stroke.

In fig. 544 the piston is at the left hand end, and crank is at the beginning of the stroke, or, at the dead point; the steam port is open to the extent of the lead v, the exhaust port is open v_0; $x = e + v$.

In fig. 545 the steam port is full open, and the valve, at its extreme position, to the right; $x = r$.

In fig. 546 the steam port is closed and the expansion has begun; $x = e$.

In fig. 547 the exhaust is closed, and the compression has begun; $x = i$.

In these figures, K is the crank, δ, the angular advance of the eccentric radius E, and the arrows show the direction of running, and the direction of the flow of steam in the ports and passages.

* See report of judges at the Cardiff Meeting of the Royal Agricultural Society.

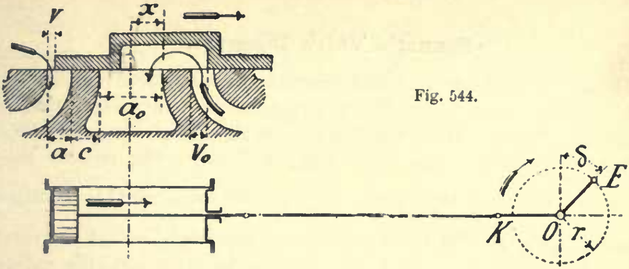


Fig. 544.

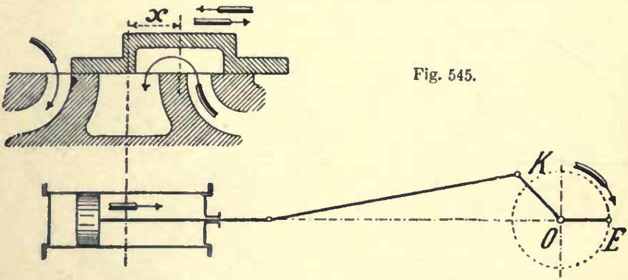


Fig. 545.

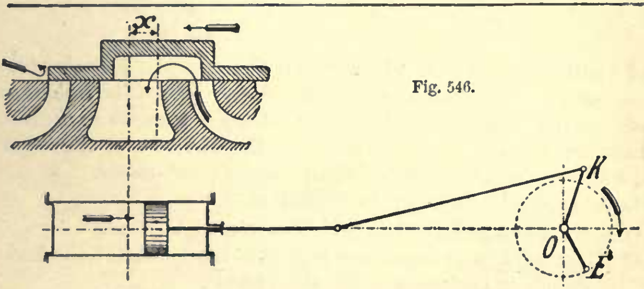


Fig. 546.

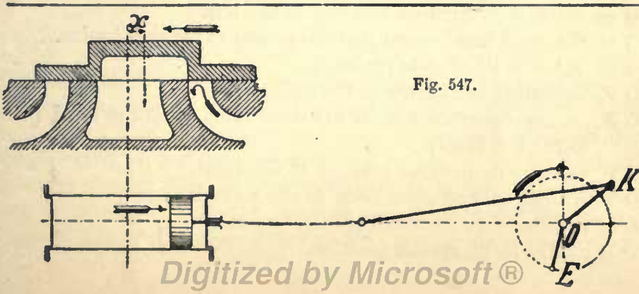


Fig. 547.

Zeuner's Valve Diagram.

By means of Zeuner's valve diagram, the relative positions of crank and valve can be shown graphically; the construction is briefly as follows: Draw the two axes at right angles to each other, O X, O Y, figs. 548, 549, then make O E = e = the outside lap; E V = v = the lead; with radius $= \frac{r}{2}$ describe the circle O V G cutting O and V, then M will be the centre of this circle, called the valve circle, O G will = r, and will give the position of the eccentric radius,

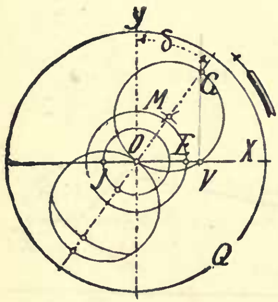


Fig. 548.

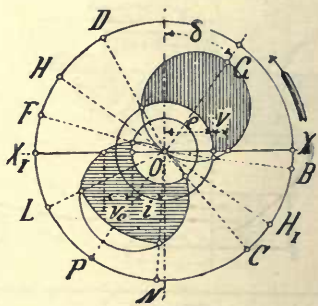


Fig. 549.

and δ will be the angle of advance on O G produced. A circle of the same radius is then drawn below O X. Make O J = i the inside lap, and draw the crank circle from centre O of any convenient size. Note that in this diagram the crank moves in the direction of the arrow· the contrary way to that in which the engine crank moves.

Looking at fig. 549, when the crank is at—

O B, the steam port is beginning to open at right hand.

O X, the right hand steam port is open to the extent of the lead v (crank at dead point to the right hand).

O G, valve at its furthest position to the left.

O D, the right hand steam port closes and expansion begins.

O H, valve in its middle position.

O F, beginning of exhaust at the right hand.

O X_1, exhaust port open v_0 at the right hand (crank at dead point to the left hand).

O P, valve at its furthest position to the right.

O L to O N, exhaust open fully to right hand.

O C, exhaust closes, beginning of compression.

O H_1, valve in its middle position.

From P to G (in direction of arrow), through H_1, the valve moves to the left. From G to P, through H, the valve moves to the right.

The half stroke or radius of eccentricity equals the port + the lap.

$$r = a + e\text{; but } r \text{ can be} \lessgtr a + e.$$

Fig. 550.

$$r < a + e = a_1 + e$$
$$a = 12$$
$$a_1 = 10$$
$$e = 18$$
$$i = 7$$
$$v = 4$$
$$r = 10 + 18 = 28$$

Fig. 551.

$$r > a + e = a + e + m$$
$$a = 12$$
$$m = 2$$
$$e = 18$$
$$i = 7$$
$$v = 4$$
$$r = 12 + 18 + 2 = 32$$

Fig. 550.

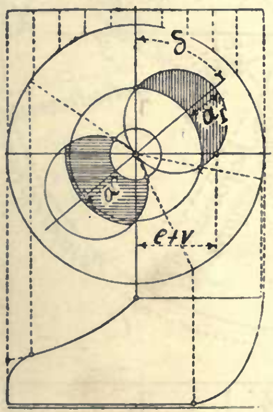
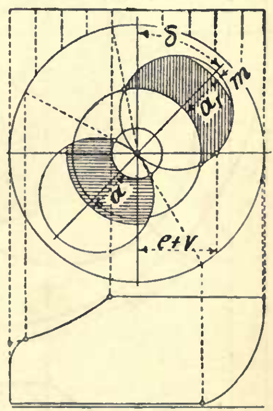


Fig. 551.



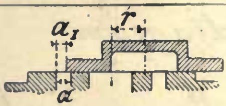


Fig. 552.

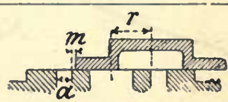
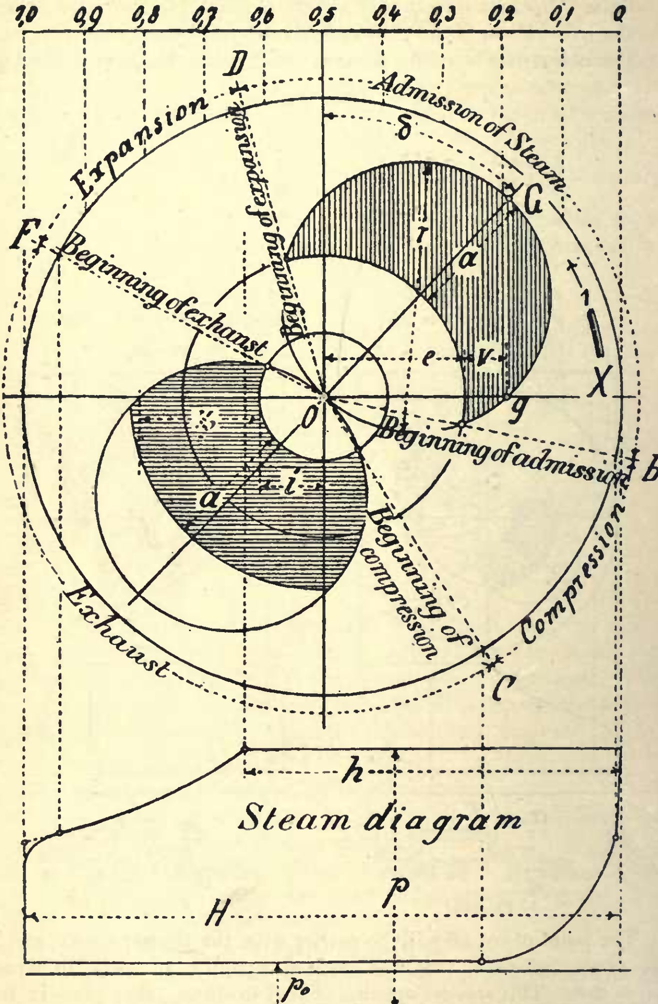


Fig. 553.

The point of cut off will be earlier with the shorter stroke, and it is often sufficient that the greatest opening of port to steam $a_1 = \cdot 8\, a$. This smaller opening of port to steam, takes place in the single valve gears connected with a governor of the Hartnell type, in order to keep the range of the governor as short as possible, and can be arranged to give very good admission and exhaust, as the exhaust side of the valve allows of full opening of the port.

Figs. 554 and 555.



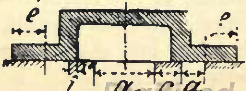


Fig. 556.

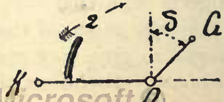


Fig. 557.

Valve Diagram for a Simple Valve Gear.

O X, direction of valve motion.

O, centre of the diagram.

e, outside lap (circle 2 e diameter about centre O, called lap circle).

i, inside lap (circle 2 i diameter about centre O).

v, lead of the valve, O $g = e + v$.

$r = a + e$ the radius or half stroke of the eccentric (valve circle through g and O).

H, diameter of the crank circle (any convenient size)

Then in figures 554 to 557—

v_0 = inner or exhaust lead.

O G, position of the eccentric radius.

δ = the angle of advance.

O B, the position of the crank for beginning of admission.

O D,	,,	,,	,,	of expansion.
O F,	,,	,,	,,	of exhaust.
O C,	,,	,,	,,	of compression.

h = the position of cut off.

If there is no inside lap, then the beginning of compression and exhaust take place on a line at right angles to the original position of the eccentric radius.

The diagram crank moves in the direction of the arrow 1, in fig. 554 ; the engine crank moves in the direction of the arrow 2, in fig. 557 ; the radius of the eccentric is in advance of the crank to the extent of $90° + \delta$; O K is the position of the crank, O G that of the eccentric, fig. 557. In order to understand this valve diagram, it is better to draw the steam diagram under, as shown in fig. 555.

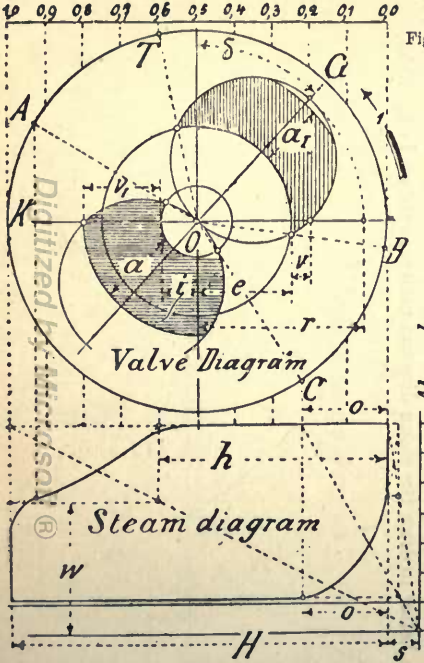
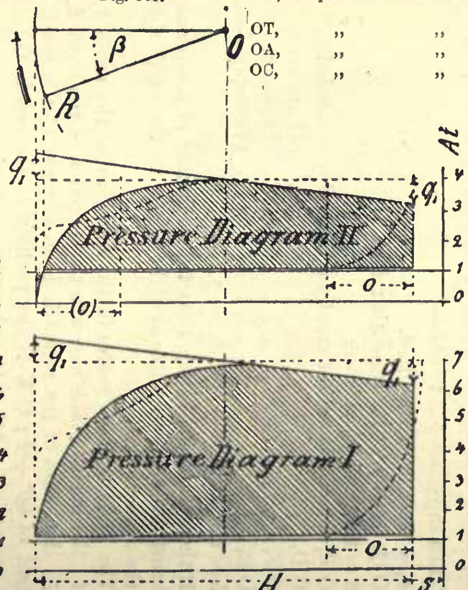


Fig. 558.

Fig. 559.

Fig. 560.



Valve Diagram (Fig. 558).

OB, the position of the crank when valve begins to open the steam port.

OT, „ „ „ closes the steam port.

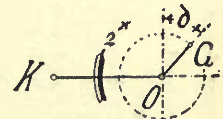
OA, „ „ „ opens exhaust.

OC, „ „ „ closes exhaust.

Pressure Diagrams I. and II.

q_1, the centrifugal force of the reciprocating parts of the engine expressed in terms of the piston area taken at the dead point, and with an infinitely long connecting rod.

Fig. 561.



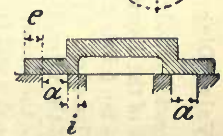


Fig. 562.

Arrow 1, fig. 558, direction of movement of the diagram crank.

Arrow 2, fig. 561, direction of movement of the engine crank.

TABLE 53.—**Valve Diagram for Simple Valve Gear** (Fig. 558).

Stroke	H	12	16	20	24	28
Diameter of piston	D	6	8	10	12	14
Revolutions per minute	n	125	110	100	90	85
Width of steam ports in port face	a	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1
Necessary width of opening to steam	a_1	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{13}{16}$
Outside lap	e	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{4}$
Inside lap	i	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$
Radius of eccentric	r	$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{3}{4}$	$2\frac{1}{4}$
Lead, outer	v	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{4}$
Lead, inner	v_1	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$
Cut off	h	·6	·6	·6	·6	·6
Exhaust	f	·07	·07	·07	·07	·06
Compression	o	·2	·2	·2	·2	·2
Angle of advance in degrees	δ	42	41	41	41	41

All Dimensions are given in Inches.

From the pressure diagram I it will be seen that with 7 atmospheres absolute pressure and full load the direction of pressure in the connecting rod changes at the dead point, but with 4 atmospheres absolute pressure, diagram II, it changes before the dead point at O R, angle β.

Slide Valves.

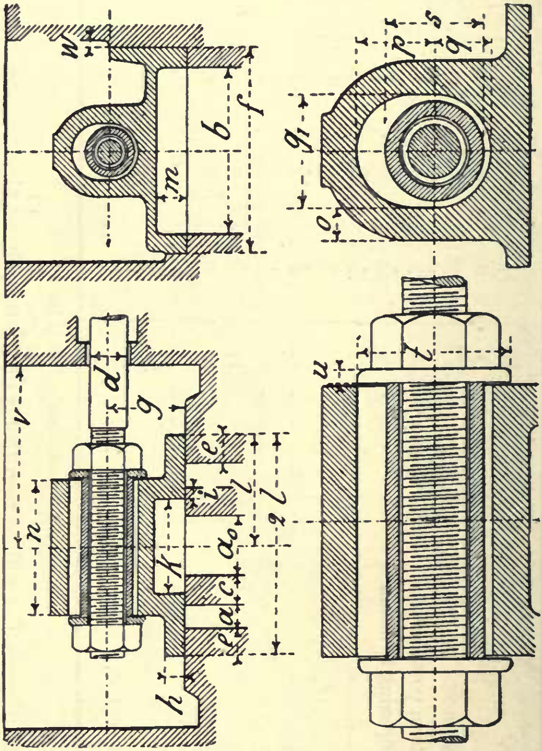


Fig. 564. Fig. 566. Fig. 563. Fig. 565.

The above figures show a simple slide valve corresponding to the valve diagram, fig. 558, and a table of dimensions is given on next page.

TABLE 54.—**Dimensions of Simple Slide Valve** (Figs. 563—566, and for Valve Diagram, see Fig. 558).

Engine. H	Engine. D	a_0	a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	q_1	s	t	u	v	w	r
12	6	$\frac{3}{4}$	$\frac{3}{8}$	$3\frac{1}{4}$	$\frac{9}{16}$	$\frac{7}{8}$	$\frac{7}{16}$	$4\frac{1}{4}$	$1\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$1\frac{5}{8}$	$1\frac{13}{16}$	$\frac{5}{8}$	$2\frac{1}{2}$	$\frac{3}{8}$	1	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{1}{4}$	$1\frac{1}{2}$	$\frac{3}{16}$	3	$\frac{3}{16}$	$\frac{3}{4}$
16	8	$1\frac{1}{8}$	$\frac{1}{2}$	$4\frac{3}{8}$	$\frac{5}{8}$	1	$\frac{5}{8}$	$5\frac{3}{8}$	2	$\frac{1}{2}$	$\frac{1}{4}$	2	$2\frac{3}{8}$	$\frac{5}{8}$	$2\frac{3}{4}$	$\frac{1}{2}$	1	$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	2	$\frac{3}{16}$	$3\frac{3}{4}$	$\frac{3}{16}$	$1\frac{1}{8}$
20	10	$1\frac{1}{2}$	$\frac{5}{8}$	$5\frac{5}{8}$	$\frac{11}{16}$	$1\frac{1}{8}$	$\frac{3}{4}$	$6\frac{3}{4}$	$2\frac{1}{4}$	$\frac{5}{8}$	$\frac{5}{16}$	$2\frac{3}{8}$	$2\frac{7}{8}$	$\frac{3}{4}$	$3\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{7}{8}$	$1\frac{7}{8}$	$1\frac{5}{8}$	$2\frac{3}{8}$	$\frac{3}{16}$	$4\frac{5}{8}$	$\frac{3}{16}$	$1\frac{3}{8}$
24	12	$2\frac{1}{8}$	$\frac{3}{4}$	$6\frac{3}{4}$	$\frac{3}{4}$	$1\frac{3}{16}$	1	$8\frac{3}{8}$	$2\frac{3}{8}$	$\frac{11}{16}$	$\frac{3}{8}$	$2\frac{7}{8}$	$3\frac{9}{16}$	1	$4\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{4}$	1	$1\frac{7}{8}$	$1\frac{5}{8}$	$2\frac{5}{8}$	$\frac{1}{4}$	6	$\frac{5}{16}$	$1\frac{3}{4}$
28	14	$2\frac{3}{8}$	1	$8\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	10	$2\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{16}$	$3\frac{3}{4}$	$4\frac{9}{16}$	$1\frac{1}{4}$	5	$\frac{3}{4}$	$1\frac{1}{4}$	1	$1\frac{7}{8}$	$1\frac{5}{8}$	$2\frac{3}{4}$	$\frac{1}{4}$	$7\frac{1}{4}$	$\frac{5}{16}$	$2\frac{1}{4}$

All Dimensions are given in Inches.

Valve Gears with Two Valves.

Meyer's Valve Gear.

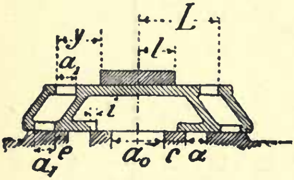


Fig 567.

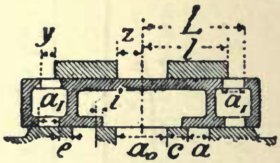


Fig. 568.

Fig. 567 shows the two valves arranged for a fixed rate of expansion; fig. 568 for variable expansion, with the following reference letters.

a, width of steam ports in port face.

a_1, width of steam ports in main valve.

a_0, width of exhaust port.

c, width of bars between ports.

e, outside lap.

i, inside lap.

v, outside lead.

v_0, inside lead or exhaust lead.

r, half stroke of main valve.

r_1, half stroke of cut-off valve.

δ, angle of advance of main valve eccentric.

δ_1, angle of advance of cut-off valve eccentric.

$y = L - l$, distance between working edges of valves; in fig. 568 an ideal middle position is taken for any selected point of cut off.

z, the horizontal distance from centre line of the back edge of cut-off valve for any selected point of cut off.

I, the valve circle of main valve, fig. 569.

II, the valve circle of cut-off valve.

III, the relative valve circle.

Valve diagram, fig. 569.

The outside lap circle, $2\,e$ diameter.

The inside lap circle, $2\,i$ diameter.

The valve circle I, with a diameter of $O\,G = r =$ half stroke of main valve, is drawn the same as for the simple valve gear. The advance angle of the cut-off valve eccentric δ_1 may be taken from 60° to 90°, and the cut-off valve circle II described with diameter O E $= r_1 =$ half stroke of cut-off valve eccentric. Draw G P parallel to O E, and O P parallel to E G, then O P will be the diameter of the relative valve circle III. The chord of valve circle III gives the distance between the centres of the two valves which is greatest when the crank is in the position O P, and then is equal to the line O P; for a fixed period of admission or point of cut off, for example ·7, O S $=$ L $-$ l, and the shaded part of the diagram shows the opening of the port to steam. When the crank is at O N, the port in the main valve is fully open; when at O m, the port is

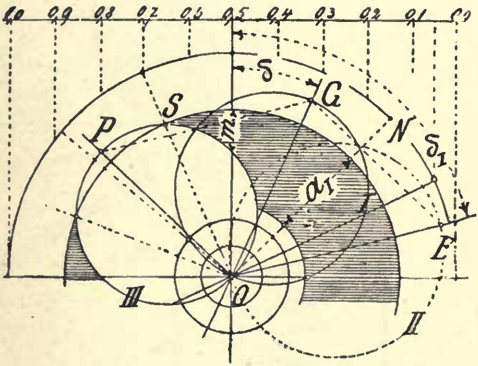


Fig. 569.

open to the extent of m; when at O S, the port is closed: the greater the distance between the working edges of the cut-off plates or valves, the greater is l, and so L $-$ l is less. For very early cut off, L $-$ l is negative; example, for a cut off at ·05, the crank centre line cuts the circle III in the lower quadrant.

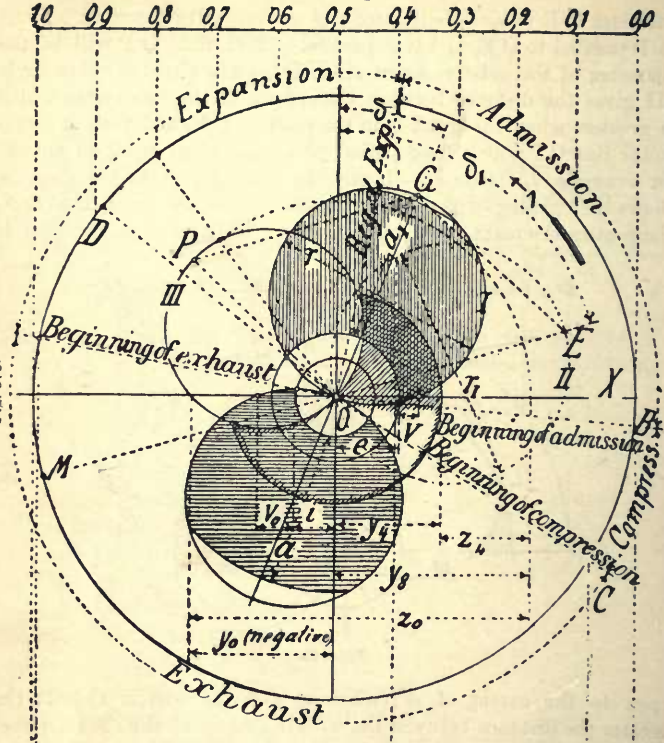


Fig. 570.

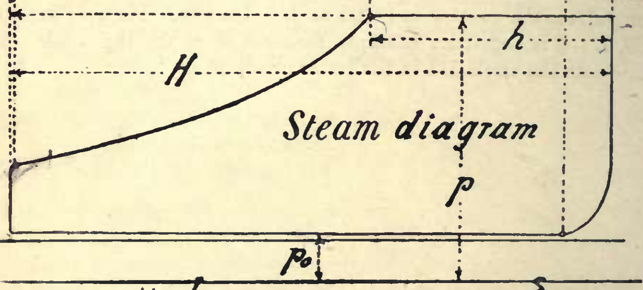
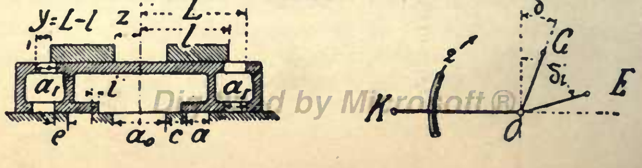


Fig. 571.



Figs. 572 and 573.

Valve Diagram for Meyer's Valve Gear (Figs. 570—573).

O X, direction of the valve motion, that is, plane of port face.

e, outside lap.

i, inside lap.

v, lead.

r, half stroke of main valve eccentric.

O E, position of cut-off valve eccentric radius.

r_1 half stroke of cut-off valve eccentric.

G P_1, parallel to O E.

O P_1, diameter of relative or resultant valve circle.

V_0, inside lead.

δ, angle of advance of main valve eccentric.

δ_1, angle of advance of cut-off valve eccentric.

$y_8 = L - l$, for cut-off at ·8 ; chord of circle III.

$y_4 = L - l$, for cut-off at ·4 ; chord of circle III.

$y_0 = L - l$, for cut-off at ·0 ; chord of circle III., negative.

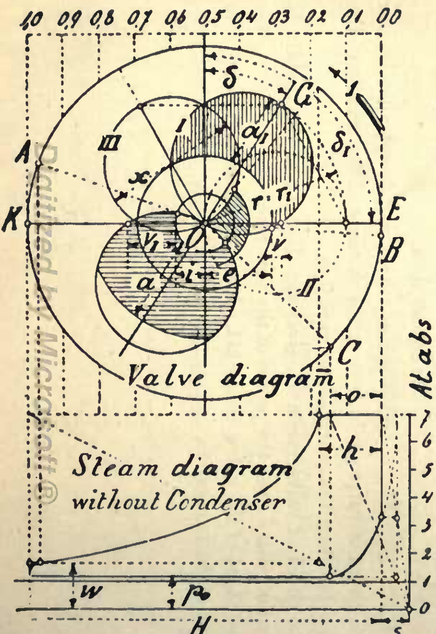
·8 is taken as the latest cut-off.

O B,	position of crank		for beginning of admission.
O, ·4	,,	,,	for beginning of expansion for cut-off at ·4.
O D,	,,	,,	when main valve closes the steam port.
O F,	,,	,,	when exhaust opens.
O M,	,,	,,	when port in main valve reopens to steam for cut off at ·4.
O C,	,,	,,	when exhaust closes and compression begins.

The chord of valve circle I. gives the distance of the middle of the main valve from the centre of the port face.

The chord of valve circle II. gives the distance of the middle of the cut-off valve from the centre of the port face.

The chord of the valve circle III. gives the distance between the centres of the two valves.



Figs. 574 and 575.

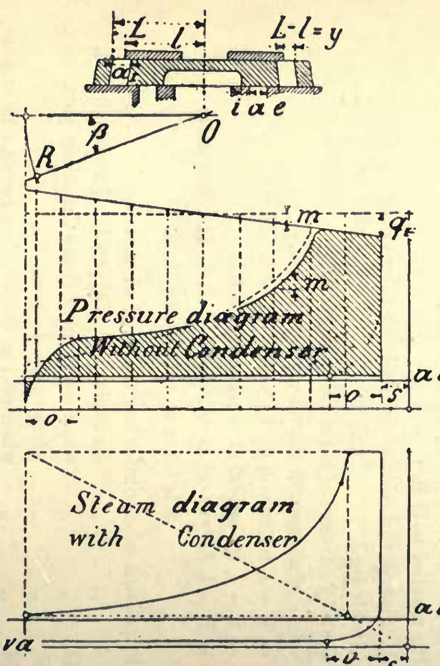


Fig. 576—578.

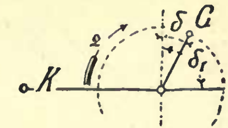


Fig. 579.

OK, position of crank.
OG, " of main valve eccentric.
OE, " of cut-off valve eccentric.

Valve Diagram.

OB, beginning of admission
OA, " of exhaust.
OC, " of compression.

Arrow 1, direction of movement of diagram crank.

Arrow 2, direction of movement of engine crank.

TABLE 55.—**Valve Diagram for Meyer's Valve Gear** (Figs. 574—578).

Stroke		H	16	20	24	28	32	36	40	44
Diameter of piston		D	8	10	12	14	16	18	20	22
Revolutions per minute		n	110	100	90	85	80	78	75	72
Width of steam ports		a	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$
Width of ports in main valve		a_1	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$
Outside lap		e	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$
Inside lap		i	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$
Half stroke of both valves		$r=r_1$	$\frac{3}{4}$	1	$1\frac{5}{16}$	$1\frac{5}{8}$	$1\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{1}{2}$	$2\frac{3}{4}$
Lead, outside		v	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{4}$
Lead, inside		v_1	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{7}{8}$	$\frac{7}{8}$
Angle of advance, main valve eccentric in degrees		δ	33	33	33	33	33	33	33	33
Angle of advance, cut-off eccentric in degrees		δ_1	90	90	90	90	90	90	90	90
Greatest distance between both valves		x	$\frac{3}{4}$	$\frac{15}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{3}{4}$
The variable distance L $-$ l between the working edges, y, is for different points of cut-off	cut-off at ·73	y	$\frac{3}{4}$	$\frac{15}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{3}{4}$
	„ ·60	y	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{13}{16}$	$2\frac{1}{8}$	$2\frac{5}{16}$	$2\frac{5}{8}$
	„ ·50	y	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{8}$	$2\frac{3}{8}$
	„ ·40	y	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{15}{16}$	$1\frac{3}{16}$	$1\frac{7}{16}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{1}{8}$
	„ ·30	y	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{16}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$
	„ ·20	y	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{7}{8}$	1	$1\frac{1}{8}$
	„ ·10	y	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$
	„ ·00	y	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$

All Dimensions are given in Inches.

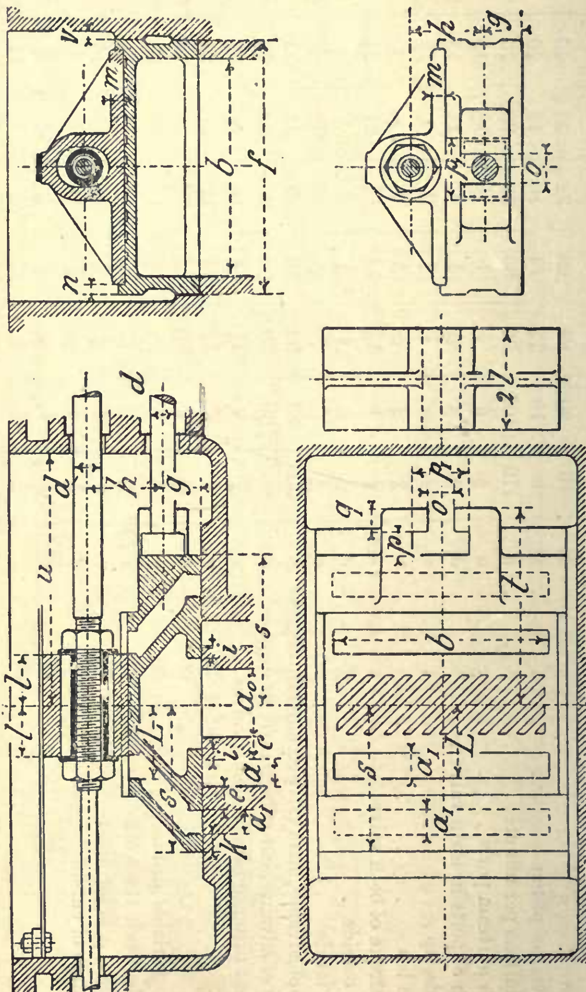


Fig. 582.

Fig. 583.

Fig. 580.

Fig. 581.

TABLE 56.—**Double Valves, for Fixed Expansion** (Figs. 580—583).

H	D	a_0	a	a_1	b	c	d	e	f	g	h	i	k	m	n	o	p	p_1	q	s	t	u	v	$r=r_1$
16	8	1	$\frac{1}{2}$	$\frac{3}{8}$	$4\frac{3}{8}$	$\frac{5}{8}$	1	$\frac{3}{8}$	$5\frac{3}{8}$	1	3	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{5}{8}$	$2\frac{1}{4}$	$\frac{3}{4}$	3	$4\frac{3}{8}$	$5\frac{5}{8}$	$\frac{1}{4}$	$\frac{3}{4}$
20	10	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$5\frac{5}{8}$	$\frac{11}{16}$	$1\frac{1}{8}$	$\frac{1}{2}$	$6\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{11}{16}$	$2\frac{3}{8}$	$\frac{7}{8}$	$3\frac{5}{8}$	$5\frac{3}{16}$	$6\frac{5}{8}$	$\frac{1}{4}$	1
24	12	$1\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$6\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	$8\frac{3}{8}$	$1\frac{3}{8}$	$3\frac{3}{8}$	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{1}{4}$	$\frac{3}{4}$	$2\frac{5}{8}$	1	$4\frac{3}{8}$	$6\frac{1}{8}$	8	$\frac{3}{8}$	$1\frac{5}{16}$
28	14	$2\frac{3}{8}$	1	$\frac{7}{8}$	$8\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{3}{4}$	10	$1\frac{5}{8}$	$3\frac{5}{8}$	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{3}{8}$	$\frac{7}{8}$	$2\frac{3}{4}$	1	$5\frac{5}{16}$	$7\frac{3}{16}$	$9\frac{1}{4}$	$\frac{3}{8}$	$1\frac{5}{8}$
32	16	$2\frac{3}{4}$	$1\frac{1}{8}$	1	10	1	$1\frac{3}{8}$	$\frac{7}{8}$	$11\frac{5}{8}$	$1\frac{3}{4}$	$3\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	1	3	$1\frac{1}{8}$	$6\frac{1}{8}$	$8\frac{1}{4}$	$10\frac{5}{8}$	$\frac{3}{8}$	$1\frac{7}{8}$
36	18	$3\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$11\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	1	$13\frac{3}{8}$	$1\frac{3}{4}$	4	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{16}$	$1\frac{5}{8}$	1	$3\frac{1}{4}$	$1\frac{1}{4}$	$6\frac{13}{16}$	$9\frac{1}{16}$	12	$\frac{3}{8}$	$2\frac{1}{8}$
40	20	$3\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$13\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$15\frac{1}{4}$	2	$4\frac{1}{4}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{1}{8}$	$3\frac{3}{8}$	$1\frac{3}{8}$	$7\frac{15}{16}$	$10\frac{7}{16}$	$13\frac{5}{8}$	$\frac{3}{8}$	$2\frac{1}{2}$
44	22	4	$1\frac{5}{8}$	$1\frac{1}{2}$	$14\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{1}{4}$	$16\frac{3}{4}$	$2\frac{1}{4}$	$4\frac{3}{8}$	$\frac{5}{8}$	1	$\frac{3}{4}$	$\frac{1}{2}$	2	$1\frac{1}{4}$	$3\frac{5}{8}$	$1\frac{1}{2}$	$8\frac{5}{8}$	$11\frac{3}{8}$	15	$\frac{3}{8}$	$2\frac{3}{4}$

All Dimensions are given in Inches

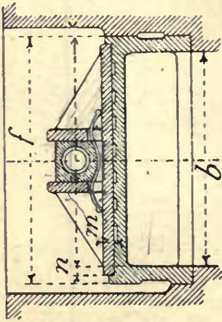


Fig. 586.

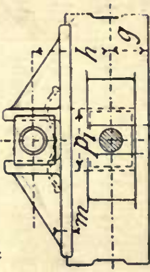


Fig. 587.

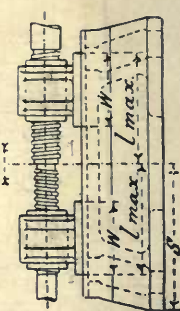


Fig. 588.

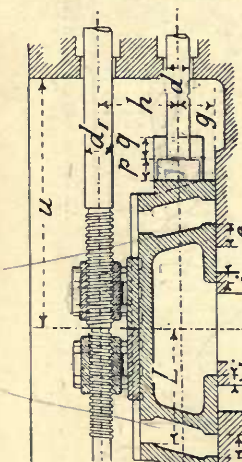


Fig. 584.

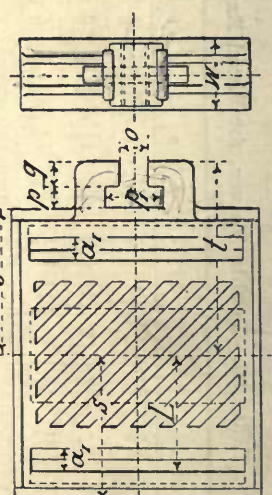
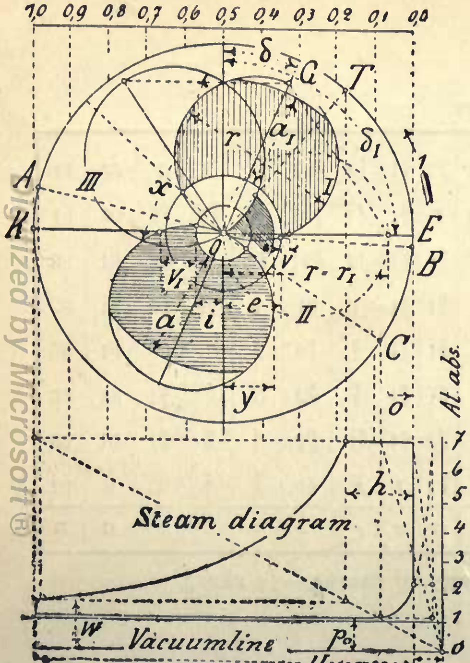


Fig. 585.

TABLE 57.—**Valves for Meyer's Variable Expansion Gear** (Figs. 584—588).

H	D	a_0	a	a_1	b	c	d	d_1	e	f	g	h	i	k	m	n	o	p	p_1	q	s	t	u	v	$r=r_1$
16	8	1	$\frac{1}{2}$	$\frac{3}{8}$	$4\frac{3}{8}$	$\frac{5}{8}$	1	$1\frac{3}{16}$	$\frac{3}{8}$	$5\frac{3}{8}$	1	3	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{5}{8}$	$2\frac{1}{4}$	$\frac{3}{4}$	3	$4\frac{3}{8}$	$5\frac{5}{8}$	$\frac{1}{4}$	$\frac{3}{4}$
20	10	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$5\frac{5}{8}$	$\frac{11}{16}$	$1\frac{1}{8}$	$1\frac{5}{16}$	$\frac{1}{2}$	$6\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{1}{4}$	$\frac{11}{16}$	$2\frac{3}{8}$	$\frac{7}{8}$	$3\frac{5}{8}$	$5\frac{3}{16}$	$6\frac{5}{8}$	$\frac{1}{4}$	1
24	12	$1\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$6\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{8}$	$8\frac{3}{8}$	$1\frac{3}{8}$	$3\frac{3}{8}$	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{1}{4}$	$\frac{3}{4}$	$2\frac{5}{8}$	1	$4\frac{3}{8}$	$6\frac{1}{8}$	8	$\frac{3}{8}$	$1\frac{5}{16}$
28	14	$2\frac{3}{8}$	1	$\frac{7}{8}$	$8\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$\frac{3}{4}$	10	$1\frac{5}{8}$	$3\frac{5}{8}$	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{3}{8}$	$\frac{7}{8}$	$2\frac{3}{4}$	1	$5\frac{5}{16}$	$7\frac{3}{16}$	$9\frac{1}{4}$	$\frac{3}{8}$	$1\frac{5}{8}$
32	16	$2\frac{3}{4}$	$1\frac{1}{8}$	1	10	1	$1\frac{3}{8}$	$1\frac{5}{8}$	$\frac{7}{8}$	$11\frac{5}{8}$	$1\frac{3}{4}$	$3\frac{7}{8}$	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	1	3	$1\frac{1}{8}$	$6\frac{1}{8}$	$8\frac{1}{4}$	$10\frac{5}{8}$	$\frac{3}{8}$	$1\frac{7}{8}$
36	18	$3\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$11\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{7}{8}$	1	$13\frac{3}{8}$	$1\frac{3}{4}$	$4\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{16}$	$1\frac{5}{8}$	1	$3\frac{1}{4}$	$1\frac{1}{4}$	$6\frac{13}{16}$	$9\frac{1}{16}$	12	$\frac{3}{8}$	$2\frac{1}{8}$
40	20	$3\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$13\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$1\frac{1}{8}$	$15\frac{1}{4}$	2	$4\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{1}{8}$	$3\frac{3}{8}$	$1\frac{3}{8}$	$7\frac{15}{16}$	$10\frac{7}{16}$	$13\frac{3}{8}$	$\frac{3}{8}$	$2\frac{1}{2}$
44	22	4	$1\frac{5}{8}$	$1\frac{1}{2}$	$14\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{1}{4}$	$16\frac{3}{4}$	$2\frac{1}{4}$	$4\frac{3}{8}$	$\frac{5}{8}$	1	$\frac{7}{8}$	$\frac{1}{2}$	2	$1\frac{1}{4}$	$3\frac{5}{8}$	$1\frac{1}{2}$	$8\frac{3}{8}$	$11\frac{3}{8}$	15	$\frac{3}{8}$	$2\frac{3}{4}$

All Dimensions are given in Inches.

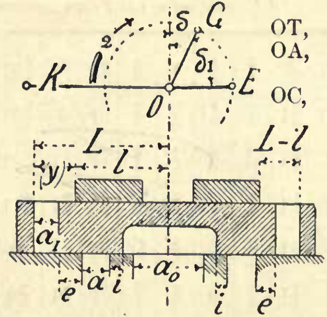


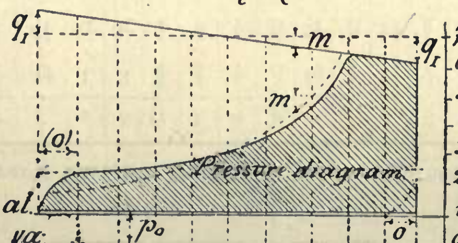
Figs. 589 and 590.

OK, position of crank (figs. 589, 591).
OG, „ of main valve eccentric.
OE, „ of cut-off valve eccentric.
OB, position of crank at beginning of admission.
OT, „ „ at cut-off, *h*.
OA, „ „ at beginning of exhaust.
OC, „ „ at beginning of compression.

Pressure Diagram (fig. 593).

q_1 is the centrifugal force of the reciprocating parts expressed in terms of the piston area at the dead point with indefinitely long connecting-rod.





Figs. 591—593.

Arrow 1, fig. 589, the direction of the movement of the diagram crank.

Arrow 2, fig. 591, the direction of the movement of the engine crank.

TABLE 58.—Diagram of Expansion Valves for Small Clearance Space (Figs. 589—593).

Stroke	H	16	20	24	28	32	36	40	44
Diameter of piston	D	8	10	12	14	16	18	20	22
Revolutions per minute	n	110	100	90	85	80	78	75	72
Width of port in port face	a	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$
" " in main valve	a_1	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$
Outside lap	e	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{13}{16}$
Inside lap	i	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$
Half stroke of both valves	$r=r_1$	$\frac{3}{4}$	1	$1\frac{3}{16}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{7}{16}$
Lead outside	v	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$
" inside	v_1	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$
Angle of advance, main valve eccentric in degrees	δ	26	26	25	24	24	25	25	25
" " cut-off in degrees	δ_1	90	90	90	90	90	90	90	90
Greatest distance between valves	x	$\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$1\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{7}{16}$	$2\frac{11}{16}$
Variable distance L—l between the working edges of the valves, y, in the diagram for different points of cut-off: ·76	y	$\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$1\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{7}{16}$	$2\frac{11}{16}$
·60	y	$\frac{11}{16}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$
·50	y	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{5}{16}$
·40	y	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2
·30	y	$\frac{3}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$
·20	y	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	1
·10	y	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$
·00	y	$-\frac{3}{8}$	$-\frac{1}{2}$	$-\frac{5}{8}$	$-\frac{7}{8}$	-1	$-1\frac{1}{8}$	$-1\frac{1}{4}$	$-1\frac{5}{8}$

All Dimensions are given in Inches.

The Valve Ellipse.

(a.) For Simple Valve Gear.

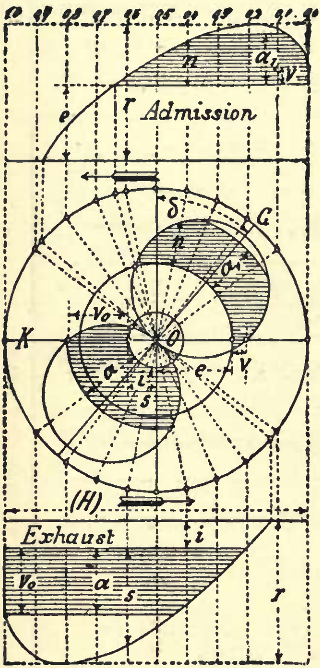


Fig. 594.

The valve ellipse, fig. 594, is a great assistance in understanding Zeuner's valve diagram, and it also shows the speed with which the valve moves at different parts of its stroke, when opening or closing the ports. The length of the connecting-rod is disregarded and is taken to be infinite. The diameter of the crank circle is divided in 10 equal parts, and perpendicular lines are drawn through these points terminating in horizontal lines as in the fig. 594 placed for convenience above and below the valve diagram, and the distances from the centre of the valve to the centre of the port face (chords of the valve circle) are laid off on the perpendicular lines, through the points thus found the curve called the valve ellipse is drawn, the lap e is laid off and the port width a_1, then the shaded part of the diagram gives the actual opening of the port by the valve; at the dead point, for example, the port is open to the extent of v, when the crank is at the position ·4 the port is open the amount n. The scale of fig. 594 is $\frac{2}{5}$ natural size:

Width of port, a_1 = ·67 inch.
" a = ·78 "
Half stroke of valve r = 1·77 inch.
Lap (outside) e = ·98 "
" (inside) i = ·35 "
Lead (outside) v = ·196 inch.
" (inside) v_o = ·78 "
Angle of advance δ = 41°.

(b.) For Rider's and Meyer's Valve Gears.

The diagram and ellipse for the main slide is constructed in the same manner as for the simple valve gear. In the figs. 595, 596, the influence of the magnitude of the angle of advance δ_1 (of the cut-off

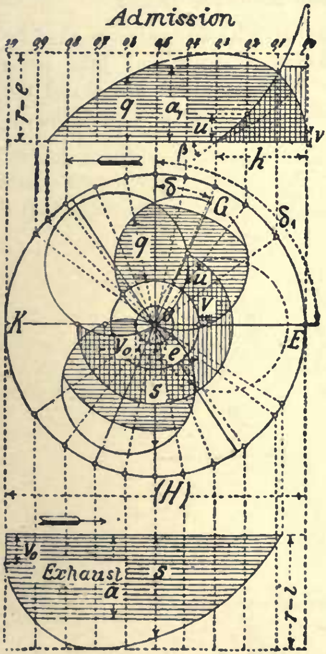


Fig. 595.

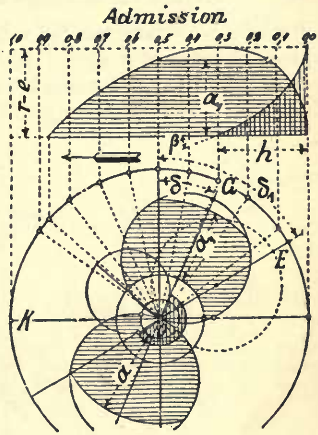


Fig. 956.

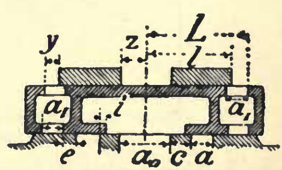


Fig. 597.

valve eccentric) on the speed with which the ports are closed is shown, and the following data are taken as starting points :—

Admission period or cut-off, h = ·3 port in back of main valve a_1 = ·98 inch ; steam port, a = 1·14 inch ; outside lap, e = ·59 inch ; inside lap, i = 2·55 inch ; half stroke of main and cut-off valves r and r_1 = 1·73 inch ; outside lead = ·12 inch angle of advance of main valve eccentric = 24° scale $\frac{2}{5}$ natural size. In fig. 595 the advance angle of cut-off valve eccentric = 90°, in fig. 596 = 60°. To show the opening of the ports in back of main valve, lay off the distances between the acting edges of the main and cut-off valves as ordinates, then the area shaded with vertical lines gives the opening of the ports, a comparison of fig. 595 with fig. 596 shows there is a better admission of steam with an advance angle of 90° than with 60°; the magnitude of the angle β shows this clearly.

Defect in Arrangement of Valve Gears.

By defective arrangement of valve gears with two valves worked by separate eccentrics, a second admission of steam may take place towards the end of the stroke before the main valve has closed the steam port. The diagram fig. 598 shows an example of this, the ex-

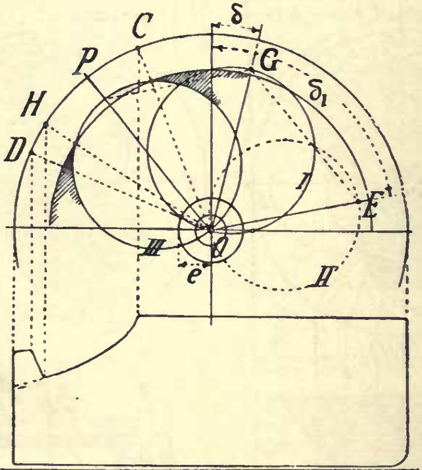


Fig. 598.

pansion will begin when the crank is at O C when the cut-off valve has closed the port; but they will reopen again when the crank is at O H, and second admission will take place as the main valve only closes the port when the crank is at O D. This fault can be avoided by a proper advance angle δ_1 and stroke of valve 2 r_1, the position of the diameter O P of the valve circle III. fig. 598 must be near the position O D the point at which the main valve closes the port. To avoid this fault requires great care in valve gears where the expansion is varied by varying the stroke of the cut-off valve eccentric as in Hartnell's expansion governor. Under indicator diagrams will be found an example which occurred in practice, but from a different cause to that above mentioned.

Arrangement for Variation of Expansion by Band.

The expansion may be varied by increasing the advance angle of the cut-off eccentric by turning the eccentric round on the shaft and fixing it by means of a bolt to the main valve eccentric as shown in fig. 599, this has usually to be adjusted when the engine is stopped, but it has also been arranged to be varied automatically. The diagram fig. 600 shows the effect of altering the advance angle δ_1 from 90° to 60° and the cut will be thus altered from ·6 to ·25 of the stroke. It should be noted that for this kind of variable expansion the two ends of the main valve must be separated from each other as in fig. 584, so that each end is independent, but for valve gears where the

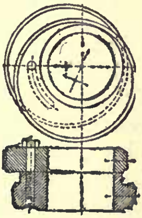


Fig. 599.

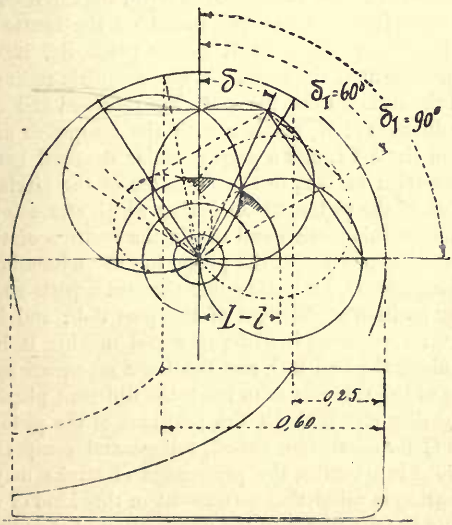


Fig. 600.

$a = 29$; $e = 15$; $r = 44$; $v = 3$.
$a_1 = 25$; $i = 6·5$; $r_1 = 44$; $L - l = 22$.
Angle of advance $\delta_1 = 90°$; Cut off $= 0·25$.
" " " $= 60°$; " $= 0·60$.

expansion is varied by altering the stroke of the cut-off eccentric the two ends need not be independent of each other, so that all the ports in the back of the main valve are used for each end of the cylinder.

Valve Diagram for Valve Gears with Two Slide Valves, each Worked by a Separate Eccentric.

The construction of the diagram is as follows:—

First describe a circle (fig. 601) whose diameter equals the stroke of the main valve eccentric and draw the horizontal and vertical diameters, R Q = N T. On the horizontal, R Q set off the lap of the valve, plus the lead, O L ; from O L erect a perpendicular, cutting the circle VI. in K, draw the radial line, O K, then O K will be the position of the main valve eccentric radius when the crank is at one of the dead points R, and the angle K O N, will be the "angle of advance" of main eccentric. The steam port in cylinder may be represented by the perpendiculars drawn below R Q from the points P and J, the distance O P being equal to the lap of the valve. If now the radial line O K be imagined to move round on the centre O, in the direction of the arrow VI. from K to the point K_1, it will show the angular movement of the crank-pin and eccentric radius from the dead point of the former until the valve has returned and closed the port. If the distance K K_1 is taken with the compasses and set off from R to x_0 in circle VI. and a perpendicular dropped cutting R Q in x_2, the proportion of R x_2 to the diameter of the circle VI. will give the portion of the piston-stroke during which steam is admitted. It is easy to follow this motion and obtain the position of the crank, and from thence the position of the piston for the admission, cut-off, release and compression, by setting out the steam ports and exhaust in their proper position at the right of the port P J ; and by setting out the valve on a separate piece of paper and moving it backwards and forwards along the line R Q, and holding a set-square against the left-hand edge of the valve so as to mark the different phases on the circle or on the diameter R Q, all the positions of the piston may be marked on R Q for admission, cut-off, release and compression, and if R Q be divided into tenths, the percentage of stroke may be read off ; always bearing in mind that no account in this kind of valve diagram is taken of the difference in the distribution of steam caused by the angle of the connecting and eccentric rods.

For valve gears with two valves, having the cut-off valve worked on the back of the main valve by a separate eccentric, a further construction is necessary. The example given is one with a "gridiron" or many-ported cut-off valve, and with a varying stroke, the maximum stroke being taken as equal to the stroke of the main valve.

On K in the circle VI., describe a circle XII. of a diameter equal to the width of one port in back of main valve, and touching this circle, draw the perpendiculars C D, F G. On a point in circle VI. describe a circle equal in diameter, to width of one port in cut-off valve, so that this circle overlaps the line of port in main valve F G, by a small amount, say, $\frac{1}{16}$th of an inch, more or less; this may be called the lead of cut-off valve. The line O A, gives the position of the cut-off

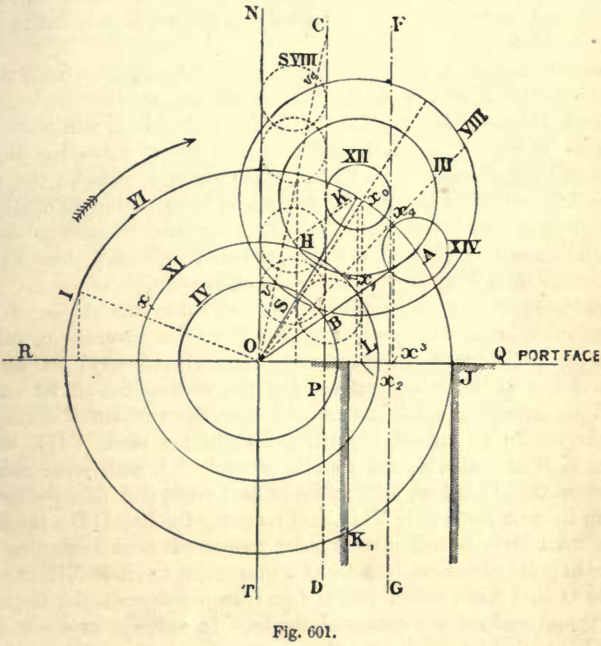


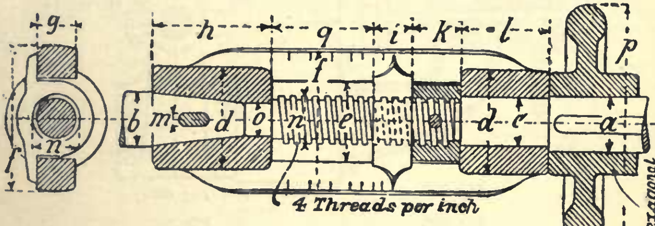
Fig. 601.

eccentric radius, when the crank is at one of the dead points R, and angle N O A "angle of advance" of cut-off valve eccentric.

From point K at radius K A, draw circle III. If closely observed, it will be seen that when the system of circles XII. and XIV. are revolved round on centre O, at radius O K, and at the fixed distance from each other K A, they will assume different positions with regard to the observer looking down on them. In the position on the diagram, circle XIV. is below and to the right of circle XII. and when K reaches K_1, XIV. will be to the left of XII. and above it, so in the complete revolution, the two circles XII. and XIV. will revolve about one another, and their relative motion with regard to each other will be the same if one, say, XII., is fixed, and XIV. revolving round it at radius K A, and, taking this to be the case, all the positions of the cut-off valve ports with relation to those on the back of the main valve can be determined; and, further, the position of crank at admission by cut-off valve, at cut-off, and at reopening of main valve ports, can be laid out in percentages of stroke or otherwise.

Describe the circle IV. on centre O at radius equal to K A, then, taking radius of port circle XIV. in compasses, describe circle at H on circle III. touching the line C D. The radius K H will represent position of cut-off eccentric radius when cut-off valve has closed ports on back of main valve, that is, when steam is cut-off; the distance A H will show the angular movement from beginning of stroke to point of cut-off by cut-off valve. Transfer this distance to circle IV. and draw the radius O V cutting the circle VI. at V, drop a perpendicular from V on to R Q and read off percentage of stroke. In the present case of an expansion gear controlled by the stroke of the cut-off valve being decreased or increased by the governor or other means, this will be the latest cut-off corresponding here to the maximum stroke of the cut-off valve. For the earliest cut-off, let circle XIV. be moved along line O A and take the position B giving a shorter stroke to cut-off eccentric, describe the circle VIII. with radius K B on centre K, and describe a circle XI. with same radius on centre O. As before, take radius of port circle and describe circle S with its centre on circle VIII. and touching the line C D; the distance from B to S will give angular movement from beginning of stroke to point of cut-off, transfer this movement to circle XI. From centre O to I draw radial line O I and drop perpendicular from I_1 to R Q and read off percentage of stroke. In valve gears where the cut-off is varied by shortening the stroke of cut-off valve eccentric, there is a danger of the ports reopening too soon and admitting steam again late in the stroke before the lap of the main valve has covered the steam port. To find where reopening occurs, describe port circle S VIII. with centre on circle VIII. and touching line C D, transfer the angular motion from B to S VIII. to circle XI. as before, starting on line R Q and cutting the circle XI. at x_1. Draw radial line O x_1 cutting circle VI. in x_4, drop perpendicular on the line R Q, note where it cuts at x_3, then if x_4 is on the right hand of L, the point where the main valve closes the port, the reopening of cut-off ports will then take place after the main valve has closed the steam port, and no second admission of steam will take place; if, however, x_3 is on the left hand side of L, second admission will take place, and the arrangement will have to be altered. Further, the bars of metal between cut-off ports can be taken from the greatest distance between the main and cut-off eccentric centres, that is, in this case, from K to B, so that the radius of circle VIII. will give the greatest relative movement of the valves on each other and the least possible distance between nearest edges of ports; about $\frac{1}{8}$th of an inch or more must be added in practice for safety.

Hand Adjustments for Meyer's Valve Gear.



Figs. 601*, 602.

TABLE 59.—**Dimensions of Hand Adjustments for Meyer's Valve Gear (Figs. 601*, 602).**

H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q**
16	8	1	$1\frac{1}{8}$	1	$1\frac{7}{8}$	$1\frac{1}{2}$	$2\frac{3}{4}$	$\frac{3}{4}$	$2\frac{3}{8}$	$\frac{3}{4}$	1	$1\frac{5}{8}$	$\frac{1}{4}$	1	$\frac{11}{16}$	$4\frac{7}{8}$	$1\frac{1}{8}$
20	10	$1\frac{1}{8}$	$1\frac{1}{4}$	1	$2\frac{1}{8}$	$1\frac{1}{2}$	3	$\frac{3}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{3}{4}$	$\frac{1}{4}$	1	$\frac{3}{4}$	$5\frac{1}{4}$	$1\frac{3}{8}$
24	12	$1\frac{1}{4}$	$1\frac{5}{16}$	1	$2\frac{1}{4}$	$1\frac{5}{8}$	$3\frac{1}{4}$	$\frac{7}{8}$	$2\frac{1}{2}$	$\frac{3}{4}$	1	2	$\frac{5}{16}$	1	$\frac{3}{4}$	$5\frac{1}{2}$	$1\frac{3}{4}$
28	14	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$2\frac{3}{8}$	$1\frac{3}{4}$	$3\frac{3}{8}$	1	$2\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$2\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{3}{4}$	$5\frac{3}{4}$	$2\frac{1}{16}$
32	16	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$2\frac{1}{2}$	2	$3\frac{5}{8}$	1	$2\frac{7}{8}$	1	$1\frac{1}{4}$	$2\frac{1}{4}$	$\frac{3}{8}$	$1\frac{1}{4}$	1	$6\frac{3}{8}$	$2\frac{5}{8}$
36	18	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{1}{8}$	$3\frac{3}{4}$	$1\frac{1}{8}$	$3\frac{1}{4}$	1	$1\frac{3}{8}$	$2\frac{3}{8}$	$\frac{3}{8}$	$1\frac{1}{4}$	1	$6\frac{3}{4}$	$2\frac{1}{2}$
40	20	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$4\frac{1}{4}$	$1\frac{1}{4}$	$3\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{1}{2}$	$\frac{7}{16}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$7\frac{1}{8}$	$2\frac{5}{8}$
44	22	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{7}{8}$	$2\frac{3}{8}$	$4\frac{1}{2}$	$1\frac{1}{4}$	4	$1\frac{1}{4}$	$1\frac{1}{2}$	$2\frac{3}{4}$	$\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$7\frac{3}{8}$	3

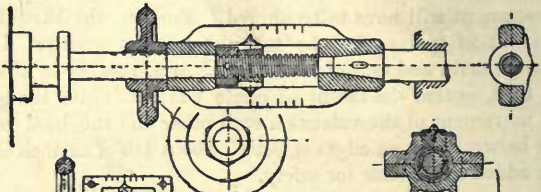
All Dimensions are given in Inches.

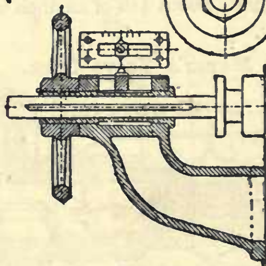
In the above design (fig. 601*) the hand-wheel is rather small; the boss is therefore made hexagonal, so that a spanner may be used if required.

* q is for cut-off varying from 0 to ·73.

Hand Adjustment for Meyer's and Rider's Valve Gears.

Fig. 603.





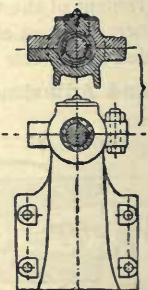
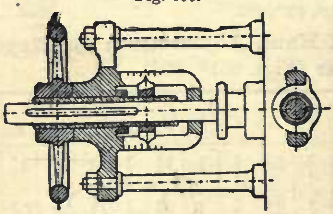


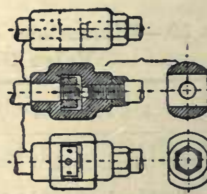
Fig. 604.

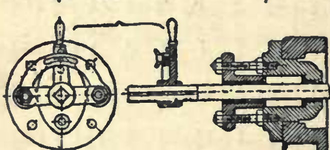
Fig. 605.

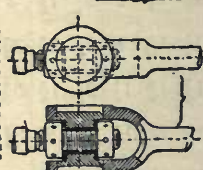
Fig. 606.

Fig. 607.









For Riders Valve gear

Fig. 608.

Fig. 609.

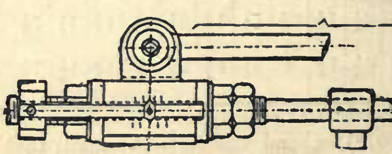
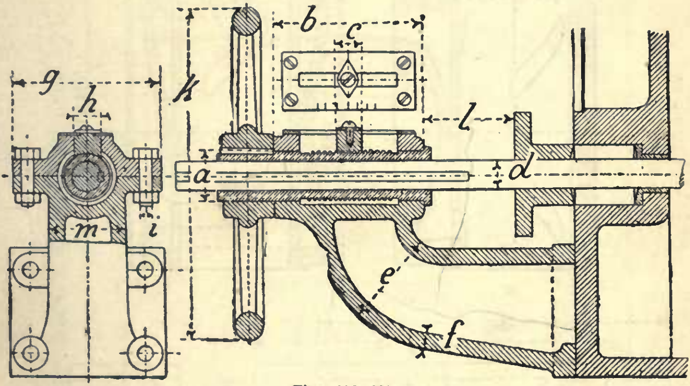


Fig. 610.

Fig. 611.

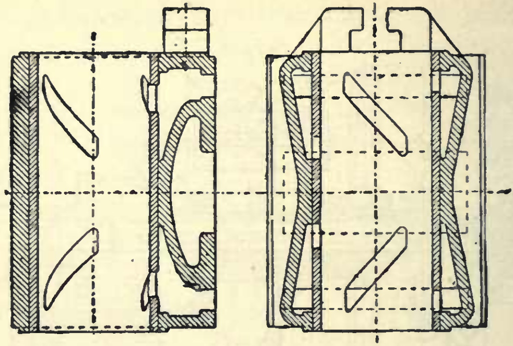


Figs. 612, 613.

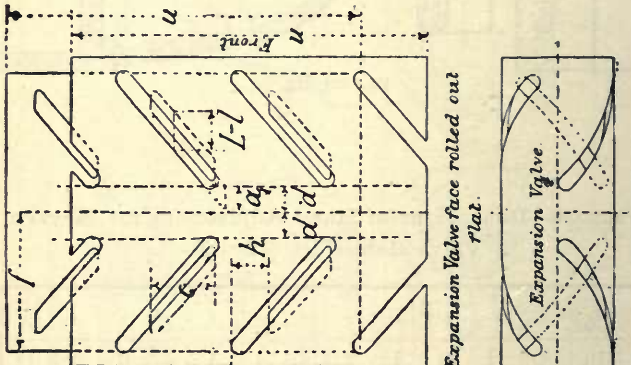
TABLE 60.—**Dimensions of Hand Adjustment for Meyer's Valve Gear** (Figs. 612, 613).

H	D	d	a	b	c	e	f	g	h	i	k	l	m
20	10	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$\frac{3}{4}$	2	$\frac{3}{8}$	4	1	$\frac{3}{8}$	6	2	$1\frac{1}{2}$
24	12	$\frac{7}{8}$	$1\frac{3}{8}$	$3\frac{3}{4}$	$\frac{7}{8}$	$2\frac{3}{8}$	$\frac{3}{8}$	$4\frac{3}{8}$	1	$\frac{1}{2}$	$7\frac{5}{8}$	$2\frac{5}{8}$	$1\frac{7}{8}$
28	14	1	$1\frac{1}{2}$	$4\frac{3}{8}$	1	$2\frac{3}{4}$	$\frac{3}{8}$	$4\frac{3}{4}$	$1\frac{1}{8}$	$\frac{1}{2}$	$9\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{1}{4}$
32	16	$1\frac{1}{8}$	$1\frac{3}{4}$	5	$1\frac{1}{8}$	$3\frac{1}{4}$	$\frac{1}{2}$	$5\frac{1}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	$11\frac{1}{4}$	$3\frac{1}{4}$	$2\frac{1}{2}$
36	18	$1\frac{1}{4}$	$1\frac{7}{8}$	$5\frac{5}{8}$	$1\frac{1}{4}$	$3\frac{5}{8}$	$\frac{9}{16}$	$5\frac{5}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$12\frac{3}{4}$	$3\frac{5}{8}$	$2\frac{7}{8}$
40	20	$1\frac{3}{8}$	$2\frac{1}{8}$	$6\frac{3}{8}$	$1\frac{1}{4}$	$4\frac{1}{4}$	$\frac{5}{8}$	6	$1\frac{3}{8}$	$\frac{5}{8}$	$14\frac{3}{8}$	4	$3\frac{1}{4}$
44	22	$1\frac{3}{8}$	$2\frac{1}{4}$	$7\frac{1}{4}$	$1\frac{3}{8}$	$4\frac{3}{4}$	$\frac{3}{4}$	$6\frac{3}{8}$	$1\frac{3}{8}$	$\frac{5}{8}$	16	$4\frac{3}{8}$	$3\frac{5}{8}$

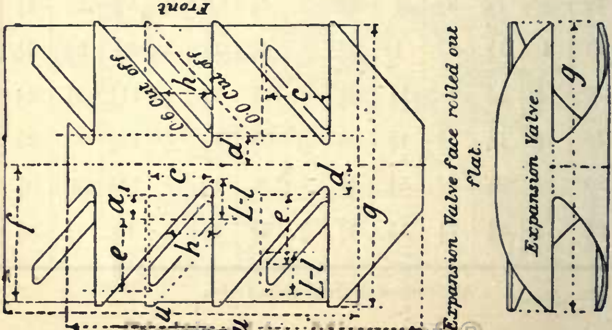
All Dimensions are given in Inches.



Figs. 618, 619.



Figs. 616, 617.



Figs. 614, 615.

TABLE 61.—**Dimensions of Cut-off Valve Faces, &c., for Rider's Valve Gear** (Figs. 614—619).

Stroke	H	20	24	28	32	36	40	44
Diameter of piston	D	10	12	14	16	18	20	22
Diameter of expansion valve	m	$3\frac{1}{4}$	4	$4\frac{3}{4}$	$5\frac{5}{8}$	$6\frac{5}{8}$	$7\frac{3}{8}$	$8\frac{3}{8}$
Circumference of expansion valve	u	$10\frac{3}{16}$	$12\frac{1}{2}$	$14\frac{7}{8}$	$17\frac{5}{8}$	$20\frac{3}{4}$	$23\frac{1}{16}$	$26\frac{3}{16}$
$L-l$ for ·0 cut-off		$\frac{7}{16}$	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{3}{4}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$
$L-l$ for ·6 cut-off		$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{13}{16}$	$2\frac{1}{8}$	$2\frac{5}{16}$	$2\frac{5}{8}$
Width of port	a_1	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$
Width of port	b	$5\frac{5}{8}$	$6\frac{3}{4}$	$8\frac{3}{8}$	10	$11\frac{5}{8}$	$13\frac{1}{4}$	$14\frac{3}{4}$
Height of port (c=about $\frac{b}{3}$)	c	$1\frac{3}{4}$	$2\frac{5}{16}$	$2\frac{3}{4}$	$3\frac{3}{8}$	4	$4\frac{3}{8}$	5
Dimension	d	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$
Dimension	e	$2\frac{1}{8}$	$2\frac{5}{8}$	$3\frac{1}{4}$	4	$4\frac{5}{8}$	$5\frac{1}{8}$	$5\frac{3}{4}$
Half length of valve seat	f	$4\frac{1}{4}$	5	6	7	8	$8\frac{3}{4}$	$9\frac{3}{4}$
Length of cut-off valve	g	$8\frac{5}{8}$	$10\frac{3}{8}$	$12\frac{3}{8}$	$14\frac{3}{8}$	$16\frac{3}{8}$	$18\frac{3}{8}$	$20\frac{3}{8}$
Turning movement of valve	h	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{5}{8}$	3	$3\frac{3}{8}$
Cut-off at mean position of governor		·13	·13	·13	·13	·13	·13	·13

All Dimensions are given in Inches.

Rider's Valve Gear.

Rider's valves differ from Meyer's in having the working edges of the cut-off valve formed as a right and left hand screw, the valve itself being turned partially round in its seat, so as to give the same effect as the separate valves of Meyer moved further apart or nearer

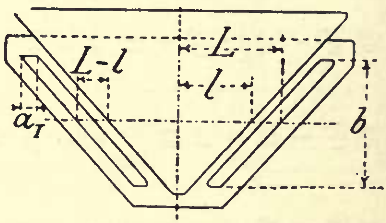


Fig. 620.

together by means of the right and left hand screws and nuts. The effect is the same if the valve is a flat plate with edges inclined right and left, fig. 620; if the plate be moved down the distance $L - l$ is decreased, and if the plate be moved upwards increased.

Fig. 621. Fig. 622.

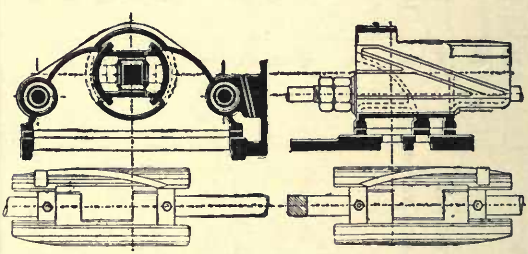
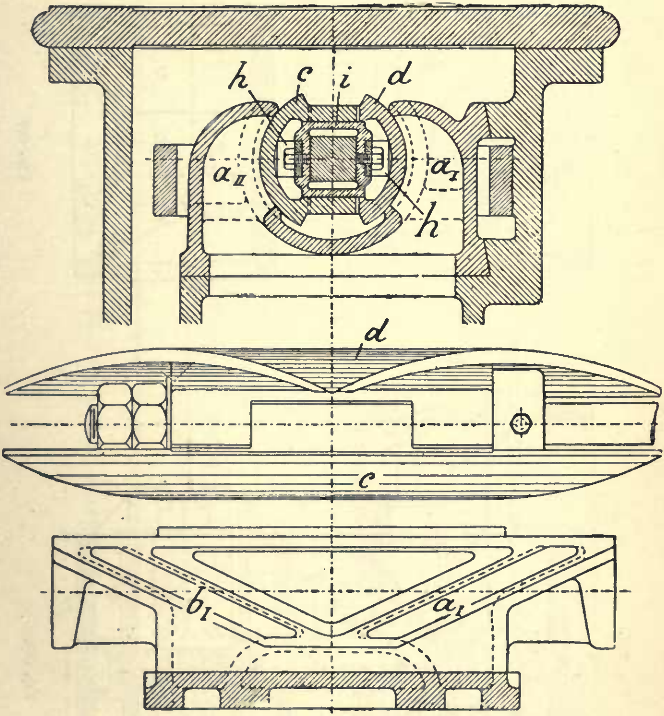


Fig. 623.

A modification of Rider's valve by Leutert of Halle is shown in figs. 621—626. The cut-off valve is in two parts, *c* and *d*, figs. 624—626, this giving short steam ports, and also rendering the necessary adjustment for variation of expansion of less range. On the back of the main valve are four ports, a_1, a_{11}, b_1, b_{11}; the first two unite into one port at the right hand end of the valve, and the two latter into one port at the left hand end of the valve; the cut-off valve has a square spindle by which it is turned in its seat to vary the cut-off. The two parts of the cut-off valve are kept up to their seat by springs, *h*.



Figs. 624—626.

These valves are usually controlled by the governor. The two halves of the valve form a kind of piston valve, and should work with very little friction.

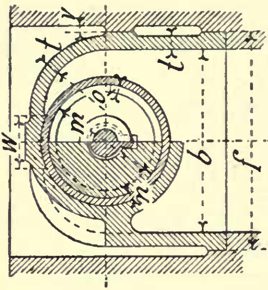


Fig. 629.

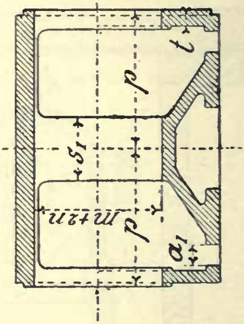


Fig. 630.

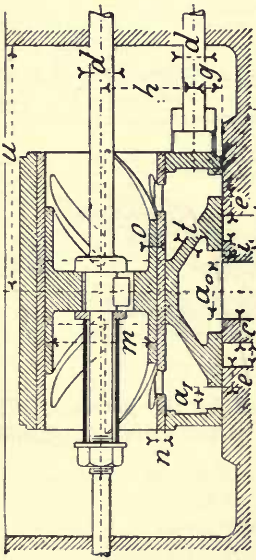


Fig. 627.

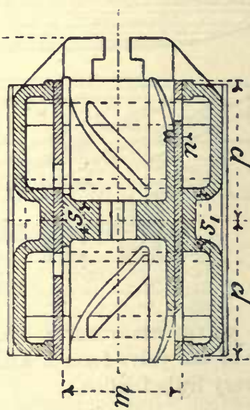


Fig. 628.

TABLE 62.—**Dimensions of Rider's Closed Valves** (Figs. 627—630).

H	D	a_0	a	a_1	b	c	d	e	f	g	h	i	m	n	o	p	q	s	s_1	t	u	v	w	$r=r_1$
20	10	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$5\frac{5}{8}$	$\frac{11}{16}$	$1\frac{1}{8}$	$\frac{1}{2}$	$6\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{4}$	$\frac{1}{4}$	$3\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$4\frac{1}{4}$	$5\frac{13}{16}$	$1\frac{3}{8}$	$2\frac{3}{8}$	$\frac{1}{2}$	$7\frac{1}{4}$	$\frac{1}{4}$	$1\frac{1}{2}$	1
24	12	$1\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$6\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	$8\frac{3}{8}$	$1\frac{3}{8}$	$3\frac{5}{8}$	$\frac{5}{16}$	4	$\frac{3}{8}$	$\frac{3}{8}$	5	$6\frac{3}{4}$	$1\frac{5}{8}$	$2\frac{5}{8}$	$\frac{1}{2}$	$8\frac{5}{8}$	$\frac{3}{8}$	$1\frac{3}{4}$	$1\frac{5}{16}$
28	14	$2\frac{3}{8}$	1	$\frac{7}{8}$	$8\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{3}{4}$	10	$1\frac{5}{8}$	$4\frac{3}{8}$	$\frac{5}{16}$	$4\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{2}$	6	$7\frac{7}{8}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$\frac{9}{16}$	$10\frac{1}{4}$	$\frac{3}{8}$	2	$1\frac{5}{8}$
32	16	$2\frac{3}{4}$	$1\frac{1}{8}$	1	10	1	$1\frac{3}{8}$	$\frac{7}{8}$	$11\frac{5}{8}$	$1\frac{3}{4}$	5	$\frac{3}{8}$	$5\frac{5}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	7	$9\frac{1}{8}$	$1\frac{7}{8}$	3	$\frac{9}{16}$	$11\frac{5}{8}$	$\frac{3}{8}$	$2\frac{1}{4}$	$1\frac{7}{8}$
36	18	$3\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$11\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	1	$13\frac{3}{8}$	$1\frac{3}{4}$	$5\frac{3}{4}$	$\frac{1}{2}$	$6\frac{5}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	8	$10\frac{1}{4}$	2	$3\frac{1}{4}$	$\frac{5}{8}$	$13\frac{1}{4}$	$\frac{3}{8}$	$2\frac{3}{8}$	$2\frac{1}{8}$
40	20	$3\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$13\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$15\frac{1}{4}$	2	$6\frac{3}{8}$	$\frac{1}{2}$	$7\frac{3}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$8\frac{3}{4}$	$11\frac{1}{4}$	$2\frac{1}{8}$	$3\frac{3}{8}$	$\frac{11}{16}$	$14\frac{3}{8}$	$\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{1}{2}$
44	22	4	$1\frac{5}{8}$	$1\frac{1}{2}$	$14\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{1}{4}$	$16\frac{3}{4}$	$2\frac{1}{4}$	$7\frac{1}{4}$	$\frac{5}{8}$	$8\frac{3}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$9\frac{3}{4}$	$12\frac{1}{2}$	$2\frac{1}{4}$	$3\frac{5}{8}$	$\frac{3}{4}$	16	$\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{3}{4}$

All Dimensions are given in Inches.

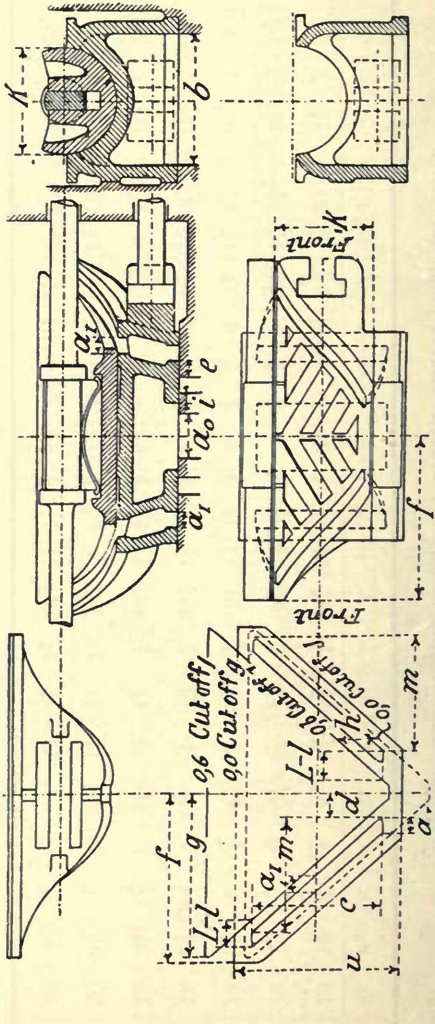


Fig. 631. Fig. 632. Fig. 633. Fig. 634. Fig. 635. Fig. 636.

TABLE 63.—**Dimensions of Rider's Open Valves** (Figs. 631—636).

Stroke	H	16	20	24	28	32
Diameter of piston	D	8	10	12	14	16
Diameter of cut-off valve	k	$2\frac{3}{4}$	$3\frac{5}{8}$	$4\frac{5}{8}$	$5\frac{5}{8}$	$6\frac{5}{8}$
Half circumference	u	$8\frac{5}{8}$	$11\frac{5}{16}$	$14\frac{1}{2}$	$17\frac{5}{8}$	$20\frac{3}{4}$
Width of port	a_1	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{7}{8}$	1
$L-l$ for ·0 cut-off		$-\frac{3}{8}$	$-\frac{7}{16}$	$-\frac{1}{2}$	$-\frac{11}{16}$	$-\frac{3}{4}$
$L-l$ for ·6 cut-off		$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{13}{16}$
Height of port c=about ·8 of b	c	$3\frac{5}{8}$	$4\frac{3}{8}$	$5\frac{5}{8}$	$6\frac{3}{4}$	$8\frac{3}{8}$
Dimension	d	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2
Dimension	m	$3\frac{1}{8}$	4	5	$6\frac{1}{8}$	$7\frac{1}{2}$
Half length of main valve	f	$4\frac{3}{4}$	6	$7\frac{3}{8}$	$9\frac{3}{8}$	$11\frac{3}{8}$
Half length of cut-off valve	g	$4\frac{5}{8}$	$5\frac{3}{4}$	7	9	11
Turning movement of cut-off valve	h	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{3}{8}$	$2\frac{3}{4}$
Cut-off with governor in mean position		·13	·13	·13	·13	·13

All Dimensions are given in Inches

Connection between Rider's Valve Gear and the Governor.

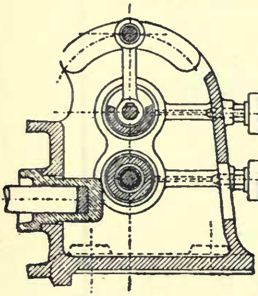


Fig. 637.

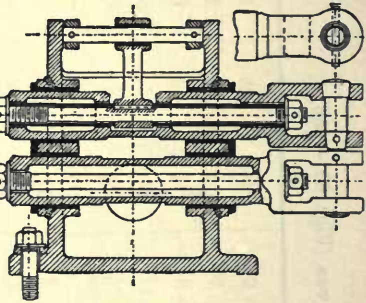


Fig. 638.

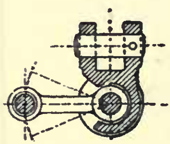


Fig. 639.

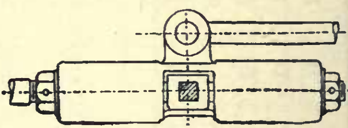


Fig. 640.

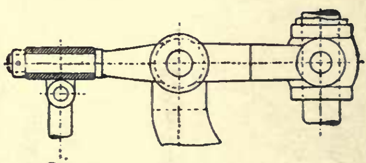


Fig. 641.

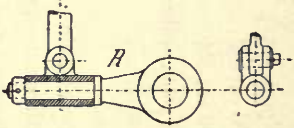


Fig. 642.

The connection between Rider's valve gear and the governor is shown in various ways in figs. 637—642. Figs. 637, 638, show a method by Starke and Hoffman, with cast-iron guides to the valve rods; figs. 639, 640, a method by Leutert; figs. 641, 642, a method of connection with a kind of universal joint, to allow for the backwards and forwards movement of the valve rod.

Connection between Rider's Valve Gear and Governor.

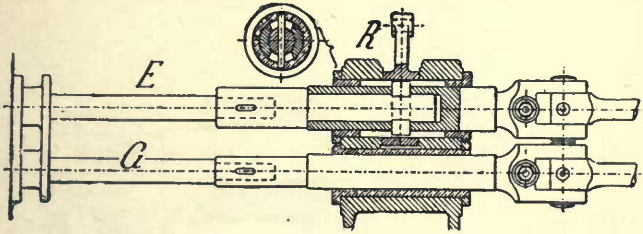


Fig. 643.

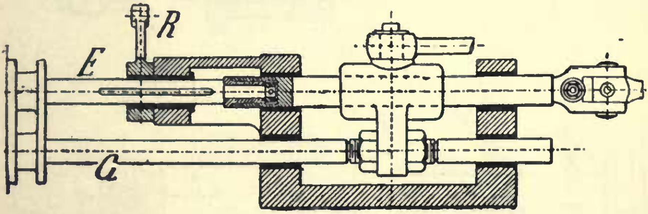


Fig. 644.

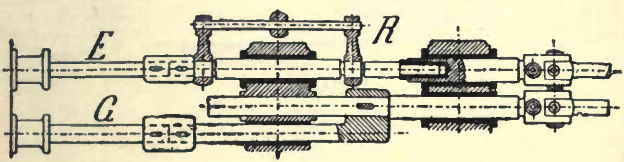


Fig. 645.

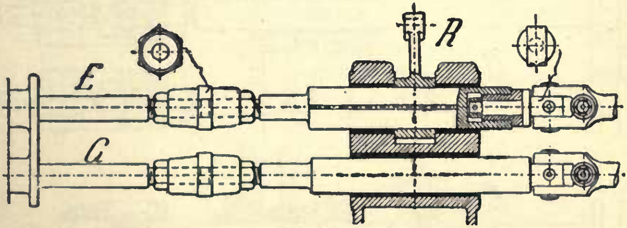


Fig. 646.

G, the main valve rod: E, the cut-off valve rod; R, the arm to which the governor rod is attached.

Connection between Rider's Valve Gear and Governor.

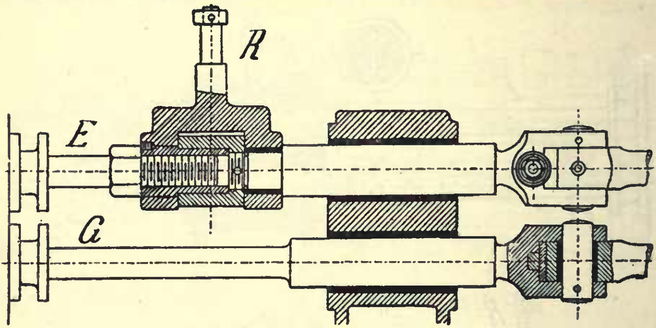


Fig. 647.

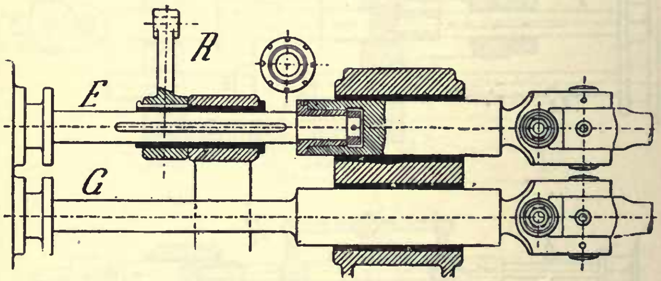
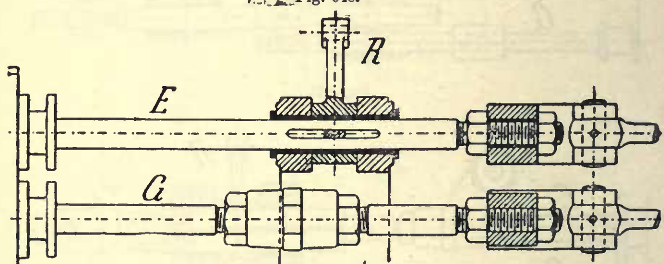
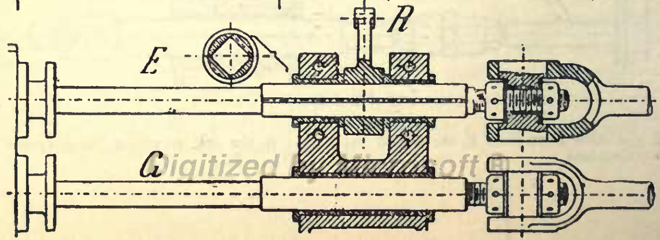


Fig. 648.





Remarks on the Governor Bracket (Figs. 651—663).

The combined governor bracket and valve rod guide on the next page may be taken as an example, a somewhat complicated piece of designing, but at the same time a substantial fixing for the governor, driving gear and guides. In engine design, especially with horizontal girder engines, it is well to have as few fixings on the girder as possible, for neatness and other obvious reasons. The whole of the governor gear and attachment to the valve rod, in this case, can be fitted up together, ready to bolt on the engine girder. In cases where a feed-pump is fitted to the engine, the barrel can often be made in the same casting as the valve-rod guides with the necessary valve boxes bolted on. This forms a very compact arrangement, and if, as may happen, the pump centre comes far out from the girder, a stay of wrought iron can be fitted from the pump casting to the slide valve chest to take the thrust of the pump.

By making the flanges at both ends of the pump barrel of the same size, the pump can be used for both right- and left-hand engines, a very important point in the economical manufacture of steam engines, and in many other machines.

In figs. 651—663 one of the valve-rod guides is shown bushed with brass or bronze. If both guide-plunger and guide are of cast iron, there is certainly no necessity for this, although it is often convenient for the owner of an engine to have such guides bushed, even if it is only with cast iron, in order that new bushes can be supplied by the maker, without having to supply a new bracket, with a risk of variation in the centres, causing trouble and delay.

Combined Bracket for carrying Governor and Valve-Rod Guides (Figs. 651—663).

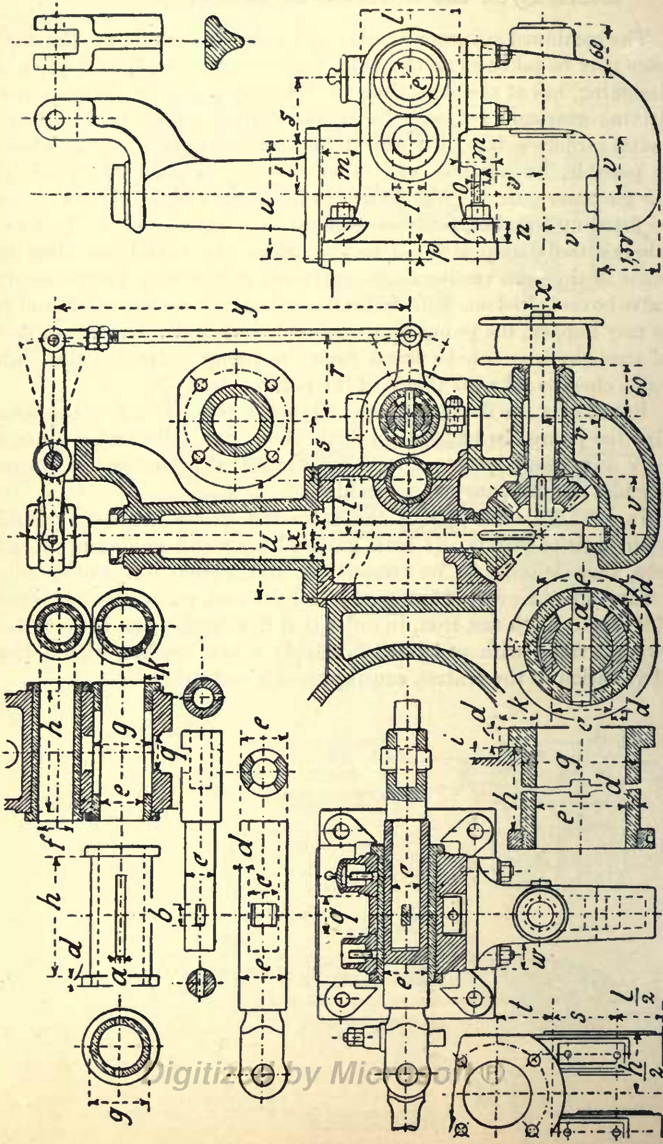


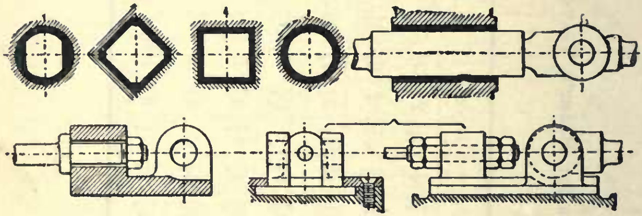
TABLE 64.—Dimensions of Valve-Rod Guides and Governor Stands for Rider's Valve Gear (Figs. 651—663).

H	D	a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	r	s	t	u	v	w	x	y
20	10	$\frac{1}{2}$	1	$1\frac{3}{8}$	$\frac{3}{8}$	$2\frac{1}{4}$	$1\frac{5}{8}$	3	$6\frac{1}{2}$	$\frac{1}{4}$	1	5	2	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{3}{4}$	4	$3\frac{1}{4}$	$2\frac{3}{4}$	$5\frac{5}{8}$	3	$1\frac{1}{4}$	$1\frac{3}{8}$	16
24	12	$\frac{1}{2}$	1	$1\frac{1}{2}$	$\frac{7}{16}$	$2\frac{3}{8}$	$1\frac{3}{4}$	$3\frac{3}{8}$	7	$\frac{1}{4}$	$1\frac{1}{8}$	$5\frac{1}{4}$	$2\frac{1}{2}$	1	$\frac{3}{4}$	$\frac{7}{8}$	2	$4\frac{1}{4}$	$3\frac{5}{8}$	3	$6\frac{3}{8}$	$3\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{3}{8}$	18
28	14	$\frac{9}{16}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$\frac{1}{2}$	$2\frac{5}{8}$	2	$3\frac{5}{8}$	$7\frac{3}{4}$	$\frac{1}{4}$	$1\frac{1}{4}$	6	$2\frac{3}{4}$	1	$\frac{3}{4}$	1	$2\frac{1}{8}$	$4\frac{1}{2}$	$4\frac{3}{8}$	$3\frac{1}{8}$	$7\frac{1}{4}$	$3\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{2}$	20
32	16	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{7}{8}$	$\frac{1}{2}$	$2\frac{7}{8}$	$2\frac{1}{8}$	4	$8\frac{1}{4}$	$\frac{1}{4}$	$1\frac{1}{4}$	$6\frac{1}{2}$	$3\frac{1}{4}$	$1\frac{1}{4}$	$\frac{7}{8}$	1	$2\frac{3}{8}$	$4\frac{5}{8}$	5	$3\frac{1}{4}$	8	$3\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{2}$	22
36	18	$\frac{5}{8}$	$1\frac{1}{4}$	2	$\frac{9}{16}$	$3\frac{1}{8}$	$2\frac{3}{8}$	$4\frac{1}{4}$	$9\frac{1}{4}$	$\frac{3}{8}$	$1\frac{3}{8}$	7	$3\frac{3}{4}$	$1\frac{3}{8}$	1	$1\frac{1}{8}$	$2\frac{1}{2}$	$4\frac{3}{4}$	$5\frac{3}{4}$	$3\frac{3}{8}$	9	$3\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$	24
40	20	$\frac{3}{4}$	$1\frac{3}{8}$	$2\frac{1}{4}$	$\frac{5}{8}$	$3\frac{3}{8}$	$2\frac{1}{2}$	$4\frac{5}{8}$	$9\frac{3}{4}$	$\frac{3}{8}$	$1\frac{3}{8}$	$7\frac{3}{4}$	4	$1\frac{3}{8}$	1	$1\frac{1}{4}$	$2\frac{5}{8}$	$5\frac{1}{4}$	$6\frac{3}{8}$	$3\frac{5}{8}$	9	$3\frac{7}{8}$	2	$1\frac{3}{4}$	26
44	22	$\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{3}{8}$	$\frac{5}{8}$	$3\frac{5}{8}$	$2\frac{3}{4}$	$4\frac{7}{8}$	10	$\frac{3}{8}$	$1\frac{1}{2}$	8	$4\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$2\frac{3}{4}$	$5\frac{3}{4}$	$7\frac{1}{4}$	$3\frac{3}{4}$	10	4	$2\frac{3}{8}$	$1\frac{3}{4}$	28

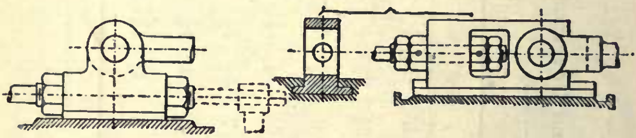
All Dimensions are given in Inches.

Valve Rod Guides and Joints

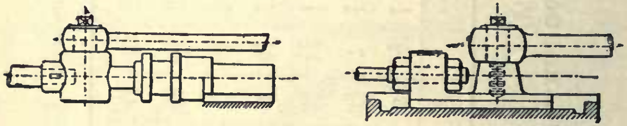
Figs. 664—668.



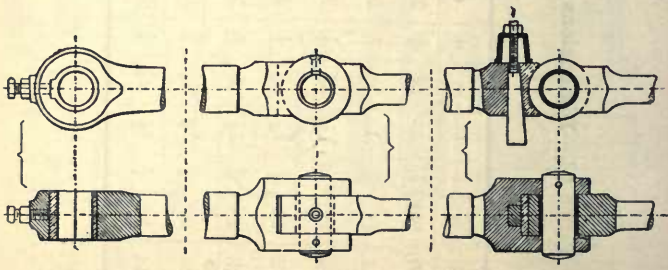
Figs. 669—671.



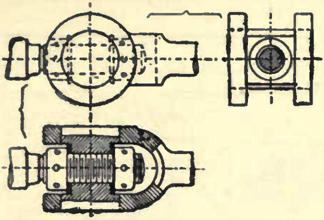
Figs. 672—675.



Figs. 676, 677.



Figs. 678—681.



Figs. 682—684.

Figs. 664 to 685 show a variety of examples of valve-rod joints and guides. That shown in fig. 685, is a very useful form when the

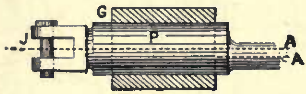


Fig. 685.

eccentric rod and valve rod are not in line, the guide plunger, P, valve rod and joint, J, being in one forging, and turned in the lathe on two centres, A, A ; this form is used in locomotives.

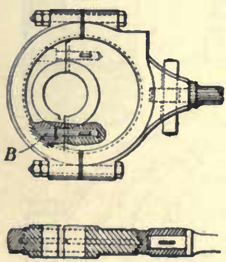


Fig. 685a.

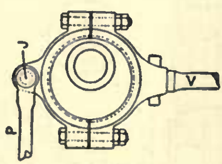
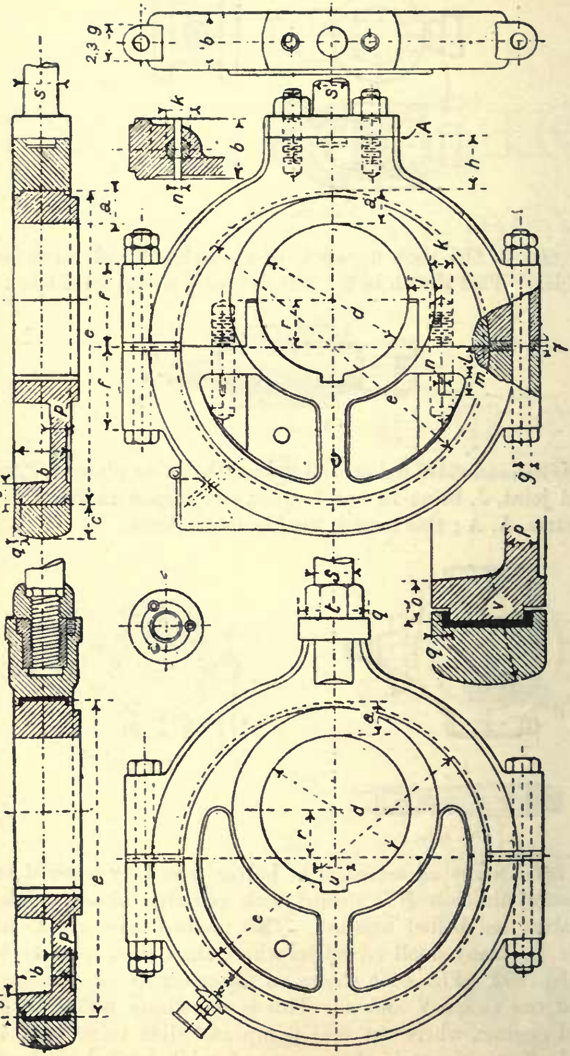


Fig. 685*b*.

Fig. 585*a* shows an eccentric in halves joined by cottered bolts; the head of the bolt *B* is turned with eccentric sheave when the two halves are bolted together. This method saves room, and is suitable for many small eccentrics where there is no space for bolts, as in fig. 693. Fig. 685*b* shows an arrangement for working two rods off one strap, V and *a*. This is sometimes useful in small vertical engines, where the feed pump and slide valve have to be worked off one eccentric, the pump rod being jointed, as shown, to the outer end of the strap.

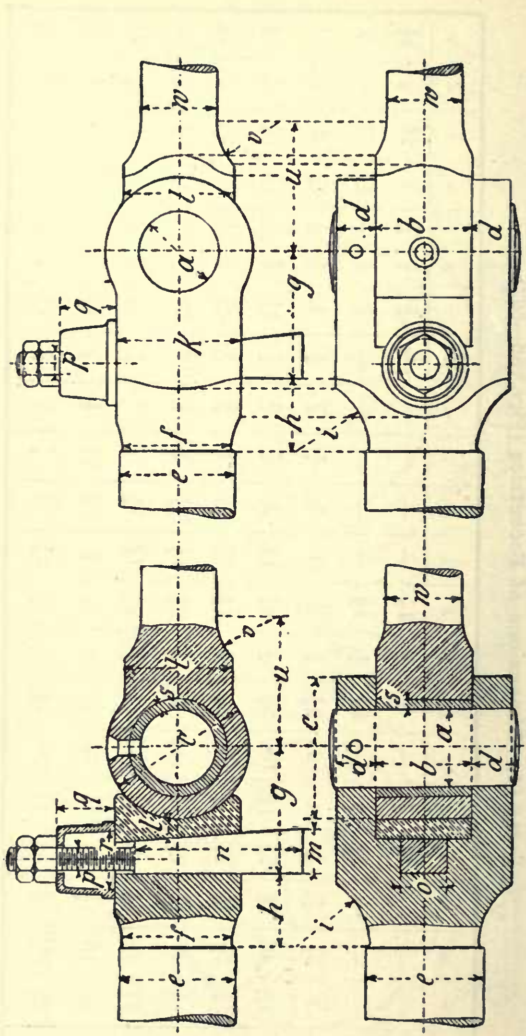


Figs. 686—693.

TABLE 65.—**Dimensions of Eccentrics** (Figs. 686—693).

H	D	r	d	a	b	c	e	f	g	h	i	k	l	m	n	o	p	q	s	t	u	v
12	6	—	—	—	$1\frac{1}{2}$	$\frac{7}{8}$	—	2	$\frac{1}{2}$	$1\frac{1}{2}$	—	—	—	—	—	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	1	$2\frac{5}{8}$	$\frac{1}{2}$	$1\frac{1}{2}$
16	8	—	—	—	$1\frac{3}{4}$	1	—	$2\frac{3}{8}$	$\frac{1}{2}$	$1\frac{3}{4}$	—	—	—	—	—	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{1}{4}$	$1\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{8}$	2
20	10	1	$5\frac{1}{4}$	$\frac{7}{8}$	2	1	9	$2\frac{3}{4}$	$\frac{5}{8}$	2	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{4}$	1	$\frac{5}{8}$	$\frac{1}{4}$	$1\frac{3}{8}$	3	$\frac{5}{8}$	$2\frac{1}{2}$
24	12	$1\frac{5}{16}$	6	$1\frac{1}{8}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$10\frac{7}{8}$	$3\frac{1}{4}$	$\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{4}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{5}{16}$	$1\frac{1}{2}$	$3\frac{1}{4}$	$\frac{5}{8}$	$2\frac{1}{2}$
28	14	$1\frac{5}{8}$	$7\frac{3}{8}$	$1\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$13\frac{1}{8}$	$3\frac{5}{8}$	$\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$	1	$\frac{3}{4}$	$\frac{5}{16}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{5}{16}$	$1\frac{5}{8}$	$3\frac{3}{8}$	$\frac{3}{4}$	$2\frac{3}{4}$
32	16	$1\frac{7}{8}$	8	$1\frac{3}{8}$	$2\frac{3}{4}$	$1\frac{1}{4}$	$14\frac{1}{2}$	4	$\frac{7}{8}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{3}{8}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{16}$	$1\frac{3}{4}$	$3\frac{5}{8}$	$\frac{3}{4}$	3
36	18	$2\frac{1}{8}$	$8\frac{5}{8}$	$1\frac{3}{8}$	3	$1\frac{3}{8}$	$15\frac{5}{8}$	$4\frac{5}{8}$	$\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{16}$	2	4	$\frac{3}{4}$	$3\frac{1}{4}$
40	20	$2\frac{1}{2}$	10	$1\frac{1}{2}$	$3\frac{1}{4}$	$1\frac{1}{2}$	18	$4\frac{3}{4}$	1	3	2	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{8}$	$2\frac{1}{8}$	$4\frac{1}{2}$	$\frac{3}{4}$	$3\frac{1}{2}$
44	22	$2\frac{3}{4}$	$11\frac{1}{4}$	$1\frac{5}{8}$	$3\frac{1}{2}$	$1\frac{1}{2}$	20	$5\frac{1}{4}$	1	$3\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{4}$	1	$\frac{3}{8}$	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{8}$	$2\frac{1}{4}$	$4\frac{1}{2}$	$\frac{7}{8}$	$3\frac{1}{2}$

All Dimensions are given in Inches.

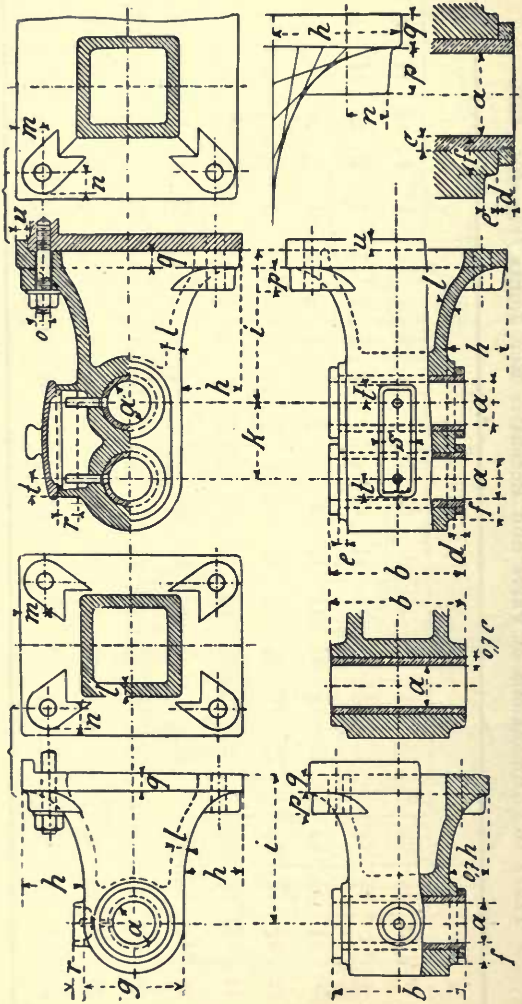


Figs. 694—697.

TABLE 66.—**Dimensions of Valve and Eccentric Rod Joints** (Figs. 694—697).

H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*
12	6	7/8	1 3/8	2 1/8	1/2	1 1/2	1 1/4	1 3/4	1 1/8	1	1 5/8	1/4	3/4	2 3/8	5/8	3/8	7/8	7/8	1/8	1 1/4	1 5/8	3/4	3/4
16	8	1	1 1/2	2 1/4	5/8	1 1/2	1 3/8	2	1 1/4	1 1/8	1 3/4	1/4	3/4	2 1/2	3/4	3/8	1	1	3/16	1 3/8	1 3/4	3/4	7/8
20	10	1 1/8	1 1/2	2 3/8	5/8	1 5/8	1 1/2	2 1/8	1 1/4	1 1/4	2	3/8	3/4	2 5/8	3/4	1/2	1	1	3/16	1 1/2	2	7/8	1
24	12	1 1/4	1 5/8	2 5/8	5/8	1 7/8	1 3/4	2 1/4	1 1/4	1 1/4	2 1/8	3/8	3/4	2 3/4	3/4	1/2	1 1/8	1	3/16	1 3/4	2 1/8	1	1 1/8
28	14	1 3/8	1 3/4	2 3/4	3/4	2	1 7/8	2 3/8	1 3/8	1 3/8	2 1/4	3/8	7/8	3	7/8	1/2	1 1/4	1 1/8	3/16	1 7/8	2 1/8	1	1 3/8
32	16	1 3/8	1 7/8	2 3/4	3/4	2 1/8	2	2 1/2	1 3/8	1 3/8	2 3/8	3/8	7/8	3 1/8	7/8	1/2	1 1/4	1 1/8	1/4	2	2 1/2	1 1/8	1 1/2
36	18	1 1/2	2	3	3/4	2 3/8	2 1/8	2 5/8	1 1/2	1 1/2	2 1/2	3/8	7/8	3 1/4	1	1/2	1 3/8	1 1/4	1/4	2 1/8	2 5/8	1 1/4	1 5/8
40	20	1 1/2	2 1/8	3 1/8	3/4	2 1/2	2 3/8	2 3/4	1 1/2	1 1/2	2 3/4	3/8	7/8	3 3/8	1	1/2	1 3/8	1 1/4	1/4	2 3/8	2 3/4	1 3/8	1 7/8
44	22	1 5/8	2 1/4	3 1/4	7/8	2 3/4	2 3/8	2 3/4	1 5/8	1 5/8	2 7/8	3/8	1	3 5/8	1	1/2	1 1/2	1 1/4	1/4	2 3/8	3	1 3/8	2

All Dimensions are given in Inches.



Figs. 698—705.

TABLE 67.—**Dimensions of Valve-Rod Guides for Simple and Meyer's Valve Gears** (Figs. 698—705).

H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*
12	6	1 1/2	2 3/4	1/4	5/16	1/4	5/8	3 5/8	—	—	—	—	—	—	—	—	—	—	—	—	—
16	8	1 1/2	3 1/2	1/4	5/16	1/4	5/8	3 3/4	—	—	—	—	—	—	—	—	—	—	—	—	—
20	10	1 5/8	4 1/2	5/16	3/8	1/4	3/4	4 1/4	2	4 3/4	3 1/4	1/2	1 1/4	1	5/8	3/4	3/4	3/4	1 1/4	3/4	3/8
24	12	1 3/4	5 1/4	5/16	3/8	1/4	7/8	4 5/8	2 3/8	5 3/4	3 3/8	9/16	1 1/4	1	5/8	7/8	3/4	3/4	1 1/4	3/4	3/8
28	14	2	6	3/8	1/2	3/8	1	5	2 3/4	6 3/4	3 5/8	9/16	1 3/8	1	3/4	1	7/8	1	1 1/2	3/4	1/2
32	16	2 1/8	6 3/4	3/8	1/2	3/8	1	5 1/4	3 1/4	8	3 7/8	5/8	1 1/2	1	3/4	1	1	1	2	7/8	1/2
36	18	2 3/8	7 3/4	3/8	1/2	3/8	1 1/8	5 5/8	3 5/8	9	4 1/4	11/16	1 1/2	1 1/4	7/8	1 1/8	1	1 1/4	2 1/2	7/8	5/8
40	20	2 1/2	8 1/2	3/8	1/2	3/8	1 1/4	6	4	10	4 3/8	3/4	1 5/8	1 1/4	7/8	1 1/4	1 1/8	1 3/8	2 3/4	1	5/8
44	22	2 3/4	8 3/4	3/8	1/2	3/8	1 1/4	6 3/8	4 1/4	10 1/2	4 5/8	3/4	1 3/4	1 1/4	1	1 1/4	1 1/4	1 3/8	3	1	3/4

All Dimensions are given in Inches.

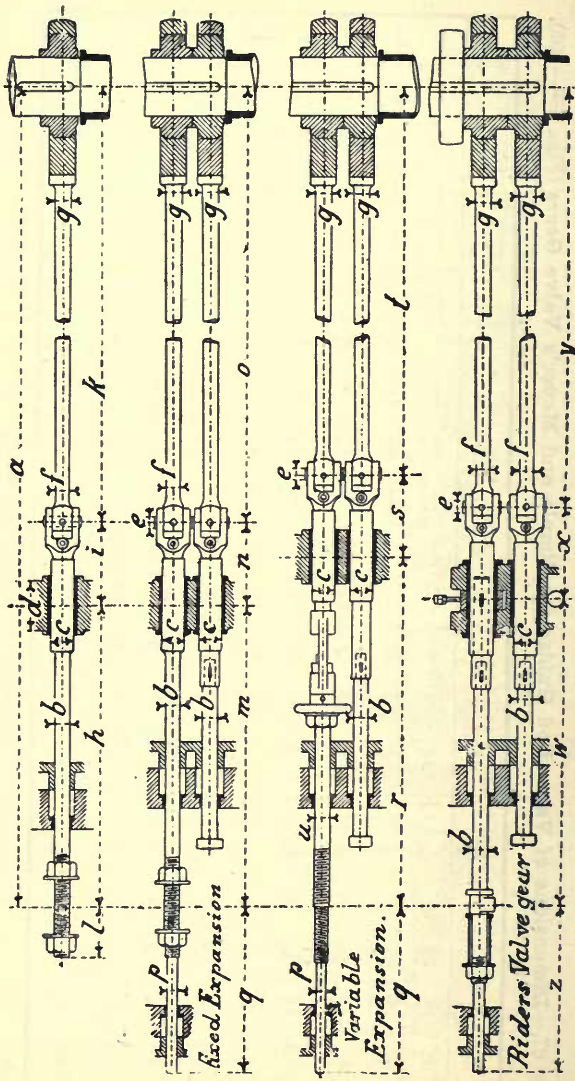


Fig 706 Fig. 707. Fig. 708. Fig. 709.

Figs. 706—709.

TABLE 68.—**Dimensions of Eccentric Rods and Valve Rods** (Figs. 706—709).

		Dimensions common to figs. 706—709.							Dimen. for fig. 706.				Dimensions for fig. 707.					Dimensions for fig. 708, for Meyer's valve gear adjusted by hand.					Dimensions for fig. 709, for Rider's valve gear adjusted by governor.			
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	Threads per in.	*w*	*x*	*y*	*z*
12	6	58	$\frac{7}{8}$	$1\frac{1}{2}$	$2\frac{3}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	1	30	5	23	$2\frac{5}{8}$	—	–	–	–	–	–	–	–	–	–	–	–	..	–
16	8	$74\frac{5}{8}$	1	$1\frac{1}{2}$	$3\frac{1}{2}$	1	$\frac{7}{8}$	$1\frac{1}{4}$	$33\frac{3}{4}$	$5\frac{7}{8}$	35	3	35	$5\frac{7}{8}$	33	$\frac{3}{4}$	$13\frac{1}{4}$	$41\frac{3}{4}$	$5\frac{7}{8}$	27	$1\frac{1}{4}$	4	–	–	–	–
20	10	90	$1\frac{1}{8}$	$1\frac{5}{8}$	$4\frac{1}{2}$	$1\frac{1}{8}$	1	$1\frac{3}{8}$	35	7	48	$3\frac{5}{8}$	$37\frac{3}{8}$	$6\frac{5}{8}$	46	$\frac{3}{4}$	15	$43\frac{3}{8}$	$6\frac{5}{8}$	40	$1\frac{3}{8}$	4	38	8	44	$14\frac{3}{8}$
24	12	$106\frac{1}{4}$	$1\frac{3}{16}$	$1\frac{3}{4}$	$5\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{2}$	$38\frac{1}{4}$	8	60	$4\frac{3}{8}$	$42\frac{3}{4}$	$7\frac{1}{2}$	56	$\frac{7}{8}$	$16\frac{5}{8}$	$48\frac{3}{4}$	$7\frac{1}{2}$	50	$1\frac{1}{2}$	4	$42\frac{1}{4}$	9	55	17
28	14	122	$1\frac{1}{4}$	2	6	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{5}{8}$	43	9	70	$4\frac{7}{8}$	47	8	67	1	$18\frac{3}{4}$	53	8	61	$1\frac{5}{8}$	4	$46\frac{1}{4}$	$9\frac{3}{4}$	66	$19\frac{1}{4}$
32	16	138	$1\frac{3}{8}$	$2\frac{1}{8}$	$6\frac{3}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	–	–	–	–	49	9	80	$1\frac{1}{8}$	$20\frac{3}{4}$	57	9	72	$1\frac{3}{4}$	4	$50\frac{1}{2}$	$10\frac{1}{2}$	77	$21\frac{5}{8}$
36	18	154	$1\frac{3}{8}$	$2\frac{3}{8}$	$7\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$	2	–	–	–	–	54	10	90	$1\frac{1}{4}$	$23\frac{1}{4}$	63	10	81	$1\frac{7}{8}$	3	$56\frac{1}{2}$	$11\frac{1}{2}$	86	24
40	20	$169\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{1}{2}$	$8\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{1}{8}$	–	–	–	–	$59\frac{3}{4}$	11	100	$1\frac{3}{8}$	$24\frac{3}{4}$	$64\frac{3}{4}$	11	94	2	3	$60\frac{3}{4}$	12	97	$25\frac{5}{8}$
44	22	$186\frac{3}{4}$	$1\frac{5}{8}$	$2\frac{3}{4}$	$8\frac{3}{4}$	$1\frac{5}{8}$	2	$2\frac{1}{4}$	–	–	–	–	$61\frac{3}{4}$	12	113	$1\frac{3}{8}$	$26\frac{3}{4}$	$70\frac{3}{4}$	12	104	$2\frac{1}{8}$	3	$64\frac{3}{4}$	13	109	28

All Dimensions are given in Inches.

Fig. 710.

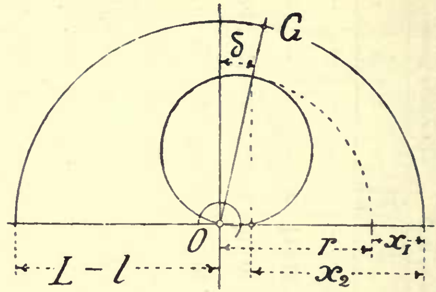


Fig. 711.

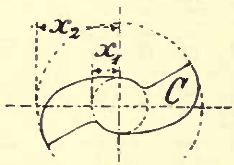
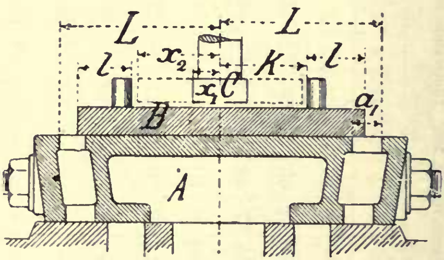


Fig. 712.



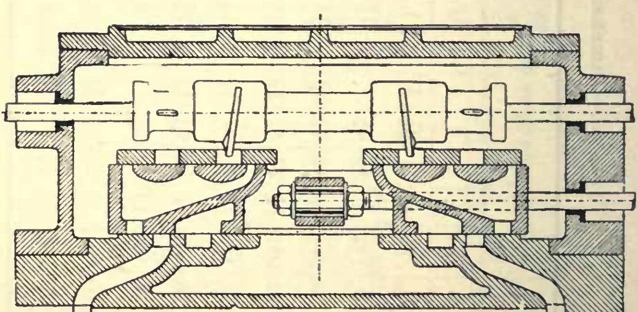


Fig. 713

Farcot's Valve Gear.

The expansion valve B, fig. 712, is here loose on the back of the main valve A, and is dragged along by friction with the latter; the distance to which the expansion valve is carried is determined by the position of the cam C, figs. 711 and 712 ; the range of variation in this gear can be from ·0 to ·4. The limit of the latest cut-off is when the crank is at O G, fig. 710, and is dependent on the angle of advance. The valve diagram is constructed as follows :—

δ = the angle of advance of main valve eccentric ; r = the radius of eccentricity of main valve eccentric ; x = the smallest diameter of the cam, $\frac{3}{8}''$ to $1''$; $L - l = r + x_1$; then

$x_2 = L - l - r \sin. \delta$ = the greatest diameter of the cam.

$2k = 2(L - l) - a_1$.

$a_1 = < 2(L - l - x_2)$.

When these equations are satisfied, the port, a_1, will be fully open when x is at its least value. The opening of a_1 is usually too small, so that two or more ports are required.

Guhrauer's valve gear is shown in fig. 713, and is, in effect, similar to that of Meyer.

Piston Valves.

Piston valves have been the subject of much ingenuity for many years, and at length are fairly well established in marine and other engines, where steam of very high pressure is used, especially for the high pressure cylinder of triple expansion engines.

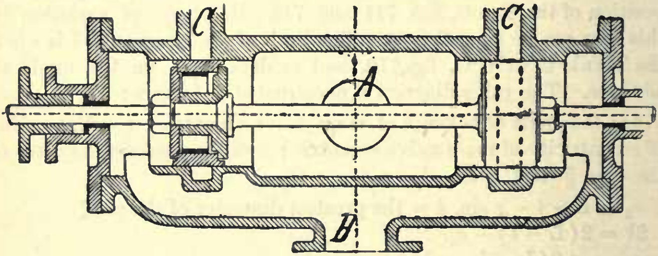


Fig. 714.

An example is shown in fig. 714 for a simple valve gear: C C are the steam ports, one at each end of the cylinder; B, the steam

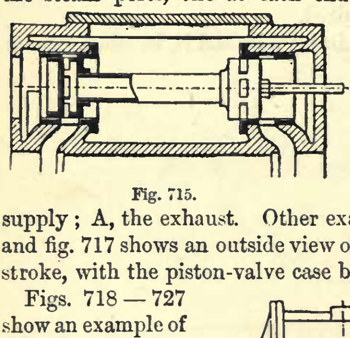


Fig. 715.

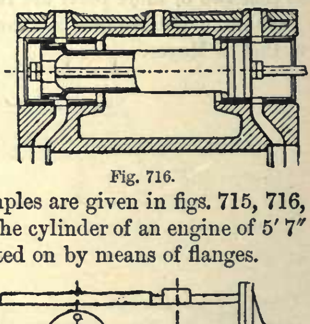


Fig. 716.

supply; A, the exhaust. Other examples are given in figs. 715, 716, and fig. 717 shows an outside view of the cylinder of an engine of 5′ 7″ stroke, with the piston-valve case bolted on by means of flanges.

Figs. 718 — 727 show an example of a piston valve for Rider's valve gear for an engine of 5′ 7″ stroke, 2′ 7″ diameter of piston, and 58 revs. per minute. The variation of cut-off is from ·0 to ·6, and the valve is turned through an angle of 36° by the governor.

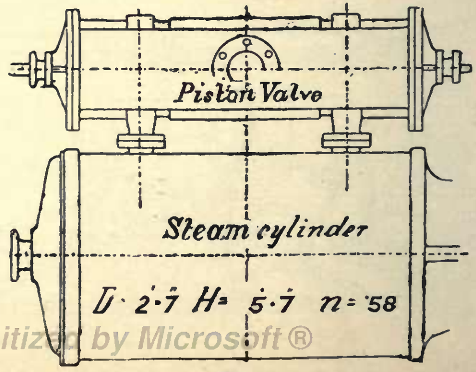
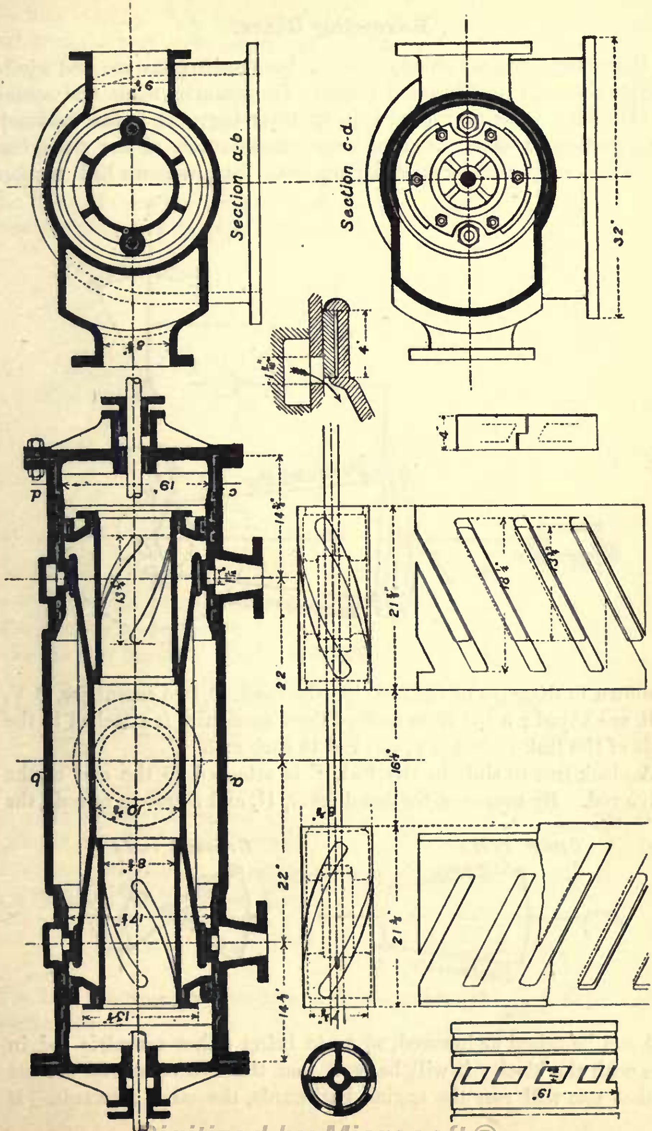


Fig. 717.



Figs. 718—727.

Reversing Gears.

Reversing gears are chiefly used on locomotives, marine and winding engines, also on traction engines for common roads and steam road-rollers. The prevailing type of reversing gear is the ordinary link motion of Stephenson or some modification of the same, but other types are used for special purposes. Stephenson's link motion

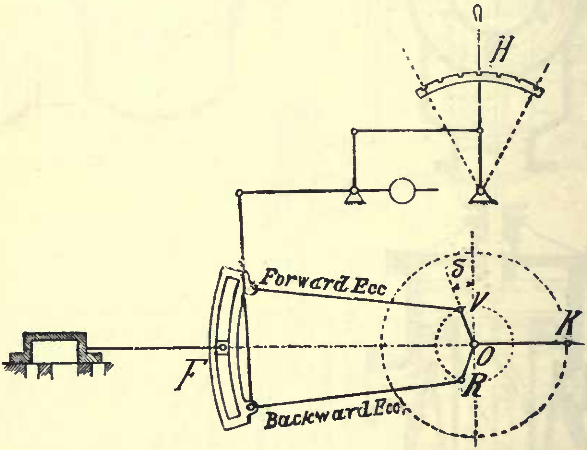


Fig. 728.

is shown in diagram in fig. 728; on the shaft, O, two eccentrics, O V, O R, are keyed; a rod from each of these eccentrics is attached to the ends of the link or sector F, one rod to each end.

A block free to slide in the link F is attached to the end of the valve rod. By means of the hand lever, H, and a system of rods, the

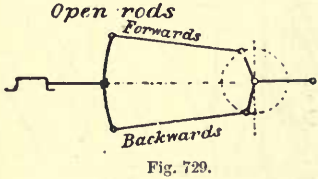


Fig. 729.

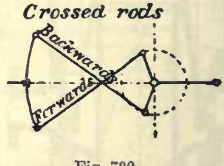


Fig. 730.

link can be raised or lowered, so as to bring either eccentric rod in line with the block; it will be seen that the two eccentrics are set so that one will run the engine backwards, the other forwards. If the link is in its highest or lowest position, the valve will receive the full stroke of one of the eccentrics ; if the link is in any other position, the stroke of the valve will be reduced, if the link is in a mean position, then the valve will receive the minimum amount of stroke. There are two ways of setting this valve gear, one, with open rods, fig. 729, the other, with crossed rods, fig. 730 ; the angle of advance, δ, is usually made the same for both forward and backward eccentric. The effect of open and crossed rods on the distribution of steam is shown in the valve diagrams, figs. 732, 733.

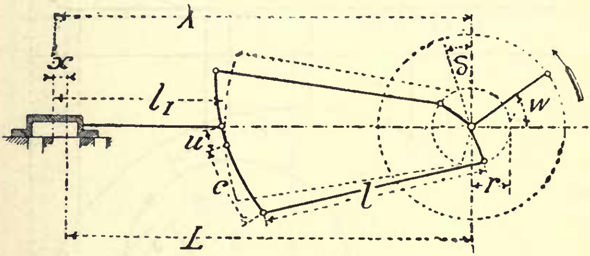


Fig. 731.

In fig. 731,

r = the radius or half stroke of both eccentrics.

δ = the angle of advance.

c = the half length of the link measured from the centre line to the point where the eccentric rod is attached.

u = the distance from the centre of the block to the centre of the link, u is positive when the link is down and negative when the link is up.

l = the length of the eccentric rod.

q = the radius of the link.

l_1 = the length of the valve rod.

for a chosen position of the crank corresponding to an angular movement W, gives for X_m (mean X)—

$$X_m = l + l_1 - \frac{r^2}{2\,l} \cos.^2 \delta + (c^2 - u^2)\,\frac{l - q}{2\,l\,q} = L.$$

The link should be curved to a radius equal to the length of the eccentric rod.

Stephenson's Link Motion with Open Rods.

($\frac{1}{2}$ natural size.)

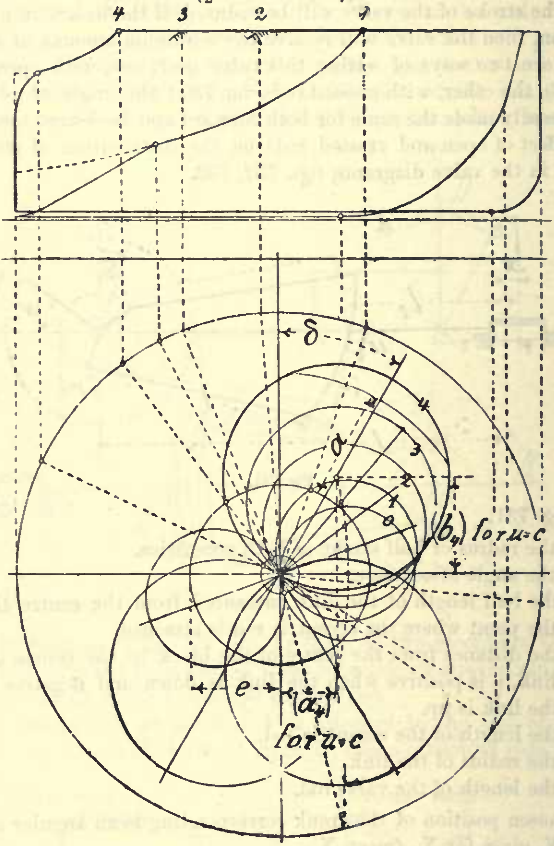


Fig. 732

(a = 1·18 ins. ; r = 2·36 ins. ; l = 55 ins. ; c = 5·9 ins. ; e = ·95 ins. ; i = ·27 ins.)

$$(a) = \frac{1}{2}\, r \left(\sin.\, \delta + \frac{c^2 - u^2}{c\, l} \cos.\, \delta\right) \qquad (b) = \frac{1}{2}\, r \frac{u}{c} \cos.\, \delta$$

For u max. = c, fig. 731 :—

Full gear forwards.	Mid gear.	Full gear backwards.
$(a_4) = \frac{1}{2}\, r \sin.\, \delta$	$(a_0) = \frac{1}{2}\, r \sin.\, \delta + \frac{1}{2}\, r \frac{c}{l} \cos.\, \delta$	$(a_4) = \frac{1}{2}\, r \sin.\, \delta$
$b_4) = \frac{1}{2}\, r \cos.\, \delta$	$(b_0) = 0$	$(b_4) = -\frac{1}{2}\, r \cos.\, \delta$

Stephenson's Link Motion with Crossed Rods.

($\frac{1}{2}$ natural size.)

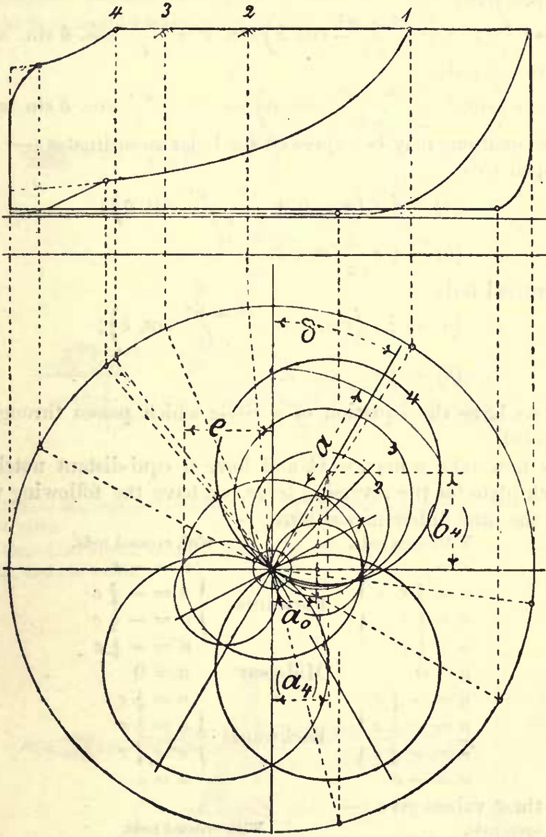


Fig. 733

(a = 1·18 ins.; R = 2·36 ins.; l = 55 ins.; c = 5·9 ins.; e = ·95 ins.; $\imath$ = ·27 ins.)

$$(a) = \tfrac{1}{2} r \left(\sin. \delta - \frac{c^2 - u^2}{c\, l} \cos. \delta\right) \qquad (b) = \tfrac{1}{2} r \frac{u}{c} \cos. \delta$$

For u max. $= c$, fig. 731 :—

Full gear forwards.	Mid gear.	Full gear backwards.
$(a_4) = \frac{1}{2} r \sin. \delta$	$(a_0) = \frac{1}{2} r \sin. \delta - \frac{1}{2} r \frac{c}{l} \cos. \delta$	$(a_4) = \frac{1}{2} r \sin. \delta$
$(b_4) = \frac{1}{2} r \cos. \delta$	$(b_0) = 0$	$(b_4) = -\frac{1}{2} r \cos. \delta$

For the distance, x, from the centre of the port face to the centre of the valve, fig. 731, we have the following approximate formulæ:—

For open rods,

$$x = r\left(\sin.\ \delta + \frac{c^2 - u^2}{c\ l}\cos.\ \delta\right)\cos.\ w + \frac{u\ r}{c}\cos.\ \delta\sin.\ w.$$

For crossed rods,

$$x = r\left(\sin.\ \delta - \frac{c^2 - u^2}{c\ l}\cos.\ \delta\right)\cos.\ w - \frac{u\ r}{c}\cos.\ \delta\sin.\ w.$$

These equations may be expressed for Polar co-ordinates:—

For open rods,

$$(a) = \tfrac{1}{2}\,r\left(\sin.\ \delta + \frac{c^2 - u^2}{c\ l}\cos.\ \delta\right);$$

$$(b) = \tfrac{1}{2}\,r\,\frac{u}{c}\cos.\ \delta.$$

For crossed rods,

$$(a) = \tfrac{1}{2}\,r\left(\sin.\ \delta - \frac{c^2 - u^2}{c\ l}\cos.\ \delta\right);$$

$$(b) = \tfrac{1}{2}\,r\,\frac{u}{c}\cos.\ \delta.$$

Thus we have the equation of a circle which passes through the centre point.

If we now take u max. $= C$, and have 9 equi-distant notches in the notch plate for the reversing lever, we have the following values of u for the nine different positions:—

With open rods.		With crossed rods.
$u = c$	Forwards	$u = -c$
$u = \frac{3}{4}c$	Forwards	$u = -\frac{3}{4}c$
$u = \frac{1}{2}c$	Forwards	$u = -\frac{1}{2}c$
$u = \frac{1}{4}c$	Forwards	$u = -\frac{1}{4}c$
$u = 0$	Mid-gear	$u = 0$
$u = -\frac{1}{4}c$	Backwards	$u = \frac{1}{4}c$
$u = -\frac{1}{2}c$	Backwards	$u = \frac{1}{2}c$
$u = -\frac{3}{4}c$	Backwards	$u = \frac{3}{4}c$
$u = -c$	Backwards	$u = c$

further these values give:—

With open rods.	With crossed rods.
Full gear forwards.	
$(a) = \frac{1}{2}\,r\sin.\ \delta,$	$(a) = \frac{1}{2}\,r\sin.\ \delta,$
$(b) = \frac{1}{2}\,r\cos.\ \delta.$	$(b) = \frac{1}{2}\,r\cos.\ \delta.$
Mid gear.	
$(a) = \frac{1}{2}\,r\sin.\ \delta + \frac{1}{2}\,r\frac{c}{l}\cos.\ \delta,$	$(a) = \frac{1}{2}\,r\sin.\ \delta - \frac{1}{2}\,r\frac{c}{l}\cos.\ \delta,$
$(b) = 0.$	$(b) = 0.$
Full gear backwards.	
$(a) = \frac{1}{2}\,r\sin.\ \delta,$	$(a) = \frac{1}{2}\,r\sin.\ \delta,$
$(b) = -\frac{1}{2}\,r\cos.\ \delta.$	$(b) = -\frac{1}{2}\,r\cos.\ \delta.$

From the diagram fig. 732, it will be seen that the port is only full open to steam when the link is in either extreme position, in any intermediate position the port is only partially opened; on this account it is usual to make the ports very wide.*

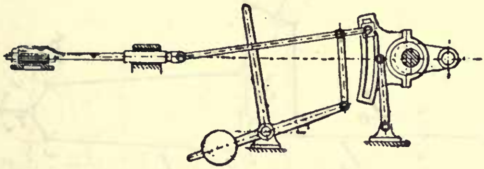


Fig. 734.

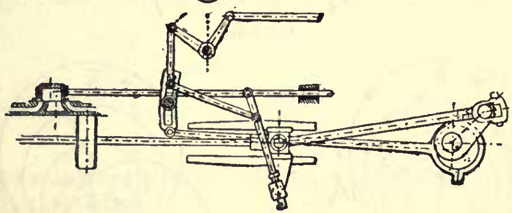


Fig. 735.

Fig. 734.—Link motion reversing gear, by Pius Fink, with one eccentric, one link, and one valve.

Fig. 735.—Link motion reversing gear, by Hensinger Von Waldegg, with one eccentric, one link, and one valve.

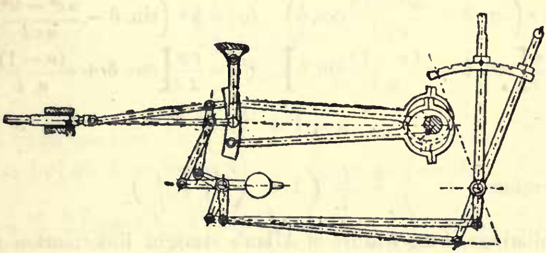


Fig. 736.—Link motion reversing gear by Polonceau, with two links and two valves.

* A full description and analysis of these link motions will be found in Zeuner's "Treatise on Valve Gears," translated by Moritz Müller. London: Spon.

Allan's Straight Link Motion.

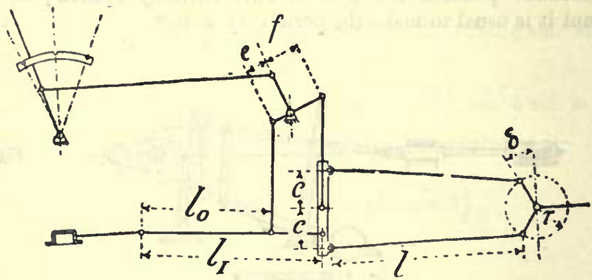


Fig. 737.

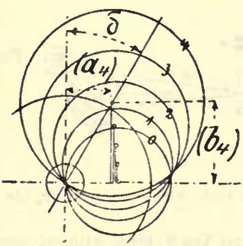


Fig. 738.

With open rods.

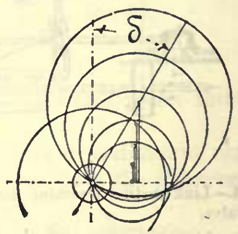


Fig. 739.

With crossed rods.

$$(a) = \tfrac{1}{2}\, r \left(\sin.\, \delta + \frac{nc^2 - u^2}{n c\, l} \cos.\, \delta\right) \qquad (a) = \tfrac{1}{2}\, r \left(\sin.\, \delta - \frac{n c^2 - u^2}{n c\, l} \cos.\, \delta\right)$$

$$(b) = \frac{r\, u}{2\, c}\left[\cos.\, \delta - c\frac{(n-1)}{n\, l} \sin.\, \delta\right] \qquad (b) = \frac{r\, u}{2\, c}\left[\cos.\, \delta + c\frac{(n-1)}{n\, l} \sin.\, \delta\right]$$

thus,
$$n = 1 + \frac{l_1}{l_0}\frac{e}{f}$$

and therefore,
$$\frac{e}{f} = \frac{l_0}{l}\left(1 + \sqrt{1 + \frac{l}{l_1}}\right).$$

The distinguishing feature of Allan's straight link motion reversing gear, fig. 737, is that the link is a straight bar, not a curved sector, and that link is raised and the block lowered simultaneously by a system of levers, the diagrams, fig. 738, 739, show that the distribution of steam is much the same as with the Stevenson's link motion, but the lead is more nearly constant for all positions of the link.

Gooch's Link Motion.

This link motion reversing gear, fig. 740, is sometimes called the "fixed link" motion, as the link is suspended from a fixed point, and is not moved up and down, the reversing action being obtained by moving the block up and down in the link.

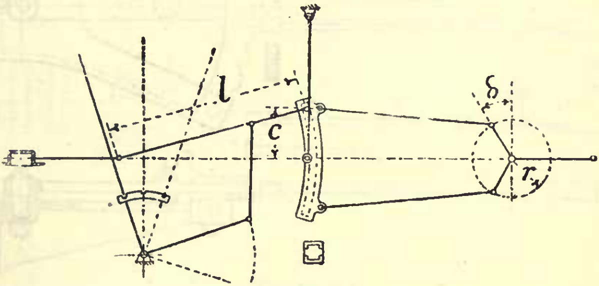


Fig. 740.

Crossed rods give the same distribution of steam, and are seldom used with this motion.

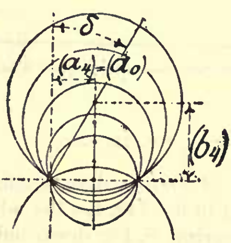


Fig. 741.

With open rods. With crossed rods.

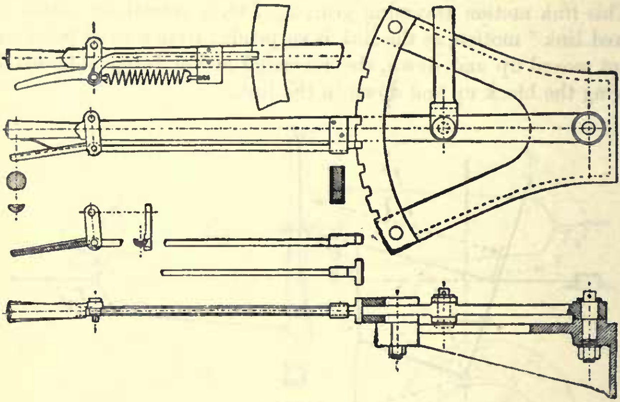
$$(a) = \tfrac{1}{2} r \left(\sin. \delta + \frac{c}{l} \cos. \delta\right), \quad (a) = \tfrac{1}{2} r \left(\sin. \delta - \frac{c}{l} \cos. \delta\right),$$

$$(b) = \frac{r}{2} \frac{u}{c} \left(\cos. \delta - \frac{c}{l} \sin. \delta\right). \quad (b) = \frac{r}{2} \frac{u}{c} \left(\cos. \delta + \frac{c}{l} \sin. \delta\right)$$

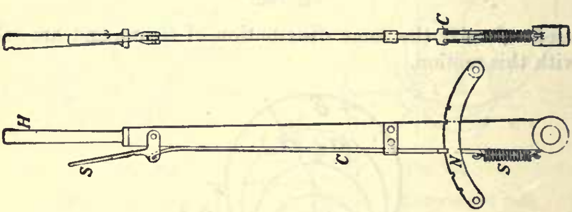
$u = +c$ Full gear forwards $u = -c$,
$u = 0$ Mid gear $u = 0$,
$u = -c$ Full gear backwards $u = +c$.

In the diagram, fig. 741, for open rods all the centres of the valve circles lie in a straight line, therefore the lead and cut-off are constant.

Reversing Gears.

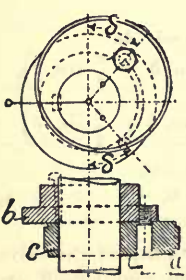


Figs 742--747



Figs 748, 749.

Reversing levers are of various designs: figs. 742 and 749 show two ordinary types, that in fig. 749, has the advantage of having the spring, S, low down out of the way and also the hand lever, H, is carried up well above the handle of the catch, S, so that both hands may be used on the lever.



Figs. 750, 751.

A simple arrangement for reversing the direction of running in small engines, such as agricultural engines, but one that requires the engine to be stopped to effect the reversal, is that of having a disc keyed on the shaft, *b*, fig. 751, and a bolt, *a*, to fix the eccentric to the disc; a slot in the eccentric allows it to be put over in position for forward or backward running.

One form of reversing gear which has been used on small marine engines consists of a sleeve, *a*, fig. 752, loose on the crank shaft, the eccentric, *b*, is fixed on the sleeve, *a*, and a stud in *c* engages in a thread cut in the shaft, so that when *c* is slid along on the shaft it causes the eccentric to turn round sufficient to reverse the engine.

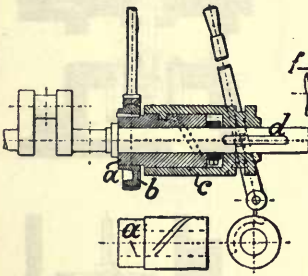


Fig. 752.

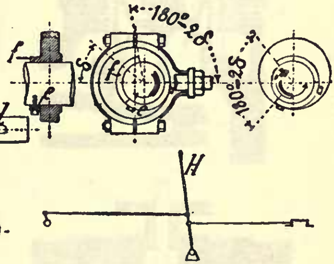


Fig. 753.

Another form of reversing gear with a loose eccentric, fig. 753, has been largely used in oscillating paddle engines. The eccentric is free to revolve on the shaft between limits formed by a stop, *e*, the eccentric is moved for reversing by means of the lever, H, the valve rod being disengaged from the eccentric rod first by a device known as a "gab" and catch ; it is usual to balance the weight of the eccentric by a plate or disc on the opposite side of the shaft, in order to lessen the labour of reversing.

Valve Gears with Mushroom and Double-beat Valves.

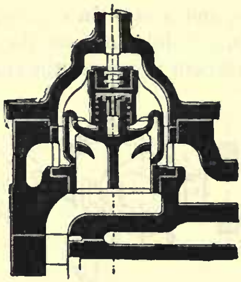
These valve gears would seem to be elaborations of the original Cornish double-beat valve gears, they are usually rather complicated, and are therefore not very largely used ; the valves of this kind are used in large winding engines, and are capable of being controlled either automatically or by hand. Sections of the valves and seats are given in figs. 754 to 761.

These valves were formerly made of gun metal, but now almost exclusively of cast iron ; the valve seating, also of cast iron, should be especially massive, to avoid the possibility of their getting out of truth. The seatings are frequently cast in one with the cylinder, and require care in the disposition of the metal, in order to secure sound castings. Some examples of the ordinary arrangements of these valves are shown by figs. 762—766 in outline.

Mushroom and Double-beat Valves.*

Schweiz Locomotive and Engine Works.

Fig. 754.



Socin and Wick, Basel.

Fig. 756.

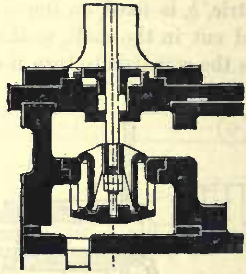
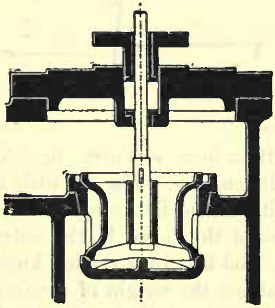
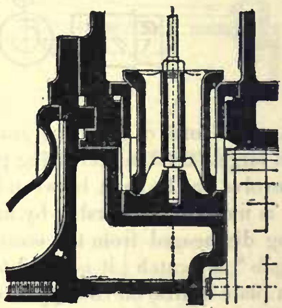


Fig 755



Société Anonyme de Marcinelle et Cowllet, Belgium.

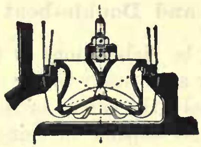
Fig. 757.



Société de l'Home, St. Julien.

Görlitzer Engine Works.

Fig 758.



C. Nolet, Ghent.

Fig. 760

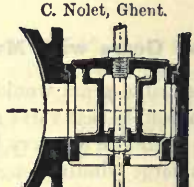
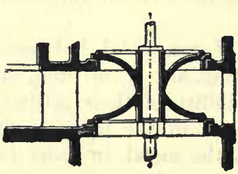
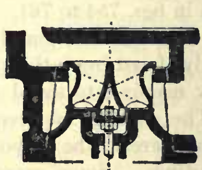


Fig. 759.



Pusey, Jones & Co., New York.

Fig. 761.



Görlitzer Engine Works.

* These sections are from Uhland's "Corliss Valve Engines."

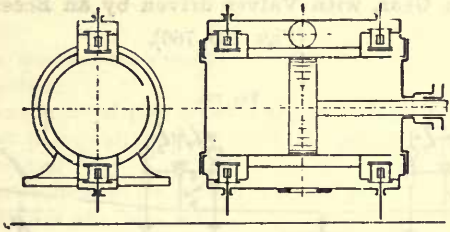


Fig. 762.

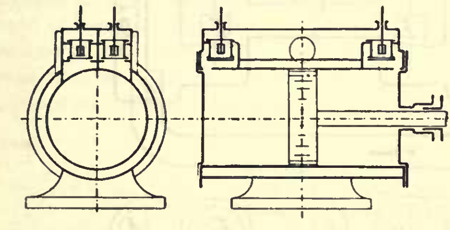


Fig. 763.

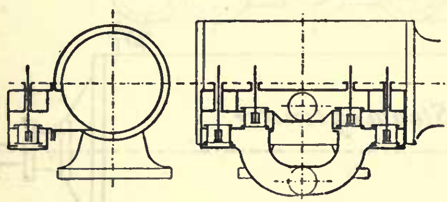


Fig. 764.

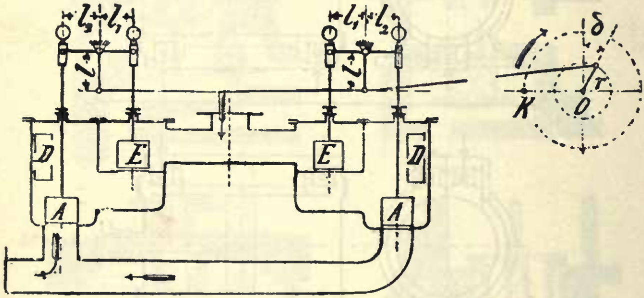
Fig. 762.—The two admission valves above, two exhaust valves below. A very common arrangement—for example, Görlitz, Sulzer, &c.

Fig. 763.—The two admission valves and the two exhaust valves on the top of the cylinder.

Fig. 764.—The case for the valves bolted on to the side of the cylinder.

Valve Gear, with Valves driven by an Eccentric (Figs. 765, 766).

Fig. 765.



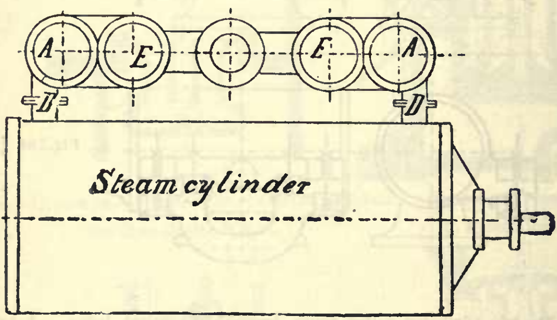


Fig. 766.

The above arrangement (figs. 765, 766) were much used in winding engines—

E E, the steam valves

D D, the steam ports.

The valve motion is obtained by means of an eccentric and bell crank levers. The valve spindles are fitted with weights on their upper ends, to ensure prompt closing. As this valve gear works exactly in the same manner as an ordinary slide valve gear, a link may be introduced, and the whole used as a reversing gear or for a variable cut-off.

In fig. 765,

h = the stroke of the valves.

v = the lead.

s = the amount of play between the end of the lever and the slot in the valve spindle when the eccentric is in the mean position.

d_1 = the diameter of the admission valve.

h_1 = stroke of admission valve.

l_1 = length of lever for admission valve.

l = length of lever, see fig. 765.

d_2 = diameter of the exhaust valve.

h_2 = stroke of exhaust valve.

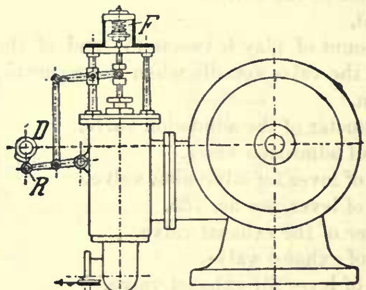
l_2 = length of lever for exhaust valve.

then we have—

$$\frac{l_2}{l_1} = \frac{h_2}{h_1}; \; r = \frac{l}{l_1}(h + s); \; r \sin. \delta = \frac{l}{l_1}(s + v); \; \sin. \delta = \frac{l}{l_1}\frac{(s + v)}{r}$$

Valve Gear with Cam Motion.

Fig 767.



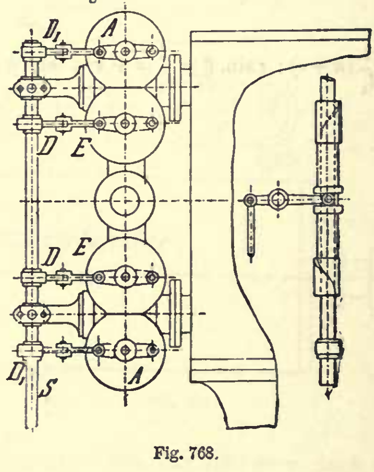


Fig. 768.

In figs. 767, 768,

A A, are the exhaust valves.
E E, the admission valves.
D_1 D_1, the cams for the exhaust valves.
D D, the cams for the admission valves.
S, the cam spindle.
R, the rollers at end of valve levers and resting on the cams.
F, the springs for closing the valves.

Construction of the Cams (Figs. 769—773).

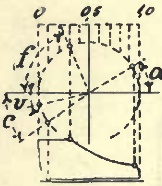


Fig 769.

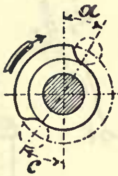


Fig. 770.

Fig. 771.

Fig. 772.

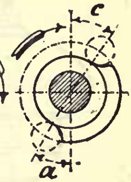


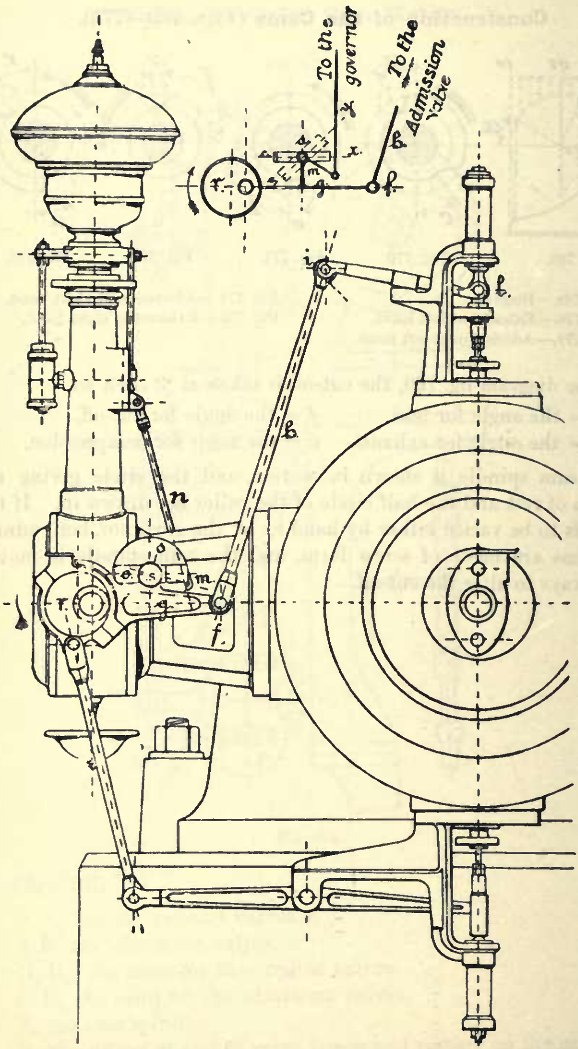
Fig. 773.

Fig. 769.—Diagram.
Fig. 770.—Exhaust at left hand.
Fig. 771.—Admission at left hand.
Fig. 772.—Admission at right hand.
Fig. 773.—Exhaust at right hand.

In the diagram fig. 769, the cut-off is taken at ·3 then let

v = the angle for lead. f = the angle for cut-off.
a = the angle for exhaust. c = the angle for compression.

The cam spindle is shown in section, and the circle giving the position of rest and the half circle of the roller are drawn in. If the cut-off is to be varied either by hand or by the governor, both admission cams are made of screw form, and the cam spindle is moved lengthways to alter the cut-off.



Figs. 774, 775.

Mushroom Valve Gear with Positive Motion to the Valves.

Hartung's Patent, made by the Harzer Co., Nordhausen, and the Buckau engine works, Magdeburg. Fig. 775.

The arrangement of this valve gear is as follows : an eccentric, *r*, with an elongated ring, *g*, is keyed on the valve shaft ; on the elongation of the eccentric ring is a pin, *s*, with a block which slides in the link, *c*, this link is moved as shown in dotted lines, figs. 774, 775, by means of the rod, *n*, in connection with the governor ; at the end of *g* is a pin, *f*, to which the rod, *b*, is attached ; the upper end of the rod, *b*, is attached to the lever, *i*, *h*, this lever being connected to the admission valve spindle, but resting only on the fulcrum bracket. In fig. 775, the engine is supposed to be at the dead point, and the valve open only to the extent of the lead, and the governor is at its lowest position corresponding to the latest cut-off, ·9 of the stroke ; the pin, *s*, is so arranged that the link may be moved without affecting the lead.

By the motion of the valve shaft when the engine is running, the eccentric moves in the direction of the arrow, and causes the end with the pin, *f*, to move up and down, thus opening and closing the valve by means of the rod, *b*, and lever, *i*, *h*. By this motion the pin, *s*, slides in the link, *c*, and if the governor rises, the link is turned from right to left, by means of the rod, *n*, and then the pin, *s*, is no longer guided in a horizontal but in an inclined direction, causing the path of the point, *f*, to be altered, and not to extend so far below the horizontal line during the opening of the valve ; this difference causes the valve to close sooner, and to give an earlier cut-off ; when the governor is at its highest position, the point, *f*, does not go below the horizontal line, and consequently there is no admission of steam.

Widnmann's Patent Mushroom Valve Gear with Positive Motion to the Valves.

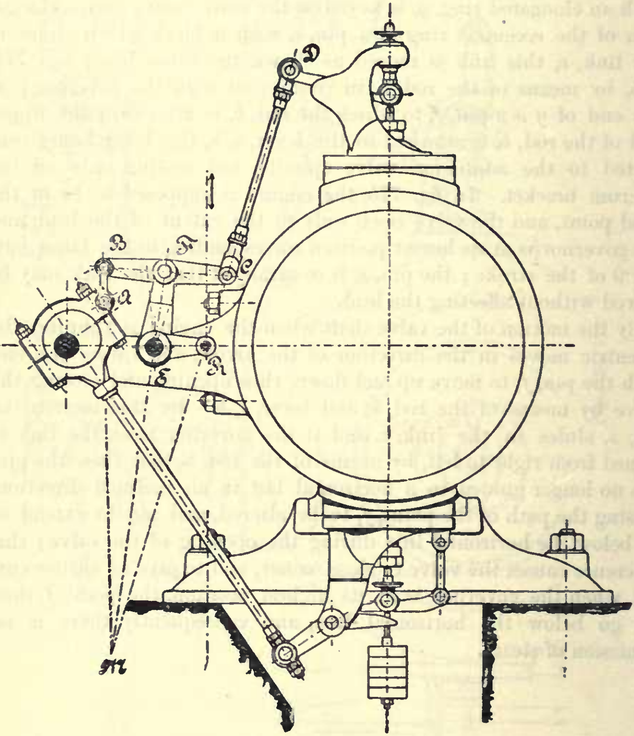


Fig. 770

Mushroom Valve Gear with Positive Motion to Valves.

H. Widnmann's Patent, Munich. Fig. 776.

The eccentric for giving motion to the valves in this gear is keyed on a side shaft, and has a long rod connected to a lever turning on a fixed point for working the exhaust valve; this renders any motion taken from the eccentric ring to be definite; a short pair of links, A B, fig. 776, are jointed to the upper side of the eccentric ring, and to the lever, B F C, and the rod, C D, gives motion to the admission valve through a rod and lever. The lever arm, E F, is keyed on a spindle, and this spindle is turned by the governor. By turning the spindle, E, a varying inclination is given to the arm, E F, and therefore also to the links, A B, and by this means the period during which the valve is open is altered, and an earlier or later cut-off obtained.

In order that the lead may be as nearly constant for all variations in the cut-off; the centre of the spindle, E, should be the centre of a circle, which approximately represents the path of the point, F.

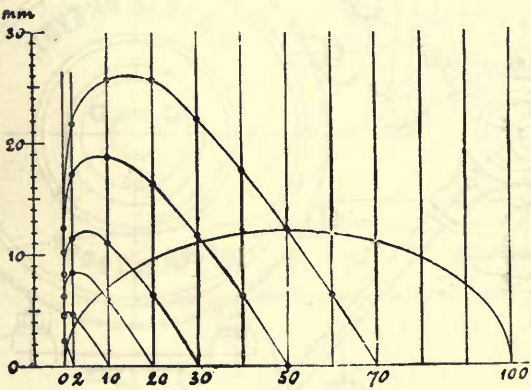


Fig. 777

O. Recke's Patent Valve Gear.

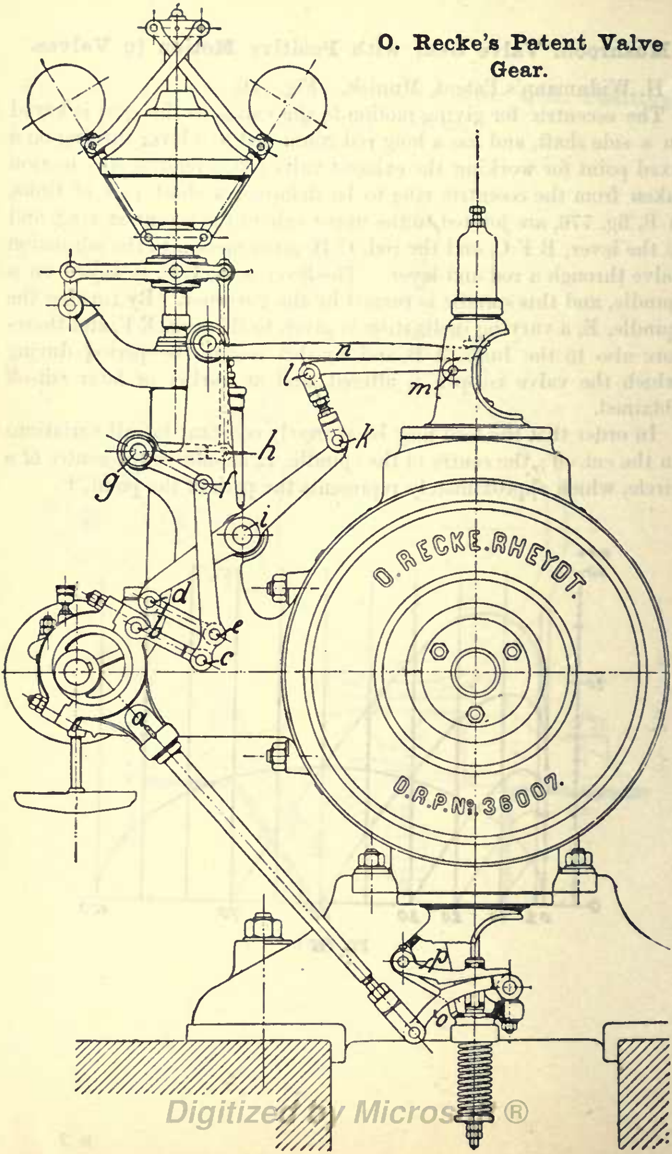


FIG. 778.

Mushroom Valve Gear with Positive Motion to the Valves.

The eccentric, *a*, fig. 778, O. Recke's Patent, moves the exhaust valve by means of the roller levers, *o* and *p*, and the admission valve through the pin, *b*.

The governor lever, *g h*, turns on the pin, *g*, which also carries the lever, *g f*, the point *f* forms a turning point for the lever, *f e c*, connected to eccentric ring at *b*, by the rod *b c*, and also connected by *d e* to the lever *d i k*; at *k* this lever is connected with the upper lever, *l m n*, which moves the admission valve; *g f*, *d e*, and *b c*, are of equal length, and in the opening position of the valve parallel to one another.

This valve gear gives a very nearly constant lead, and a sufficient opening of the valves for the normal cut-off without excessive opening for the early cut-off, and very little back pressure on the governor.

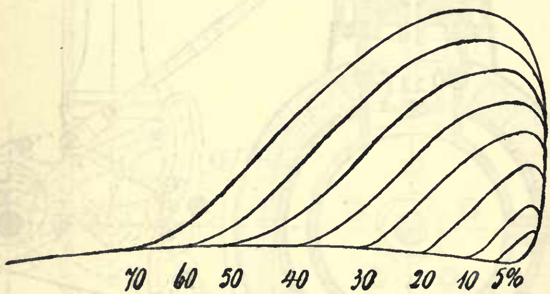
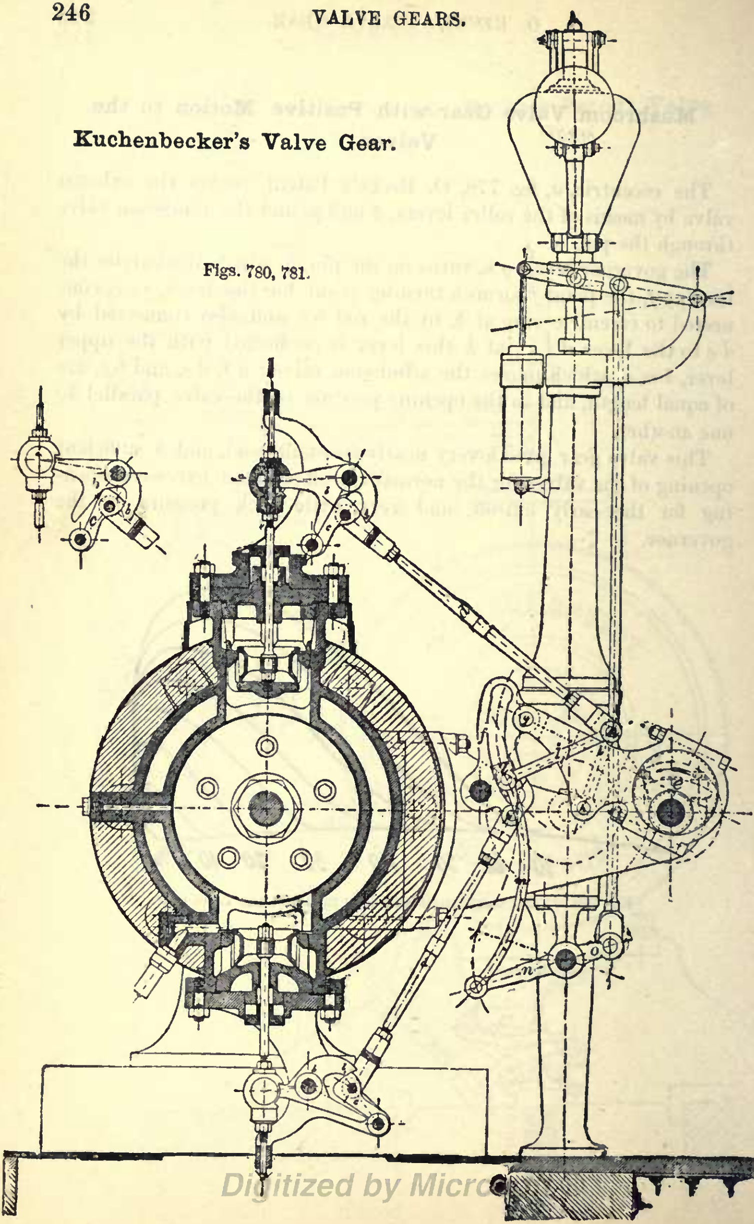


Fig 779.—Valve diagram, drawn by a model of the valve gear.

Kuchenbecker's Valve Gear.

Figs. 780, 781.



Kuchenbecker's Patent Mushroom Valve Gear with Positive Motion to the Valves.

The shaft, *e*, fig. 780, is turned in the direction of the arrow by gearing from the crank shaft, and carries an eccentric which gives a vibrating motion to the link, *f*. The sliding block, *l*, in this link, by means of the rods, i_1, *i*, and *d*, gives the necessary motion to the admission valve.

The double lever, *b*, b_1, communicates the motion by means of the arm, *c*, which is connected to the rod, *d*; the arm, *c*, acting against *b*, opens the valve, and acting against b_1, closes the valve; fig. 780 shows the position of the levers with the valve open, fig 781 with the valve closed. The rod, *d*, is connected to the link at *p*, for opening and closing the exhaust valve.

The governor alters the position of the block in the link, *f*, by means of the lever, *n o*, and the rod, *m*, and varies the cut-off.

The movement of the valves is very quick, and is positive, without the aid of dash pots or springs for either opening or closing.

Proell's Patent Mushroom Valve Gear with Positive Motion to the Valves.

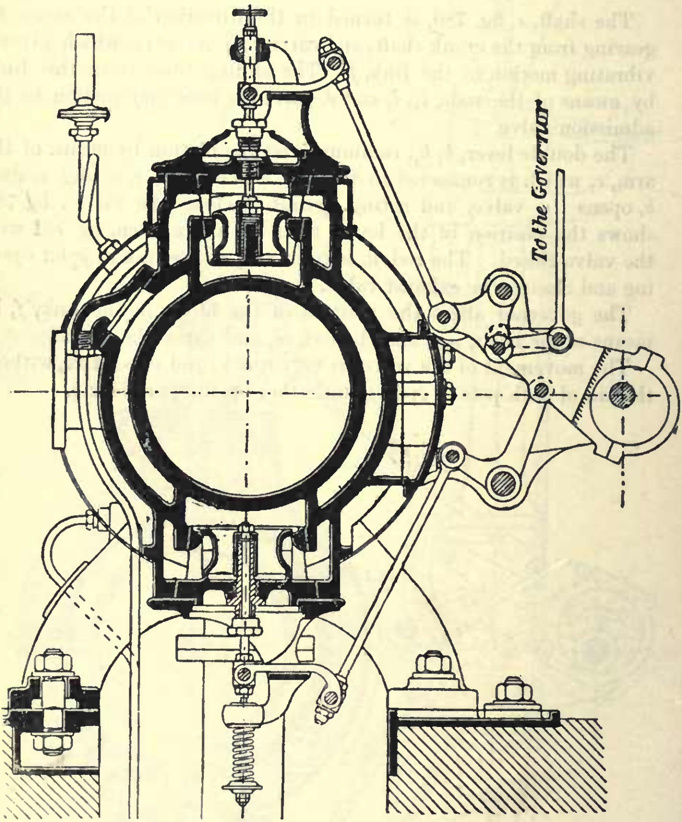


Fig. 782.

This gear consists essentially of valves driven by an eccentric ; a point in the eccentric ring moves in a closed curve, and by means of an adjustable lever connected to the governor the cut-off is varied ; by this peculiar motion the valves move with varying speed to and from their seatings, and open and close quietly. The exhaust valve is by levers attached to the eccentric ring, as shown in fig. 782

E. König's Mushroom Valve Gear, with Positive Motion to the Valves.*

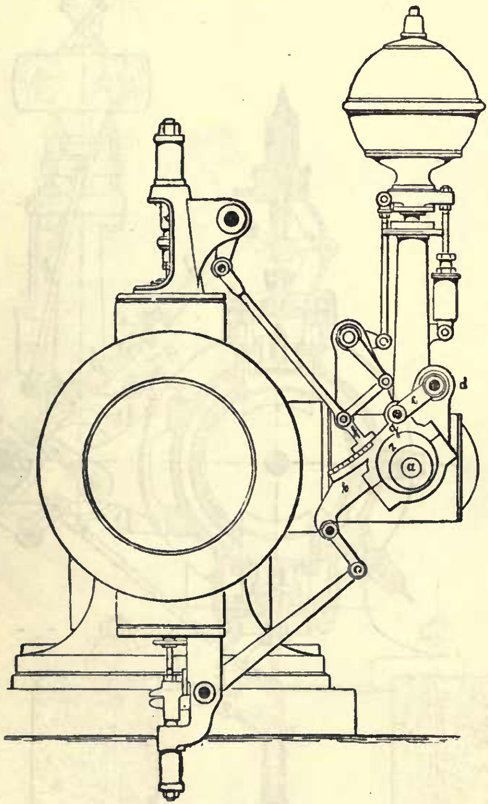


Fig. 783.

The eccentric ring in fig. 783 is made with a projection, *b*, and is jointed at *g* to an arm, *c*, which swings on a fixed point, *d*; the admission valve is moved by the rod, *f*, resting against the eccentric ring, *b*, the position of *f* is altered by the governor, and varies the cut-off

* Dingler's "Polytechnic Journal," No. 3, 1888.

Gamerith's Patent Valve Gear, with Trip Arrangement by Stark & Hoffman, of Hirschberg.

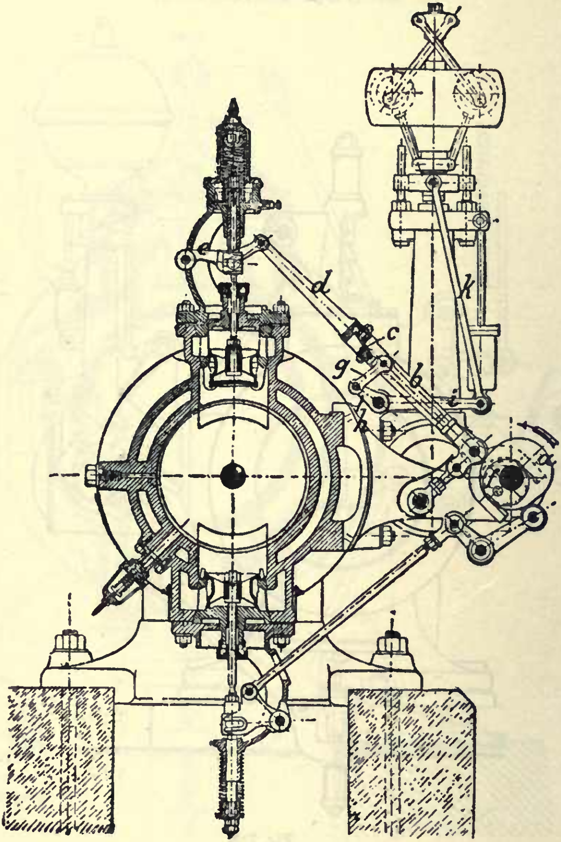


Fig. 784.

The valves in fig. 784 are worked by the cam, *a*, the admission valve is lifted by means of the catch, *c*, this slides away and allows the valve to fall at different positions determined by the governor, which raises the piece on which the catch acts by means of the rods *g*, *i* and *k*, and thus alters the cut-off. The exhaust valve is worked by the same cam, *a*.

Sulzer's Valve Gear, with Trip Arrangement (Sulzer Bros., Winterthur).

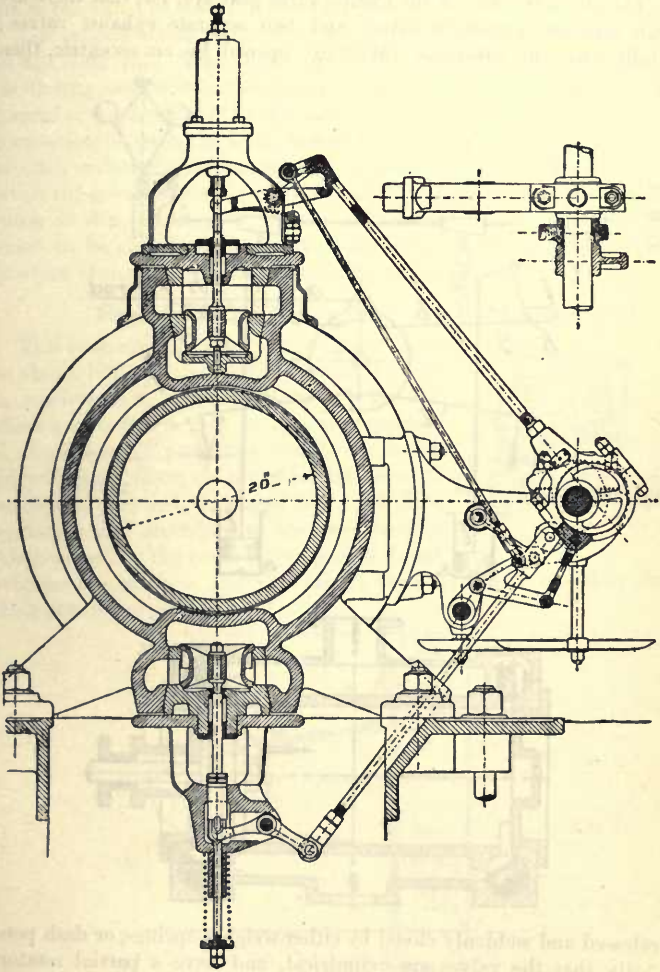
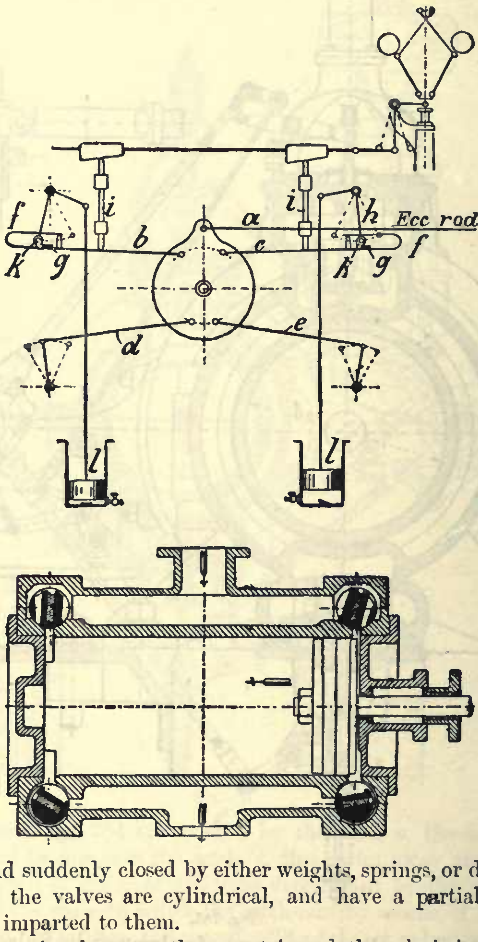


Fig. 785.

Fig. 785 shows the arrangement of valve gear by Sulzer, in which the valves are opened by an eccentric, and provided with trip gear by which they are allowed to close by means of a spring.

Corliss Valve Gear.

The characteristics of the Corliss valve gear are, 1st, that there are two separate admission valves and two separate exhaust valves; 2ndly, that the admission valves are opened by an eccentric, then



Figs. 786, 787.

released and suddenly closed by either weights, springs, or dash pots; 3rdly, that the valves are cylindrical, and have a partial rotatory movement imparted to them.

The connection between the eccentric and the admission valve is generally in two parts, one being connected with the valve, the other with the wrist plate, which receives its motion from the eccentric rod ; during the opening of the valves the two parts are coupled, but to allow the valve to close the two parts are suddenly uncoupled.

The general action of the Corliss valve gear is shown in diagram in figs. 786, 787. The centre disc, or "wrist plate," receives an oscillating motion from the eccentric rod, *a*, the two rods, *b* and *c*, are formed into springs at *f*, *f*. The lever, *h*, is connected to the valve, the connection between the lever, *h*, and the rod, *c*, is by means of a kind of catch or detent arrangement, *k*, *g*, the governor acts by the rods, *i*, *i* ; when the governor rises, the rods, *i*, *i*, are pushed down, and cause the catch to slip, leaving the lever, *h*, disconnected, and allowing the valve to be closed suddenly by dash pots, *l*, or by weights. The position of the governor determines the time of release.

Harris' Corliss Valve Gear (Figs. 788, 789).

This is an arrangement very much the same as above, but modified in the details, the rod, *a*, receives motion from the wrist plate, the short arm is free to turn on the valve spindle, *d*, at *g* is a small projection against which the curved arm, *f*, rests ; the backward and forward motion of the rod, *a*, causes the *f* to slide against *g*, and according to the position of *g* (determined by the governor) depresses *f* and releases the catch, *e*, and thus allows the valve to be closed by the dash pot or weights.

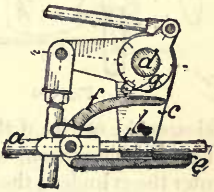


Fig. 788.

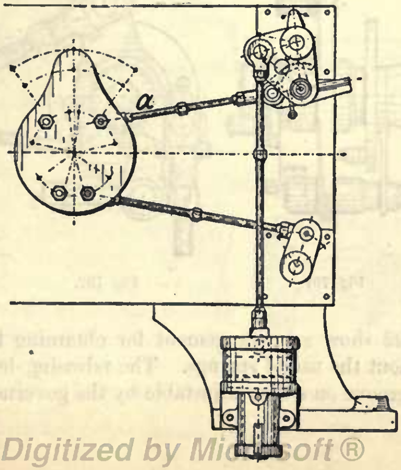


Fig. 789.

Reynold's Corliss Valve Gear (Fig. 790).

The arrangement is the same with respect to the general features as the original Corliss gear, the wrist plate receiving its motion from an eccentric. The lever, *c*, fig. 790, is loose on the valve spindle, *b*, and, together with *d*, is moved by the rod, *a*; from the wrist plate on *d* is a piece of steel, *e*, and on *e* the block *f* rests, this block is on lever *g*, latter being fast on the valve spindle *b*. The spring, *h*, serves to keep the forked lever, *d*, pressed up against *f*. Again, on the valve spindle, *b*, is the governor lever with the projection, I, against which by the motion the bell crank lever presses and thus releases the catch; if the governor ceases to act by the strap breaking, then the projection, *k*, releases the valve, and no more steam can enter the cylinder; the usual dash pot, or weight, for closing the valve is attached by the rod, *i*.

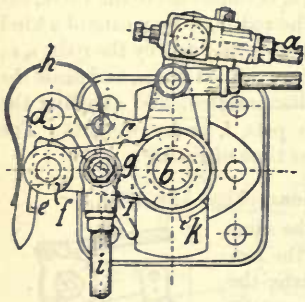


Fig. 790

Fig 791.

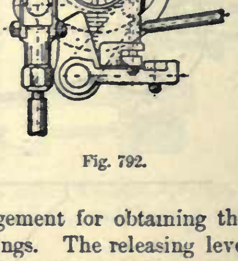


Fig. 792.

Figs. 791, 792 show an arrangement for obtaining the release of the valve without the use of springs. The releasing lever has a pin working in a groove on a disc adjustable by the governor.

Wheelock Corliss Valve Gear (Figs. 793, 794).

In this arrangement all the four ports are brought to the ends of the cylinder, the exhaust valves are driven direct from the wrist

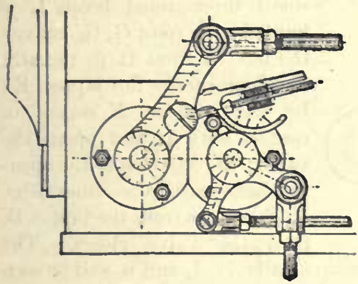


Fig. 793.

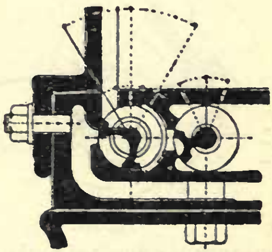


Fig. 794.

plate and are hollow, fig. 794. The admission valves act as expansion valves.

Corliss Valve Gear, by J. R. Frikart, Paris * (Fig. 795).

The exhaust valves are worked by the eccentric and levers as in fig. 795, and a five-armed lever takes the place of the wrist plate

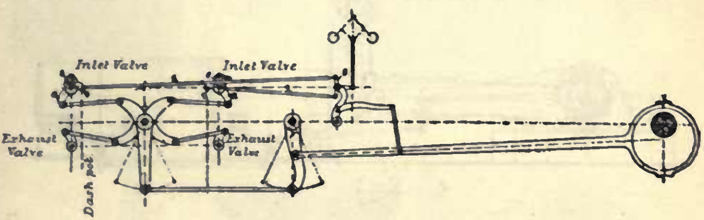


Fig. 795.

in other types of Corliss gear The upper arms of this lever work on the double arms, A, A, these are free to turn on the bush which forms the bearing of the valve spindle, C, fig. 796. On C the tappet lever, D, D, is firmly fixed ; this lever, D, D, has on one side a plate of hardened steel, and on the other a joint and rod connected to the dash pot piston for closing the valve.

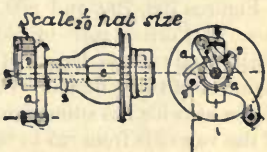


Fig 796.

* "Zeitschrift der Verein deutsch. Ingenieur," 1890, page 917.

Another tappet, E, is fixed to the short arm, F, fig. 796, which is free to turn on a pin fixed to A; a small rod connects the angle lever, H, with the eccentric rod, and at the upper end of H a small three-armed lever, I, is jointed; the rods, G, G, convey the motion from H to the arm F, which carries the tappet, E; by this motion E comes in contact with D, and opens the valve; the duration of the opening lasts until the inner edge of E escapes from the tappet D, then the valve closes. The governor is connected to the three-armed lever, I, and it will be seen from fig. 797 that the release will take place earlier or later according to the position of the lever, F, and with it the tappet, E, the alteration of position for the variation of cut-off being effected by the governor. The curves in chain dotted lines show the movement for the earliest and latest cut-off.

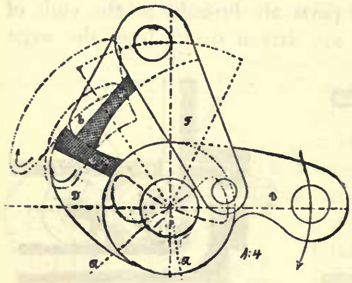


Fig. 797.

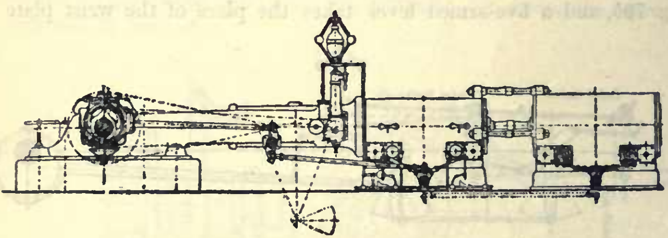


Fig. 798.—Corliss Engine by Thomas Powell, Rouen.*

Figures 798, 799, and 800, give examples of the general arrangement of Corliss gear by American and French engineers. With Corliss valves the length of the ports is very nearly equal to the diameter of the cylinder. The area of the ports is about ·07 of the piston area for the admission, and ·10 for the exhaust. The diameter of the valves is from ·25 to ·32 of the cylinder diameter.

* "Zeitschrift der Verein deutsch. Ingenieur," 1890 page 924

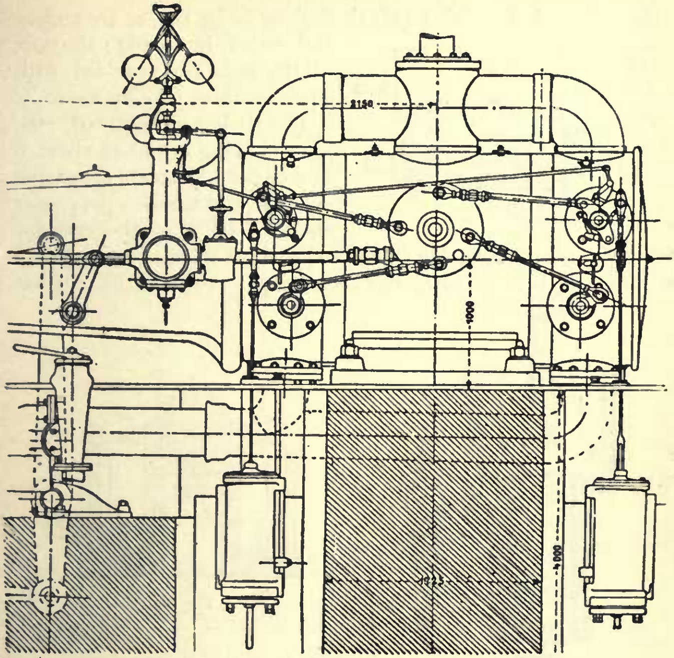


Fig. 799.—Corliss Engine by J. Farcot, St. Ouen.*

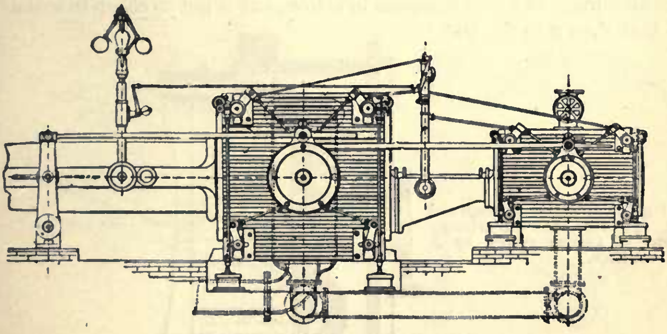
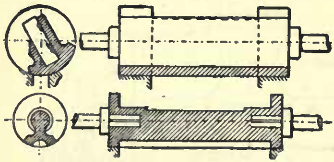


Fig. 800.--American Corliss Engine.†

* "Zeitschrift der Verein deutsch. Ingenieur," 1890, page 924.
† *Engineering*, 1891, page 750.

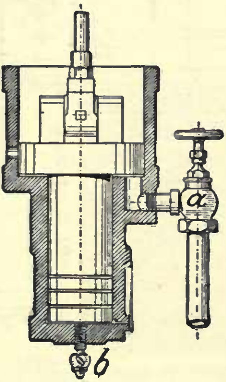
The valve spindles either pass through as in fig. 801, or are merely gudgeons as in fig. 802; the ends of the rods are provided with adjustments to take up wear.



Figs. 801, 802.

The dash pots, figs. 803, 804, have working cylinders about ·6 to ·5 of the diameter of the engine cylinder, and the dash pot proper about ·4 to ·3 of the cylinder diameter. Small adjustable valves are provided at *a* and *b*, to regulate the action of both pistons; the arrangement shown in fig. 804 has

Fig. 803.



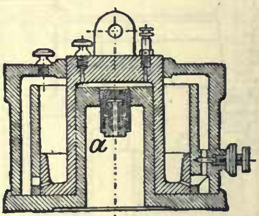


Fig. 804.

the advantage of being noiseless in action, but is not so cheap to make as that shown in fig. 803.

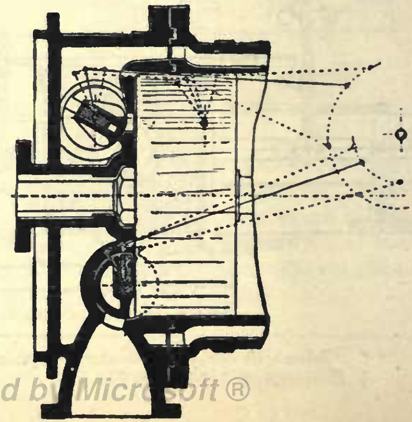


Fig. 805.—Section through end of cylinder showing arrangement of Corliss valves, by J. Farcot Bros.

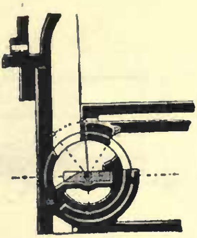


Fig. 806.—Section of Corliss valve, by Douglas & Grant, Kirkcaldy.

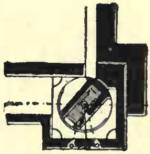
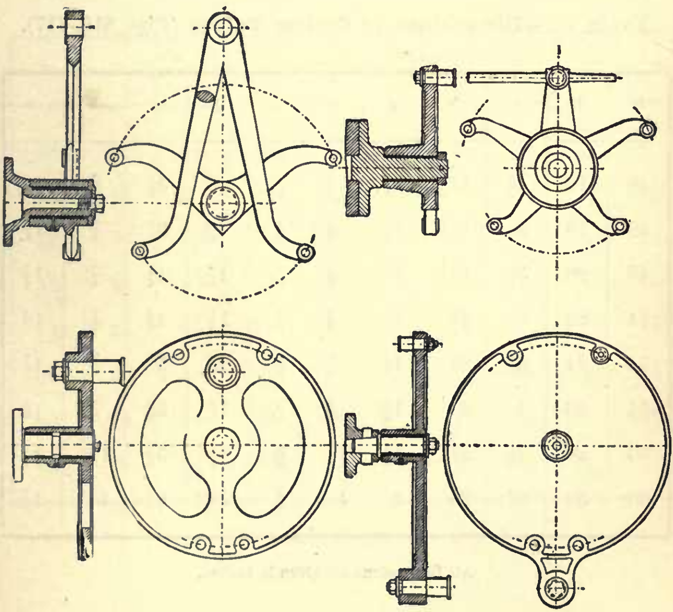
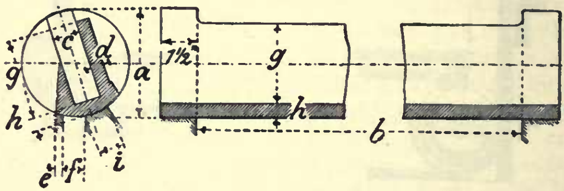


Fig. 807.—Section of Corliss valve by the Sangerhauser Engine Works.



Figs. 808—815.—Various designs for wrist-plates and multiple-armed levers for Corliss gear

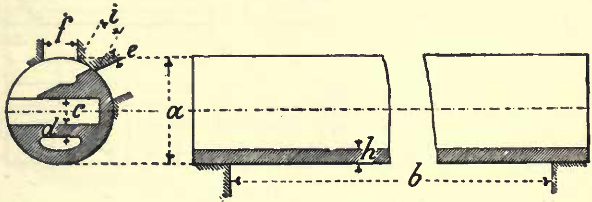


Figs. 816, 817.—Sections of Corliss Valves.

TABLE 69.—**Dimensions of Corliss Valves** (Figs. 816, 817).

H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*
48	16	$4\frac{1}{2}$	$14\frac{1}{2}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{5}{16}$	$\frac{7}{8}$	$3\frac{1}{4}$	$\frac{3}{4}$	1
48	18	5	$16\frac{1}{2}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{15}{16}$	$3\frac{7}{8}$	$\frac{3}{4}$	$1\frac{1}{8}$
48	20	$5\frac{1}{2}$	$18\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{7}{16}$	$1\frac{1}{16}$	$4\frac{1}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$
54	22	6	21	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{16}$	$1\frac{1}{8}$	$4\frac{5}{8}$	$\frac{7}{8}$	$1\frac{3}{8}$
54	24	$6\frac{1}{2}$	23	$1\frac{5}{8}$	$\frac{7}{8}$	$\frac{1}{2}$	$1\frac{3}{16}$	5	$\frac{7}{8}$	$1\frac{1}{2}$
54	26	7	25	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{9}{16}$	$1\frac{3}{8}$	$5\frac{3}{8}$	$\frac{7}{8}$	$1\frac{5}{8}$
60	28	$7\frac{1}{2}$	27	$1\frac{7}{8}$	1	$\frac{5}{8}$	$1\frac{1}{2}$	$5\frac{3}{4}$	1	$1\frac{3}{4}$
60	30	$8\frac{1}{2}$	29	2	1	$\frac{3}{4}$	$1\frac{15}{16}$	$6\frac{3}{8}$	1	$1\frac{7}{8}$

All Dimensions are given in Inches.

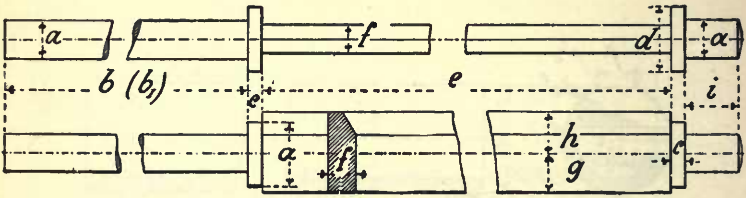


Figs. 818, 819.—Sections of Corliss Valves.

TABLE 70.—**Dimensions of Corliss Valves** (Figs. 818, 819).

H	D	a	b	c	d	e	f	h	i
48	16	$4\frac{1}{2}$	$14\frac{1}{2}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{1}{8}$	$1\frac{7}{16}$	$\frac{3}{4}$	1
48	18	5	$16\frac{1}{2}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{3}{16}$	$1\frac{9}{16}$	$\frac{3}{4}$	$1\frac{1}{8}$
48	20	$5\frac{1}{2}$	$18\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{3}{16}$	$1\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$
54	22	6	21	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{16}$	$1\frac{7}{8}$	$\frac{7}{8}$	$1\frac{3}{8}$
54	24	$6\frac{1}{2}$	23	$1\frac{5}{8}$	$\frac{7}{8}$	$\frac{1}{4}$	2	$\frac{7}{8}$	$1\frac{1}{2}$
54	26	7	25	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{4}$	$2\frac{1}{8}$	$\frac{7}{8}$	$1\frac{5}{8}$
60	28	$7\frac{1}{2}$	27	$1\frac{7}{8}$	1	$\frac{1}{4}$	$2\frac{5}{16}$	1	$1\frac{3}{4}$
60	30	$8\frac{1}{2}$	29	2	1	$\frac{5}{16}$	$2\frac{1}{2}$	1	$1\frac{7}{8}$

All Dimensions are given in Inches.



Figs. 820, 821.—Corliss Valve Spindles.

TABLE 71.—**Dimensions of Corliss Valve Spindles** (Figs. 820, 821).

H	D	*a*	*b*	b_1	*c*	*d*	*e*	*f*	*g*	*h*	*i*
48	16	1½	15⅞	11⅜	½	2⅝	17½	1⅛	1⅜	1⅞	2¼
48	18	1⅝	17⅛	12⅛	⅝	2¾	19½	1¼	1⅝	2¼	2½
48	20	1¾	18¾	13¼	⅝	2⅞	21½	1⅜	1⅞	2⅜	2¾
54	22	1⅞	19¾	14	¾	3	24	1½	2	2⅝	3
54	24	2	20¾	14¾	¾	3¼	26	1⅝	2¼	2¾	3¼
54	26	2⅛	21¾	15½	⅞	3½	28	1¾	2½	2⅞	3½
60	28	2¼	22¾	16¼	⅞	3⅝	30	1⅞	2⅝	3⅛	3⅞
60	30	2⅜	24	17	1	3¾	32	2	3⅛	3¼	4¼

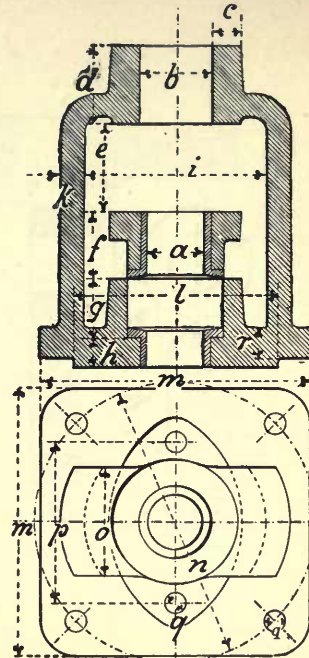
All Dimensions are given in Inches.

b for admission valves.
b_1 for exhaust valves.

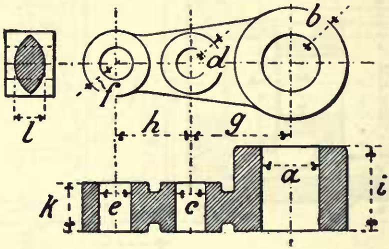
TABLE 72.—**Dimensions of Glands and Brackets for Corliss Valve Spindles.**

H	D	a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	r
48	16	1½	1⅞	13/16	1⅞	3½	1½	1½	¾	5½	9/16	4⅞	7	7⅛	2⅞	4½	⅝	¾
48	18	1⅝	2	⅞	2	3½	1¾	1⅝	¾	6	⅝	5⅜	7½	7⅝	3	5¼	⅝	13/16
48	20	1¾	2¼	15/16	2¼	3⅝	1⅞	1⅞	⅞	6½	11/16	5⅞	8	8¼	3¼	5⅜	¾	15/16
54	22	1⅞	2⅜	1	2½	3⅝	2	2	1	6¾	¾	6⅜	8½	8⅞	3½	5½	¾	1
54	24	2	2⅝	1 1/16	2¾	3⅝	2¼	2⅛	1	7	13/16	6⅞	9¼	9⅝	3¾	5¾	¾	1 1/16
54	26	2⅛	2¾	1⅛	2⅞	3¾	2⅜	2¼	1⅛	7¼	⅞	7⅜	10	10¼	4	6	⅞	1⅛
60	28	2¼	3	1 3/16	3	3¾	2⅝	2⅜	1¼	7½	15/16	7⅞	10⅜	11	4¼	6¼	⅞	1¼
60	30	2⅜	3⅛	1¼	3¼	3¾	2¾	2½	1¼	7¾	1	8⅞	11¼	11¾	4½	6½	⅞	1 5/16

All Dimensions are given in Inches.



Figs. 822, 823. – Brackets and Glands for Corliss Valve Spindles.

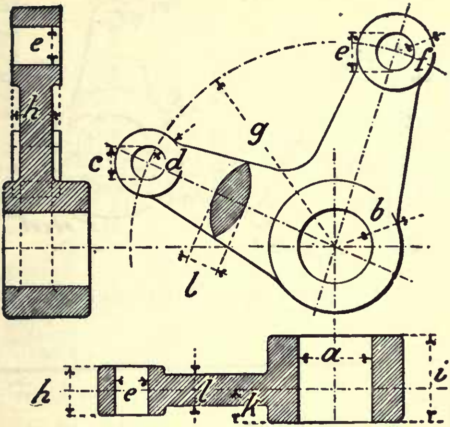


Figs. 824—826.—Lever Arms for Corliss Valves.

TABLE 73.—**Dimensions of Lever Arms for Corliss Valves** (Figs. 824—826).

H	D	a	b	c	d	e	f	g	h	i	k	l
48	16	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{15}{16}$	$\frac{7}{16}$	$2\frac{3}{4}$	2	$2\frac{1}{4}$	$1\frac{1}{4}$	$\frac{13}{16}$
48	18	$1\frac{5}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{16}$	1	$\frac{1}{2}$	3	$2\frac{1}{8}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$\frac{7}{8}$
48	20	$1\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{1}{2}$	$1\frac{1}{16}$	$\frac{9}{16}$	$3\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$\frac{15}{16}$
54	22	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{9}{16}$	$3\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{7}{8}$	$1\frac{5}{8}$	1
54	24	2	$\frac{15}{16}$	$\frac{16}{16}$	$\frac{9}{16}$	$1\frac{3}{16}$	$\frac{5}{8}$	$3\frac{1}{2}$	$2\frac{5}{8}$	3	$1\frac{3}{4}$	$1\frac{1}{16}$
54	26	$2\frac{1}{8}$	1	1	$\frac{9}{16}$	$1\frac{1}{4}$	$\frac{5}{8}$	$3\frac{3}{4}$	$2\frac{3}{4}$	$3\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{1}{8}$
60	28	$2\frac{1}{4}$	$1\frac{1}{16}$	$1\frac{1}{16}$	$\frac{5}{8}$	$1\frac{5}{16}$	$\frac{11}{16}$	4	3	$3\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{3}{16}$
60	30	$2\frac{3}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$\frac{5}{8}$	$1\frac{3}{8}$	$\frac{11}{16}$	$4\frac{1}{8}$	$3\frac{1}{8}$	$3\frac{1}{2}$	2	$1\frac{1}{4}$

All Dimensions are given in Inches.

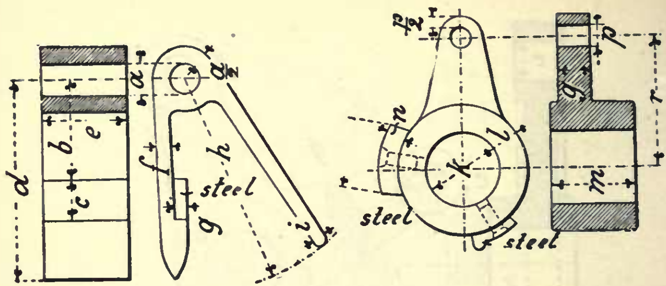


Figs. 827—829. Levers for Corliss Valves.

Table 74.—Dimensions of Levers for Corliss Valves (Figs. 827—829).

H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*
48	16	$1\frac{7}{8}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{7}{16}$	$\frac{15}{16}$	$\frac{1}{2}$	$5\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$\frac{3}{4}$	$\frac{13}{16}$
48	18	2	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{16}$	1	$\frac{1}{2}$	$5\frac{1}{2}$	$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{7}{8}$	$\frac{7}{8}$
48	20	$2\frac{1}{4}$	$\frac{15}{16}$	$\frac{15}{16}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{9}{16}$	6	$1\frac{3}{8}$	$2\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$
54	22	$2\frac{3}{8}$	1	1	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{9}{16}$	$6\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{7}{8}$	1	1
54	24	$2\frac{5}{8}$	$1\frac{1}{16}$	1	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{5}{8}$	7	$1\frac{5}{8}$	3	$1\frac{1}{8}$	$1\frac{1}{16}$
54	26	$2\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$\frac{9}{16}$	$1\frac{5}{16}$	$\frac{5}{8}$	$7\frac{1}{2}$	$1\frac{3}{4}$	$3\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$
60	28	3	$1\frac{3}{16}$	$1\frac{3}{16}$	$\frac{5}{8}$	$1\frac{3}{8}$	$\frac{11}{16}$	8	$1\frac{7}{8}$	$3\frac{1}{4}$	$1\frac{1}{4}$	$\frac{13}{16}$
60	30	$3\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	$8\frac{1}{2}$	2	$3\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$

All Dimensions are given in Inches.

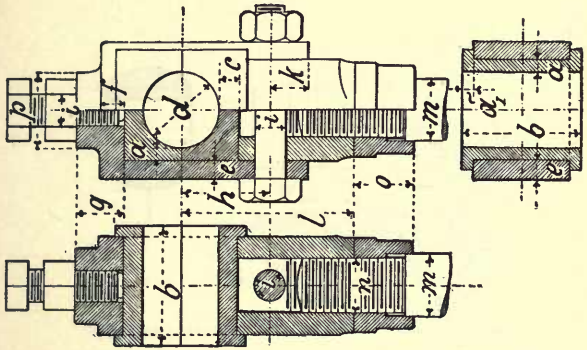


Figs. 830–833.—Trip gear levers for Corliss valves.

TABLE 75.—Dimensions of Trip-gear Levers for Corliss Valves (Figs. 830—833).

H	D	a	b	c	d	e	f	g	h	i	k	l	m	n	p	r
48	16	$\frac{7}{8}$	$2\frac{5}{8}$	1	$5\frac{1}{8}$	$2\frac{1}{4}$	$\frac{7}{16}$	$\frac{5}{16}$	$5\frac{5}{8}$	$\frac{5}{16}$	$1\frac{7}{8}$	$\frac{13}{16}$	$2\frac{1}{4}$	$\frac{5}{8}$	$\frac{5}{8}$	$3\frac{1}{2}$
48	18	$\frac{15}{16}$	$2\frac{7}{8}$	$1\frac{1}{8}$	$5\frac{5}{8}$	$2\frac{1}{2}$	$\frac{7}{16}$	$\frac{5}{16}$	$6\frac{1}{8}$	$\frac{5}{16}$	2	$\frac{7}{8}$	$2\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$3\frac{3}{4}$
48	20	1	$3\frac{1}{8}$	$1\frac{1}{4}$	6	$2\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$6\frac{5}{8}$	$\frac{3}{8}$	$2\frac{1}{4}$	$\frac{5}{16}$	$2\frac{3}{4}$	$\frac{11}{16}$	$\frac{3}{4}$	4
54	22	$1\frac{1}{16}$	$3\frac{3}{8}$	$1\frac{1}{4}$	$6\frac{1}{2}$	$2\frac{7}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	7	$\frac{3}{8}$	$2\frac{3}{8}$	1	$2\frac{7}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$4\frac{1}{4}$
54	24	$1\frac{1}{8}$	$3\frac{3}{8}$	$1\frac{3}{8}$	$6\frac{7}{8}$	3	$\frac{9}{16}$	$\frac{7}{16}$	$7\frac{1}{2}$	$\frac{7}{16}$	$2\frac{5}{8}$	$1\frac{1}{16}$	3	$\frac{3}{4}$	$\frac{3}{4}$	$4\frac{1}{2}$
54	26	$1\frac{1}{4}$	$3\frac{3}{4}$	$1\frac{3}{8}$	$7\frac{1}{4}$	$3\frac{1}{8}$	$\frac{9}{16}$	$\frac{7}{16}$	8	$\frac{7}{16}$	$2\frac{3}{4}$	$1\frac{1}{8}$	$3\frac{1}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$4\frac{3}{4}$
60	28	$1\frac{5}{16}$	4	$1\frac{1}{2}$	$7\frac{3}{4}$	$3\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$8\frac{1}{2}$	$\frac{1}{2}$	3	$1\frac{3}{16}$	$3\frac{1}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	5
60	30	$1\frac{3}{8}$	$4\frac{1}{4}$	$1\frac{1}{2}$	$8\frac{1}{4}$	$3\frac{1}{2}$	$\frac{5}{8}$	$\frac{1}{2}$	9	$\frac{1}{2}$	$3\frac{1}{8}$	$1\frac{1}{4}$	$3\frac{1}{2}$	$\frac{7}{8}$	$\frac{7}{8}$	$5\frac{1}{2}$

All Dimensions are given in Inches.

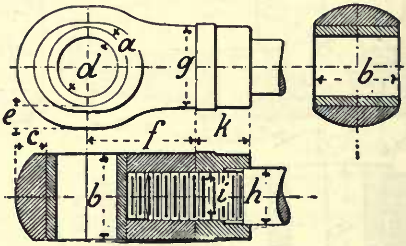


Figs. 834—836.—Valve Rod-End for Corliss Valves.

Table 76.—Dimensions of Valve Rod-Ends for Corliss Valves. (Figs. 834—836).

d	b	a	a_1	c	e	f	g	h	i	k	l	m	n	o	p
1	$1\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	$2\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$
$1\frac{1}{16}$	$1\frac{5}{8}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$2\frac{5}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{4}$
$1\frac{3}{16}$	$1\frac{3}{4}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{4}$	$1\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$2\frac{3}{4}$	1	$\frac{7}{8}$	1	$1\frac{1}{4}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$2\frac{7}{8}$	1	$\frac{7}{8}$	1	$1\frac{1}{4}$
$1\frac{3}{8}$	2	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{4}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$3\frac{1}{8}$	$1\frac{1}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{2}$
$1\frac{7}{16}$	$2\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$3\frac{1}{4}$	$1\frac{1}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{2}$
$1\frac{1}{2}$	$2\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$3\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$
$1\frac{5}{8}$	$2\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	1	$1\frac{7}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$3\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$

All Dimensions are given in Inches.



Figs. 837—839.—Valve Rod-End for Corliss Valves.

TABLE 77.—**Dimensions of Valve Rod-Ends for Corliss Valves** (Figs. 837—839).

d	*b*	*a*	*c*	*e*	*f*	*g*	*h*	*i*	*k*
$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{16}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{8}$
$\frac{7}{8}$	$1\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{15}{16}$	$1\frac{1}{4}$	$\frac{1}{8}$	$\frac{7}{16}$	$\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$
1	$1\frac{7}{16}$	$\frac{1}{8}$	$\frac{7}{16}$	$\frac{3}{8}$	$1\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
$1\frac{1}{16}$	$1\frac{1}{2}$	$\frac{5}{32}$	$\frac{1}{2}$	$\frac{3}{8}$	$1\frac{7}{8}$	$1\frac{3}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
$1\frac{1}{8}$	$1\frac{1}{2}$	$\frac{3}{16}$	$\frac{1}{2}$	$\frac{3}{8}$	2	$1\frac{3}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
$1\frac{3}{16}$	$1\frac{3}{4}$	$\frac{3}{16}$	$\frac{9}{16}$	$\frac{7}{16}$	$2\frac{1}{4}$	$1\frac{5}{8}$	1	$\frac{7}{8}$	1
$1\frac{1}{4}$	$1\frac{3}{4}$	$\frac{3}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$2\frac{3}{8}$	$1\frac{5}{8}$	1	$\frac{7}{8}$	1

All Dimensions are given in Inches.

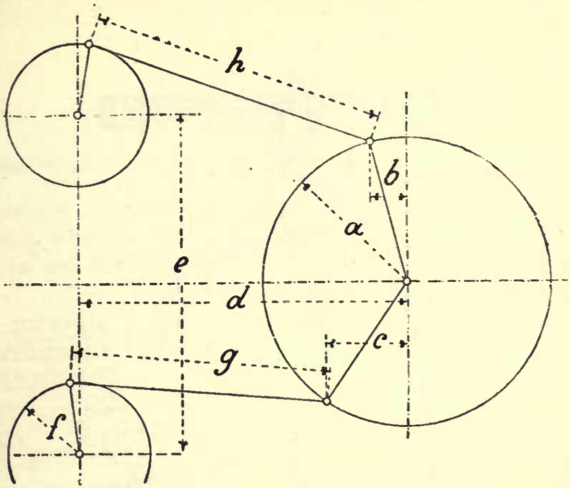
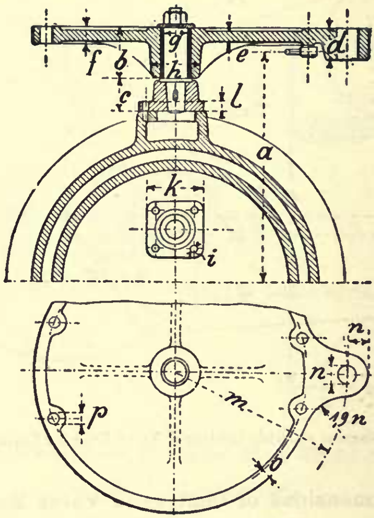


Fig. 840.—Diagram showing Centres of Valve Rods for Corliss Valve Gear.

TABLE 78.—**Dimensions of Centres of Valve Rods for Corliss Valve Gear** (Fig. 840).

H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*
48	16	10	$3\frac{1}{4}$	$4\frac{1}{4}$	27	$23\frac{1}{2}$	$5\frac{1}{4}$	$23\frac{1}{2}$	$23\frac{3}{4}$
48	18	11	$3\frac{1}{4}$	$4\frac{1}{2}$	$27\frac{1}{4}$	$25\frac{1}{2}$	$5\frac{1}{2}$	$23\frac{1}{2}$	$24\frac{1}{2}$
48	20	12	$3\frac{3}{8}$	$4\frac{3}{4}$	$27\frac{1}{2}$	$28\frac{1}{4}$	6	$21\frac{1}{2}$	25
54	22	13	$3\frac{1}{2}$	5	$30\frac{3}{4}$	$30\frac{1}{2}$	$6\frac{1}{2}$	$28\frac{1}{4}$	27
54	24	14	$3\frac{7}{8}$	$5\frac{3}{8}$	31	$33\frac{1}{2}$	7	$26\frac{1}{4}$	$27\frac{5}{8}$
54	26	15	4	$5\frac{3}{4}$	$31\frac{1}{2}$	$36\frac{1}{4}$	$7\frac{1}{2}$	$26\frac{1}{2}$	28
60	28	16	$4\frac{1}{4}$	$6\frac{1}{4}$	$34\frac{1}{2}$	$38\frac{3}{4}$	8	29	$30\frac{3}{4}$
60	30	17	$4\frac{1}{2}$	$6\frac{1}{2}$	$35\frac{1}{4}$	$41\frac{3}{4}$	$8\frac{1}{2}$	30	$31\frac{3}{4}$

All Dimensions are given in Inches.



Figs. 841, 842.—Wrist Plate for Corliss Gear.

TABLE 79.—**Dimensions of Wrist Plate for Corliss Valve Gear** (Figs. 841, 842).

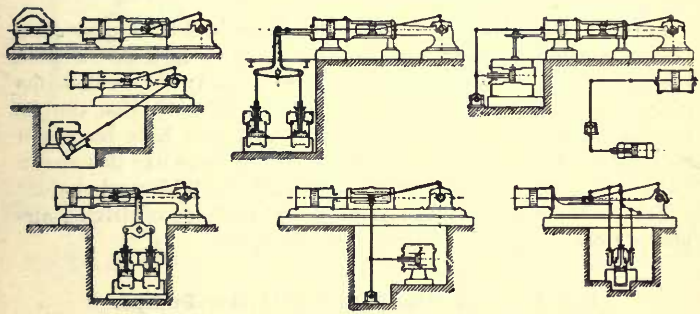
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*
54	24	$25\frac{1}{2}$	$4\frac{7}{8}$	$4\frac{1}{4}$	$3\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$2\frac{1}{2}$	$5\frac{1}{4}$	$\frac{3}{4}$	$6\frac{3}{4}$	1	$19\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{3}{4}$	$1\frac{5}{16}$
54	26	$27\frac{5}{16}$	$5\frac{5}{8}$	$4\frac{1}{2}$	$3\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$5\frac{1}{2}$	$\frac{7}{8}$	7	$1\frac{1}{8}$	$20\frac{1}{2}$	$2\frac{1}{2}$	2	$1\frac{5}{16}$
60	28	$28\frac{7}{8}$	$5\frac{5}{8}$	$4\frac{3}{4}$	4	$1\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{7}{8}$	$5\frac{3}{4}$	$\frac{7}{8}$	7	$1\frac{1}{8}$	$21\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{1}{4}$	$1\frac{3}{8}$
60	30	$30\frac{1}{2}$	6	$4\frac{3}{4}$	$4\frac{1}{4}$	$1\frac{3}{5}$	2	3	6	$\frac{7}{8}$	7	$1\frac{1}{8}$	$23\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{3}{4}$

All Dimensions are given in Inches.

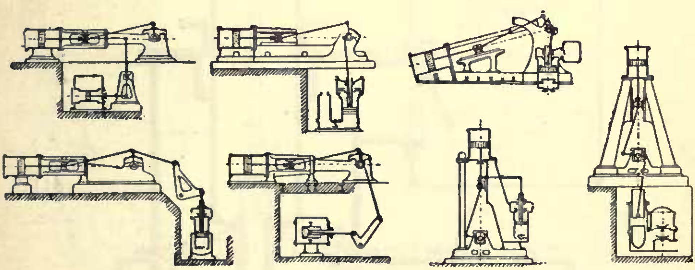
SECTION V.

CONDENSERS, AIR-PUMPS, AND FEED-PUMPS.

Condensers are divided into two principal classes: (*a*) *Injection Condensers,* where the condensing water acts in direct contact with the steam, and therefore mixes with the water from the condensed



Figs. 843—849.



Figs. 850—856.

steam; (*b*) *Surface Condensers,* where the condensing water acts on one side of a thin wall of metal, the steam being in contact with the other, and thus the water from the condensed steam is kept separate from the condensing water; these are the two varieties most in use.

A few special methods of condensing steam are occasionally used ; of these some may be said to belong to the class (*a*), others to the class (*b*).

The earliest examples of condensers and air-pumps were vertical and single-acting, and were generally applied to beam engines. When horizontal engines came into fashion, the air-pumps were often vertical, worked by means of a bell-crank lever ; but owing to the number of parts required to drive a vertical air-pump from a horizontal engine, the simpler method of putting the air-pump surrounded by the condenser horizontal, and in the same line as the piston-rod, and driving direct from the tail rod, has come into use to a large extent.

An objection to horizontal air-pumps driven in this way, has often been urged against their use, namely, that with high piston-speed engines, the speed of air-pump bucket, necessarily the same, is too high for efficiency. By good proportions this objection can be removed, and even speeds of 800 feet per minute have been used with good results, especially with plunger air-pumps like that shown in fig. 876, taken from the Allen engine made by Whitworth & Co.

Figs. 843 to 856 show in diagram several methods of driving air-pumps from both horizontal and vertical engines.

(a.) Injector Condensers with Air-Pumps.

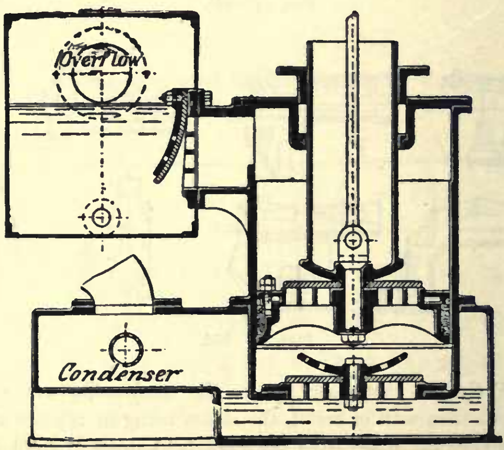
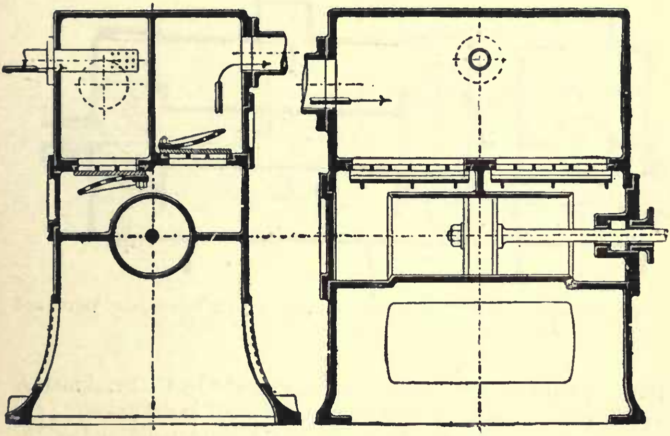
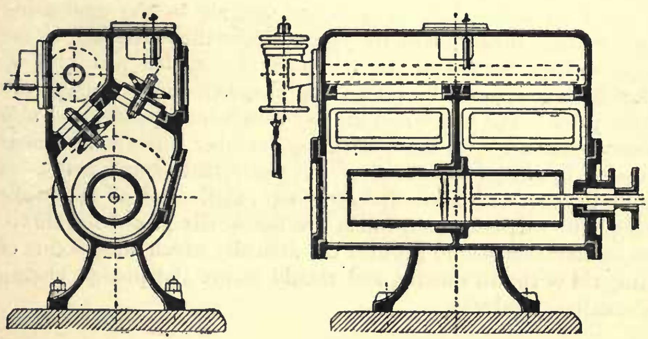


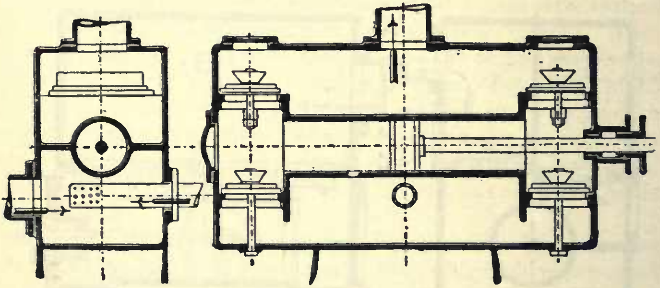
Fig. 857.—Vertical Bucket and Plunger Air-Pump of ordinary type.



Figs. 858, 859.—Horizontal Air Pump and Condenser with Rectangular Valves.

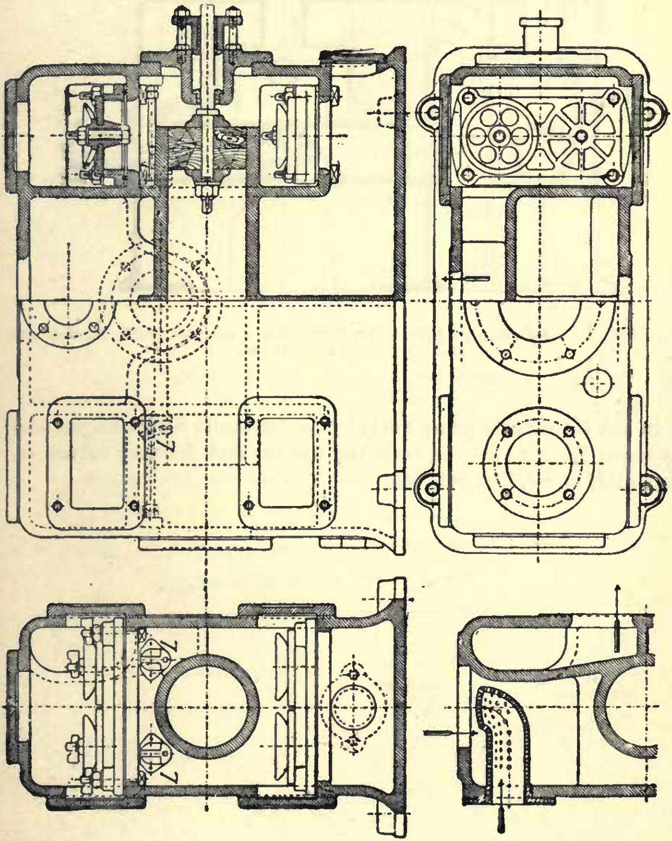


Figs. 860, 861.—Horizontal Air Pump and Condenser with Rectangular Valves, made by the Prince Rudolf Iron Works, Dülmen.

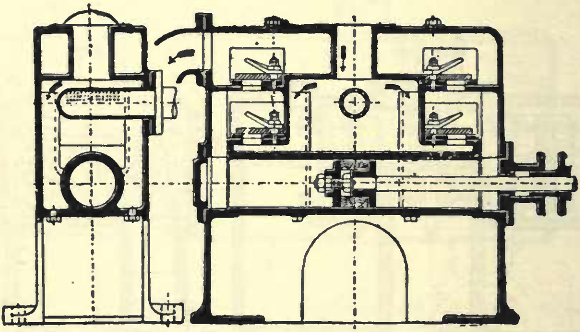


Figs. 862, 863.—Horizontal Air Pump and Condenser, with Rectangular Valves and Injection Nozzle, and Exhaust Inlet below the Barrel.

Horn's patent air pump and condenser, made by G. Brinkman & Co. of Witten, is shown in section, figs. 864—867 and 870—872 ; the air pump piston or bucket is shown at the right hand end of the barrel and is just starting back to the left, at this moment the space between the suction and delivery valves and piston is full of water, as the piston continues to move to the left the water level sinks, (whilst the delivery valve remains closed by the external atmospheric pressure,) and maintains a vacuum in the space between the surface of the water and delivery valve ; as the water level sinks it allows the small air valves, L L, to open, and any air in the condensing chamber passes through into the space left by the movement of the piston, without having to pass through the suction valve below, whilst the water from the condensing chamber passes through the suction valve. The water and air are thus kept separate, and also the air is drawn out of the condensing chamber through the small valves, L L, at the beginning of the piston stroke, and not, as is usually the case with horizontal air pumps, at the end of the stroke through the suction valves, which are necessarily placed low down. This construction should get over the difficulty which often occurs of getting rid of the air quietly, and should secure the prompt closing of the delivery valves.

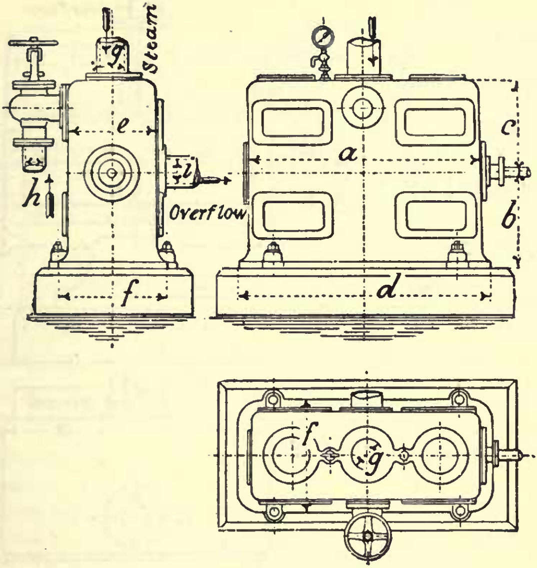


Figs. 864—867.



Figs. 868, 869.—Ed. König's Patent Air Pump and Condenser, with Suction and Delivery Valves above Barrel.

In Ed. König's air pump made by the Chemnitz Steam Engine and Spinning Machine Works, both the suction and delivery valves are above the barrel, figs. 868, 869.

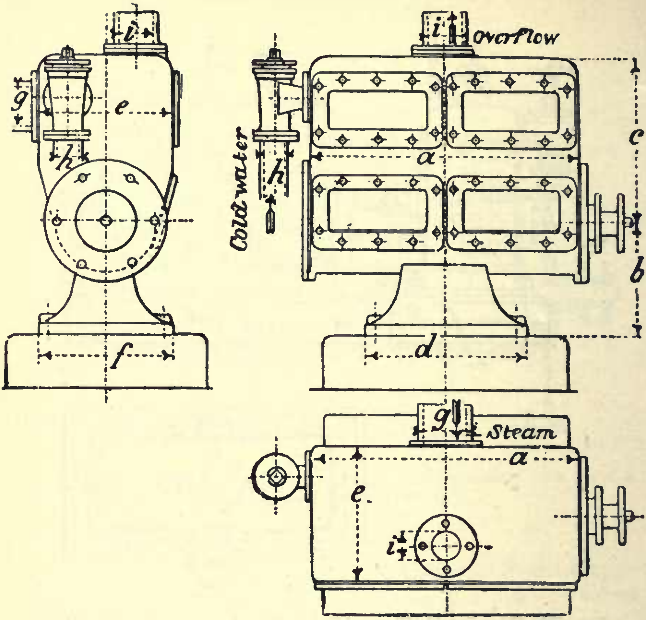


Figs. 870—872.—Outside Views of Horn's Patent Air Pump and Condenser.

TABLE 80.—**Dimensions of Horn's Patent Air Pump and Condenser** (Figs. 864—867 and 870—872).

Engine.		Air Pump and Condenser.								Pipes.		
H	D	H	D	*a*	*b*	*c*	*d*	*e*	*f*	Exhaust. *g*	Injection. *h*	Overflow. *i*
$19\frac{5}{8}$	$11\frac{3}{4}$	$19\frac{5}{8}$	5	$33\frac{1}{2}$	$15\frac{3}{4}$	$15\frac{3}{4}$	$37\frac{1}{2}$	$13\frac{3}{4}$	$17\frac{5}{8}$	$3\frac{1}{2}$	$2\frac{3}{8}$	4
$23\frac{5}{8}$	$13\frac{3}{4}$	$23\frac{5}{8}$	$6\frac{1}{4}$	$39\frac{1}{2}$	$16\frac{1}{2}$	17	$43\frac{1}{2}$	15	19	$4\frac{3}{8}$	3	$4\frac{3}{4}$
$27\frac{1}{2}$	$15\frac{3}{4}$	$27\frac{1}{2}$	7	$47\frac{1}{4}$	$17\frac{5}{8}$	$17\frac{5}{8}$	$51\frac{1}{4}$	16	20	5	4	$5\frac{1}{2}$
$31\frac{1}{2}$	$17\frac{5}{8}$	$31\frac{1}{2}$	$7\frac{3}{4}$	$49\frac{1}{4}$	$19\frac{5}{8}$	$19\frac{5}{8}$	$54\frac{1}{4}$	$19\frac{5}{8}$	$23\frac{5}{8}$	$5\frac{1}{2}$	$4\frac{3}{4}$	$6\frac{1}{4}$
$35\frac{1}{2}$	$19\frac{5}{8}$	$35\frac{1}{2}$	$8\frac{3}{4}$	65	$19\frac{5}{8}$	$20\frac{1}{2}$	$69\frac{1}{4}$	$22\frac{1}{2}$	$27\frac{1}{2}$	$6\frac{1}{4}$	$5\frac{1}{2}$	7
$39\frac{1}{2}$	$21\frac{5}{8}$	3[illegible]$\frac{1}{2}$	$9\frac{3}{4}$	65	$27\frac{1}{2}$	$25\frac{1}{2}$	71	$27\frac{1}{2}$	$35\frac{1}{2}$	7	$6\frac{1}{4}$	8

All Dimensions are given in Inches.

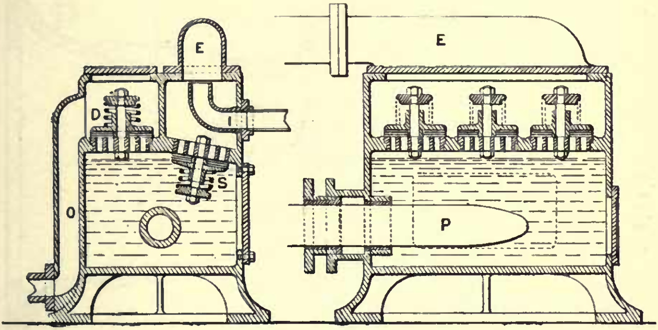


Figs. 873—875.—Outside Views of the Dülmen Air Pump and Condenser.

TABLE 81.—**Dimensions of Air Pump and Condenser** (Figs. 873—875 and 860, 861).

Engine.		Air Pump and Condenser.								Pipes.		
H	D	H	D	*a*	*b*	*c*	*d*	*e*	*f*	Ex-haust. *g*	Injec-tion. *h*	Over-flow. *i*
$19\frac{5}{8}$	$11\frac{3}{4}$	$19\frac{5}{8}$	$4\frac{3}{8}$	$33\frac{1}{2}$	$13\frac{3}{4}$	$19\frac{5}{8}$	$23\frac{5}{8}$	$16\frac{1}{2}$	$15\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{3}{4}$
$23\frac{5}{8}$	$13\frac{3}{4}$	$23\frac{5}{8}$	5	$37\frac{3}{8}$	$15\frac{3}{4}$	$20\frac{5}{8}$	$27\frac{1}{2}$	$16\frac{1}{2}$	$17\frac{5}{8}$	4	$2\frac{3}{8}$	$3\frac{1}{4}$
$27\frac{1}{2}$	$15\frac{3}{4}$	$27\frac{1}{2}$	$5\frac{1}{2}$	$43\frac{3}{8}$	$19\frac{5}{8}$	$23\frac{5}{8}$	$33\frac{1}{2}$	$19\frac{3}{4}$	$21\frac{5}{8}$	5	$2\frac{3}{8}$	$3\frac{1}{2}$
$31\frac{1}{2}$	$17\frac{5}{8}$	$31\frac{1}{2}$	$6\frac{1}{4}$	49	$23\frac{5}{8}$	$24\frac{5}{8}$	$39\frac{3}{8}$	$19\frac{3}{4}$	$27\frac{1}{2}$	6	$3\frac{1}{4}$	4
$35\frac{1}{2}$	$19\frac{5}{8}$	$35\frac{1}{2}$	7	55	$23\frac{5}{8}$	$26\frac{1}{2}$	$43\frac{3}{8}$	$19\frac{3}{4}$	$31\frac{1}{2}$	6	$3\frac{1}{2}$	5
$39\frac{1}{2}$	$21\frac{5}{8}$	$39\frac{1}{2}$	8	59	$27\frac{1}{2}$	$28\frac{1}{2}$	$47\frac{1}{4}$	$21\frac{5}{8}$	$31\frac{1}{2}$	7	4	5

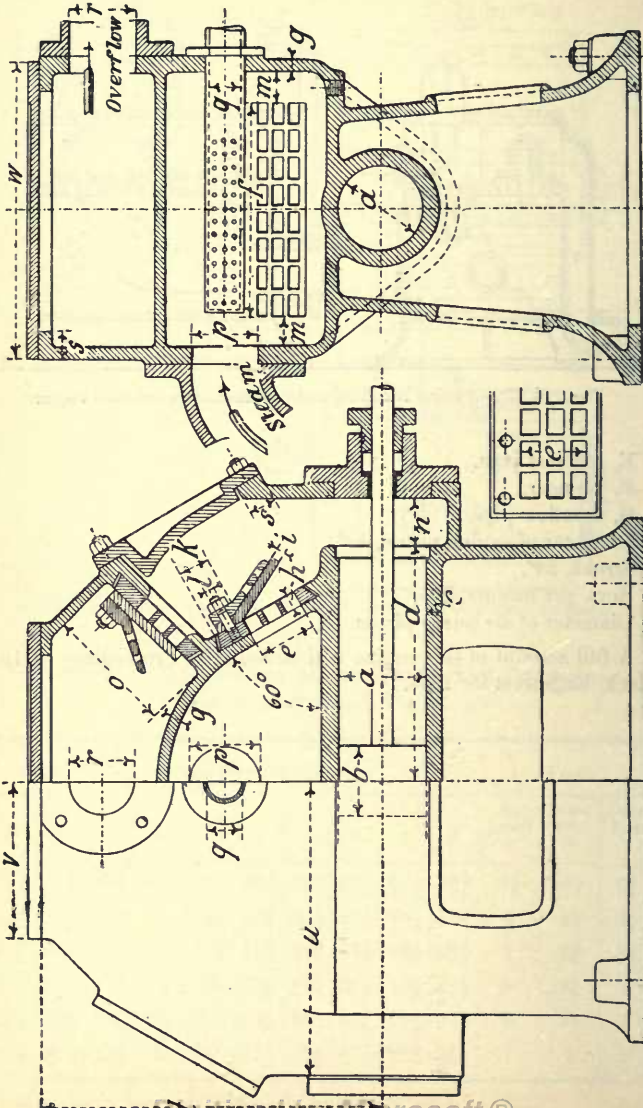
All Dimensions are given in Inches.



Figs. 876, 877.—Plunger Air Pump and Condenser from the Allen Engine.

E, exhaust pipe.
P, plunger.
O, overflow pipe.
Diameter of engine piston, 12″.
Stroke, 24″.
Revs. per minute, 200.
Diameter of air pump piston, 5″.

A full account of this engine will be found in Proceedings of Inst. Mech. Engineers for 1868.



Figs. 878—880.

TABLE 82.—**Dimensions of Air Pump and Condenser** (Figs. 878—880).

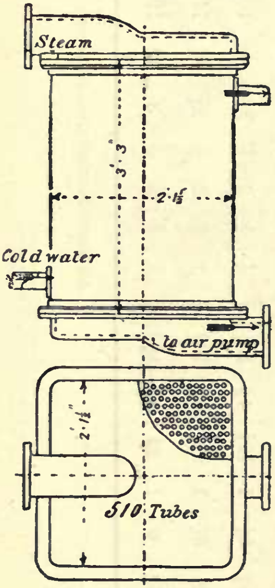
H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*	*t*	*u*	*v*	*w*
24	12	$4\frac{3}{4}$	4	$\frac{7}{8}$	$12\frac{5}{8}$	$3\frac{1}{4}$	$11\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{3}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$2\frac{5}{8}$	$6\frac{3}{4}$	$3\frac{5}{8}$	$2\frac{3}{8}$	$3\frac{1}{4}$	$1\frac{1}{2}$	18	16	$8\frac{3}{8}$	16
28	14	$5\frac{5}{8}$	$4\frac{3}{8}$	$\frac{7}{8}$	$14\frac{3}{4}$	$3\frac{5}{8}$	$12\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{3}{4}$	$\frac{5}{8}$	$1\frac{3}{4}$	3	8	$4\frac{3}{8}$	3	$3\frac{3}{4}$	$1\frac{1}{2}$	$20\frac{3}{4}$	$18\frac{5}{8}$	$9\frac{3}{4}$	$17\frac{5}{8}$
32	16	$6\frac{3}{8}$	$4\frac{3}{4}$	$\frac{7}{8}$	17	4	14	$\frac{5}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	2	$\frac{3}{4}$	$2\frac{1}{8}$	$3\frac{3}{8}$	9	5	$3\frac{5}{8}$	$4\frac{3}{8}$	$1\frac{1}{2}$	$22\frac{3}{4}$	$21\frac{1}{4}$	$11\frac{1}{4}$	$19\frac{5}{8}$
36	18	$6\frac{3}{4}$	$5\frac{1}{4}$	1	$19\frac{3}{8}$	$4\frac{3}{4}$	$15\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{5}{8}$	2	$\frac{3}{4}$	$2\frac{1}{2}$	4	10	$5\frac{5}{8}$	$4\frac{1}{4}$	5	$1\frac{1}{2}$	$25\frac{1}{4}$	$23\frac{3}{4}$	$12\frac{3}{4}$	$21\frac{5}{8}$
40	20	8	$5\frac{5}{8}$	1	$21\frac{5}{8}$	$5\frac{5}{8}$	$16\frac{3}{4}$	$\frac{3}{4}$	1	$\frac{5}{8}$	2	$\frac{3}{4}$	$2\frac{3}{4}$	$4\frac{3}{8}$	11	$6\frac{3}{4}$	$4\frac{3}{4}$	$5\frac{5}{8}$	$1\frac{3}{4}$	$27\frac{5}{8}$	$26\frac{3}{4}$	$14\frac{3}{8}$	24
44	22	$8\frac{3}{4}$	6	1	24	$6\frac{3}{8}$	$18\frac{3}{8}$	$\frac{3}{4}$	1	$\frac{5}{8}$	2	$\frac{3}{4}$	3	$4\frac{3}{4}$	12	$7\frac{1}{4}$	$5\frac{1}{4}$	6	$1\frac{3}{4}$	30	$29\frac{5}{8}$	16	26

All Dimensions are given in Inches.

The quantity of condensing water required depends in some measure upon the temperature, but a rule which will serve as a basis upon which to judge of the amount required is from 20 to 30 times the amount of feed water used; an actual example taken by measuring the overflow water as it left the condenser gave from 61 to 80 gallons per indicated HP per hour; the temperature of the injection water was 56°, and that of the overflow 100°; the engine had a 12″ piston and 15″ stroke, the boiler pressure being 55 lbs., and at the time when the water was measured indicated from 20 to 24 HP; the air pump was a vertical plunger pump.

(b.) Surface Condensers.

Surface condensers are now almost exclusively used in marine work, and enable the boilers to be worked at much higher pressures



Figs. 881, 882.

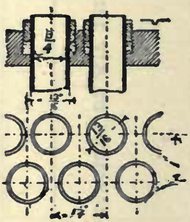
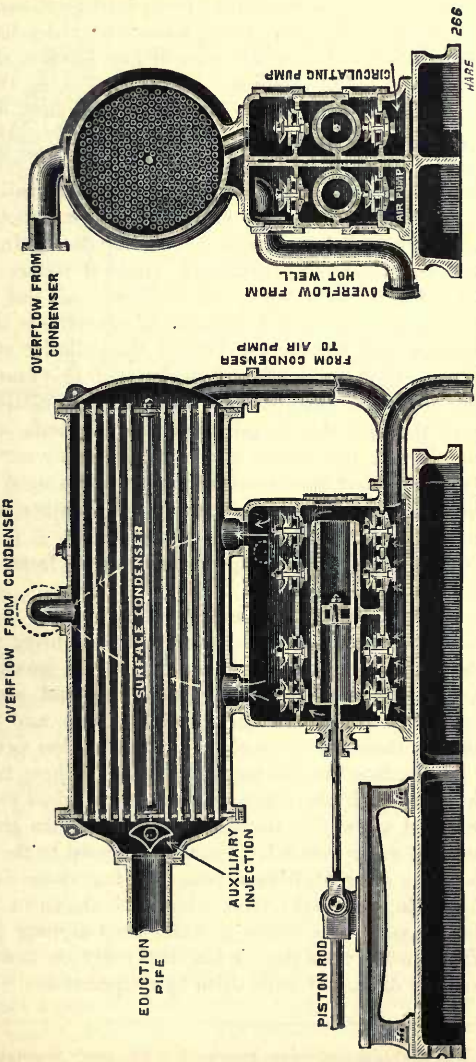


Fig. 883.

on account of the purity of the feed water, only a small amount of auxiliary feed being used direct from the sea; they are also increasing in use on land, especially where water is plentiful but bad; their



Figs. 884, 885.—Horizontal Surface Condenser and Air Pump, by Messrs. Ruston, Proctor & Co., Lincoln.

first cost is much greater than an ordinary jet condenser, but there is much to be said as to their suitability for special positions, such as by the side of brackish rivers, or rivers where the water has a bad action on boiler plates. An outside view of one form is shown in figs. 881, 882, and fig. 883 shows one method of fixing the tubes. The tubes are usually of solid drawn brass of $\frac{5}{8}''$ or $\frac{3}{4}''$ or $\frac{13}{16}''$ diameter, the tube plates of brass or copper. $\frac{5}{8}''$ tubes run $10\frac{3}{4}''$ to the lb., $\frac{3}{4}''$ tubes, $9\frac{1}{2}''$ to the lb., the thickness being 18 B W G.

The method shown in fig. 883, of fixing the tubes by small screwed glands and packing allows of free expansion and contraction, and also admits of a tube being drawn for examination. In marine surface condensers the cooling surface is about 3 square feet per indicated HP. A surface condenser combined with air and circulating pumps, by Messrs. Ruston & Proctor of Lincoln, is shown in figs. 884, 885, arranged for placing behind the cylinder of a horizontal engine, on an extension of the bed plate ; in this example the exhaust steam is passed through the tubes, the circulating water being pumped through the chamber containing them, and thus coming in contact with the outside of the tubes ; many surface condensers are made so that the circulating water is pumped through the tubes, the exhaust steam passing through the chamber, and coming into contact with the outside of the tubes.

The quantity of circulating water required varies from 55 to 90 gallons per indicated HP per hour.

In places where the supply of water for condensing is limited, various devices have been used for cooling the condensing water, so that it may be used over again with the addition of a small amount of make up. Shallow cooling ponds have been and are in use successfully in many places, but these take up large and valuable ground space, and their use is therefore limited. Other devices are now coming largely into use, the most important of these being the Cooling Tower, in which the condensing water is raised to the top of a high tower and allowed to trickle down over wooden gratings so as to distribute the water over a large surface, exposed to the air, and in most cases to the draught from a fan. An ingenious device by Theisen,* of the Engine Works, Grevenbroich, is shown in fig. 886 ; here the water is cooled by a series of metal discs dipping partly in water and kept slowly revolving, a fan draws dry air through the spaces between the discs, and cools them by evaporation.

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* A full account of this arrangement appeared in Dingler's Journal, Vol. 267, Part 13, page 586.

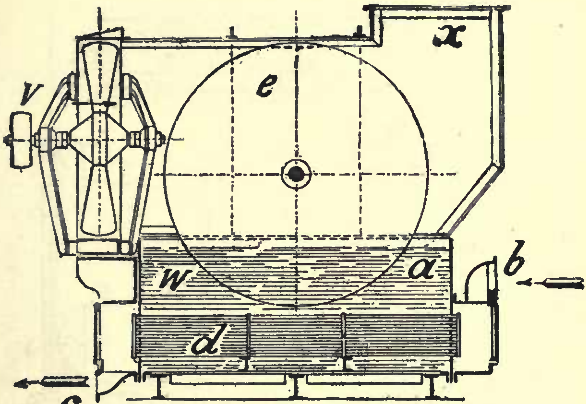


Fig. 886.—Theisen's Cooling Apparatus for Condensing Water.

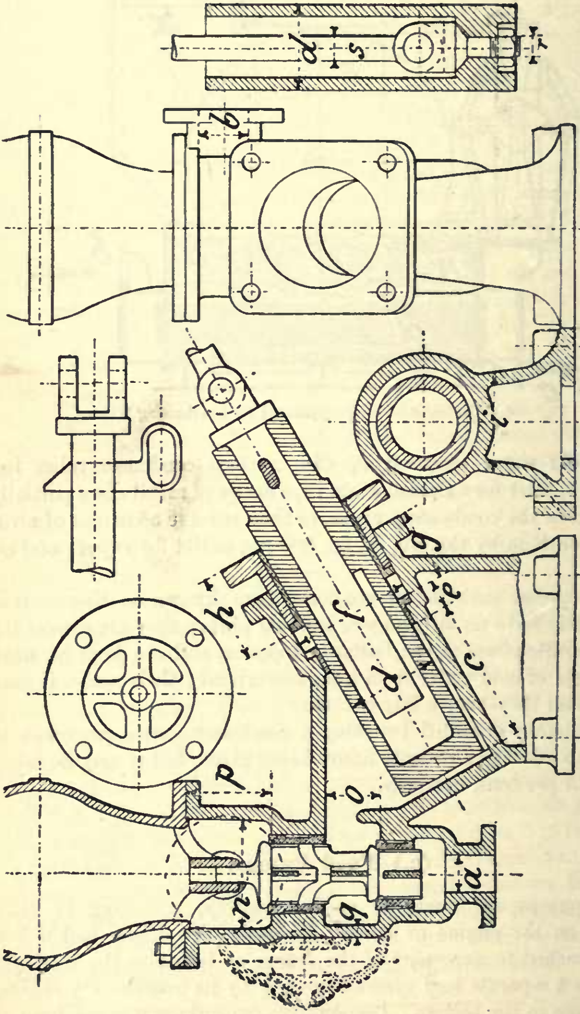
The cold water reservoir *a*, with surface condenser tubes immersed : *b*, inlet for exhaust steam ; *e*, a series of metal discs partially immersed in the condensing water, as they revolve a current of air is forced over them by the fan, V, fig. 886 ; *x*, outlet for vapour and air from fan.

Other special varieties of condenser are known as Ejector condensers, and have no air pump or moving parts ; they are almost the same in construction as the Giffards Injector, and are kept at work by a stream of cold water from a tank overhead ; their action is continuous, and they keep a fair vacuum.

The economy obtained by using a condenser varies too much in practice to allow of a definite figure being given, but it may be taken as from 15 per cent. upwards.

(c.) Feed Pumps.

Feed pumps are made in endless variety, according to their position on the engine or boiler ; formerly all engines had a feed pump attached to some part of the frame or bed, now the tendency is to have a separate feed system, worked by its own donkey engine, placed close to the boilers. Locomotives depend upon some form of injector, and very often on large portable or under-type engines an auxiliary feed by injector or donkey pump is provided. When a



Figs. 887—891.

TABLE 83.—**Dimensions of Feed Pump** (Figs. 887—891).

Engine.		Feed-pump.																		
H	D	*d*	H	*a*	*b*	*c*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*s*
12	6	1¼	1¾	¾	¾	1¾	1	2¼	2	2⅜	1¼	5¼	1	1¼	2	1⅝	2⅜	½	—	—
16	8	1⅝	2	1¼	1	1⅞	1¼	2¾	2⅝	2¾	1⅝	5⅝	1⅜	1¾	2⅜	2	2¾	½	—	—
20	10	2	2⅝	1⅜	1¼	2½	1¼	2⅜	3¼	3⅜	2	6⅜	1½	2	2¾	2	3¼	½	—	—
24	12	2¾	3¼	1½	1⅜	3⅛	1⅜	4¼	3¾	4	2¾	7¼	2	2⅜	3⅜	2¾	3⅝	½	1	1¼
28	14	3¼	3⅝	1¾	1¾	3⅝	1½	4¾	4⅜	4⅜	3¼	8	2¼	2⅝	4	3	4	½	1¼	1⅜
32	16	3⅝	4	2	2	4	1¾	5¼	5	4¾	3⅝	8⅜	2⅜	2⅞	4⅜	3¼	4⅜	½	1¼	1⅝
36	18	4	4⅜	2⅜	2⅜	4⅜	2	5⅝	5⅝	5¼	4	9¼	2¾	3⅜	4¾	3⅜	4¾	½	1⅜	1¾
40	20	4⅜	5	2¾	2¾	4¾	2⅜	6	6	5⅜	4⅝	10	3¼	3¾	5⅜	3⅝	5¼	½	1⅜	1¾
44	22	4¾	5⅝	3¼	3¼	5⅜	2¾	6⅜	6⅜	5⅝	5¼	10¾	3⅝	4¼	6	4	5⅝	⅝	1⅜	2

range of boilers is fed from one pump, there must be a feed escape valve, placed so as to command the whole feed system, and loaded so as to blow off a little above the boiler pressure, and this valve should be in a conspicuous position ; if any obstruction occur this valve will blow off and show that there is something resisting the flow of the water.

An ordinary feed pump for bolting down to the engine foundation, and worked by an eccentric, is shown in fig. 887. A method of disengaging the plunger, is shown in upper fig., this method is often

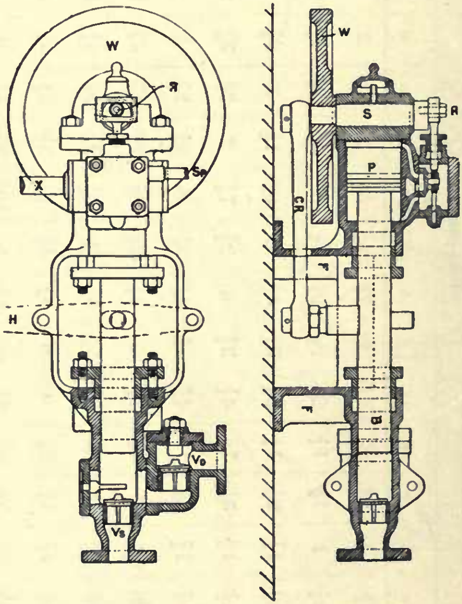


Fig. 892. Fig. 893.

used in slow moving pumps when required to do intermittent work, but is not often applied to boiler feed pumps.

Figs. 892—893, show a simple boiler feeding donkey pump.

F F, the frame by which the pump is bolted to the wall ; W, the fly wheel ; S, the crank shaft ; R, a small crank pin for working the slide valve ; P, the steam piston ; B, the pump plunger ; V *d*, the delivery valve ; V *s*, the suction valve ; H, a hand lever for working the pump by hand when required.

NOTE. See also page 446A.

(d.) Feed-Water Heaters.

Feed-water heaters are extensively used, and effect great economy in fuel. They may be considered as a reversed surface condenser, the exhaust steam being used to warm the water as it passes from the

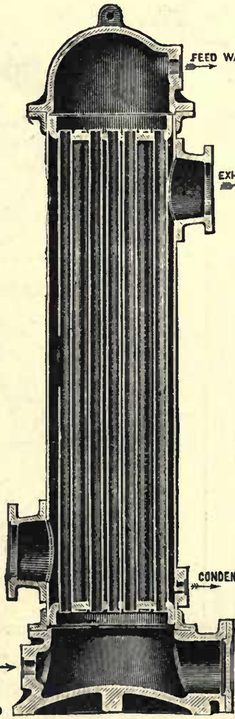
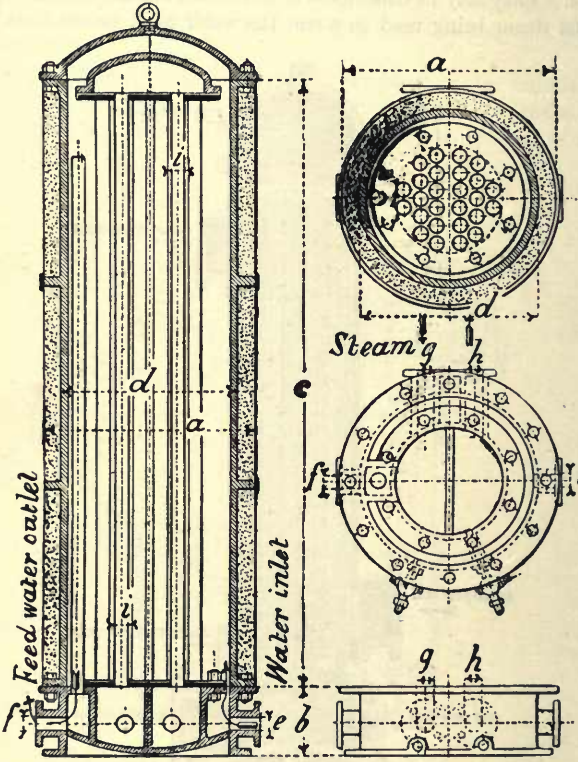


Fig. 894.—Feed-Water Heater, by Messrs. Marshall Sons & Co. Ltd., Gainsboro.

pump to the boiler, two examples are shown in figs. 894—895. Feed-water heaters of this type are sometimes placed horizontally. If pure water could be used, the heaters would work efficiently for long periods, but as water always contains mineral matters which are partially precipitated when the water is warmed, the heater tubes become coated with scale, and have to be cleaned. Occasionally, the feed water seems to select certain tubes for its flow, and not infrequently, if the heaters have been neglected, only some of the tubes are deeply scaled,* the rest being comparatively free; baffle plates, to ensure good distribution of the water, would possibly prevent this



Figs. 895—898.—Feed-heater, by the Prince Rudolf Iron Works, Dülmen.

selection. From the scaling it is obvious that feed-water heaters are to a certain extent feed-water purifiers, and if of ample size, must act beneficially in this direction. In some feed-heaters, the tubes are re-

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* An instance of this occurred in practice where a vertical heater of the above type, with about 65 tubes, had been in use for a long time without cleaning, upwards of 50 of the tubes were totally blocked up, and the remainder comparatively clear.

placed by an arched branch of exhaust pipe, or this arched branch may be made up of a number of tubes through which the exhaust passes, and the feed water is pumped through the case containing the tubes. These heaters are now very much used, and have proved to be quite satisfactory.

Other types of heaters depend on a small jet of water from the feed pump used as an injector to carry a part of the exhaust steam into a tank, and thus heating the water ; this arrangement is used on agricultural engines, but it has the disadvantage of introducing grease from the exhaust into the feed water.

The maximum speed of the feed water in any part of the feed system should not exceed from 120 to 160 feet per minute.

TABLE 84.—**Dimensions of Feed-Water Heater**
(Figs. 895—898).

Nominal H.P.	Heating-surface in sq. ft.	*a*	*b*	*c*	*d*	Feed-water.		Steam.		Tubes	
						Iulet	Outlet	Inlet	Outlet	No.	*i*
10	7·53	$17\frac{3}{4}$	$11\frac{3}{4}$	42	$13\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{3}{8}$	4	2
15	10·76	$17\frac{3}{4}$	$11\frac{3}{4}$	61	$13\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{5}{8}$	$2\frac{3}{4}$	$2\frac{3}{4}$	4	2
25	16·14	$19\frac{3}{4}$	$13\frac{3}{4}$	59	$15\frac{3}{4}$	2	2	$3\frac{1}{8}$	$3\frac{1}{8}$	6	2
40	26·90	26	$13\frac{3}{4}$	67	$21\frac{5}{8}$	2	2	$3\frac{1}{2}$	$3\frac{1}{2}$	10	2
60	37·66	28	$13\frac{3}{4}$	67	$23\frac{5}{8}$	2	2	4	4	14	2
80	64·56	28	$13\frac{3}{4}$	$86\frac{1}{2}$	$23\frac{5}{8}$	$2\frac{3}{8}$	$2\frac{3}{8}$	5	5	18	2
100	91·46	$34\frac{1}{4}$	$15\frac{3}{4}$	$74\frac{1}{2}$	$29\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{3}{8}$	6	6	30	2
125	107·6	$34\frac{1}{4}$	$15\frac{3}{4}$	$86\frac{1}{2}$	$29\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{3}{4}$	6	6	30	2
150	177·5	$44\frac{1}{2}$	$17\frac{3}{4}$	71	$39\frac{3}{8}$	$3\frac{1}{8}$	$3\frac{1}{8}$	7	7	6	2

All Dimensions are given in Inches except the Heating-surface.

SECTION VI.

EXAMPLES OF VARIOUS TYPES OF HORIZONTAL AND VERTICAL ENGINES FROM ACTUAL PRACTICE.

With Tables of Dimensions, Weights, &c.

Horizontal Engines of the more usual Types.

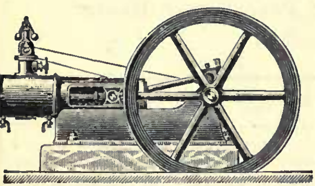


Fig. 899.

Engine with forked or Y frame, bent or slotted out crank shaft, arranged to carry the fly-wheel outside the bearings on either side of the engine, frequently known as "self-contained" horizontal engines. These engines are usually made with cylinders from 5 to 12 inches diameter, the stroke being from $1\frac{1}{4}$ to $1\frac{1}{2}$ times the diameter.

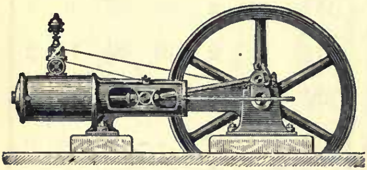


Fig. 900.

Girder or "Corliss" Frame Engines, with cylinder bolted on to end of, and overhanging the girder. Frame unsupported except by a foot at each end. Crosshead guides cast in the frame for small and medium-sized engines, and either bored or with planed flat surfaces.

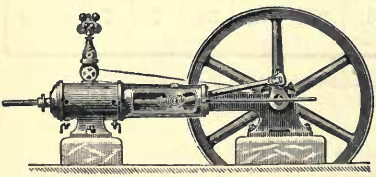


Fig. 901.

Girder type of engine supported by feet on cylinder, and at end of frame under crank shaft bearings. Piston rod extended and guided through back cylinder cover. Crosshead guides cast in one with frame in small and medium-sized engines.

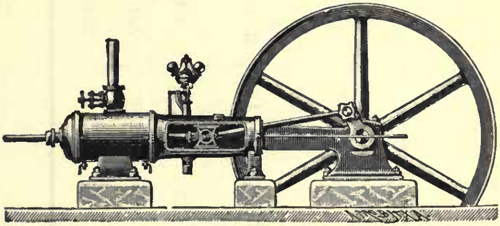


Fig. 902

Girder type of engine with extra supporting foot under centre of frame.

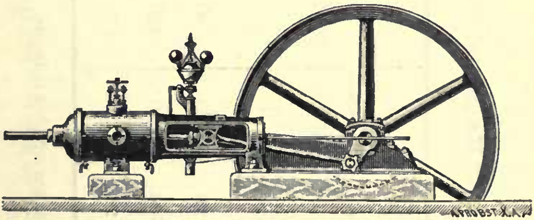


Fig. 903.

Semi-Girder type of engine, with half the length of frame bolted down to foundation, and with support under cylinder.

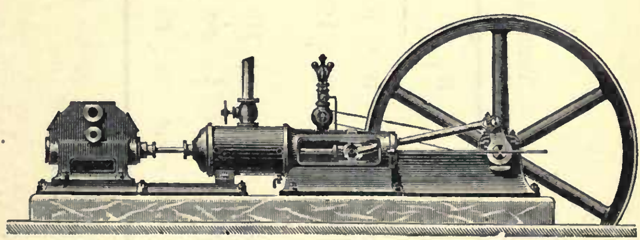


Fig. 904.

Condensing engine, with air-pump at back of cylinder and worked by tail rod.

TABLE 85.—**Dimensions of Horizontal Steam Engines.**

Stroke . . . H	16		20		24		28		32		36		40		44		48	
Diameter of cylinder . D	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
Ratio . . H : D	2	1·78	2	1·82	2	1·85	2	1·87	2	1·88	2	1·89	2	1·90	2	1·91	2	1·92
Revolutions per minute n	110		100		90		85		80		78		75		72		70	
Piston speed in feet per minute . . .	283		334		360		396		426		468		500		528		560	
Diameter of fly-wheel .	66		84		96		120		138		156		180		192		216	

All Dimensions are given in Inches.

TABLE 86.—**Dimensions of Horizontal Corliss Engines.**

Stroke . . H	30		36		42			48			54			60		66	
Diameter of cylinder . . D	10	12	12	14	14	16	18	16	18	20	22	24	26	28	30	32	34
Ratio . . H : D	3	2·67	3	2·57	3	2·62	2·33	3	2·66	2·40	2·45	2·25	2·07	2·15	2·00	2·06	1·97
Revolutions per minute . n	80	75		72		70		68			65			60		55	
Piston speed in feet per minute	400	375	465	432	505	490	490	544			585			600		605	
Diameter of fly-wheel .	108	120	120	132	144	156	168	168	180	192	204	216	240	264	288	312	336

All Dimensions are given in Inches.

TABLE 87.—Dimensions of Vertical Steam Engines.

Stroke . . H	6		8		10		12		16		20		24		30		36		42	
Diameter of cylinder . . D	4	5	6	7	8	9	10	12	14	16	18	20	22	24	28	30	34	38	42	46
Ratio . H : D	1·5	1·2	1·33	1·14	1·25	1·11	1·20	1·00	1·14	1·00	1·11	1·00	1·09	1·00	1·07	1·00	1·06	·95	1·00	·91
Revolutions per minute . . n	300		280		260		245		210		180		160		130		110		100	
Piston speed in feet per minute . .	300		374		434		490		560		600		640		650		660		700	
Diameter of fly-wheel . . .	42		48		54		60		72		84		96		120		144		168	

All Dimensions are given in Inches.

TABLE 88.—**Dimensions of Compound Engines.**

Stroke . . H	20		24		28		32		36		40		44		48	
Diameter of high pressure cylinder . .	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
Diameter of low pressure cylinder . .	16	17½	19	20½	22	24	25	27	29	30	32	33	35	37	38	40
Ratio of capacities . .	2·56	2·53	2·51	2·49	2·46	2·56	2·44	2·52	2·60	2·49	2·56	2·46	2·46	2·58	2·51	2·56
Revolutions per minute	90		80		75		70		68		65		62		60	
Piston speed in feet per minute . . .	300		320		350		373		408		433		455		480	
Diameter of fly-wheel .	78		96		120		138		162		180		198		216	

All Dimensions are given in Inches.

TABLE 89.—**Dimensions of Compound Corliss Engines.**

Stroke . . . . . H	30		36		42			48			54			60	
Diam. of high-pressure cylinder	10	12	12	14	14	16	18	16	18	20	22	24	26	28	30
Diam. of low-pressure cylinder	16	19	19	22	22	25	29	25	29	32	35	38	42	44	48
Ratio of capacities	2·56	2·51	2·51	2·46	2·46	2·44	2·60	2·44	2·60	2·56	2·46	2·51	2·60	2·46	2·56
Revolutions per minute . .	80	75		72		70		68			65			60	
Piston speed in feet per minute	400	375	465	432	505	490	490	544			585			600	
Diameter of fly-wheel	108	120		132	144	156	168	180	192	204	216	240	264	288	312

All Dimensions are given in Inches.

Leading Dimensions of Single Cylinder Horizontal Engines, with and without Condensers.

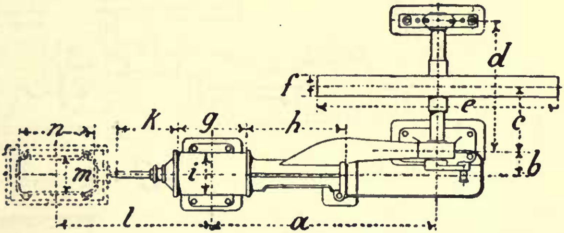


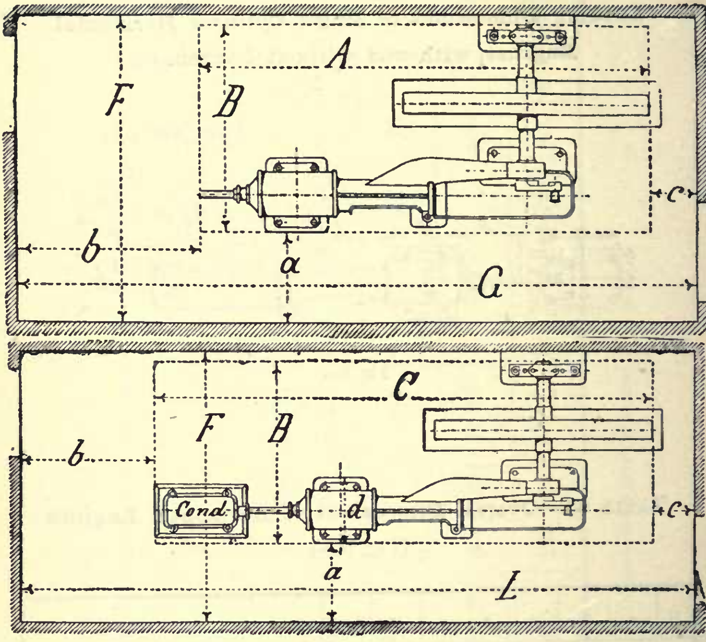
Fig. 905.

Table 90.—Overall Dimensions of Horizontal Engines (Fig. 905).

H	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*
16	75	9	$27\frac{3}{4}$	51	75	8	22	$36\frac{1}{2}$	$15\frac{3}{4}$	—	—	—	—
20	90	$10\frac{3}{4}$	$29\frac{3}{4}$	57	93	10	$26\frac{3}{4}$	42	$18\frac{1}{2}$	$30\frac{1}{2}$	71	$17\frac{3}{4}$	$33\frac{1}{2}$
24	105	$12\frac{1}{2}$	$31\frac{1}{2}$	63	110	12	$31\frac{1}{2}$	48	$21\frac{1}{4}$	36	77	19	39
28	122	14	34	69	130	$14\frac{3}{4}$	$36\frac{1}{2}$	$54\frac{1}{2}$	$23\frac{3}{4}$	$40\frac{1}{2}$	82	20	47
32	136	$15\frac{1}{2}$	$35\frac{1}{2}$	75	150	17	41	60	26	45	98	$23\frac{3}{4}$	49
36	151	17	$39\frac{1}{2}$	83	165	20	$45\frac{1}{2}$	66	$28\frac{3}{4}$	50	108	$27\frac{1}{2}$	65
40	168	$18\frac{1}{2}$	$43\frac{1}{2}$	90	185	$21\frac{3}{4}$	50	$73\frac{1}{2}$	$30\frac{3}{4}$	$55\frac{1}{2}$	118	36	65

All Dimensions are given in Inches.

Floor Space required for Single Cylinder Horizontal Engines, with and without Condensers.



Figs. 906, 907.

TABLE 91.—**Floor Space Dimensions for Horizontal Engines** (Figs. 906, 907).

Without condenser, fig. 906.									With condenser fig. 907.	
H	A	B	*a*	*b*	*c*	*d*	F	G	C	L
16	126	40	32	63	20	24	110	209	—	—
20	181	50	36	67	20	28	130	268	231	318
24	196	60	$39\frac{1}{2}$	71	24	32	144	291	264	359
28	245	69	$43\frac{1}{2}$	75	24	36	160	344	300	399
32	277	119	47	79	28	40	173	384	342	449
36	307	130	51	83	28	44	189	418	384	495
40	333	142	55	87	28	48	205	448	420	535

All Dimensions are given in Inches.

Floor Space required, Horizontal Coupled Engines, with and without Condensers.

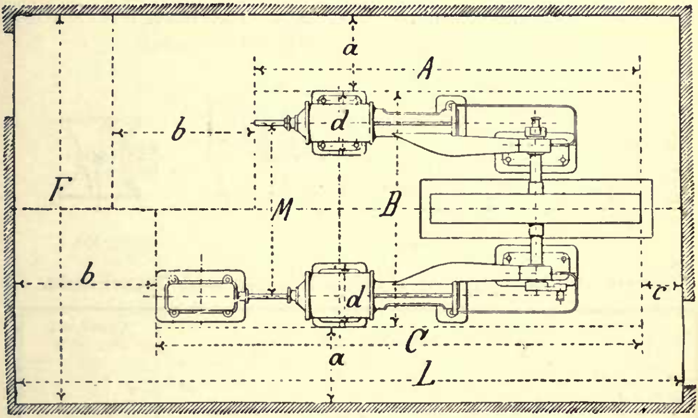


Fig. 908.

TABLE 92.—Floor Space Dimensions for Horizontal Coupled Engines (Fig. 908).

H	A	B	M	*a*	*b*	*c*	*d*	F	C	L
16	—	106	82	32	63	20	24	169	—	—
20	181	118	90	36	67	20	28	189	231	318
24	196	130	96	39½	71	24	32	209	264	359
28	245	142	104	43½	75	24	36	228	300	399
32	277	152	112	47	79	28	40	246	342	449
36	307	169	118	51	83	28	44	274	384	495
40	333	173	126	55	87	28	48	283	420	535

All Dimensions are given in Inches.

Weights of Engines in Detail.

In the following tables—

CI = Cast iron. St = Mild steel.
WI = Wrought iron. CS = Cast steel.
M = White metal. G = Gun metal.
B = Bolts.

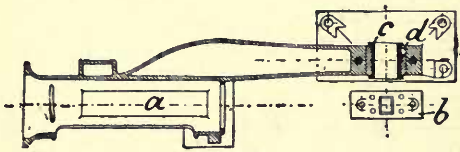


Fig. 909.

Fig. 910.

TABLE 93.—Weights of Engine Frames and Crossheads.

Engine.		Frame (Fig. 909).									Crosshead (Fig. 910).		
Stroke. H	Diameter of Piston. D	Frame. a	Bearing Cap. b	Brasses. c	Wedges. d	Frame complete with Bearing Cap.					Crosshead. e	Pin. f	Total.
		CI	CI	G	WI	CI	G	WI	B	Total.	CI	St.	
12	6	700	—	—	—	700	—	—	—	700	15	2	17
16	8	830	30	13	—	860	12	—	8	880	24	2	26
20	10	1060	57	18	11	1117	18	11	11	1157	42	4	46
24	12	1540	90	26	15	1630	26	15	15	1685	64	7	71
28	14	2100	140	37	17	2240	37	17	17	2310	90	11	101
32	16	2816	220	53	19	3036	53	19	18	3125	120	15	135
36	18	3740	290	70	22	4030	70	22	23	4145	155	19	174
40	20	5060	370	88	24	5430	88	24	23	5565	194	24	218
44	22	6600	470	110	28	7070	110	28	26	7234	234	31	265

Dimensions in Inches, Weights in Pounds.

Weights of Engines.

Pistons, and Piston Rods.

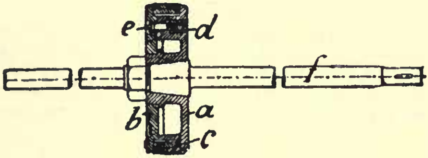


Fig. 911.

Table 94.—Weights of Pistons and Piston Rods.

Engine.								Fig. 911.			
Stroke.	Diameter of piston.	Piston body. *a*	Cover. *b*	Inner ring. *c*	Nuts. *d*	Screws. *e*	Piston rod. *f*	Piston complete with rod. *a* to *f*			
H	D	CI	CI	CI	G	St	St	CI	G	St	Total.
12	6	15	4	9	—	—	7	28	—	7	35
16	8	20	7	13	—	—	13	40	—	13	53
20	10	26	10	18	—	—	24	55	—	24	79
24	12	33	18	24	—	—	57	75	—	57	132
28	14	44	26	30	4	4	88	100	4	92	196
32	16	66	40	40	4	4	121	146	4	125	275
36	18	92	57	57	7	7	154	206	7	161	374
40	20	121	73	68	7	7	200	262	7	207	476
44	22	179	95	92	9	9	286	366	9	295	570
48	24	220	128	125	11	11	396	473	11	407	891
—	28	330	187	191	13	15	726	708	13	741	1462
—	32	450	253	260	18	20	1188	963	18	1208	2189
—	36	572	330	338	22	26	1738	1240	22	1764	3026
—	40	720	418	429	26	33	2420	1567	26	2453	4046

Dimensions in Inches, Weights in Pounds.

Weights of Engines.

Crank Shafts, Fly Wheels and Outside Bearing.

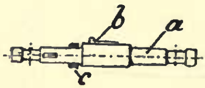


Fig. 912.

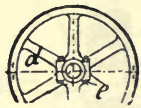


Fig. 913.

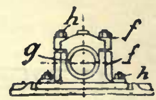


Fig. 914.

TABLE 95.—**Weights, Crank Shafts, Fly Wheels and Outside Bearings.**

Engine.		Crank Shaft (Fig. 912).				Fly Wheel (Fig. 913).			Outside Bearing (Fig. 914).			
Stroke.	Diameter of Piston.	Shaft. *a*	Key. *b*	Governor Pulley. *c*	*a+b+c.*	Fly Wheel. *d*	Bolts and Rings. *e*	*d+e.*	Block and Cap. *f*	Brasses. *g*	Bolts *h*	*f+g+h.*
H	D	St	St	CI	Total.	CI	WI	Total.	CI	G	WI	Total.
12	6	—	—	—	—	385	—	385	—	—	—	—
16	8	—	—	—	—	1045	—	1045	—	—	—	—
20	10	325	5·5	27·5	358	1650	—	1650	110	14	11	135
24	12	528	8	39	575	2530	—	2530	242	24	18	284
28	14	836	10	54	900	3520	44	3564	352	35	23	410
32	16	1255	14	66	1335	5012	48	5060	462	48	32	542
36	18	1760	15	80	1855	6548	52	6600	554	62	44	660
40	20	2244	18	92	2354	8963	57	9020	660	77	55	792
44	22	3256	24	110	3390	13138	62	13200	770	88	66	824

Dimensions in Inches, Weight in Pounds.

Weights of Engines.

Rider's Valves and Valve Rods.

Fig. 915.

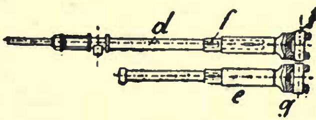


Fig. 916.

TABLE 96.—**Weight of Rider's Valves and Valve Rods.**

Engine.		Rider's valve, fig. 915.				Valve rods, fig. 916.				
Stroke.	Diameter of piston.	Main valve. *a*	Case. *b*	Expansion valve. *c*	$a+b+c$.	Valve rod. *d*	Guide piece. *e*	Cotters and bolts. *f*	Adjusting pieces. *g*	$d+e+f+g$.
H	D	CI	CI	CI	Total.	St	WI	St	G	Total.
12	6	—	—	—	—	7	9	1	1	18
16	8	—	—	—	—	9	13	1	1	24
20	10	42	6·6	4·4	53	13	22	2	1	38
24	12	66	13	9	88	18	35	2	1	56
28	14	100	22	15	137	24	52	3	3	82
32	16	145	33	26	204	31	70	4	5	110
36	18	194	46	35	275	40	88	7	5	140
40	20	242	62	48	352	48	105	7	5	165
44	22	290	84	66	440	57	123	8	5	193

Dimensions in Inches, Weights in Pounds.

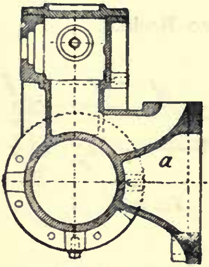


Fig. 917.

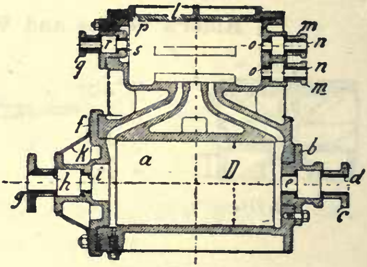


Fig. 918.

TABLE 97.—**Weights of Cylinders with**

Engine.		Cylinder.	Front cover.	Gland.	Bush.	Bush.	Back cover.	Gland.	Bush.	Bush.	Casing.	Valve chest covers.
Stroke	Diameter of piston.	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*
H	D	CI	CI	CI	G	G	CI	CI	G	G	CI	CI
12	6	286	13	4	0·7	0·4	22	—	—	—	—	13
16	8	475	22	7	1·2	0·7	40	—	—	—	—	28
20	10	690	35	9	1·5	1·2	59	—	—	—	—	72
24	12	1090	48	11	2·2	1·5	90	11	2·5	1·1	22	75
28	14	1540	64	14	2·8	2·2	143	15	3·5	1·5	26	134
32	16	2125	80	15	3·5	2·6	200	19	4	1·9	40	194
36	18	2860	94	18	4·2	3·3	308	24	4·5	2·4	48	275
40	20	3520	112	22	4·8	3·7	440	30	5	3·0	57	335
44	22	4225	132	26	5·5	4·4	572	37	5·5	3·7	66	400

Dimensions in Inches, Weights in Pounds.

Weights of Cylinders, Covers and Stuffing Boxes.

Note: in Table 97, Cylinders from 12″ to 16″ stroke have simple valve gear; those from 20″ to 44″ stroke have Rider's valve gear with closed valves.

Covers and Stuffing Boxes (Figs. 917, 918).

Gland. m	Bush. n	Bush. o	Stuffing-box. p	Gland. q	Bush. r	Bush. s	Casing. t	Cylinder complete with covers and glands. $a+\&c.+t$				
CI	G	G	CI	GI	G	G	WI	CI	G	WI	B	Total.
—	—	—	—	—	—	—	7	338	1·1	7	11	357
—	—	—	—	—	—	—	13	572	1·9	13	22	608
7	1·3	0·9	15	3	0·4	0·4	22	860	5·7	22	31	919
7	1·5	1·1	24	4	0·4	0·4	40	1382	11	40	42	1475
9	1·8	1·1	30	5	0·7	0·4	57	1980	14	57	55	2106
11	2·0	1·3	37	6	0·9	0·4	81	2727	17	81	70	2895
13	2·2	1·3	46	7	0·9	0·7	105	3693	19	105	88	3905
15	2·4	1·5	52	8	1·1	0·7	132	4591	22	132	102	4847
18	2·6	1·5	62	9	1·1	0·9	160	5547	25	160	119	5851

Note.—Cylinders with jackets weigh 18 per cent. more.

Weights of Engines.

Connecting Rods, Cranks, and Crank Pins.

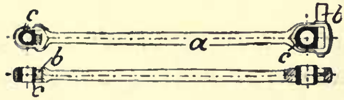


Fig. 919

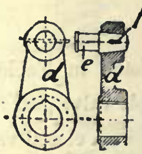


Fig. 920.

TABLE 98.—**Weights of Connecting Rods, Cranks, and Crank Pins.**

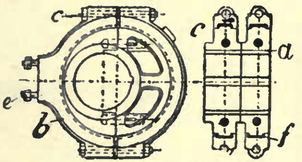
Engine.		Connecting-rods, fig. 919.				Cranks and crank-pins, fig. 920.			
Stroke.	Diameter of piston.	Connecting-rod. *a*	Cotter. *b*	Brasses. *c*	*a+b+c.*	Crank. *d*	Crank-pin. *e*	Cotter. *f*	*d+e+f.*
H	D	WI	St	G	Total.	WI	St	St	Total.
12	6	24	3	3	30	—	—	—	—
16	8	42	3	5	50	—	—	—	—
20	10	66	7	9	82	70	12	2	84
24	12	100	7	11	118	119	17	2	138
28	14	142	9	13	164	176	23	4	203
32	16	187	11	18	216	238	34	6	278
36	18	265	13	24	302	308	52	6	366
40	20	319	15	33	367	376	55	9	440
44	22	385	15	44	444	462	70	11	543

Note.—Cast-iron cranks are about 20% heavier.

Dimensions in Inches, Weights in Pounds.

Weights of Engines.

Eccentrics and Eccentric Rods.



Figs. 921, 922.

Fig. 923.

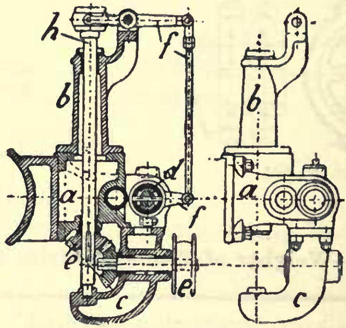
TABLE 99.—Weights of Two Eccentrics and Rods.

Engine.		Two eccentrics, figs. 921, 922.							Two eccentric rods, fig. 923.
Stroke.	Diameter of piston.	Eccentric sheaves. *a*	Eccentric rings. *b*	Bolts. *c*	Packings. *d*	Bolts. *e*	Linings. *f*	Two eccentrics complete *a* to *f*	*g*
H	D	CI	CI	WI	G	WI	M	Total.	WI
12	6	10	19	—	0·4	1·1	—	30·5	9
16	8	12	25	—	0·4	2·6	—	40	15
20	10	18	35	1·5	0·7	3	6·6	64·8	22
24	12	31	64	2	0·9	4	13	115	38
28	14	44	109	2	1	5	20	172	60
32	16	62	145	3	1	7	26	244	92
36	18	77	202	4	1	7	30	321	125
40	20	97	257	5	1·4	7·6	40	408	158
44	22	121	320	5	1·5	8·5	44	500	194

Dimensions in Inches, Weights in Pounds.

Weights of Engines.

Valve Rod Guides, with Governor Bracket, Bevel Wheels, Spindles, and Levers, exclusive of Governor and Spindle.



Figs. 924, 925.

Table 100.—Weights of Governor Bracket, &c. (Figs. 924, 925).

Engine. Stroke.	Engine. Diameter of piston.	Valve-rod guides. *a*	Governor bracket. *b*	Bracket for footstep. *c*	Bevel wheels and pulley. *e*	Lever. *f*	Spindle. *g*	Bushes. *d*	Weight complete, as in figs. 924, 925. *a* to *g*					
H	D	CI	CI	CI	CI	WI	St	G	CI	G	WI	St	B	Total.
20	10	70	26	33	18	31	11	7	160	18	11	7	4	200
24	12	121	40	40	24	37	15	7	238	24	15	7	7	291
28	14	165	50	46	33	44	22	9	305	33	22	9	9	378
32	16	215	66	53	40	53	26	9	387	40	26	9	11	473
36	18	268	84	62	46	59	33	11	473	46	33	11	13	576
40	20	325	103	68	53	64	37	11	560	53	37	11	15	676
44	22	375	121	77	62	70	44	13	643	62	44	13	18	780

Dimensions in Inches, Weights in Pounds.

Weights of Engines.

Foundation Bolts and Plates, Guard Rail, and Barring Apparatus.

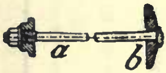


Fig. 926.

Fig. 927.

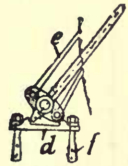


Fig. 928.

Table 101.—Weights of Foundation Bolts, &c.

Engine.		Foundation bolts and plates, fig. 926.			Guard rail, fig. 927.	Barring apparatus, fig. 928.			
Stroke.	Diameter of piston.	Plates. *b*	Bolts. *a*	*b+a*	*c*	Bracket. *d*	Lever. *e*	Lewis bolts. *f*	Barring apparatus complete. *d+e+f*
		CI	WI	Total.	WI	CI	WI	WI	Total.
12	6	44	35	79	—	—	—	—	—
16	8	44	48	92	—	—	—	—	—
20	10	140	66	206	11	44	22	13	79
24	12	176	132	308	15	44	22	13	79
28	14	330	198	528	22	55	26	18	99
32	16	396	264	660	29	55	26	18	99
36	18	616	352	968	35	77	35	26	158
40	20	616	440	1056	44	77	35	26	158
44	22	748	550	1290	55	77	35	26	158

Dimensions in Inches, Weights in Pounds.

TABLE 102.—**Summary of the Tables of Weights, 92 to 101.**

Engine. Stroke.	Engine. Diameter of piston.	Cast-iron.	Gun-metal.	Wrought-iron.	Steel	Bolts.	Weight without fly-wheel or governor.	Governor.	Fly-wheel.	Sundry fittings, valves, &c.	Weight with Rider's valve gear without condenser.	Condenser.	Weight with Condenser.	Feed-pump with eccentric.
H	D	CI	G	WI	St	WI	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.
20	10	2663	66	326	405	75	3535	110	1650	66	5361	—	—	88
24	12	4073	97	532	651	100	5453	176	2530	88	8247	1760	10007	110
28	14	5787	140	774	1020	125	7846	242	3564	110	11762	3080	14842	155
32	16	7806	185	1038	1502	158	10689	308	5060	155	16212	3520	19732	220
36	18	10428	233	1355	2075	200	14291	374	6600	176	21441	4400	25841	308
40	20	13188	287	1676	2625	230	18006	462	9020	198	27686	5500	33186	418
44	22	16380	343	2045	3782	264	22814	550	13200	220	36734	6600	43384	528
Per cent. of material		75·5	1·8	9·2	11·4	2·1	100	Without fly-wheel for engines up to 20″ stroke.						
		71·9	1·5	8·9	16·5	1·2	100	„ „ „ „ 44″ stroke.						

Dimensions in Inches, Weights in Pounds.

TABLE 103.—**Cost of Patterns, complete with Core Boxes. For Engines of 20 to 44 Inches stroke, with Rider's Valve Gear.**

Stroke	20	24	28	32	36	40	44
Diameter of piston	10	12	14	16	18	20	22
Frame with bearing cap and brasses	250	264	285	300	320	350	390
Crosshead	25	30	38	44	50	60	65
Connecting-rod brasses	5	6	7	8	9	10	12
Cylinder with cover and glands	195	210	220	240	260	280	300
Piston with rings and cover	24	26	29	31	35	40	45
Outside main bearing	33	36	41	46	55	60	70
Rider's valve	70	75	80	85	95	105	115
Adjusting nuts for valve gear	1	1	1	2	2	2	2
Eccentrics with rings	30	35	40	45	50	57	65
Valve rod guides with governor bracket, bevel wheels and pulley	70	80	90	105	115	130	150
Foundation plates	3	3	3·5	3·5	3·5	3·5	3·5
Barring apparatus	9	9	10·5	10·5	12·5	12·5	12·5
Feed pump	120	130	140	150	160	170	185
Total cost of patterns	835	905	985	1070	1165	1280	1415

The dimensions are given in inches, the cost in marks, the value of a mark being about $11\frac{3}{4}$ English pence.
The patterns admit of the engine being built either right or left-handed.

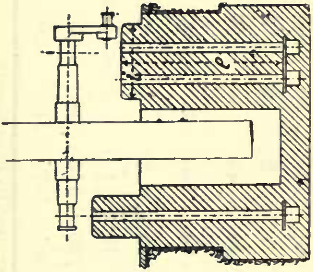


Fig. 931.

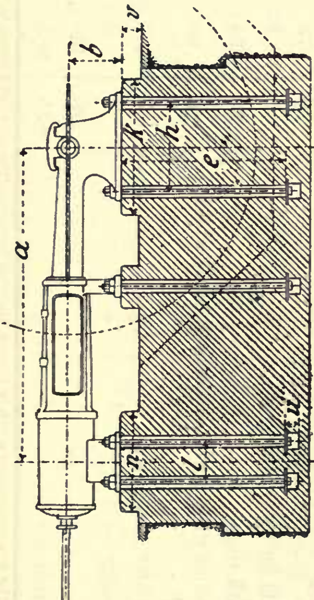


Fig. 929.

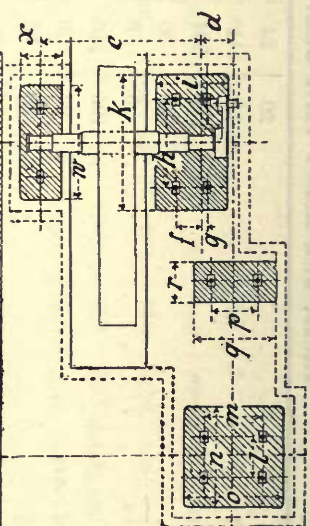


Fig. 930.

TABLE 104.—**Dimensions of Engine Foundations for Horizontal Engines** (Figs. 929—931).

H	D	*a*	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*k*	*l*	*m*	*n*	*o*	*p*	*q*	*r*	*u*	*w*	*x*	M*
20	10	76	14½	51	9	55	7	2	22	20½	36¼	—	15½	23⅝	23⅝	—	—	—	4	39½	15¾	6·5
24	12	90	18	57	10⅝	63	8	2¼	25⅝	20½	41	13¾	17¾	27¾	27¾	—	—	—	4¾	43½	16¾	10·5
28	14	106	20	63	12½	71	9	2¾	30	26	46½	15¾	20½	31½	31½	—	—	—	5½	47½	17¾	14·4
32	16	121½	22	70	14	75	10¼	3⅜	34	31	52	17¾	22¾	35½	35½	20½	35½	20	6¼	52	18¾	19·6
36	18	135	24	75	15½	79	11¾	4	37	31	57	18¾	25¼	39½	39½	22¼	38⅝	20¾	7	55	19¾	26
40	20	151	26	82	17	84	12½	4⅜	40½	36⅝	62	20	27¾	43½	43½	22¾	42	22¼	8	60	20¾	32·7
44	22	168	28	90	18½	90	14	5	44½	36⅝	69	21¾	29½	47½	47½	25¼	46	23⅝	8	64	21¾	39·3

* (M), the contents of Foundations in Cubic Yards; all other Dimensions in Inches.

Engine Foundations.

The depth of is in some measure determined by the nature of the ground. It may be necessary to go down a great depth to reach a solid basis whereon to build, and to fill up with coarse concrete up to the level of the foundation bolts and then begin the foundation proper. The material depends on the district; in some districts concrete is the most ready material, in others brick or masonry. In cases where a building for manufacturing purposes has to be erected on very loose ground and where concrete is easily obtained, it is a good plan to floor the whole area occupied by factory, engine and boiler house with a bed of concrete of sufficient thickness, and to build all structures on this bed. Settlement is very unlikely to take place with such a system, as the weight is distributed over the largest possible area of the loose ground.

For brick foundations each course should be grouted in thoroughly with good cement, and all bolt holes formed by inserting wood bars about 3″ to 4″ square, the lower end of these resting in the foundation plates, and the upper ends in a template fixed in places; the wood bars are afterwards withdrawn and the bolts inserted. When the engine is in place and bolted down, the spaces round the bolts may be filled in with cement grout. In concrete the same plan with regard to the bolt holes may be used.

With masonry foundations the holes are usually cut through. A top course of stone is very good on brick foundations, but is expensive in districts far from the quarries.

When the first motion drive from the engine is through spur or bevel wheels, great care should be taken with the foundations, and if of concrete as much time as possible should be allowed after building before the engine is started. The most convenient form of foundation bolt is perhaps the cottar bolt, as it can be passed down the hole in the foundation without trouble, and should such an accident happen as a defective bolt, it can be replaced. In small work ordinary bolts with heads and nuts are much used, being built in the foundation.

Note.—The particulars given in Tables 85 to 104, and the corresponding illustrations, are from engines of continental make, and are given exactly as they appear in the 3rd edition of M. Haeder's book, "Die Dampfmaschine," except that the dimensions are converted into English measures.

SECTION VII.

PARTICULARS OF STANDARD ENGINES BY ENGLISH MAKERS.

The following Tables, 105 to 129, give the particulars of small and medium-sized engines by English makers who manufacture large numbers of these engines, and have given every attention to details of construction and design; they may therefore be taken as examples of best English practice in the various types of which tables and illustrations are given.

Descriptions of some of the special valve gears applied to these engines are given in the Section on Valve Gears, and examples of indicator diagrams from actual practice are given in the Section on Indicator Diagrams.

The particulars in each table were supplied by the makers themselves.

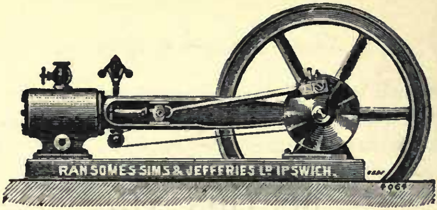


Fig. 932.

Table 105.—**Particulars of Messrs. Ransomes, Sims & Jefferies' Long-Stroke Horizontal Engine without Condenser** (Fig. 932).

Diameter of cylinder	10	11	12	13	14	15	16	17	18	19
Length of stroke	20	20	24	24	28	28	32	32	36	36
Revolutions per minute	110	110	90	90	80	80	70	70	60	60
Boiler-pressure per □″	80	80	80	80	80	80	80	80	80	80
Best average H.P.	30	40	45	55	65	75	85	95	100	115
Diameter of crank-shaft	4	$4\frac{1}{4}$	$4\frac{3}{4}$	5	$5\frac{5}{8}$	$5\frac{7}{8}$	$6\frac{1}{2}$	$6\frac{3}{4}$	$7\frac{1}{4}$	$7\frac{1}{2}$
Diameter of fly-wheel	84	84	96	96	108	108	120	120	144	144
Width of fly-wheel	8	8	10	10	12	12	14	14	16	16
Weight of fly-wheel, cwts.	24	24	46	46	60	60	90	90	110	110
Height to centre of cylinder	$16\frac{1}{2}$	$16\frac{1}{2}$	19	19	22	22	23	23	30	30
Total length, including fly-wheel	153	153	181	181	210	210	238	238	281	281
Total width to outer bearing	74	74	$82\frac{1}{2}$	$82\frac{1}{2}$	96	96	107	107	131	131
Approx. weight packed, cwts.	72	76	106	112	163	172	218	230	291	307

All Dimensions are given in Inches.

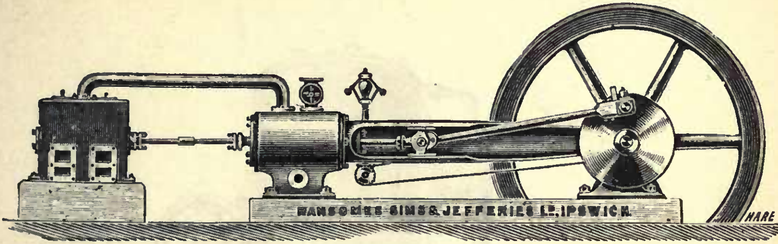


Fig. 933.

TABLE 106.—**Particulars of Messrs. Ransomes, Sims & Jefferies' Long-Stroke Engine with Condenser** (Fig. 933).

Diameter of cylinder	10	11	12	13	14	15	16	17	18	19
Length of stroke	20	20	24	24	28	28	32	32	36	36
Revolutions per minute	110	110	90	90	80	80	70	70	60	60
Boiler pressure	80	80	80	80	80	80	80	80	80	80
Best average H.P.	40	50	55	65	80	90	105	115	125	140
Diameter of fly-wheel	84	84	96	96	108	108	120	120	144	144
Width of fly-wheel	8	8	10	10	12	12	14	14	16	16
Weight of fly-wheel, cwts.	24	24	46	46	60	60	90	90	110	110
Diameter of crank-shaft	4	4¼	4¾	5	5⅝	5⅞	6½	6¾	7¼	7½
Diameter of air-pump bucket	4	4	4	4	5½	5½	5½	5½	7	7
Height to centre of cylinder	16½	16½	19	19	22	22	23	23	30	30
Total length, including fly-wheel	241	241	272	272	313	313	345	345	420	420
Width to outer bearing	78½	78½	87	87	102	102	111	111	135	135
Approx. weight packed, cwts.	89	93	126	132	188	197	257	269	348	364

All Dimensions are given in Inches.

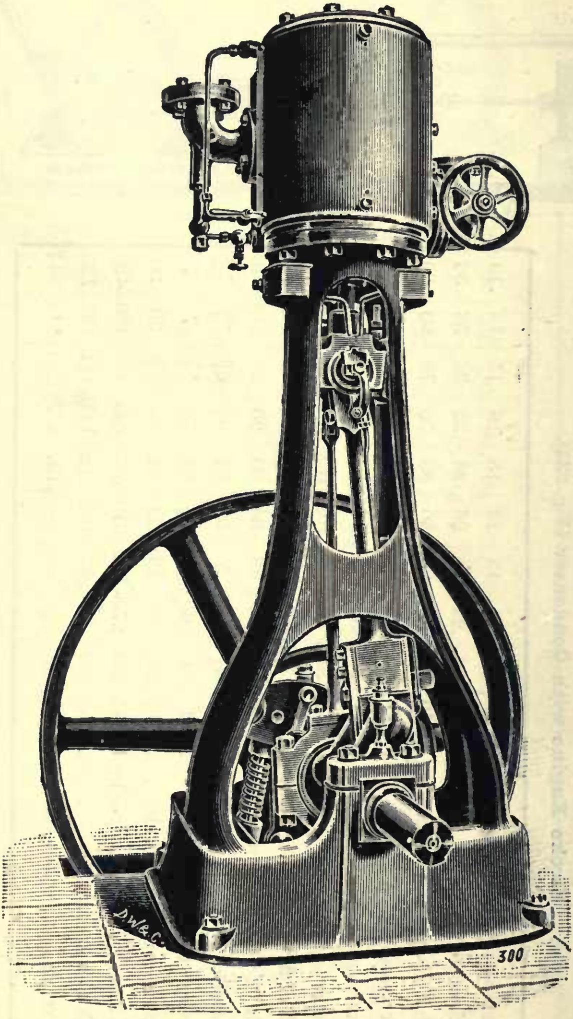


Fig. 934.

TABLE 107.—**Particulars of Messrs. Marshall's Vertical Short-Stroke Engines** (Fig. 934).

Diameter of cylinder	$4\frac{1}{2}$	$5\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{2}$	8	9	10	11	12	13
Stroke	8	8	10	10	12	12	14	14	16	16
Revolutions per minute	260	260	210	210	175	175	150	150	130	130
Boiler-pressure, lbs. per □″	100	100	100	100	100	100	100	100	100	100
Best average H.P.*	5	$7\frac{1}{2}$	$10\frac{1}{2}$	$13\frac{3}{4}$	16	20	25	$30\frac{1}{2}$	36	42
Diameter of fly-wheel	32	32	$41\frac{1}{2}$	$41\frac{1}{2}$	$52\frac{1}{2}$	$52\frac{1}{2}$	60	60	66	66
Width of fly-wheel	5	5	6	6	$7\frac{1}{2}$	$7\frac{1}{2}$	$9\frac{1}{2}$	$9\frac{1}{2}$	$11\frac{1}{4}$	$11\frac{1}{4}$
Weight of fly-wheel, cwts.	$2\frac{1}{4}$	$2\frac{1}{4}$	$4\frac{3}{4}$	$4\frac{3}{4}$	$7\frac{3}{4}$	$7\frac{3}{4}$	11	11	17	17
Diameter of crank-shaft	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{1}{2}$	$4\frac{1}{2}$
Total height of engine	$55\frac{5}{16}$	$55\frac{5}{16}$	$66\frac{1}{4}$	$66\frac{1}{4}$	$78\frac{1}{8}$	$78\frac{1}{8}$	$89\frac{1}{4}$	$89\frac{1}{4}$	$105\frac{1}{8}$	$105\frac{1}{8}$
Floor space, width†	32	32	$41\frac{1}{2}$	$41\frac{1}{2}$	$52\frac{1}{2}$	$52\frac{1}{2}$	60	60	66	66
Floor space, length†	$43\frac{1}{2}$	$43\frac{1}{2}$	52	52	63	63	70	70	70	70
Height of centre from ground	$8\frac{1}{2}$	$8\frac{1}{2}$	10	10	$11\frac{1}{2}$	$11\frac{1}{2}$	$13\frac{1}{2}$	$13\frac{1}{2}$	15	5

*Effective H.P. † Including space for fly-wheel.

All Dimensions are given in Inches.

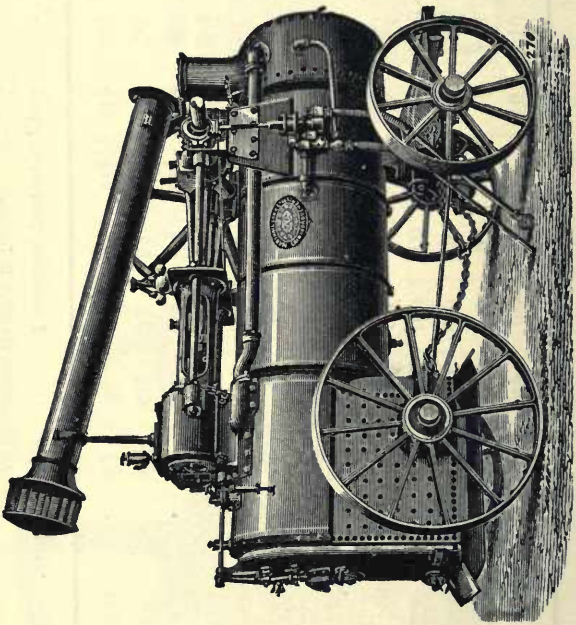


Fig. 935.

TABLE 108.—**Particulars of Messrs. Marshall's Double Cylinder Portable Engines (Fig. 935).**

Nominal horse-power	14	16	20	25	30	35	40
Diameter of cylinders (each)	9	10	11	12	13	13¾	14½
Length of stroke	14	14	16	16	18	18	20
Revolutions per minute	115	115	95	95	85	85	80
Boiler pressure per □″	90	90	90	90	90	90	90
Best average H.P.	35	40	50	63	75	88	100
Diameter of crank-shaft	4	4	4¾	5	5½	6	6¼
Diameter of fly-wheel	67	67	72	72	79	79	99
Width of fly-wheel	9½	9½	13	13	15	15	15
Weight of fly-wheel in cwts.	11	11	15¼	15¼	19	19	30
Total heating-surface of boiler, ordinary, sq. feet	280·52	314·59	389·81	422·62	471·092	560·907	622·685
Total heating-surface of boiler, colonial, sq. feet	285·73	321·94	395·32	429·241	479·177	569·114	633·683
Area of grate, ordinary	9·04	9·68	12·69	13·810	15·265	17·747	20·035
Area of grate, colonial	10·74	11·93	14·34	15·910	17·747	20·285	23·411
Diameter of boiler barrel	40¾	42¾	45	49⅛	51	51	55
Total length of boiler, ordinary	150	158	174¼	178	189	203	211
Total length of boiler, colonial	157	167	180	185	197	211	221
Length of boiler barrel	91½	94	100¼	100	105	111	115

All Dimensions are given in Inches.

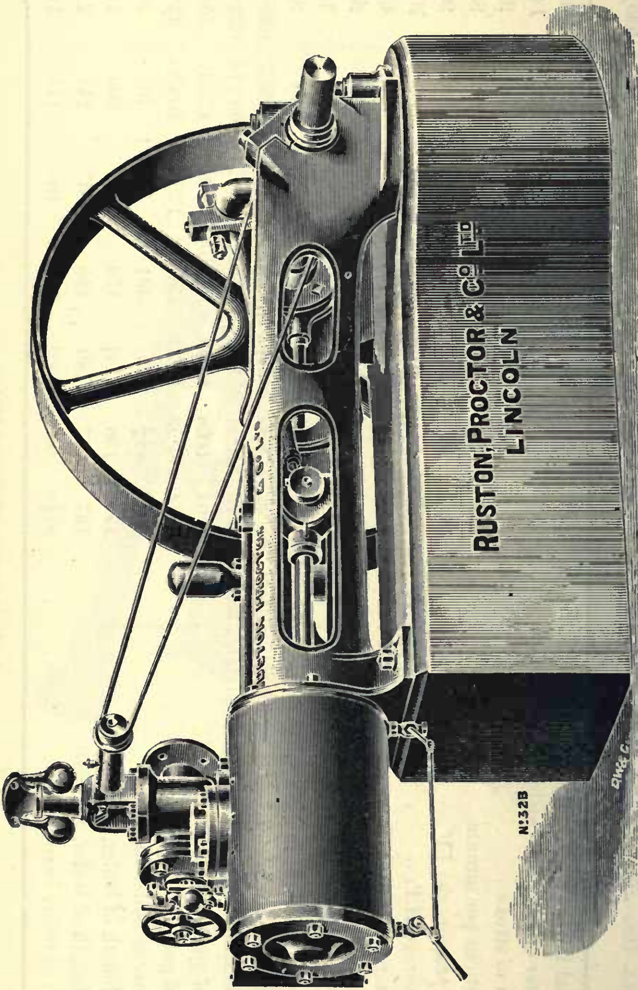


Fig. 936.

TABLE 109.—**Particulars of Messrs. Ruston & Proctor's Horizontal Self-Contained Short-Stroke Engines** (Fig. 936).

Diameter of cylinder	$5\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{4}$	8	$8\frac{3}{4}$	$9\frac{1}{2}$	$10\frac{1}{2}$	$11\frac{1}{2}$
Length of stroke	8	10	10	12	12	14	14	16
Revolutions per minute	190	160	160	140	140	125	125	110
Boiler pressure, lbs. per square inch	80	80	80	80	80	80	80	80
Best average H.P.	7	11	13	17	20	24	30	36
Diameter of crank-shaft	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$
Diameter of fly-wheel	33	36	39	40	45	48	54	60
Width of fly-wheel	$4\frac{1}{2}$	$4\frac{1}{2}$	5	5	6	6	7	$7\frac{1}{2}$
Weight in cwts.	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	$4\frac{1}{4}$	$5\frac{1}{4}$	$6\frac{3}{4}$	8
Height to centre	17	18	18	28	28	30	30	30
Total length	64	74	76	88	89	100	103	116
Total width	32	36	36	42	44	50	52	60
Approx. weight unpacked, cwts.	7	9	11	15	17	24	27	35

All Dimensions are given in Inches.

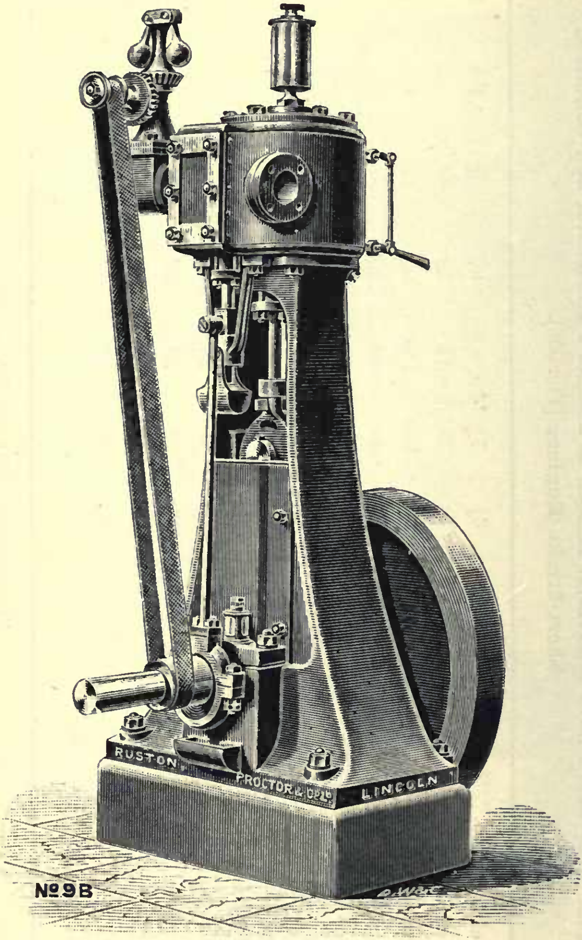


Fig. 937.

TABLE 110.—Particulars of Messrs. Ruston & Proctor's Vertical Short-Stroke Engines (Fig. 937).

Diameter of cylinder	6			$6\frac{3}{4}$			$7\frac{1}{2}$			$8\frac{1}{2}$			$9\frac{1}{2}$			$10\frac{1}{2}$		
Length of stroke	6			6			8			8			10			10		
Revolutions per minute	150	200	250	150	200	250	150	200	250	150	200	250	150	200	250	150	200	250
Best average H.P.	5	7	9	7	9	11	10	14	18	14	18	23	22	29	36	27	35	44
Boiler pressure, lbs. per sq. in.	80			80			80			80			80			80		
Diameter of fly-wheel	30			30			33			33			42			42		
Width of fly-wheel	4			5			$4\frac{1}{2}$			6			6			8		
Weight of fly-wheel in cwts.	2			$3\frac{1}{2}$			5			6			$7\frac{1}{2}$			9		
Diameter of crank-shaft	$2\frac{1}{4}$			$2\frac{1}{2}$			3			$3\frac{1}{4}$			$3\frac{1}{2}$			$3\frac{3}{4}$		
Total width of engine	30			30			33			33			42			42		
Total height from ground	57			57			66			66			78			78		
Total length	34			34			40			40			48			48		
Approx. weight in cwts.	12			14			16			17			20			22		

All Dimensions are given in Inches.

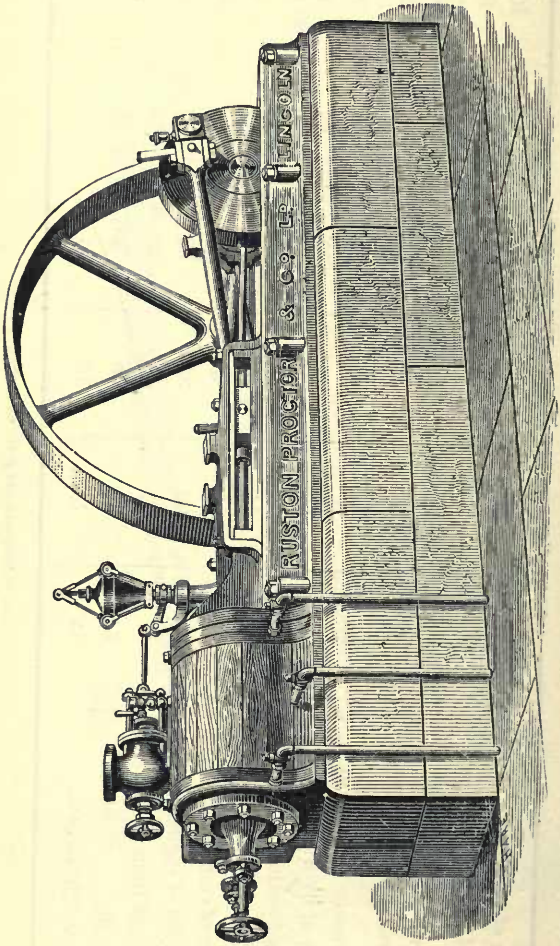


Fig. 938.

TABLE 111.—**Particulars of Messrs. Ruston & Proctor's Long-Stroke Horizontal Engines** (Fig. 938).

Diameter of cylinder	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	26	27
Length of stroke . .	16	16	20	20	24	24	28	28	32	32	36	36	40	40	44	44	52	52
Revs. per minute .	135	135	110	110	90	90	80	80	70	70	65	65	60	60	60	60	60	60
Boiler press., lbs. per square inch	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Best average H.P. .	15	20	25	30	35	40	48	55	65	75	85	95	110	120	140	160	230	250
Diam. of fly-wheel .	62	66	85	87	99	101	116	116	120	120	144	144	146	146	168	168	177	177
Width of fly-wheel .	7	9	8	8	10	10	12	14	14	16	16	18	20	22	22	24	28	30
Weight of fly-wheel, cwts. . .	10	$12\frac{1}{2}$	$21\frac{1}{2}$	$21\frac{1}{2}$	$29\frac{1}{2}$	$29\frac{1}{2}$	48	54	66	74	$89\frac{1}{2}$	105	130	—	—	—	—	—
Diam. of crank-shaft	3	3	$3\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$5\frac{1}{4}$	$5\frac{1}{4}$	6	6	$6\frac{3}{4}$	$6\frac{3}{4}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$8\frac{1}{4}$	$8\frac{1}{4}$	$9\frac{3}{4}$	$9\frac{3}{4}$
Total length . .	142	142	160	160	188	188	218	218	240	240	274	274	294	294	335	335	378	378
Total width . .	75	75	84	84	93	93	102	102	114	114	123	123	132	132	141	141	150	150
Approx. weight, cwts.	48	50	76	80	106	112	156	162	204	214	310	323	338	358	473	493	700	720
Height to centre . .	36	36	36	36	36	36	36	36	36	36	36	36	36	36	36	36	36	36

All Dimensions are given in Inches.

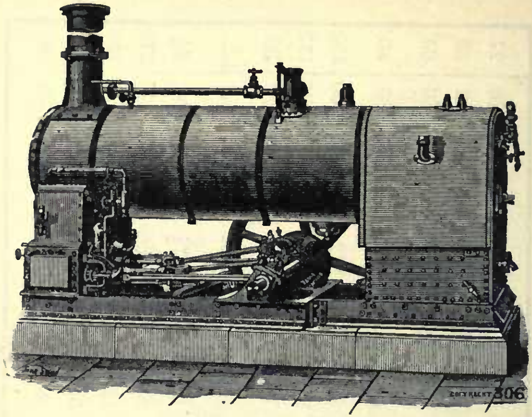


Fig. 939.

TABLE 112.—**Particulars of Compound Under-type Engines by Messrs. E. R. & F. Turner** (Fig. 939).

Diameter of small cylinder . .	7	8	9	10
Diameter of large cylinder . . .	$11\frac{1}{4}$	$12\frac{3}{4}$	14	16
Length of stroke	12	14	16	18
Revolutions per minute . . .	180	155	135	120
Piston speed in feet per minute .	360	360	360	360
Best average indicated horse-power .	38	50	63	78
Diameter of crank-shaft	$3\frac{1}{2}$	$3\frac{3}{4}$	$4\frac{1}{4}$	5
Diameter of fly-wheel	60	72	72	84
Width of fly-wheel	8	9	$10\frac{1}{2}$	12
Weight of fly-wheel, cwt. . . .	8	13	16	22
Height to centre of boiler . .	$42\frac{1}{2}$	49	58	$66\frac{1}{4}$
Height to centre of crank-shaft . .	13	$14\frac{1}{2}$	17	$19\frac{5}{8}$
Diameter of boiler-barrel . . .	$34\frac{1}{2}$	$36\frac{3}{4}$	$40\frac{1}{2}$	47
Length of boiler-barrel . . .	112	$121\frac{1}{2}$	132	142
Heating-surface in square feet . .	187·5	250	312·8	390
Area of fire-grate in square feet .	5·06	6·6	8·2	10·25
Width over all	86	91	102	116
Working pressure in lbs. per sq. in. .	140	140	140	140

All Dimensions in Inches.

N.B.—The heating-surfaces and boiler dimensions in the table are given for ordinary English coal; for inferior and colonial coal, about 25 per cent. should be added to grate area and fire-box heating surface.

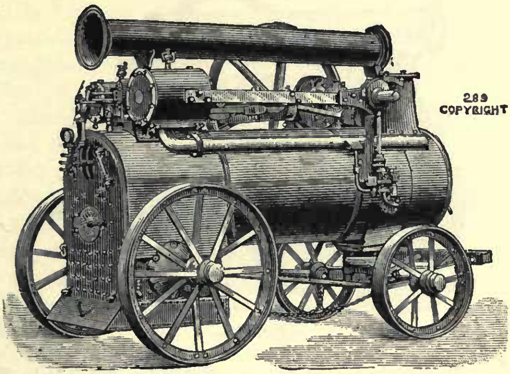


Fig. 940.

Table 113.—Particulars of Compound Portable Engines by Messrs. E. R. & F. Turner (Fig. 940).

Diameter of small cylinder	7	8	9
Diameter of large cylinder	11¼	12¾	14
Length of stroke	12	14	16
Revolutions per minute	180	155	135
Piston speed in feet per minute	360	360	360
Best average indicated horse-power	36	48	60
Diameter of crank-shaft	3¾	3¾	4¼
Diameter of fly-wheel	60	72	72
Width of fly-wheel	8	9	10½
Weight of fly-wheel in cwt.	8	13	16
Diameter of boiler-barrel	34½	36¾	40½
Length of boiler over all	147	161½	179
Heating-surface in square feet	187·5	250	312·8
Area of grate in square feet	5·06	6·6	8·2
Length over all	158	173	192
Width over axles	87	89	93
Height to top of boiler	109	118	126
Working-pressure in lbs. per sq. in.	140	140	140

Dimensions in Inches.

See remarks as to heating-surface in Table 112.

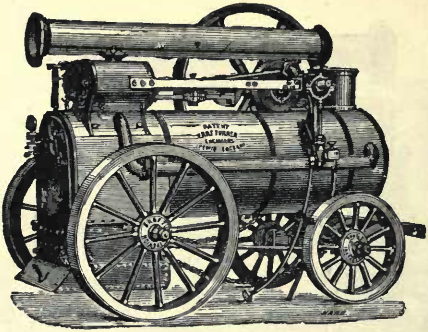


Fig. 941.

TABLE 114.—**Particulars of Single-Cylinder Portable Engines by Messrs. E. R. & F. Turner** (Fig. 941).

Diameter of cylinder .	6	$6\frac{3}{4}$	$7\frac{1}{2}$	$8\frac{1}{2}$	9	$9\frac{1}{2}$	$10\frac{1}{2}$
Length of stroke . .	9	$10\frac{1}{2}$	$10\frac{1}{2}$	12	12	12	12
Revs. per minute .	180	150	150	130	130	130	130
Piston speed in feet per minute . .	270	263	263	260	260	260	260
Best aver. indic. H.P.	6	8	10	12	14	16	20
Diam. of crank-shaft .	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	3	$3\frac{1}{4}$
Diam. of fly-wheel .	45	52	52	60	60	60	60
Width of fly-wheel .	$4\frac{1}{2}$	5	5	6	6	6	7
Weight of do. cwt.	3	$3\frac{1}{2}$	$3\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{3}{4}$	$6\frac{3}{4}$	$7\frac{1}{2}$
Diam. of boiler-barrel	24	29	29	$30\frac{3}{8}$	$33\frac{1}{2}$	$33\frac{1}{2}$	$34\frac{1}{2}$
Length of do. . .	62	67	81	83	$80\frac{5}{8}$	$85\frac{3}{8}$	$95\frac{1}{4}$
Heating-surface, sq. ft.	59·4	78·36	96·37	109·03	128	144·27	172·29
Area of fire-grate, sq. ft.	2·02	2·38	2·96	3·27	3·61	4·12	5·01
Approx. weight, cwt.	40	50	57	65	74	82	95
Length over all . .	96	112	120	126	126	132	138
Width over axles . .	58	66	66	73	77	79	82
Height to top fly-wheel	75	93	93	99	104	106	110
Working-pressure in lbs. per sq. inch .	80	80	80	80	80	80	80

Dimensions in Inches.

See remarks as to heating-surface, &c., in Table 112.

Portable engines have been and are still manufactured in large numbers in England, and have settled down into few varieties. In the first portable engines, pressures were low and there was no objection to throwing the whole strain of the working on to the barrel plates of the boiler. Now, however, with higher pressures the tendency has been to relieve the boiler as much as possible of the working strain due to the pressure of steam on the piston, by adding stay bars between the cylinder and the crank shaft bearings, or in some cases of mounting the engine on a frame of its own which is then bolted to stools rivetted on the boiler ; this would seem to commend itself on account of the facilities of packing for export, and facility of carriage up difficult countries.

From the portable engine the road locomotive or traction engine has been developed, and the number of these for farm purposes increases every year ; they do not differ greatly from the portable engine as far as the engine itself is concerned, but have the addition of gearing and travelling wheels of various designs.

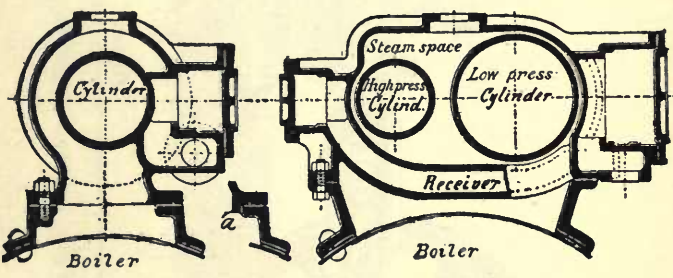


Fig. 941A.

Compound portable engines have come into favour for the larger sizes ; the cylinders of these are usually arranged side by side, Fig. 941A, and the cranks at right angles, the receiver taking the form of a pipe under the cylinders and concealed in the casing so that the receiver has the advantage of being well protected from cold air, and from its position close to the boiler top is kept in a hot atmosphere. The steam pipe in portable engines is reduced to a minimum in length and the steam jacket can be readily drained direct into the boiler ; these are great advantages and no doubt contribute to the very economical results which are obtained from portable engines.

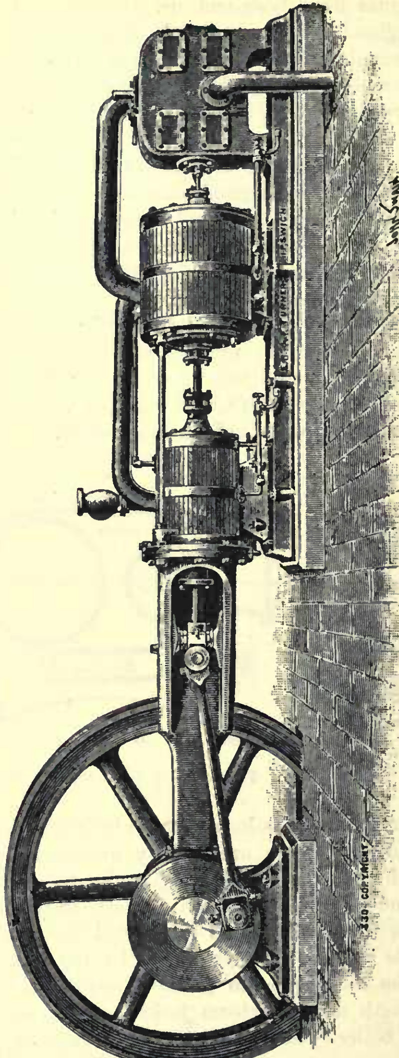


Fig. 942.

TABLE 115.—**Particulars of Tandem Compound Engines by Messrs. E R. & F. Turner** (Fig. 942).

Best average indicated H.P.	44	55	68	82	96	110	124	140
Diameter of small cylinder	9	10	11	12	13	14	15	16
Diameter of large cylinder	14	16	18	20	21	23	25	26
Length of stroke	16	20	20	24	24	28	28	36
Revolutions per minute	150	120	120	100	100	86	86	70
Diameter of fly-wheel	84	96	96	114	114	129	129	144
Width of fly-wheel	8	9	9	11	11	12	12	14
Weight of fly-wheel in cwts.	30	50	50	62	62	80	80	100
Diameter of crank-shaft at journal	$4\frac{1}{4}$	5	5	6	6	$6\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{2}$
Diameter of crank-shaft at fly-wheel	$5\frac{1}{2}$	$6\frac{3}{8}$	$6\frac{3}{8}$	$7\frac{1}{2}$	$7\frac{1}{2}$	$8\frac{1}{4}$	$8\frac{1}{4}$	$9\frac{1}{2}$
Height of centre from ground	28	30	30	32	32	34	34	36

Dimensions are given in Inches.

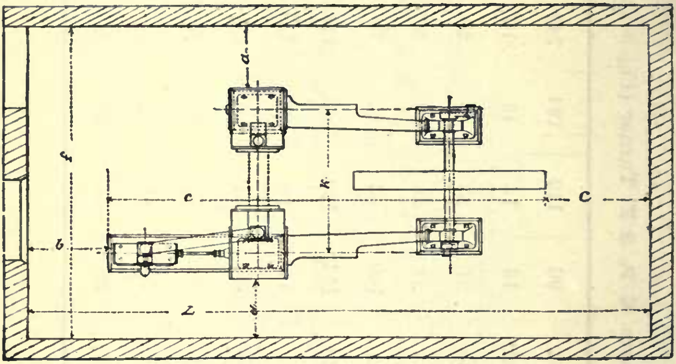


Fig. 943.

Table 116.—Particulars of Floor-Space occupied by Coupled Compound Condensing Engines by Messrs. E. R. & F. Turner (Fig. 943).

Diameter of cylinders . .	9 & 14	10 & 16	11 & 18	12 & 20	13 & 21	14 & 23	15 & 25	16 & 26
Length of stroke . .	16	20	20	24	24	28	28	36
Distance from cyls. to wall *a*	30	30	30	30	30	30	30	30
" " " *b*	36	36	36	36	36	36	36	36
" " " *c*	12	12	12	18	18	18	18	18
Total length of engine C	—	201	201	237	237	272	272	342
Total width of engine room *f*	173	176	176	188	190	199	202	210
Distance between centres of cylinders . . K	78	78	78	84	84	90	90	96
Total length of engine room L	—	249	249	285	285	320	320	390

Dimensions are given in Inches.

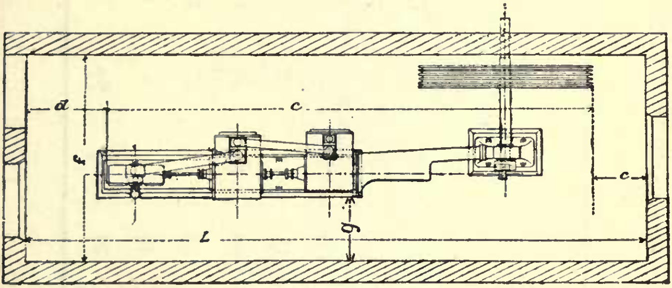


Fig. 944.

Table 117.—**Particulars of Floor-Space Occupied by Tandem Compound Condensing Engines by Messrs. E. R. & F. Turner** (Fig. 944).

Diameter of cylinders . .	9 & 14	10 & 16	11 & 18	12 & 20	13 & 21	14 & 23	15 & 25	16 & 26
Stroke of piston . . .	16	20	20	24	24	28	28	36
Total length of engine-room L	—	327	327	381	381	—	—	—
Total width of engine-room F	—	107	107	121	121	—	—	—
Distance from engine to wall *c*	15	15	15	15	15	15	15	15
,, ,, ,, *d*	36	36	36	42	42	42	42	48
,, ,, ,, *g*	30	30	30	30	30	30	30	30
Length over all of engine C	—	276	276	324	324	—	—	—

Dimensions are given in Inches.

TABLE 118.—**Particulars of Compound Coupled Condensing Engines by Messrs. E. R. & F. Turner** (see also Table 116).

Best average indicated H.P.	44	55	68	82	96	110	124	140
Diameter of small cylinder	9	10	11	12	13	14	15	16
Diameter of large cylinder	14	16	18	20	21	23	25	26
Length of stroke	16	20	20	24	24	28	28	36
Revolutions per minute	150	120	120	100	100	86	86	70
Diameter of fly-wheel	84	96	96	114	114	129	129	144
Width of fly-wheel	8	9	9	11	11	12	12	14
Weight of fly-wheel in cwts.	25	40	40	50	50	64	64	80
Height of centre from ground	24	24	24	26	26	27	27	33
Diameter of crank-shaft at journal	$3\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{1}{4}$	$4\frac{1}{4}$	5	5	6
Diameter of crank-shaft at fly-wheel	$4\frac{1}{4}$	$4\frac{3}{4}$	$4\frac{3}{4}$	$5\frac{3}{4}$	$5\frac{3}{4}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{2}$

Dimensions are given in Inches.

Fig. 945

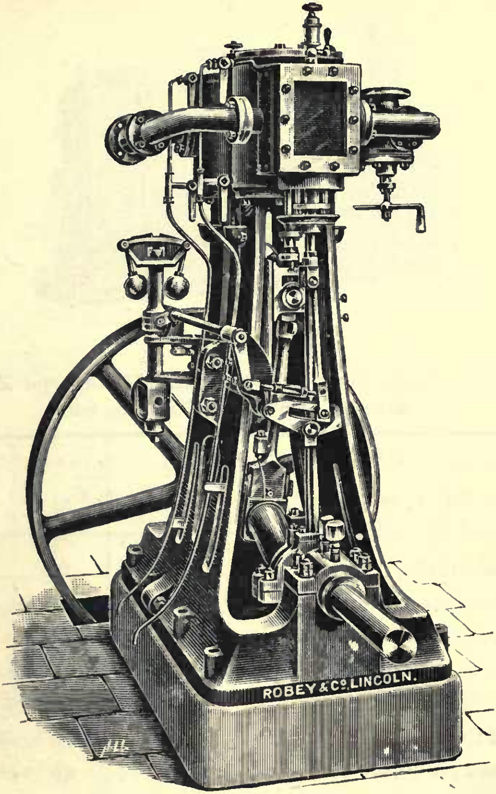


TABLE 119.—Particulars of a Double Cylinder Vertical Engine* by Messrs. Robey & Co. (Fig. 945).

Nominal horse-power	16	Width of fly-wheel	11
Diameter of cylinders, each	$9\frac{1}{2}$	Height to centre of crank from under side of standard	11
Length of stroke	12		
Revolutions per minute	150		
Best average B.H.P.	32	Total height	79
Diameter of crank-shaft	4	„ width	60
Diameter of fly-wheel	60	„ length	80
Weight of fly-wheel, cwt.	15	Approx. weight in cwt.	54

* For 80 lbs. boiler pressure. Dimensions are given in Inches.

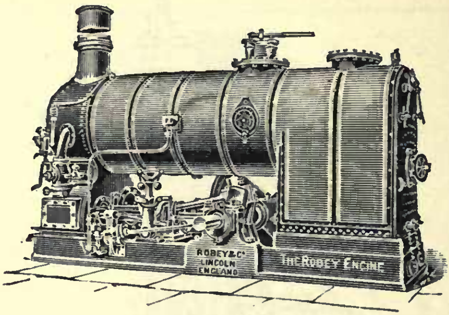


Fig. 946.

TABLE 120.—**Particulars of Compound Undertype Engines by Messrs. Robey & Co.** (Fig. 946).

Nominal H.P.	8	50
Diameter of H.P. cylinder	$5\frac{1}{2}$	$13\frac{1}{4}$
Diameter of L.P. cylinder	$9\frac{1}{2}$	23
Length of stroke	12	24
Revolutions per minute	200	100
Best average I.H.P.	24	150
Diameter of crank-shaft	$3\frac{1}{4}$	8
Diameter of fly-wheel	58	96
Width of fly-wheel	$7\frac{1}{2}$	23
Weight of fly-wheel in cwts.	$8\frac{1}{4}$	96
Height to centre of boiler	46	$94\frac{1}{2}$
Height to centre of engine	$15\frac{1}{2}$	$33\frac{1}{2}$
Diameter of boiler	$31\frac{3}{8}$	61
Length of boiler	128	231
Heating-surface in square feet	169·4	757·7
Grate area in square feet	5·8	21·8
Width over all	63	126
Approx. weight, cwts.	50	380
Boiler-pressure	140	140

Dimensions are given in Inches.

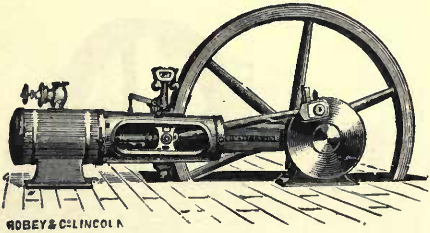


Fig. 947.

TABLE 121.—**Particulars of Horizontal Girder Engines by Messrs. Robey & Co.** (Fig. 947).

Nominal H.P.	8	30
Diameter of cylinder	9	18
Length of stroke	18	36
Revolutions per minute	133	84
Best average I.H.P.	28	142
Diameter of crank-shaft	4	8
Diameter of fly-wheel	66	144
Width of fly-wheel	8	17
Weight of fly-wheel, cwts.	18	90
Height to centre of cylinder	15	28
Total length over all	125	247
Total width over all	65	134
Total weight, approx. in cwts.	48	270
Boiler-pressure	80	80

Dimensions are given in Inches.

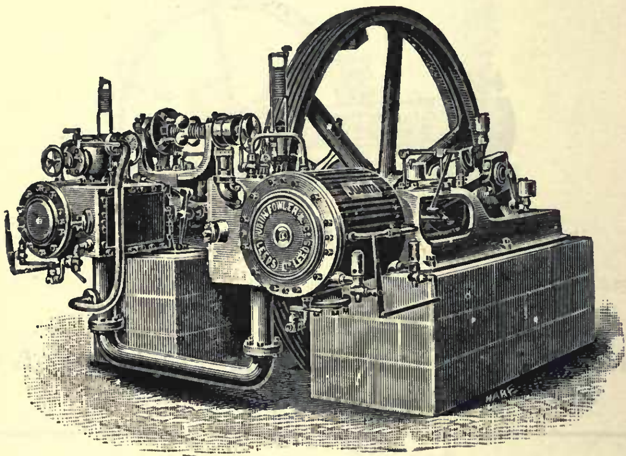


Fig. 948.

TABLE 122.—**Particulars of Compound Coupled Non-condensing Engine by Messrs. John Fowler & Co.**

Diameter of high pressure cylinder	10	11	13	15	$17\frac{1}{2}$
Diameter of low pressure cylinder	16	$17\frac{1}{2}$	23	24	$27\frac{1}{2}$
Length of stroke	20	20	24	30	36

Dimensions are given in Inches.

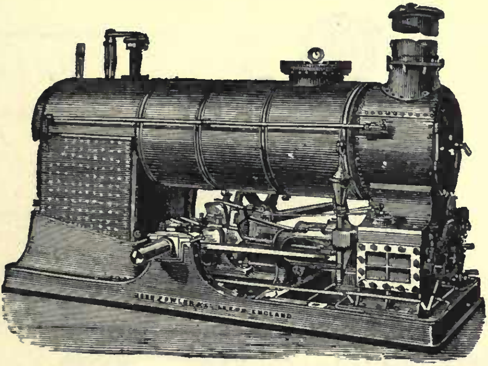


Fig. 949.

TABLE 123.—Particulars of Compound Undertype Engines by Messrs. John Fowler & Co.

Diameter of high pressure cylinder	$5\frac{3}{4}$	$6\frac{5}{8}$	$7\frac{1}{4}$	8	9	10	11	12	13	17	18
Diameter of low pressure cylinder	9	$10\frac{1}{2}$	$11\frac{1}{4}$	$12\frac{3}{4}$	14	16	$17\frac{1}{2}$	21	23	27	29
Length of stroke	12	14	14	15	16	18	18	24	24	30	30

Dimensions are given in Inches.

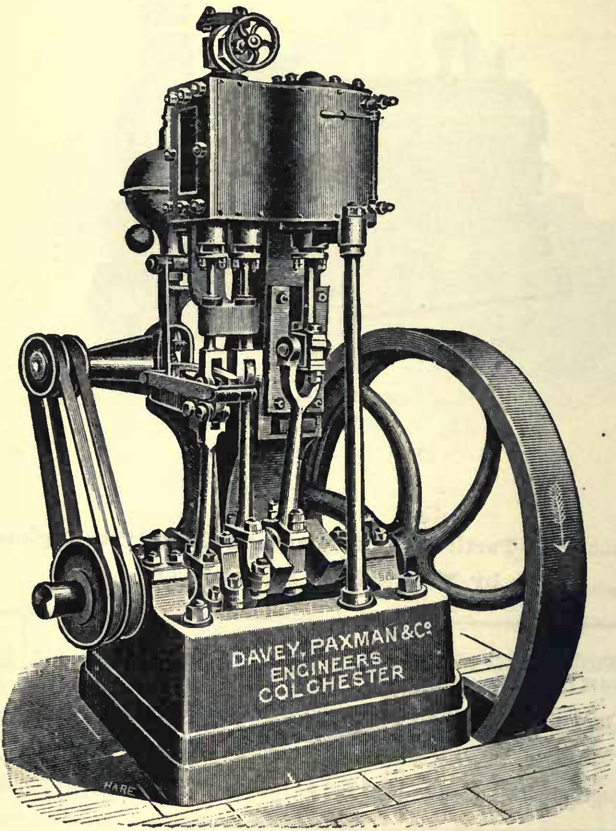


Fig 950.

TABLE 124.—**Single Cylinder Vertical Engines** (Fig. 950) **by Davey, Paxman & Co.**

Nominal H.P.	2	3	4	5	6	8	10	12
Diameter of cylinder	4½	5½	6½	7¼	8¼	9½	10½	12
Length of stroke	6	7	7	8	8	9	10	12
Revolutions per minute	300 400	260 350	260 350	225 300	225 300	200 260	180 250	160 220
Best average I.H.P.	5	7½	10	12½	15	20	25	30
Diameter of fly-wheel	42	42	42	48	48	48	54	54
Width of fly-wheel	3½	4	5	5	6	7	8	8

TABLE 125.—**Double Cylinder Vertical Engines by Davey, Paxman & Co.**

Nominal H.P.	4	6	8	10	12	16	20	25
Diameter of cylinders	4½	5½	6½	7¼	8¼	9½	10½	12
Length of stroke	6	7	7	8	8	9	10	12
Revolutions per minute	300 400	260 350	260 350	225 300	225 300	200 260	180 250	160 220
Best average I.H.P.	10	15	20	25	30	40	50	62
Diameter of fly-wheel	42	42	42	48	48	48	54	60
Width of fly-wheel	5	6	7	8	9	10	11	12

All Dimensions are given in Inches.

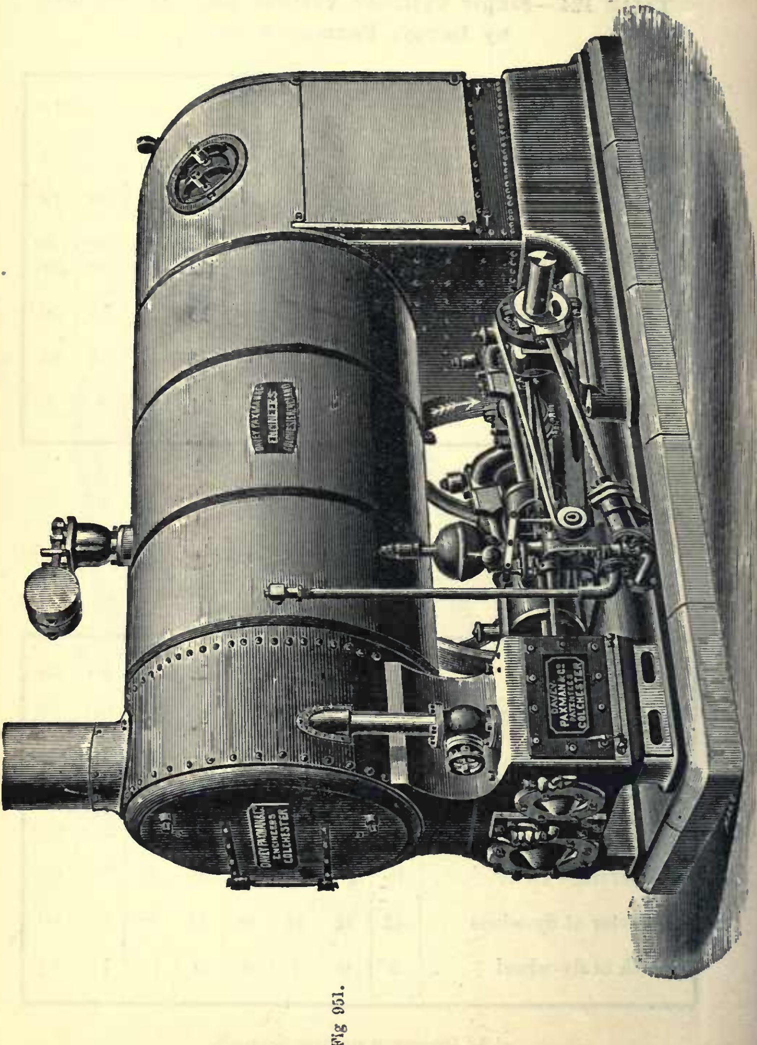

DAVEY PAXMAN & CO.
ENGINEERS
COLCHESTER ENGLAND
DAVEY PAXMAN & CO.
PATENTEES
COLCHESTER
DAVEY PAXMAN & C.
ENGINEERS
COLCHESTER

Fig 951.

Undertype Engines by Messrs. Davey, Paxman & Co.

Table 126.—Compound Engines (Fig. 951).

Nominal H.P.	8	10	12	16	20	25	30	35	40	50	60
Diameter of high pressure cylinder	5½	6½	7	8	9	10	11	12	12¾	14	16
Diameter of low pressure cylinder	9	10½	11¼	13	14¼	16	17½	18¾	20	22½	25
Length of stroke	14	14	14	14	14	18	18	24	24	24	24
Revolutions per minute	155	155	155	155	155	120	120	90	90	90	90
Best average I.H.P.	20	25	30	40	50	62	75	80	100	125	150
Diameter of fly-wheel	60	60	60	66	66	84	84	102	102	102	102
Width of fly-wheel	7	8	9	10	11	12	14	15	17	20	22

Table 127.—Single Cylinder.

Nominal H.P.	4	5	6	8	10	12
Diameter of cylinder	6½	7¼	8¼	9½	10½	12
Length of stroke	12	12	12	12	14	14
Revolutions per minute	125	125	125	125	115	115
Best average I.H.P.	6	12½	15	20	25	30
Diameter of fly-wheel	52	60	60	60	66	66
Width of fly-wheel	5	5	6	7	8	9

All Dimensions are given in Inches.

TABLE 128.—**Willans' Central Valve Engines (Non-Condensing Type).**

Mark	F	F	G	G	H	H	I	I
Number of cranks	2	3	2	3	2	3	2	3
Diameter of low pressure cylinders	12″	12″	14″	14″	17″	17″	20″	20″
Stroke	6″	6″	6″	6″	8″	8″	9″	9″
Speed, revolutions per minute	470	470	460	460	380	380	350	350
Maximum load for continuous running, I.H.P.	75	112	100	150	165	250	240	360
Boiler pressure, simple engines	—	—	—	—	100	—	—	—
,, compound engines	—	—	—	—	140	—	—	—
,, triple expansion engines	—	—	—	—	190	—	—	—

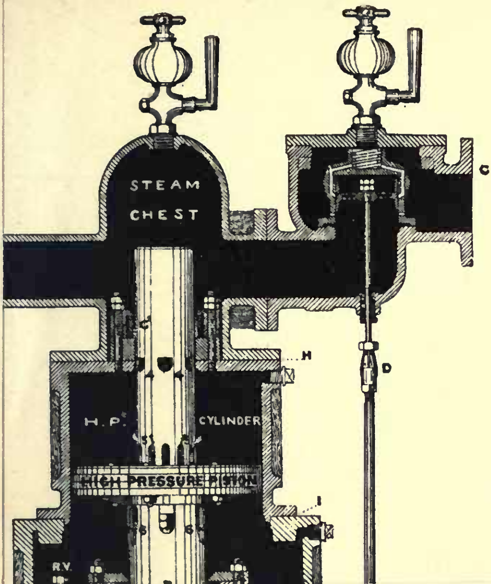

STEAM
CHEST
C
H
D
H. P.
CYLINDER
HIGH PRESSURE PISTON
I
R.V.

TABLE 129.—**Willans' Central Valve Engines (Condensing Type).**

Mark	O	O	B	B	P	P	Q	Q	R	R	S	S	T	T
Number of cranks	2	3	2	3	2	3	2	3	2	3	2	3	2	3
Diameter of low pressure cylinders	17·32		20·47		23·6		25·6		28·3		30·4		32·3	
Stroke	5·9		6·89		7·88		9·05		10·24		11·4		13·39	
Speed, revolutions per minute	450		400		360		340		320		300		270	
Maximum load for continuous running	120	180	180	270	240	360	300	450	385	575	470	705	550	825

Mark	V	VL	VS	VL	WS	WL	XS	XL	YS	YL	ZS	ZL
Number of cranks	2	2	3	3	3	3	3	3	3	3	3	3
Diameter of low pressure cylinders	37·5		37·5		43·3		48		53		67	
Stroke	16·9	22·4	16·9	22·4	20·47	27·6	23·6	31·5	28·3	37·8	33·5	44·5
Speed, revolutions per minute	240	180	240	180	220	165	200	150	180	135	160	120
Maximum load for continuous running	835	835	1250		1875		2500		3750		5000	

Boiler pressure, not less than 75 lbs. for simple engines, 110 lbs for compound engines, and 160 lbs. for triple expansion engines.

Quick-running enclosed engines are now very much in use for driving electric light and power machinery, the dynamos being usually coupled direct on to the engine shaft. There are two distinct types of these engines: single acting, of which Willans' central valve

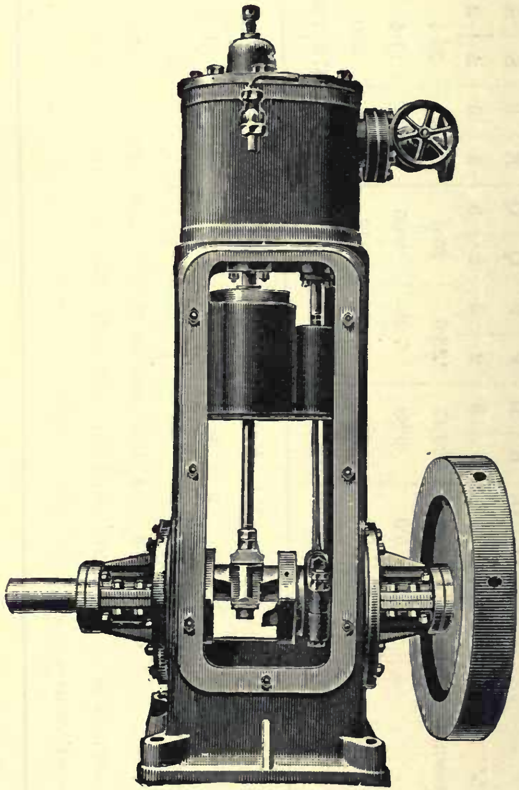


Fig. 952A.

engine may be taken as typical, and double-acting engines, of which that of Messrs. Belliss may be taken as typical.

The engine of Messrs. Willans, of the Victoria Works, Rugby, is shown in fig. 952, the cycle through which the steam goes being the same as in the old Cornish pumping engines; tables of sizes and speeds are given on pages 348 and 349.

Another example of the single-acting type is shown in fig. 952A, and is made by Messrs. Bumsted & Chandler, of Hednesford, Staffordshire. In fig. 952A the doors which enclose the engine are removed to show the arrangement; the piston and valve rods work through ordinary stuffing-boxes; these latter never being subjected to the full pressure of the steam from the boiler, each cylinder receiving but one charge of steam per revolution, the piston and valve rod stuffing-boxes at the lower end of the cylinder are subjected to exhaust pressure only; this should render the packing more durable than when subjected to steam of high pressure and temperature. By a new improvement the crank shaft brasses can be readily taken out for examination, and the crank shaft can be drawn out from either side of the engine. It is a special feature of all single-acting engines that they can be run at a very high speed with but little noise, owing to all working parts being always in compression, the strain never being reversed.

Of the double-acting type the engine of Messrs. Belliss & Morcomb Ltd., Birmingham, has proved very successful. Messrs. Belliss claim to have introduced their system of forced lubrication in 1890, and very efficient this system has proved to be. Fig. 952C shows the arrangement and construction of the engine, and the system of forced lubrication. The governing of this engine is very good, and even under the severe and sudden changes in load which occur in engines used for driving dynamos for electric traction, the variation in speed is very slight. The forced lubrication is essential to quick-running engines, and the wear and tear in engines of this kind is very small.

The distribution of steam in the double-acting engines is slightly more economical than in the single-acting Cornish cycle engine, but the difference in practice is small.

The large number of these enclosed engines now in use, and the long runs they are daily making without requiring adjustment, point rather to the fact that engines, when properly constructed and lubricated, require but little attention, and when enclosed and thus kept out of the way, run very much better than open engines, where adjustment is so easy that it is often done when not required.

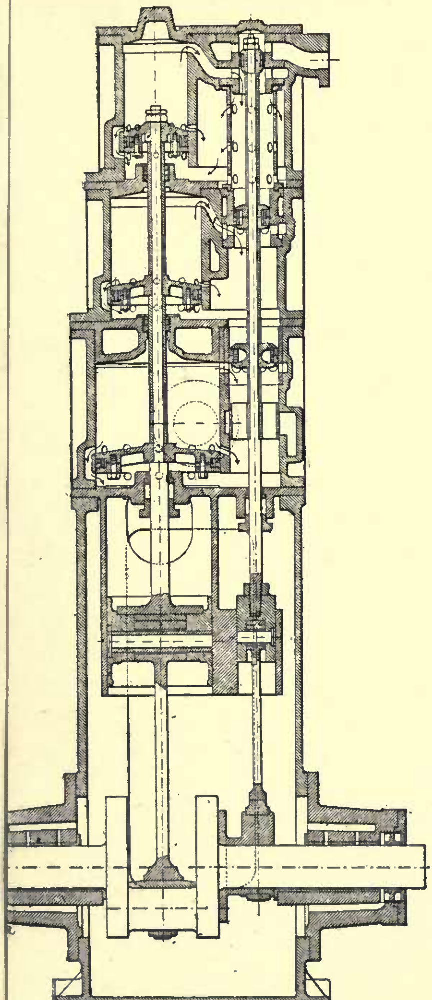
TABLE 129A.—**Particulars of Chandler's "Silent" Engines** (Fig. 952A).

Single-Crank Engines.

Diameter of low-pressure cylinder . . .	3	5	$5\frac{3}{4}$	$6\frac{1}{2}$	8	$9\frac{1}{2}$	11	13	15	18	20	24	*
Revolutions per minute	800	625	600	550	500	450	425	400	380	350	350	225	
Available brake horse power when exhausting into atmosphere at the following boiler pressures :—													
60 lbs. per square inch . . .	$\frac{3}{4}$	3	4	5	8	11	17	24	32	48	60	98	S
80 " " . . .	1	4	6	7	11	16	$22\frac{1}{2}$	32	43	64	80	131	S
90 " " . . .	$1\frac{1}{4}$	5	7	9	14	19	28	40	54	80	100	163	S
100 " " . . .			5	$6\frac{1}{2}$	$10\frac{1}{2}$	14	21	30	40	60	75	123	C
125 " " . . .			7	9	14	19	28	40	54	80	100	163	C
150 " " . . .			8	10	16	22	33	48	64	96	120	195	C
160 " " . . .						19	28	40	54	80	100	163	T

The diameter of the fly-wheel of these engines is three times the diameter of the low-pressure cylinder, and the width of the fly-wheel is ·5 of the diameter of the low-pressure cylinder.

* S means simple; C, compound; T, triple expansion. **All Dimensions are given in Inches.**



ed and Chandler. [To face p. 352.

TABLE 129B.—**Particulars of Chandler's "Silent" Engines.**

Double-Crank Engines.

													*
Diameter of low-pressure cylinder	3	5	$5\frac{3}{4}$	$6\frac{1}{2}$	8	$9\frac{1}{2}$	11	13	15	18	20	24	
Revolutions per minute	800	625	600	550	500	450	425	400	380	350	350	225	
Available brake horse power (when exhausting into atmosphere) at the following boiler pressures :—													
60 lbs. per square inch	$1\frac{1}{2}$	6	8	10	16	22	34	48	64	96	120	196	S
80 " "	2	8	12	14	22	32	45	64	86	128	160	262	S
90 " "	$2\frac{1}{2}$	10	14	18	28	38	56	80	108	160	200	326	S
100 " "			10	13	21	28	42	60	81	120	150	240	C
125 " "			14	18	28	38	56	80	108	160	200	326	C
150 " "			16	20	32	44	66	96	128	192	240	390	C
160 " "						38	56	80	108	164	200	320	T

Diameter and width of fly-wheel as in Table 129A.

The indicated H.P. of these engines is from 5 to 10 per cent. higher than the brake H.P. given in the tables according to the size of the engines. When exhausting into a condenser the powers will be about 10 per cent. higher than those given in the tables.

* S, simple; C, compound; T, triple expansion. All Dimensions are given in Inches.

TABLE 129C.—Belliss & Morcom's Quick Revolution Engines.

Compound, Non-Condensing.

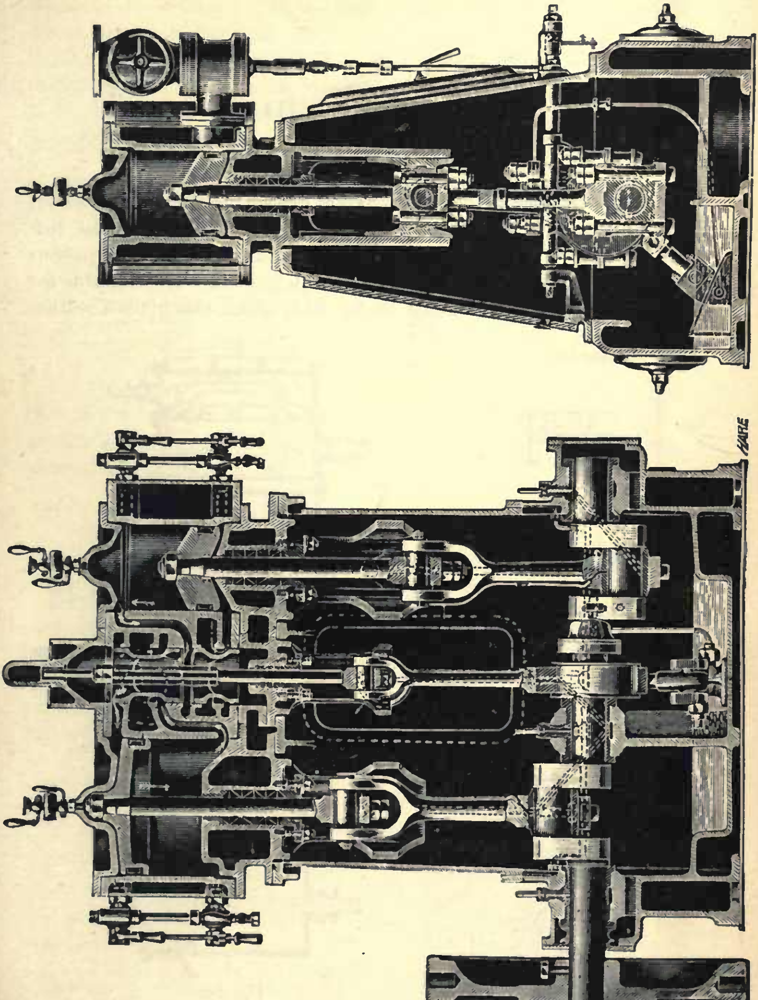
Mark	C 1	C 2	C 3	C 4	C 5	C 6	C 7	C 8	C 9	C 10	C 11	C 12	C 13	C 14	C 15	C 16	C 17
Number of cranks	2																
Speed, revolutions per minute	650	625	600	575	525	525	475	450	435	425	420	400	400	380	360	350	325
Maximum brake H.P. approximate	30	38	53	75	90	110	135	150	180	200	225	260	300	325	360	400	500

Boiler pressure, 140 lbs. to 150 lbs. per square inch.

Triple Expansion.

Mark	T 1	T 2	T 3	T 4	T 5	T 6	T 7	T 8	T 9	T 10	T 11	T 12	T 13	T 14	T 15	T 16	T 17	T 18	T 19
Number of cranks	3																		
Speed, revolutions per minute	550	500	475	450	400	375	400	340	350	330	330	280	300	300	250	270	230	250	200
Maximum brake H.P. approximate	220	220	300	325	400	430	500	500	620	720	800	800	1100	1250	1300	1525	1585	2140	2140

Boiler pressure, 170 lbs. to 200 lbs. per square inch.



Belliss & Morcom's Quick Revolution Self-Lubricating Engine.

Fig. 952D.

SECTION VIII.

COMPOUND ENGINES.

THE early compound engines by Woolf had no intermediate receiver, and the passage of the steam was direct from the high to the low pressure cylinder. In some cases, as in figs. 953, 954, both pistons moved together in the same direction. A form of valve suitable for this kind of engine is shown in fig. 954, which also gives a section

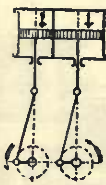


Fig. 953.

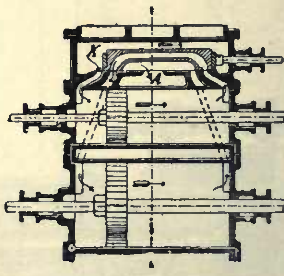


Fig. 954.

through the cylinders with arrows to indicate the course of the steam through the ports and passages. In this arrangement the passages

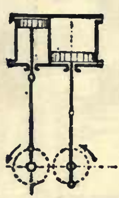


Fig. 955.

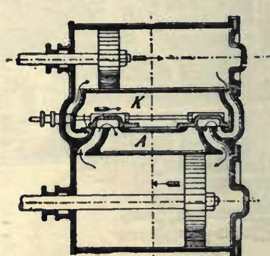


Fig. 956.

to the low pressure cylinder are long and thus give a large clearance space.

In the arrangement shewn in figs. 955, 956, the pistons move in opposite directions. The clearance space here is less than in the former example, figs. 953, 954.

The indicator diagram, fig. 957, is from an arrangement like fig. 956, with cut-off at ·4 in the high, and at ·8 in the low pressure cylinder.

In these types of compound engines, it is a good plan to have the cut-off in the high pressure cylinder controlled by the governor, and

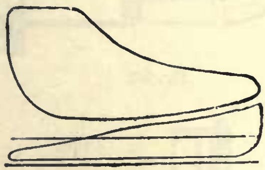


Fig. 957.

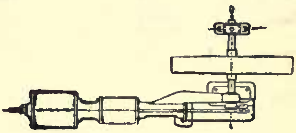


Fig. 958.

that of the low variable by hand. The side-by-side cylinders, with pistons moving together as fig. 954, have been much used for beam pumping engines.

Tandem Compound Engine.

The tandem horizontal engines, as fig. 958, often have a receiver even if it only takes the form of a large connecting pipe. These engines take up but little room in the width, but are rather long.

Examples of the various methods of connecting the high and low pressure cylinders of tandem compound engines are shewn in figs. 959--963.

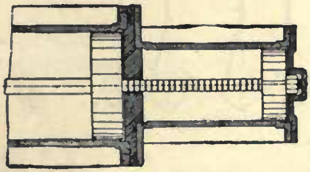


Fig. 959.—High and low-pressure cylinders for tandem engine by Simpson, of Dartmouth.*

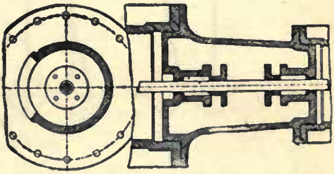


Fig. 960. — Connecting-piece for tandem cylinders used for vertical engines.

* For performance of a small engine of this type, see reports of judges of Royal Agricultural Society's meeting at Plymouth, 1890.

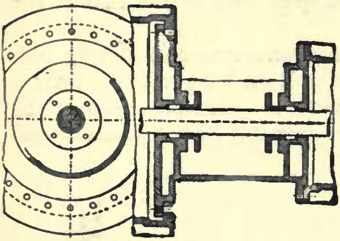


Fig. 961.

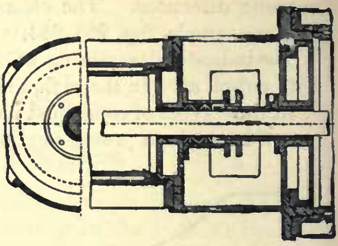


Fig. 962

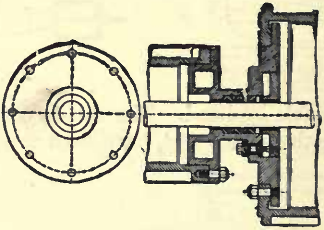


Fig. 963.

Figs. 961—963.—Various methods of arranging the connecting-piece and glands for tandem cylinder.

Compound Engines with the Cranks at 90°.

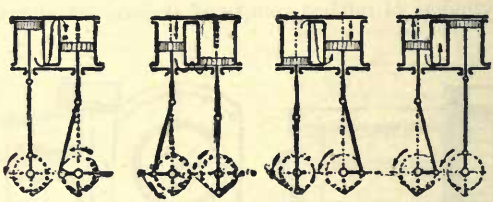


Fig. 964.

The outline diagrams, fig. 964, shew the different positions of cranks and pistons in a compound engine with two cranks at right angles to each other, and fig 965 shews the indicator diagrams corresponding to the two cylinders. H in fig 965 gives the position of the high pressure crank, and N that of the low.

In compound engines of this kind, the cut-off in the high pressure cylinder is usually controlled by the governor, and that of the low is either fixed or better varied by hand.

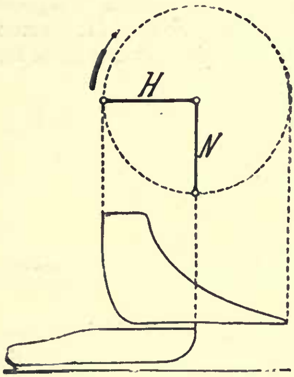


Fig. 965.

Ratio of Cylinder Volumes of Compound Engines.

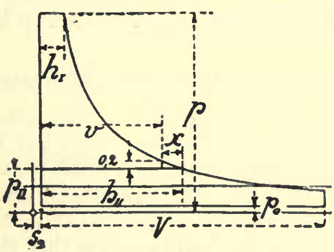


Fig. 966.

From the theoretical diagram, fig. 966, the ratio of the cylinder volumes can be determined :

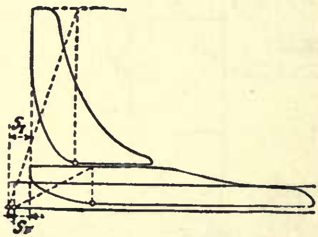
t = the temperature of the entering steam.
p = pressure of the entering steam.
t_0 = the temperature of the exhaust steam.
p_0 = pressure of the exhaust steam.

The mean temperature of the cylinders should be equal to half the total fall of temperature. Let the range of temperature difference in the two cylinders be taken as nearly equal as possible, then

$$t_{II} = \frac{t + t_0}{2}$$

From this temperature and corresponding pressure p, the line passing through the point in the curve at the height p_{ii}, will be the division line between the two cylinder diagrams, and b_{ii} as the admission line of the low pressure cylinder diagram, an allowance of about 3 lbs. pressure (·2 atmosphere) being made for the drop between

Fig. 967.



the cylinders, then the volume of the high pressure cylinder v will be a little less than b_{ii}. The diagram can be then rounded off and corrected, when it will appear as in fig. 967. Sometimes the endeavour is made to exactly equalize the work in the two cylinders, but although for many reasons it is good practice, the equality can only happen at certain loads unless the low pressure cylinder is provided with a variable expansion gear worked by hand.

Receivers for Compound Engines

The receiver was formerly merely a large pipe between the cylinders of compound engines, or in some cases a wrought-iron barrel lagged and cased, but now they are often steam jacketed, and sometimes constructed as in fig. 968 with tubes. An ordinary form is that shewn in fig. 970, which is simple and cheap to make.

One of the most important points to note is that the receiver and jacket should be efficiently drained.

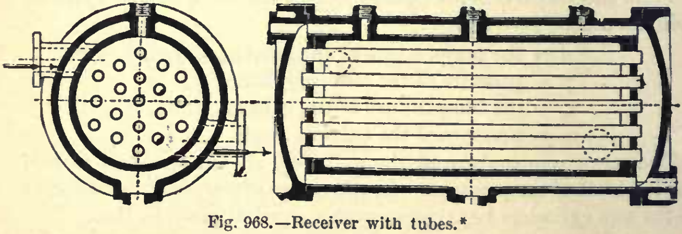


Fig. 968.—Receiver with tubes.*

* "Zeitschrift d. Verein Deutsch Ingenieur," 1888, plate 15.

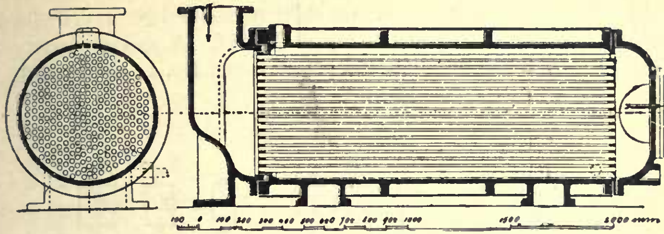


Fig. 969.—Receiver with tubes for tandem engine.*

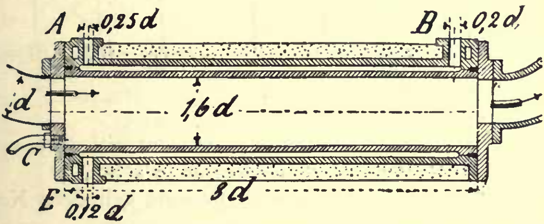


Fig 970.—Ordinary jacketed receiver.

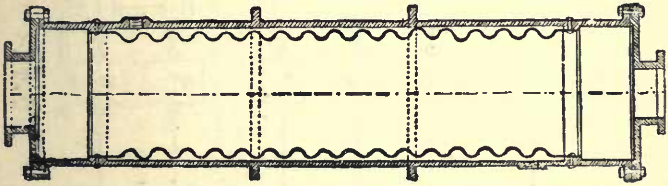


Fig 971 —American jacketed receiver.†

Triple Expansion Engines.

a. Marine Engines.

The number of triple expansion engines built increases every year, and it is now the recognized type for marine engines with boiler pressure of about 160 lbs. per square inch. Four different arrangements are in use.‡

---

* "Zeitschrift d. Verein Deutsch Ingenieur," 1890, plate 22.

† *Engineering*, 1891. p. 750.

‡ These arrangements are from a paper by Otto J. Müller, jun., "Zeitschrift d. deutsches Verein Ingenieur," 1887, page 445. See also "Proceedings of the Institution of Mechanical Engineers," July, 1891.

(1.) In fig. 972 there are cranks at right angles, and No. I. cylinder is placed tandem fashion on the top of No. II. A special variety with No. II. cylinder annular, and placed round No. I., see fig. 973.

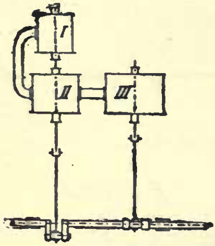


Fig. 972.

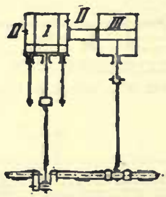


Fig. 973.

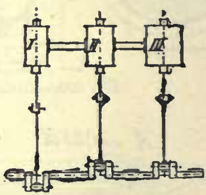


Fig. 974

(2.) Fig. 974 shews the very usual arrangement with three cranks at 120° to each other, the three cylinders side by side.

(3.) In fig. 975, there are two high pressure cylinders, Nos. I.

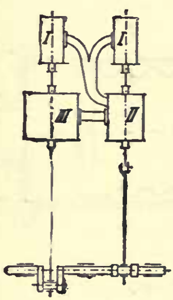


Fig. 975.

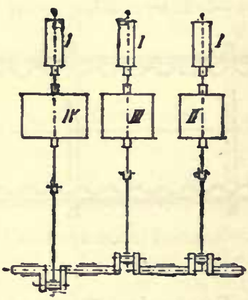


Fig. 976.

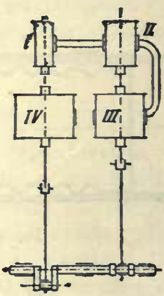


Fig. 977.

I. placed over Nos. II. and III.; in this arrangement there are two cranks at right angles to each other. An elaboration of this arrangement for quadruple expansion is given in fig. 976.

(4.) A better arrangement for quadruple expansion is shewn in fig. 977.

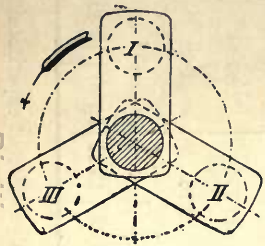


Fig. 978.

The cranks of the triple expansion marine engines may be arranged in order I., II., III., as in fig. 978, with the ship going ahead, so that the steam continuously expands in the shortest possible time while doing its work.*

* "Zeitschrift d. Verein Deutsch Ingenieur," 1886, No. 24.

TABLE 130.—**Leading Particulars of Triple-Expansion Condensing Engines, for 150 lbs. working pressure.**

Stroke of all three cranks H		16	24	32	40	48	56	64	72	80
I. Diameter of high pressure cylinder . . d_1		9	$13\frac{1}{2}$	18	$22\frac{1}{2}$	27	32	36	40	44
II. Diam. of intermediate pressure cylinder . d_2		$14\frac{1}{4}$	$21\frac{3}{4}$	29	36	44	52	59	66	72
III. Diameter of low pressure cylinder . . d_3		24	36	48	60	72	84	96	108	118
Proportion of cylinder volumes (in round numbers) . .	II. : I.	2·5	2·6	2·6	2·6	2·7	2·6	2·7	2·7	2·7
	III. : II.	2·8	2·8	2·8	2·8	2·7	2·6	2·6	2·7	2·7
	III. : I.	7·1	7·1	7·1	7·1	7·1	7·0	7·1	7·3	7·2
Revolutions per minute n		200	140	120	100	85	77	70	66	60
Piston speed in feet per minute		530	560	640	660	680	710	740	790	800
N_i with 150 lbs. pressure N_i		250	600	1200	2000	2800	4000	5600	7400	9300
Steam used in lbs. per indicated H.P. per hour .		16·1	15·63	15·63	15·00	14·6	14·1	13·7	13·45	13·23

All Dimensions are given in Inches.

Stationary Triple-Expansion Engines.

The triple-expansion engine does not offer such great advantages for use on land as for use at sea. The amount of steam used per HP is no doubt less, especially when pressures up to 160 lbs. per square inch are employed, than with simple or compound engines using lower pressures. The disadvantages of triple-expansion engines consist of an increased number of moving parts, by about one-third, and the cost of maintenance and repairs are increased. The high pressures render it less easy to keep the steam pipes and joints steam tight. The power absorbed with the engine running empty is greater than with simple and compound engines.

Professor Schöter gives an interesting account of experiments with a triple-expansion engine built by the Augsburger Engine Company in the Zeitschift d. Verein deutsches Ingenieur, 1890, No. 1.

The engine had two cranks at 90°; the (I) high and (II) intermediate cylinders were placed tandem with one piston rod common to the two, and the (III) low-pressure cylinder on the other side of the fly-wheel.

The working pressure was 150 lbs. per square inch, and the engine condensing; the normal HP at 70 revolutions per minute was 200. For the leading dimensions, see Table 131.

Table 131.—Leading Particulars of Triple-Expansion Engine.

	High Pressure Cylinder I.		Intermediate Pressure Cylinder II.		Low Pressure Cylinder III.	
	Front.	Back.	Front	Back.	Front.	Back.
Diameter of pistons	11·102		17·75		27·61	
Diameter of piston rods	2·95	3·35	3·35	0	3·35	3·35
Stroke	39·37		39·37		39·39	
Ratio of cylinder vols.	1		2·73		6·63	
Ratio of cylinder vols.	—		1		2·34	
P_m in lbs. per square inch	48·36	46·65	12·83	13·11	12·40	12·30

Dimensions in Inches.

The values of P_m are from the mean of five different experiments.

A set of indicator diagrams to a reduced scale with pressures in atmospheres is given in fig. 979.

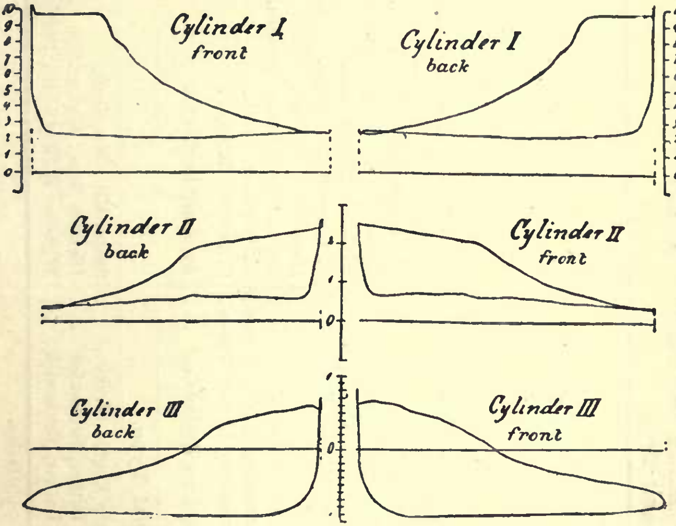


Fig. 979.

Indicated H.P.		Indicated H.P.	
Cylinder I., front . .	31·7	Cylinder I., back . .	31·4
Cylinder II., back . .	20·1	Cylinder II., front . .	19·5
Cylinder III., back . .	48·5	Cylinder III., front . .	49·3

Total Indicated H.P. . . . 200·5.

The steam was taken from a water tube boiler by Dürr & Co., Ratinger, Düsseldorf, with a permitted pressure of 190 lbs. per sq. inch, 150 lbs. only being used during the trials. The heating surface of the boiler was 1722 square feet. The steam used per I.H.P. per hour was 12·52.

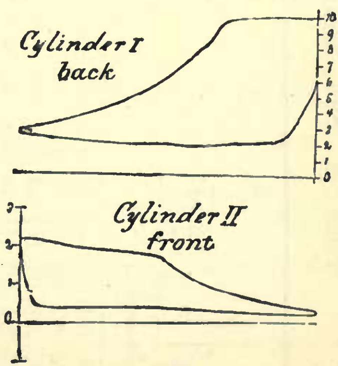


Fig. 980.

From diagrams taken with the engine running light, gave 23·8 I.H.P., about 11·5 per cent. of the normal power.

In the next table the performance of some examples of triple expansion engines is given from actual practice.

TABLE 132.—**Particulars of Triple-Expansion Engines from Actual Practice.**

Horizontal Type.

High. Cylinder Diam.	Intermediate.	Low.	Stroke.	Revolutions per Minute. n	Steam Pressure.	Indicated H.P. Ni	Steam in lbs. per I.H.P. per Hour. Guaranteed	At-tained	Heating Surface of Boiler in square feet.	Builder of Engine.	User of Engine.
23⅝	35½	53¼	63	65	150	1000	12·13	11·62	—	Sulzer Bros. . . .	Brunswick Flax Works.
21⅝	33½	48⅛	47¼	68	150	700	12·13	11·80	—	Sulzer	H. Haggemacher, Buda Pesth.
19¾	29½	47¼	55¼	65	150	700	13·78	12·45	6264	Augsburg Engine Co. . .	Thread Works, Göggingen.
18¾	29½	43½	41½	70	150	450	13·23	12·01	—	Sulzer Bros.	Kinkindaer Steam Mill.
14¼	22⅝	35½	43½	70	150	400	13·23	—	—	Görlitz Engine Works .	L. Loewe & Co., Berlin.
13½	21⅝	33⅝	39⅜	70	170	330	13·00	—	—	Görlitz Engine Works . .	Hansa Mill, Bremen.
14¾	21⅝	31½	39⅜	70	150	240	13·23	—	—	Görlitz Engine Works .	Ammunition Works, Spandau.
11	15¾	23⅝	27½	85	205	225	14·33	13·44	1614	Nuremberg Engine Works .	N. Wiederer, Fürth.
9¾	15·	22½	31½	85	175	200	14·33	13·44	1280	Nuremberg Engine Works .	C. Sclenk, Rotha, S.
11	17¾	27½	39⅜	70	150	200	—	12·4 12·0	1764	Augsburg Engine Co. . .	Augsburg Engine Co.

Vertical Type.

High. Cylinder Diam.	Intermediate.	Low.	Stroke.	Revolutions per Minute. n	Steam Pressure.	Indicated H.P. Ni	Steam in lbs. per I.H.P. per Hour. Guaranteed	At-tained	Heating Surface of Boiler in square feet.	Builder of Engine.	User of Engine.
21⅝	35	55¼	35½	85	150	825	12·7	—	6963	Sächs Engine Works . .	Steam Mill, Wansbeck.
23⅝	37¾	53¼	23⅝	90	157	700	13·0	—	—	G. Luther, Brunswick . .	Dietrich Bros., Weissenfels.
23⅝	37¾	53¼	23⅝	90	157	480	13·0	—	—	G. Luther, Brunswick .	Wittener Rolling Mills, Duisburg.

Dimensions in Inches.

Proportions of Cylinder Volumes with equal strokes for Triple-Expansion Engines.

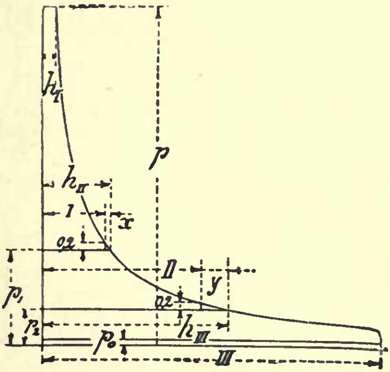


Fig. 981.

From the diagram fig. 981, the proportions of cylinder volumes can be determined :

t = temperature of entering steam.
p = pressure of the same.
t_1 = temperature of steam in inter. cylinder.
p_1 = pressure of steam in inter. cylinder.
t_2 = temperature of steam in LP cylinder.
p_2 = pressure of steam in LP cylinder.
t_0 = temperature of exhaust steam.
p_0 = pressure of exhaust steam.

Taking the mean temperature to be half the extreme difference,

$$t_1 = \frac{2t + t_0}{3}$$

$$t_2 = \frac{t + 2t_0}{3}.$$

Then taken the corresponding pressures p_1 and p_2, the diagram can be divided as with the one for compound engines, a drop being allowed of 3 lbs. per square inch. h_{ii} and h_{iii} will be respectively the points of cut-off from the inter. and LP cylinders ; the diagram may then be corrected for compression and losses.

Figs. 982, 983, shew the pistons and cranks of triple-expansion engines in various positions.

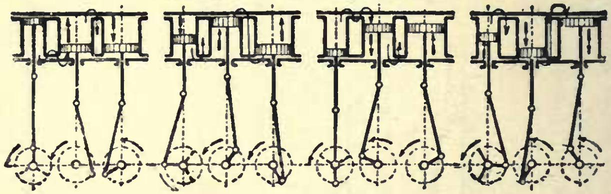


Fig. 982.—Triple-expansion engine with two receivers.

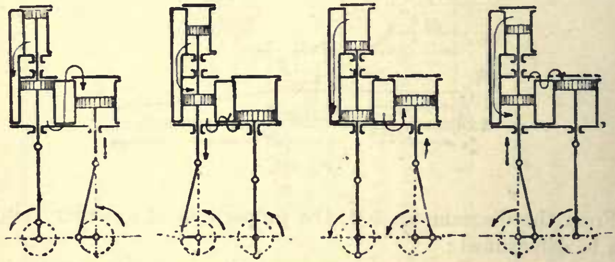


Fig. 983.—Semi-tandem triple-expansion engine with two receivers.

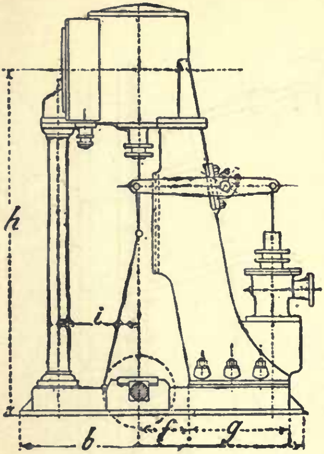


Fig. 984.

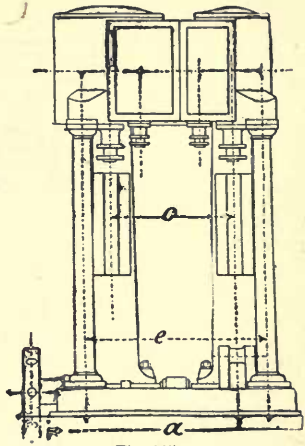


Fig. 985.

TABLE 133.—**Compound Engine for Screw Steamers**
(Figs. 984, 985).

Stroke of both pistons . .	H	$7\frac{7}{8}$	$9\frac{3}{4}$	$11\frac{3}{4}$	$13\frac{3}{4}$	$15\frac{3}{4}$	$19\frac{3}{4}$	$23\frac{3}{4}$
Diam. of high-press. piston	d	$5\frac{3}{8}$	$6\frac{3}{4}$	$7\frac{7}{8}$	$9\frac{1}{4}$	$11\frac{1}{4}$	$13\frac{3}{4}$	17
Diam. of low-press. piston .	D	$8\frac{1}{4}$	$10\frac{5}{8}$	$12\frac{1}{4}$	$14\frac{5}{8}$	$17\frac{3}{4}$	$21\frac{3}{4}$	$26\frac{1}{2}$
Proportion of cylinder vols.	$\frac{v}{d}$	2·5	2·5	2·5	2·5	2·5	2·5	2·5
Proportion of H : d . .		1·5	1·5	1·5	1·5	1·4	1·4	1·4
Revolutions per minute .	n	280	260	245	230	210	180	150
Piston speed in ft. per min.	c	365	420	470	515	530	590	590
I.H.P. with 105 lbs. boiler press.		25	50	80	120	180	270	400
Dimension	a	$23\frac{5}{8}$	$29\frac{1}{2}$	$37\frac{1}{2}$	$45\frac{1}{2}$	55	73	90
"	b	$23\frac{5}{8}$	$29\frac{1}{2}$	$37\frac{1}{2}$	$45\frac{1}{2}$	55	73	90
"	c	$9\frac{1}{2}$	$11\frac{3}{4}$	$16\frac{1}{2}$	$19\frac{3}{4}$	$23\frac{5}{8}$	$37\frac{1}{2}$	$47\frac{1}{2}$
"	e	$19\frac{3}{4}$	$24\frac{5}{8}$	$29\frac{1}{2}$	$34\frac{1}{2}$	$41\frac{1}{2}$	55	$70\frac{3}{4}$
"	f	$5\frac{1}{2}$	7	$8\frac{1}{4}$	$9\frac{3}{4}$	11	$13\frac{3}{4}$	$16\frac{1}{2}$
"	g	$10\frac{1}{4}$	$12\frac{5}{8}$	$15\frac{1}{2}$	$17\frac{3}{4}$	$20\frac{1}{2}$	$25\frac{5}{8}$	$30\frac{3}{4}$
"	h	$41\frac{1}{2}$	51	59	$70\frac{3}{4}$	$78\frac{3}{4}$	98	118
"	i	$7\frac{7}{8}$	$9\frac{3}{4}$	$12\frac{5}{8}$	15	17	$21\frac{5}{8}$	26
Weight in cwts.	G	$15\frac{3}{4}$	$31\frac{1}{2}$	55	$88\frac{1}{2}$	137	236	452

All Dimensions are given in Inches.

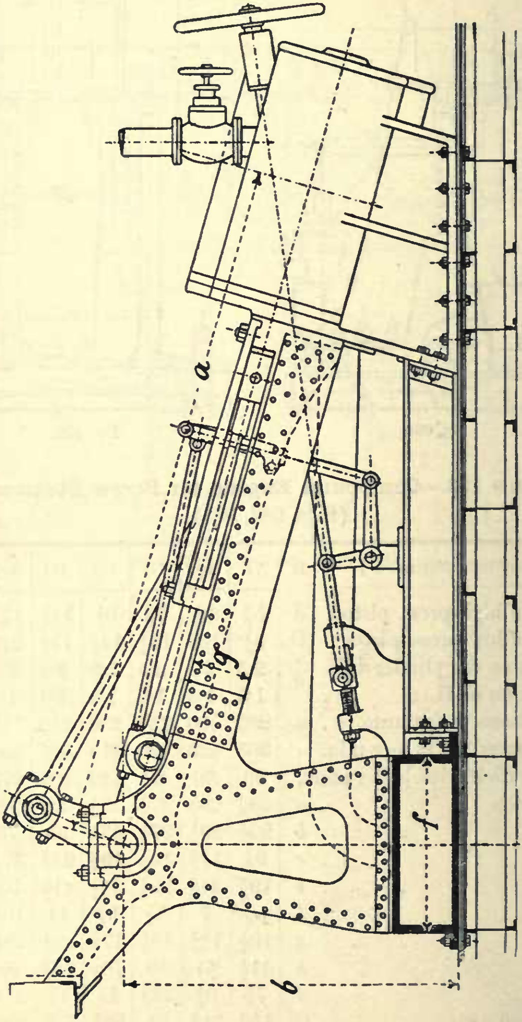


Fig. 986.

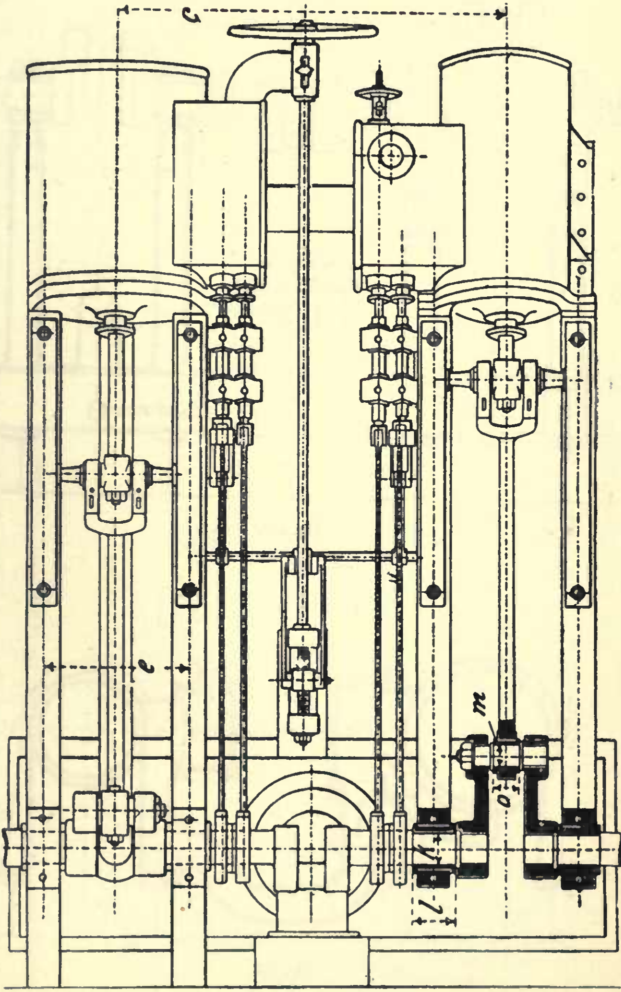


Fig. 987

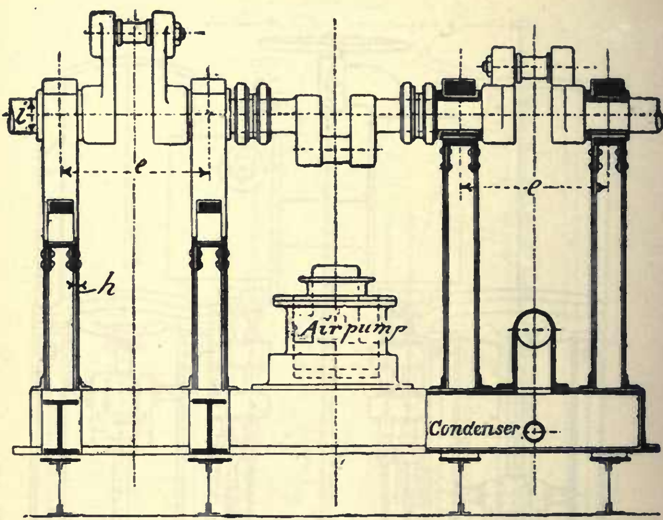


Fig. 988.

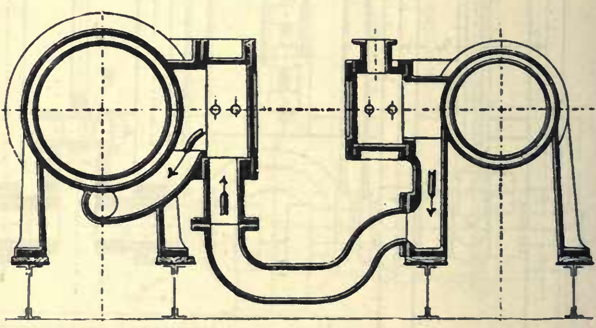


Fig. 989

TABLE 134.—**Compound Engines for Paddle Steamers, 105 lbs. Boiler Pressure** (Figs. 986—989).

					lbs. pressure.									Dia. of paddle-shaft.	Bearings.		Crank-pin.		Weight in cwts.
H	d*	D*	$\frac{V†}{v}$	n	p	H.P.	a	b	c	e	f	g	h	i	k	l	m	o	G
$19\frac{3}{4}$	$12\frac{1}{4}$	$19\frac{3}{4}$	2·5	50	120	40	$86\frac{5}{8}$	$47\frac{1}{4}$	51	$19\frac{3}{4}$	$23\frac{3}{4}$	7	$\frac{5}{16}$	$5\frac{7}{8}$	$5\frac{1}{8}$	$7\frac{7}{8}$	$3\frac{1}{2}$	$6\frac{3}{4}$	79
$23\frac{3}{4}$	$14\frac{1}{2}$	$23\frac{3}{4}$	2·5	46	120	60	$102\frac{3}{8}$	51	63	$23\frac{3}{4}$	$29\frac{1}{2}$	$9\frac{1}{2}$	$\frac{11}{32}$	$6\frac{1}{2}$	$5\frac{3}{4}$	$8\frac{5}{8}$	$4\frac{1}{8}$	$7\frac{1}{2}$	128
$31\frac{1}{2}$	18	$31\frac{1}{2}$	2·5	40	120	125	134	57	$78\frac{3}{4}$	$29\frac{1}{2}$	$33\frac{1}{2}$	$11\frac{3}{4}$	$\frac{11}{32}$	$7\frac{7}{8}$	$6\frac{3}{4}$	$9\frac{1}{2}$	$5\frac{1}{8}$	$8\frac{5}{8}$	295
$39\frac{1}{2}$	$21\frac{3}{4}$	$39\frac{1}{2}$	2·5	38	120	210	$165\frac{1}{2}$	63	$94\frac{1}{2}$	$36\frac{1}{4}$	$39\frac{1}{2}$	$15\frac{3}{4}$	$\frac{3}{8}$	$9\frac{1}{2}$	$8\frac{1}{4}$	$11\frac{1}{2}$	$6\frac{1}{2}$	$10\frac{1}{4}$	551
47	$25\frac{3}{4}$	47	2·5	35	120	350	197	67	110	$43\frac{1}{4}$	$43\frac{1}{4}$	$19\frac{3}{4}$	$\frac{7}{16}$	$11\frac{3}{4}$	$9\frac{3}{4}$	$13\frac{1}{2}$	$7\frac{7}{8}$	$12\frac{5}{8}$	925
55	$29\frac{1}{2}$	55	2·5	32	120	500	232	71	126	51	$47\frac{1}{4}$	$25\frac{5}{8}$	$\frac{15}{16}$	$13\frac{3}{4}$	$11\frac{3}{4}$	$15\frac{3}{4}$	$9\frac{1}{2}$	15	1377
63	$34\frac{3}{4}$	63	2·5	30	120	700	268	75	141	59	51	$31\frac{1}{2}$	$\frac{1}{2}$	$15\frac{3}{4}$	$13\frac{3}{4}$	18	11	17	1968

* D = Diameter of low-pressure piston; d = Diameter of high-pressure piston.
H = Stroke of both pistons.
† V = Volume of low-pressure cylinder; v = Volume of high-pressure cylinder.

All Dimensions are given in Inches.

SECTION IX.

INDICATOR AND INDICATOR DIAGRAMS.

In the beginning of section IV. on Valve gears, page 151, indicator diagrams have been mentioned, and a method of drawing an approximate expansion curve has been given; in this section the subject is given in more detail, with examples of diagrams from actual practice.

The Indicator.

The original indicator, as invented and made by James Watt, has been since improved in detail although the principle remains the same. The object of the improvements has been to decrease the weight of the moving parts and to make the stroke of the piston as short as possible.

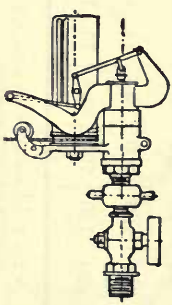


Fig. 990.—Richards' Indicator.

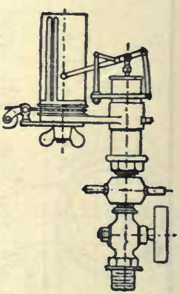


Fig. 991.—Thompson's Indicator.

Amongst the earliest improvements was that of Richards, fig. 990, who reduced the stroke of the piston but retained the longer stroke of the pencil by introducing a system of light levers which also formed a parallel motion for the same. This indicator is perhaps the most largely used, and with ordinary speeds gives very good results. The union or coupling by which this indicator is attached to the cock, is cut with threads of different pitches in the two parts, that which screws on to the indicator itself having a fine thread, and that part which screws on to the cock has a coarse thread ; the part of the cock is conical and fits a corresponding taper on the indicator ; this makes a very neat arrangement, and easy to take on and off.

Fig. 991 shews a modification of Richards' indicator by Thompson. A further modification by Crosby, fig. 992, has many points of interest, the spring fixed rigidly only to the cylinder cover, and is attached to the piston by a ball and socket joint ; the spring is also double wound, right and left hand, an arrangement intended to do away with any tendency to twist as the spring compresses. The whole indicator is much smaller than Richards', and better adapted

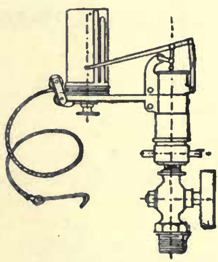


Fig. 992.—Crosby's Indicator.

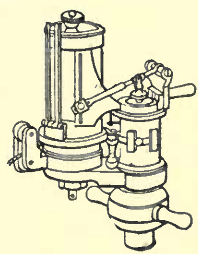


Fig. 993.—Darke's Indicator.

for very high speeds. The coupling for attaching the indicator is like that of Richards, but has a right and left hand screw instead of two right hand threads of different pitch ; this makes it much less easy to take on and off. A still smaller indicator is that of Darke, fig. 993, where the pencil is guided by a loose slide in a slot ; the coupling is the same as that of Richards, and is interchangeable with it. This indicator is arranged for very high speeds, and is fitted with a dentent or catch to hold the paper drum whilst string is being hooked on.

The stroke of the paper drums in all these indicators is necessarily much less than that of the engine. To reduce the stroke a moving lever is generally used, and care should be taken in leading the string to the indicator that its motion is always truly in the direction of the axis of the string.

A segment of wood fitted to the lever of correct proportion to give the right stroke will, if of sufficient extent, always ensure the string moving correctly. The levers may be made of wood, or better of iron. Figs. 994—997 shew an example of levers. The choice of arrangement of indicator gear necessarily depends on the style and

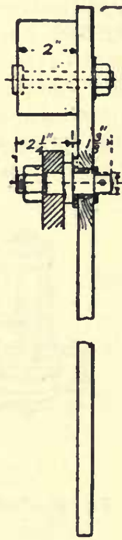


Fig. 994.

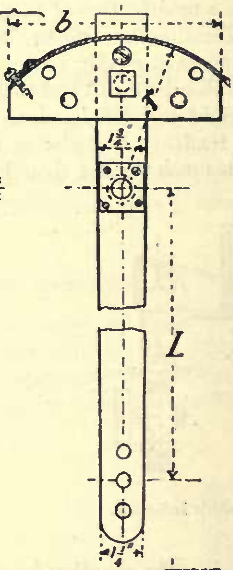


Fig. 995.

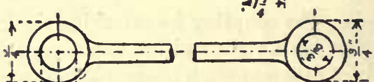


Fig. 996.

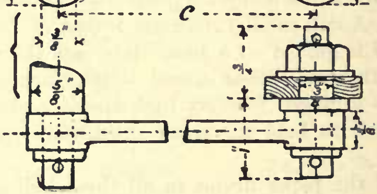


Fig. 997.

position of the engine to be indicated ; sometimes the lever may have to be attached to the walls or roof of the engine-room, but now the best makers will supply self-contained indicator gear with the engines they make.

Fig. 998 shews in outline an indicator gear of the ordinary lever kind, fitted to a horizontal girder engine. Fig. 999, a reducing gear with different sized drums pulled round by a string from an arm fixed to the cross head; these arrangements are not so simple as the

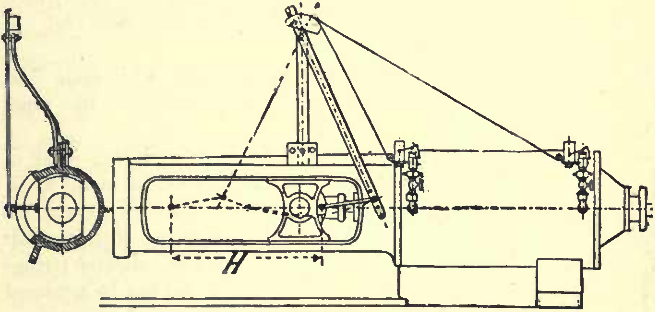


Fig. 998.

ordinary lever; the lower end of the lever should be connected to a pin in the crosshead by a link and not, as is often done, by a slot in

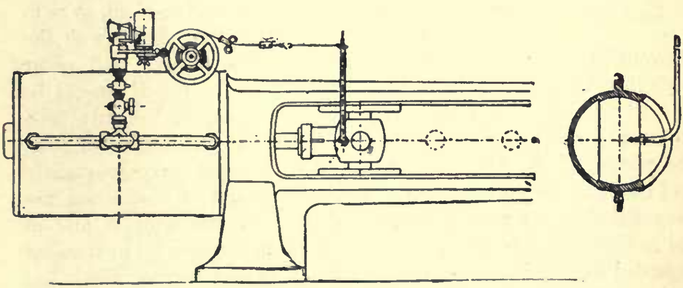


Fig. 999.

the end of the lever which gives more error to the motion of the paper due to the arc described by the end of the lever.

The practice of putting the indicator on a system of pipes from both ends of the cylinder, fig. 999, to which is fitted a 3-way cock, gives very pretty results (fig. 1000) by enabling diagrams from both ends of the cylinder to be produced on one card, but is not to be re-

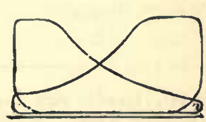


Fig. 1000.

commended, as elements of inaccuracy are thus introduced. For very exact work two indicators should be used, one for each end, and occasionally they should be changed over from one end to the other to eliminate a possible chance of error. An example of an extreme case of error from long pipes is given on page 378, fig. 1017; here the pipes were lead to the centre, and a cock fixed at each end, not even with a 3-way cock in the middle.

Fig. 1018 shews diagram taken from same engine with same load on with indicator fixed direct on each end of cylinder, the pipes having been removed.*

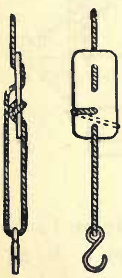


Fig. 1001.

Some operators prefer to have a sliding joint in the string by using a piece of wood or metal as in fig. 1001; this may be of use with slow speeds, but with high speeds a dead length string with a small ring at the end, and a hook on the indicator strings or *vice versa*, is the best, or the string can be arranged to pull through the edge of the wood segment on the lever and up to a knot, and can be held whilst taking the diagram, and thus the hook need not be used.

It is good before starting to indicate an engine to put the string on stretch in the engine-room, so as to get it to the same temperature and dryness as the surrounding air; if this be done, very little trouble will occur with the string whilst in use. In indicating the pencil should be pressed to the paper very gently, and if the engine has only been running for a few days or hours, that is, if the engine is new, the piston of the indicator should be taken out very frequently and the cylinder sponged out: the lower limits of a diagram are very liable to error from dirt accumulated in the cylinder, and an old indicator should always be used on a new engine; no instrument regarded as a standard should be applied to any engine which has not run itself clean from sand and dirt off the steam ports and passages. If after having taken a few diagrams the atmospheric line appears to be a trifle higher than on the first diagram it is a sure sign that the cylinder is foul, and it should be sponged out before any more diagrams are taken.

---

* Mr. Haeder says that the bosses for indicator cocks are tapped with 1" English thread. In England the custom is to tap with ¾" Whitworth thread.

To calculate the power from an indicator diagram, the average or mean pressure on the piston is found either by using a planimeter or more conveniently by the method of equidistant ordinates. The diagram is divided into ten equal parts, figs. 1002, 1003, and the mean height of each part is measured by the scale corresponding to the spring used in the indicator; the ten mean heights are then added together and divided by ten; this gives the mean pressure on the piston with fair accuracy. A quick and accurate way of measuring the added lengths or heights of the ordinates

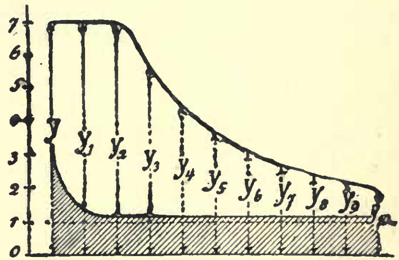


Fig. 1002.

is to take a narrow strip of paper and to mark off with a sharp pencil the length of the first division, then run the strip to the next and make a mark, and so on to the tenth, then measure the total length between the first and last mark and point off for dividing by ten; this gives the mean pressure, which multiplied by the area of the piston in square inches by twice the stroke in feet, and by the number of revolutions per minute, and divided by 33,000, gives the indicated horse-power. If diagrams be taken with a load on the engine and then with the load off, the difference

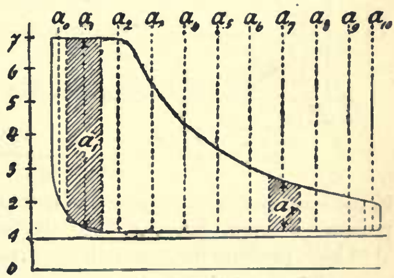


Fig. 1003.

between the two results will give the useful or effective horse power. It is not easy to get accurate measurements of power from a large engine when running without load, and therefore it is seldom possible to get the true effective load by this means; but a fairly accurate idea of the power taken by individual machines out of a large number driven by one engine may be obtained by indicating the engine with the particular machine on and then with it off, and taking the difference for the power required for driving the machine.

Indicator Diagrams.

Combined Diagrams from Compound Engines.

The proportion of the area of the combined diagrams to the area of the figure A, B, C, D, E, gives approximately the efficiency of the proportion of the efficiency of the engine. To combine the diagrams from the two cylinders of a compound engine they may be laid out

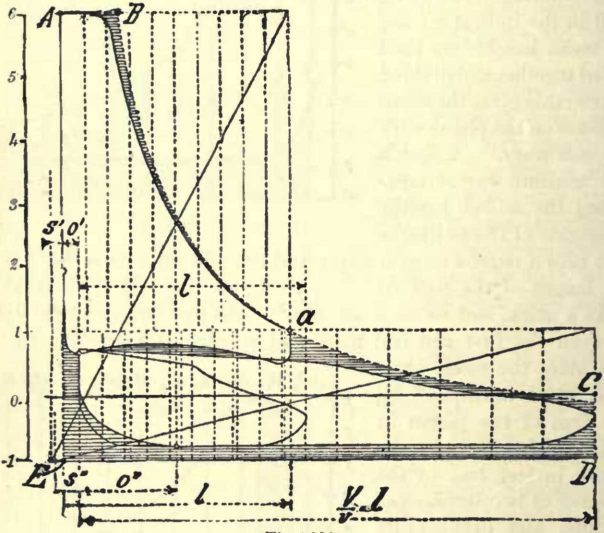


Fig. 1004.

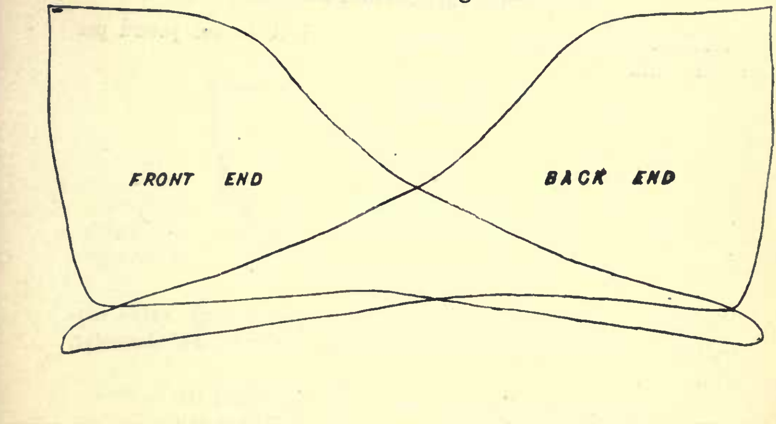
to the same scale and length, l, so that the admission line of the low pressure diagram comes just to the beginning of the compression o^1 of high pressure diagram; the low pressure diagram is then laid out to a length equal to the proportion of the cylinder volumes $\frac{V}{v} \times l$.

The theoretical expansion curve is drawn through the point a, and the shaded portion will show the theoretical loss, fig. 1004.

s' = the clearance of the high-pressure cylinder.
s'' = the clearance of the low-pressure cylinder.
o' = the compression in high-pressure cylinder.
o'' = the compression in low-pressure cylinder.

The effect of laying out combined diagrams, is to produce a diagram which should represent what would take place in one cylinder of the volume of the low-pressure cylinder of the compound engine.

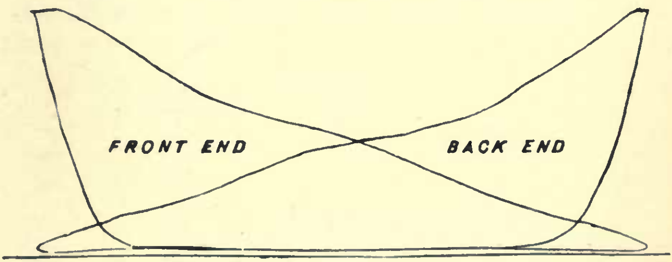
Indicator Diagrams.



Scale $\frac{1}{64}$

HIGH PRESSURE CYLINDER

Fig. 1005.



Scale $\frac{1}{32}$

LOW PRESSURE CYLINDER

Fig. 1006.

Diagrams* from compound engine with Davey Paxman's valve gear controlled by the governor. The two valves are driven by separate eccentrics, and the cut-off valve works on a false port face between the main and cut-off valves.

* See Royal Agricultural Society's report of Newcastle meeting, 1887.

Diagrams from Actual Practice.

Scales of all are given in fractions of an inch to one pound per square inch.

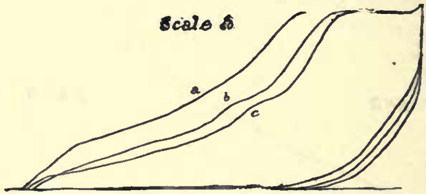
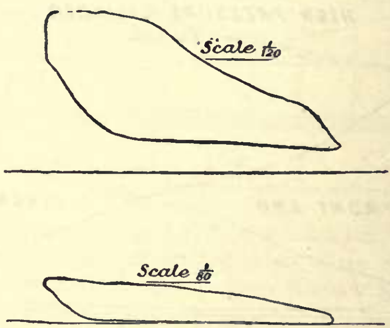


Fig. 1007.

*Diagrams from a portable engine fitted with single valve controlled by Hartnell's crank-shaft governor. Cylinder 8½″ diameter, stroke 12″.

a. 127 revolutions per minute 13·9 indicated HP.
b. 129 revolutions per minute 10·8 indicated HP.
c. 130 revolutions per minute 10·1 indicated HP.



Figs. 1008, 1009.

Diagrams from compound undertype engine with high-pressure cylinder valve controlled by Hartnell's governor low-pressure valve by simple eccentric.

High-pressure cylinder 8″ diameter.
Low-pressure cylinder 12¾″ diameter.
Stroke of both, 14″.
Revolutions per minute, 155.

* See reports of Cardiff meeting of the Royal Agricultural Society.

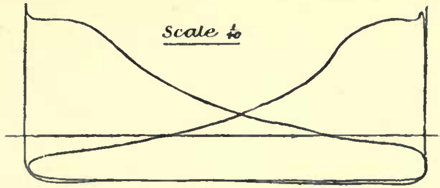


Fig. 1010.

Diagrams from condensing engine with Meyer valve gear and throttle valve. Cylinder 18″ diameter ; 30″ stroke. Revolutions per minute 60.

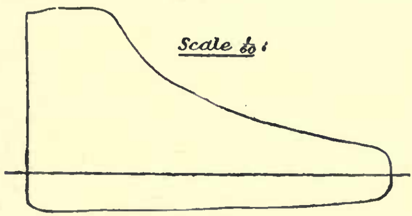
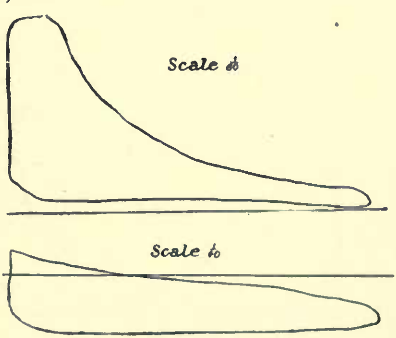


Fig. 1011.

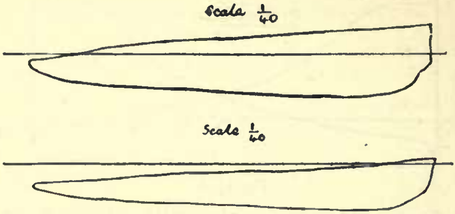
Diagram from condensing engine with cut-off valve controlled by Hartnell governor. Cylinder 16½″ diameter ; 36″ stroke. Revolutions per minute, 63.



Figs. 1012, 1013.

Diagrams from a compound horizontal engine, cranks at 90°, cut-off valve of high-pressure cylinder controlled by Hartnell's governor the valve of low-pressure cylinder worked by simple eccentric.

Diameter of high-pressure cylinder, 16″; low-pressure cylinder, 26″ Stroke of both, 36″. Revolutions per minute, 67.



Figs. 1014, 1015.

Diagrams from a side lever marine engine by Miller & Ravenhill, dated 1839, used as a stationary engine.

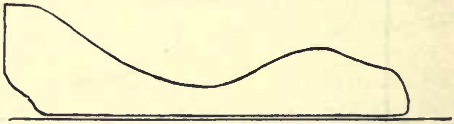
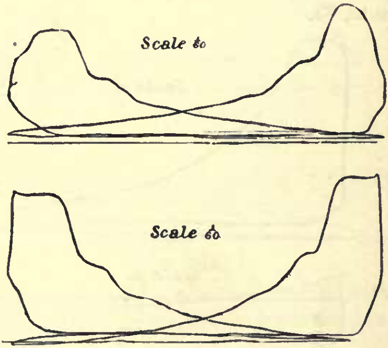


Fig. 1016.

Diagram shewing a case of accidental re-admission by main valve over-running port face. Cylinder $11\frac{1}{4}''$ diameter, $24''$ stroke.



Figs. 1017, 1018

Fig. 1017 shews the effect of long pipes between cylinder and indicator.

Fig. 1018 from same engine with same load, the indicator being fixed direct on cylinder without any pipes. Cylinder $13\frac{1}{2}''$ diameter, $24''$ stroke, 100 revolutions per minute.

Diagrams shewing the Defects in Valve Gears which may often be met with in Practice.

Example of a normal indicator diagram, from an engine with good expansion gear.

Diagram shewing the effect of throttling.

Too early admission, with light fly-wheel.

Too late admission.

Leaking valves, allowing steam to pass after cut-off.

Re-admission after cut-off, caused by re-opening cut-off ports before main valve has closed the steam-port. See page 192.

Too late opening of exhaust, or may be caused by exhaust being too small.

Load too light for size of engine.

Too little compression.

Too much compression.

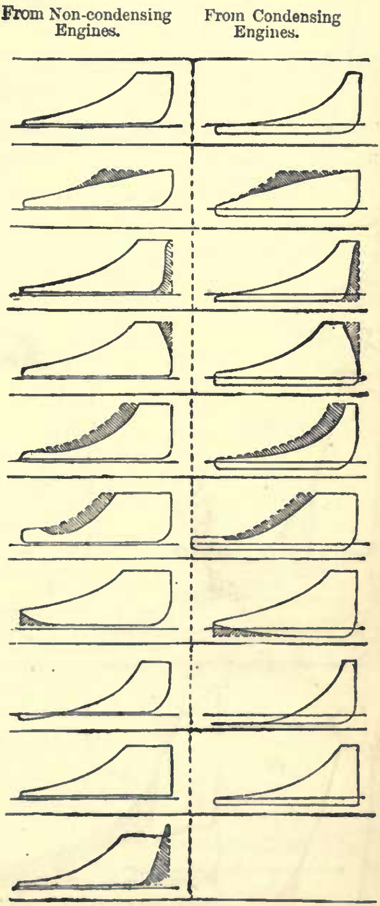


Fig. 1019.

The shaded parts in the above diagrams shew the losses.

C C

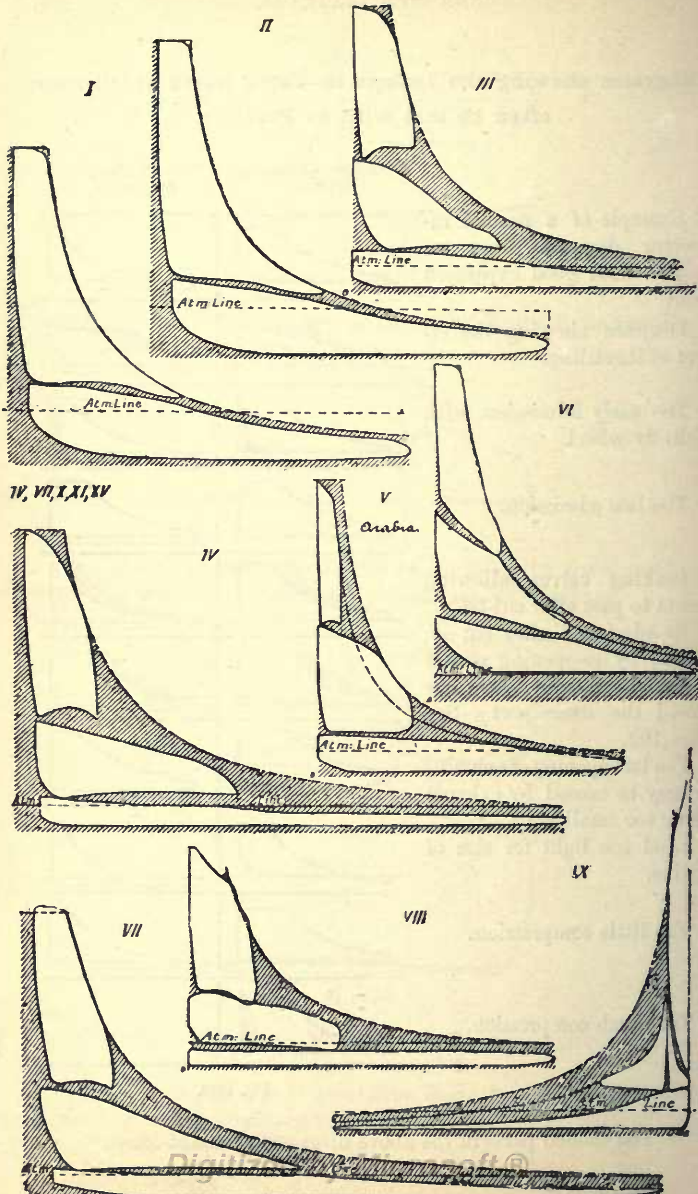
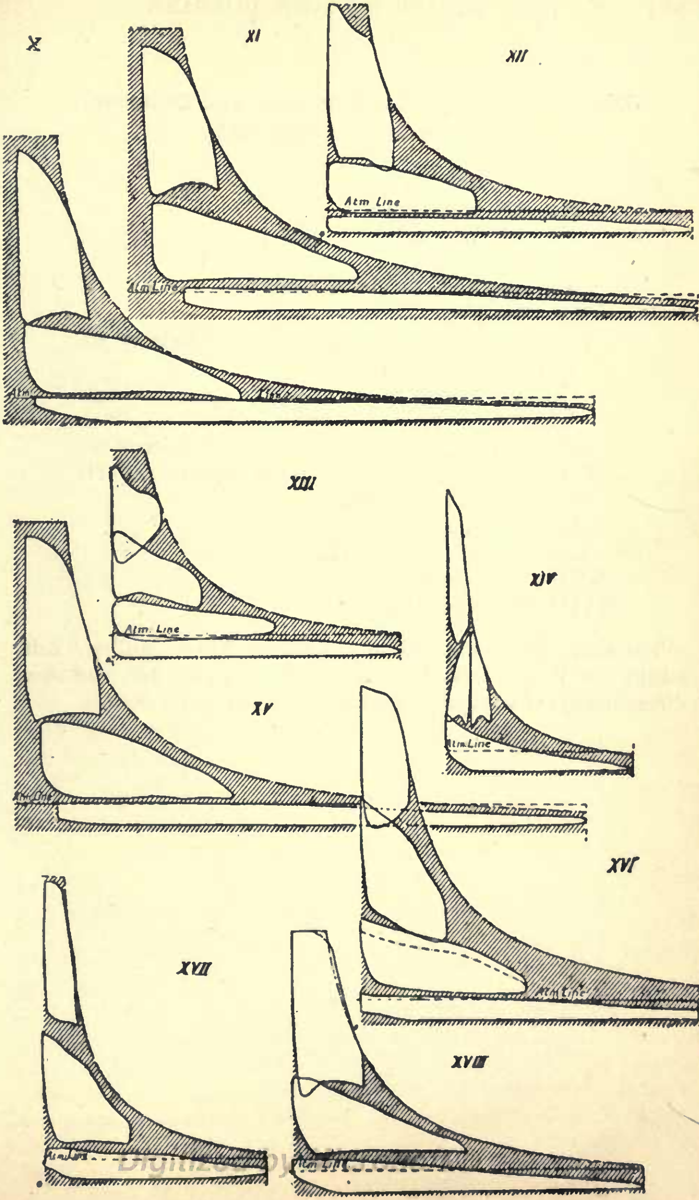

II
I
III
Atm: Line
Atm Line
Atm.Line
VI
IV, VII, X, XI, XV
V
Arabia
IV
Atm: Line
Atm: Line
IX
VIII
VII
Atm: Line
Atm
Line

Fig. 1020.


X
XI
XII
Atm Line
Atm Line
XIII
Atm. Line
XIV
Atm Line
XV
XVI
XVII
XVIII

Diagrams from Triple-Expansion and Compound Engines, pages 380, 381.

Nos. I. & II., S.S. Bachheibel.
III., S.S. Falkenburg.
IV., S.S. African.
V., S.S. Arabia.
VI., Stülchen's 3-cylinder compound.
VII., S.S. Para.
VIII., S.S. Aberdeen, with superheater efficiency ·67.
IX., S.S. Wanderer.
X., S.S. Stella.
XI., S.S. Lusitania.
XII., S.S. Aberdeen, with superheater efficiency ·65.
XIII., Adamson's quadruple-expansion efficiency ·71.
XIV., S.S. Isle of Dursey.
XV., S.S. Jungfrau.
XVI., S.S. Rimnag na Maia.
XVII., S.S. Sobraleuse.
XVIII., S.S. Nierstein.

The above diagrams are from a paper by Otto H. Müller, "Zeitschrift der Verein Deutsch. Ingenieur," 1887, page 445, and shew different ways of combining diagrams from compound engines.

SECTION X.

CALCULATIONS FOR POWER AND STEAM CONSUMPTION.

THE following terms and letters are used in these calculations :—

N_i = the indicated horse-power.
N_e = the effective horse-power.
Q = the effective surface of the piston in square inches.
H = the stroke in inches.
n = the revolutions per minute.
c = the piston speed in feet per minute.
h = the admission or cut-off when $H = 1$.
P = the initial pressure in lbs. per square inch.
k = expansion coefficient dependent upon the admission h, and the clearance s, see Table 135.
s = the clearance space in terms of piston area, so that the length of the line s on the diagrams, figs. 1022, 1023, represents the volume of the clearance space when $H = 1$.
P_m = the mean pressure on the piston in lbs. per square inch.
P_o = the back pressure in lbs. per square inch.
σ = the sum of the losses of work by wire-drawing, compression, early opening of exhaust, back pressure, etc.

$$\text{then } c = 2Hn.$$

$$N_i = \frac{\overset{A}{Q}\, c\, \overset{P}{P_m}}{33{,}000},$$

The mean pressure $P_m = kp - (P_o + \sigma)$.

The expansion coefficient $k = h + (h + p) \log_e \dfrac{1 + s}{h + s}$; see Tables 135 and 169.

The back pressure P_o of the exhaust steam is dependent on the terminal pressure ω, and the size of the exhaust passages.

Table 137 gives the value of the back pressure for usual proportions, and for a speed of about 100 feet per second of the steam in the passages, allowance also being made for the sum of the losses σ.

TABLE 135.—**Value of the Expansion Coefficient k.**

h	Clearance space s.								
	2%	3%	4%	5%	6%	7%	8%	9%	10%
0·00	0·079	0·107	0·130	0·152	0·172	0·191	0·210	0·226	0·240
0·02	0·151	0·173	0·190	0·210	0·230	0·250	0·263	0·276	0·289
0·04	0·204	0·232	0·250	0·268	0·280	0·292	0·302	0·314	0·328
0·06	0·255	0·273	0·292	0·303	0·321	0·332	0·343	0·353	0·366
0·08	0·305	0·321	0·337	0·348	0·363	0·371	0·383	0·392	0·403
0·10	0·356	0·369	0·381	0·392	0·403	0·412	0·422	0·432	0·440
0·12	0·394	0·406	0·417	0·427	0·437	0·446	0·455	0·464	0·472
0·14	0·431	0·442	0·452	0·462	0·470	0·479	0·487	0·495	0·503
0·16	0·467	0·477	0·486	0·496	0·502	0·511	0·518	0·525	0·533
0·18	0·502	0·513	0·519	0·529	0·533	0·542	0·548	0·554	0·562
0·20	0·535	0·545	0·552	0·559	0·565	0·571	0·577	0·584	0·590
0·22	0·564	0·573	0·578	0·586	0·592	0·597	0·603	0·609	0·615
0·24	0·592	0·600	0·606	0·612	0·615	0·622	0·628	0·633	0·639
0·26	0·619	0·626	0·631	0·637	0·643	0·646	0·652	0·656	0·662
0·28	0·645	0·651	0·655	0·661	0·667	0·671	0·675	0·678	0·683
0·30	0·670	0·675	0·680	0·685	0·689	0·692	0·696	0·700	0·704
0·32	0·693	0·697	0·702	0·706	0·710	0·714	0·718	0·721	0·725
0·34	0·715	0·718	0·723	0·726	0·730	0·734	0·738	0·741	0·745
0·36	0·736	0·738	0·743	0·745	0·749	0·753	0·757	0·760	0·764
0·38	0·756	0·757	0·762	0·763	0·767	0·772	0·775	0·778	0·782
0·40	0·773	0·775	0·779	0·781	0·784	0·787	0·794	0·797	0·800
0·42	0·791	0·792	0·794	0·798	0·801	0·803	0·810	0·812	0·815
0·44	0·808	0·809	0·810	0·814	0·817	0·818	0·824	0·826	0·829
0·46	0·824	0·825	0·827	0·829	0·832	0·834	0·837	0·839	0·842
0·48	0·838	0·839	0·841	0·843	0·845	0·847	0·849	0·851	0·854
0·50	0·850	0·852	0·854	0·856	0·857	0·858	0·862	0·864	0·866
0·55	0·879	0·881	0·883	0·885	0·886	0·887	0·889	0·890	0·891
0·60	0·906	0·908	0·910	0·912	0·913	0·913	0·914	0·915	0·916
0·65	0·927	0·929	0·931	0·932	0·933	0·934	0·935	0·935	0·936
0·70	0·947	0·949	0·951	0·952	0·953	0·953	0·954	0·954	0·955
0·75	0·962	0·964	0·966	0·967	0·968	0·968	0·968	0·968	0·973
0·80	0·976	0·978	0·980	0·980	0·981	0·981	0·981	0·981	0·981
0·90	0·994	0·995	0·995	0·995	0·996	0·997	0·997	0·998	0·998

Example.—Given the cut-off h = ·2, the clearance space s = 7 per cent., the expansion coefficient k = ·571.

The losses of power from wiredrawing, &c., as shown in diagrams, figs. 1022, 1023, are represented by σ, expressed in

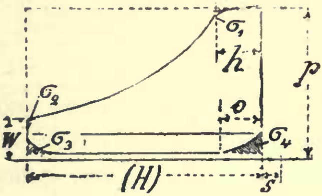


Fig. 1022.

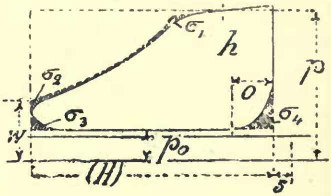


Fig. 1023.

pressure on the piston area during the whole stroke in lbs. per square inch.

σ_1 the loss due to wiredrawing or throttling the steam during admission.
σ_2 the loss due to early opening of the exhaust.
σ_3 the loss due to the back pressure of the exhaust steam.
σ_4 the loss due to compression.
σ_5 the loss due to the drop in pressure in compound engines.
$\sigma = \sigma_1 + \sigma_2 + \sigma_3 + \sigma_4 + \sigma_5 =$ the sum of all these losses.

The loss σ, by wiredrawing increases with the lateness of the cut-off h for engines with the usual valve-gears, Meyer's, Rider's, &c., and is given in Table 136.

TABLE 136.—**Values of σ_1.**

Cut-off h . .	·05	·10	·15	·20	·30	·40	·50	·60	·70
Without jacket .	1·14	1·42	1·70	2·15	2·56	2·84	3·41	3·70	4·00
With jacket .	·43	·57	·71	1·00	1·14	1·42	1·70	2·00	2·13

TABLE 137.—**Values of σ_2 in lbs. per Square Inch.**

Percentage of exhaust opening before end of stroke.	Without condenser. Final pressure w in lbs. per sq. in.				With condenser. Final press. in lbs. per sq. in.		
	17·64	29·4	44·1	58·8	14·7	29·4	44·1
·02	·000	·000	·000	·000	·000	·000	·000
·05	·000	·073	·147	·220	·073	·117	·147
·10	·043	·220	·430	·588	·294	·357	·430
·20	—	1·17	1·32	1·47	·882	1·17	1·32
·30	—	1·47	1·91	2·20	—	—	—

If the exhaust opens up 2 per cent. or less before the end of the stroke the loss is very small, some valve gears, however, will not allow of the exhaust being kept closed later than from 20 to 30 per cent. before the end of the stroke.

The values of σ_2 are given in Table 137.

TABLE 138.—**Values of Back Pressure σ_3.**

Exhaust lead. Vo	Without condenser. Final pressure w in lbs. per sq. in.				With condenser. Final press. in lbs. per sq. in.		
	18·37	29·4	44·1	58·8	14·7	29·4	44·1
$Vo = a$	·000	·043	·147	·430	·073	·147	·294
$Vo = 0·5\,a$	·073	·117	·220	·588	·147	·730	1·17
$Vo = 0·2\,a$	·294	·430	·588	1·03	·588	1·17	1·76
$Vo = 0$	·588	1·32	1·47	1·76	1·17	1·47	2·64

The values of σ_3, the back pressure or exhaust resistance caused by early closing, are dependent on the inside or exhaust lead, and the final pressure w, and are given in Table 138.

Values of Compression σ_4.

TABLE 139.—**Without Condenser** (ρ_o = 16·4 lbs. per sq. in.).

Compression 0	Clearance space in per cent. of stroke s.								
	2%	3%	4%	5%	6%	7%	8%	9%	10%
0·00	0	0	0	0	0	0	—	—	—
0·025	·228	·214	·209	·185	·171	·128	·114	·085	·071
0·050	·626	·551	·485	·415	·341	·271	·228	·185	·142
0·075	·925	·855	·782	·712	·640	·571	·485	·385	·285
0·10	1·550	1·38	1·14	·891	·792	·732	·700	·685	·670
0·15	—	2·70	2·34	1·96	1·71	1·44	1·26	1·06	·99
0·20	—	—	3·47	3·25	2·90	2·56	2·28	2·09	1·71
0·25	—	—	—	4·45	4·10	3·76	3·51	3·28	2·98
0·30	—	—	—	—	5·32	5·00	4·75	4·46	3·75

TABLE 140.—**With Condenser** (ρ_o = 3·13 lbs. per sq. in.).

0·00	0	0	0	0	0	0	0	0	0
0·025	·028	·028	·028	·028	·014	·014	·014	·014	·014
0·050	·071	·057	·043	·043	·028	·014	·014	·014	·014
0·075	·142	·128	·114	·081	·057	·043	·028	·014	·014
0·10	·284	·256	·214	·157	·128	·100	·071	·043	·028
0·15	·525	·470	·412	·341	·285	·256	·228	·209	·171
0·20	·821	·770	·670	·570	·512	·455	·400	·341	·314
0·25	1·10	1·01	·925	·810	·770	·728	·640	·527	·485
0·30	1·54	1·41	1·24	1·03	·970	·855	·810	·770	·728

TABLE 141.—**Values of the Cut-off h with a given Final Pressure, w, for Clearance Spaces of 3 to 7 per cent.**

At. ab.	$p=8$ Atmos. Absolute.			$p=7$ Atmos. Absolute.			$p=6$ Atmos. Absolute.			$p=5$ Atmos. Absolute.			$p=4$ Atmos. Absolute.		
w	$s=3$°/。	5°/。	7°/。	3°/。	5°/。	7°/。	3°/。	5°/。	7°/。	3°/。	5°/。	7°/。	3°/。	5°/。	7°/。
0·6	0·047	0·027	0·012	0·058	0·040	0·021	0·073	0·055	0·037	0·093	0·076	0·059	0·124	0·106	0·090
0·7	0·061	0·042	0·023	0·073	0·055	0·037	0·090	0·072	0·054	0·114	0·097	0·080	0·150	0·134	0·118
0·8	0·073	0·055	0·037	0·088	0·070	0·052	0·107	0·090	0·072	0·135	0·118	0·101	0·176	0·160	0·144
0·9	0·086	0·068	0·053	0·103	0·085	0·067	0·124	0·107	0·096	0·155	0·139	0·123	0·202	0·186	0·171
1·0	0·098	0·081	0·064	0·117	0·100	0·083	0·141	0·125	0·108	0·176	0·160	0·144	0·229	0·212	0·197
1·2	0·127	0·107	0·095	0·147	0·130	0·113	0·174	0·160	0·144	0·217	0·202	0·183	0·279	0·265	0·251
1·4	0·150	0·133	0·117	0·176	0·160	0·144	0·210	0·195	0·179	0·255	0·244	0·229	0·337	0·318	0·305
1·6	0·178	0·160	0·144	0·205	0·190	0·174	0·244	0·230	0·215	0·299	0·286	0·272	0·389	0·370	0·358
1·8	0·202	0·186	0·171	0·234	0·220	0·205	0·279	0·265	0·251	0·340	0·328	0·315	0·434	0·422	0·412
2·0	0·227	0·212	0·197	0·265	0·250	0·235	0·313	0·300	0·287	0·382	0·370	0·358	0·485	0·475	0·465
2·5	0·292	0·278	0·264	0·337	0·325	0·312	0·399	0·387	0·375	0·485	0·475	0·465	0·614	0·606	0·600
3·0	0·356	0·343	0·331	0·411	0·400	0·390	0·485	0·475	0·465	0·588	0·580	0·572	0·742	0·737	0·732
3·5	0·420	0·409	0·398	0·485	0·475	0·405	0·571	0·562	0·554	0·693	0·685	0·679	0·871	0·868	0·862
4·0	0·485	0·475	0·465	0·558	0·550	0·541	0·656	0·650	0·643	0·794	0·790	0·786	1·000	1·000	1·000
4·5	0·549	0·540	0·531	0·632	0·625	0·617	0·742	0·737	0·731	0·897	0·893	0·890	—	—	—
5·0	0·610	0·600	0·599	0·705	0·700	0·695	0·830	0·821	0·811	1·000	1·000	1·000	—	—	—

Pressures in Atmospheres Absolute.

Example.—Given the final pressure $w=1·2$ atmospheres absolute, the clearance $s=5$ per cent., and the initial pressure $p=7$ atmospheres absolute, then the cut-off h will $=0·13$.

Values of the Final Pressure w in Atmospheres Absolute for the most Economical Normal Horse-power.

TABLE 142.—Simple Engine.

N_i.	Non-condensing.				Condensing.			
	Initial pressure p in Atmospheres Absolute.							
	4—5·5	6—7·5	8—9	10	4—5·5	6—7·5	8—9	10
2—5	2·0	2·0	2·1	2·2	—	—	—	—
5—10	1·8	1·8	1·9	2·1	—	—	—	—
10—50	1·6	1·7	1·8	2·0	0·8	0·9	1·0	1·1
50—100	1·5	1·6	1·7	1·9	0·8	0·9	1·0	1·1
100—200	1·4	1·5	1·6	1·8	0·7	0·8	0·9	1·0
200 and upwards.	1·3	1·4	1·5	1·7	0·7	0·8	0·9	1·0

TABLE 142A.—Compound Engine

10—50	—	—	1·6	1·7	0·7	0·8	0·9	1·9
50—100	—	—	1·5	1·6	0·7	0·8	0·8	0·9
100—500	—	—	1·4	1·5	0·6	0·7	0·8	0·9
500 and upwards.	—	—	1·3	1·4	0·5	0·6	0 7	0·8

Example.—The best final pressure w for a simple condensing engine with N_i = 150 H.P., and 7 atmospheres absolute is from the Table ·8 atmospheres absolute, the corresponding cut-off from Table 133, with 7 per cent. clearance space, gives ·052, The losses σ_4 by compression given in Tables 139, 140, are dependent on the length of the compression period o, and the back pressure P_o. The clearance space depends on the kind of valve-gear used. See Table 33.

Clearance space s expressed in terms of length of stroke, for $H = 1$.

TABLE 143.

Kind of valve-gear used.			
Ordinary valve-gear.	Divided valves and piston valves.	Mushroom valves.	Corliss valves.
0·06 – 0·08	0·03 – 0·06	0·04	0·025

The above values for s are for ordinary piston speeds. For very quick running engines with higher piston speed, the clearance space may be double the value given above.

A coefficient approximately proportional to the efficiency of an engine may be found by dividing the mean pressure (taken from the indicator diagram) by the final pressure (absolute) $\frac{P_m}{w}$. For example, an engine giving a mean pressure of 37, and a final pressure absolute of 23·7, gives this coefficient $\frac{37}{23 \cdot 7} = 1 \cdot 56$. In many engines this coefficient is greater than 2, but has hardly reached 3.

The water per IHP per hour, can be approximated from the coefficient thus obtained. Divide the undermentioned constants by the coefficient, and the quotient will give the water per IHP per hour, in lbs.

When the final pressure is about 5 lbs. *below* the atmosphere, the constant = 36; with a final pressure 15 lbs. *above* the atmosphere, the constant = 34; when final pressure is 45 lbs. above atmosphere the constant = 32.

Either of these divided by the coefficient will give the quantity of water per IHP per hour, very nearly, *exclusive* of the losses by clearance and condensation, jacket, &c., and the gain by compression.

Take as an example the engine before referred to which had a final pressure of 9 lbs. above the atmosphere, and use corresponding constant 34. $\frac{34}{1{\cdot}56} = 21{\cdot}7$ lbs., add $\frac{1}{10}$ for clearance and loss exclusive of jacket = 23·87 lbs. per 1 H.P. per hour. The water measured was 24·1 lbs. per 1 H.P. per hour, which shows how near the formula is.

Let $l = l' + l''$ resistance in non-condensing engine running light;
$l = l'_c + l''_c$ " condensing " "
μ coefficient of additional friction for single cylinder engine.
μ_s " " " for double " "

The effective power for single cylinder $\frac{Qc\,(P_m - l)}{33000\,(l + \mu)}$.

" " for double cylinder $\frac{Qc\,(P_m - l)}{33000\,(1 + \mu_s)}$.

TABLE 144.

D	8	16	24	32	14
l''	1·86	1·00	·571	·428	·285
l''_c	3·14	1·57	1·00	·715	·571
μ	·18	·14	·12	·10	·08
μ_s	·20	·16	·13	·11	·10

TABLE 145.

p	59	88	118
l'	1·14	1·43	1·71
l'	1·71	2·00	2·28

p = pressure in lbs. per sq. inch.

By very careful management the coefficient μ may be reduced by 30%.

Calculation of the Power of Compound Engines.

The power of a compound engine should be the same as if the total expansion had been carried out in one large cylinder, the expression for the total expansion being

$$\frac{\text{initial absolute pressure}}{\text{final absolute pressure}} = \frac{P}{w}.$$

In figures 1024, 1025,

h = the period of admission.
s = the clearance.
w = the final pressure.
d = the diameter of the high-pressure cylinder.
h', s', w', = as above for the high-pressure cylinder.
D = diameter of low-pressure cylinder.
h'', s'', w'' = as above for the low-pressure cylinder.
$\frac{V}{v}$ = the ratio of the cylinder volumes.
Q = the surface of the low-pressure piston.
h_i = the ideal admission reduced the low-pressure cylinder volume corresponding to the total expansion.
$s = \frac{s''}{\frac{V}{v}}$ for determining the mean pressure for the ideal value of the clearance space.

Then, taking the stroke of both cylinders to be the same,

Taking no account of the clearance space.

$$\frac{V}{v \times h'} = \frac{P}{W''} = \frac{1}{h'} \times \frac{1}{h''}.$$

Allowing for the clearance space.

$$\frac{V \times (l + s'')}{v \times (h' + s')} = \frac{P}{W''} = \frac{1}{h' + s'} \times \frac{1}{h'' + s''}.$$

Example.—To calculate the nominal power of a compound condensing engine.

Diameter of high-pressure cylinder . .	d	$15\frac{3}{4}$".
Diameter of low-pressure cylinder .	D	24".
Stroke of both	H	$27\frac{1}{2}$".

Ratio of cylinder volumes . . . $\frac{V}{v} = 2{\cdot}35.$

Revolutions per minute . . . . $n = 75.$

Initial pressure absolute . . . $P = 103.$

Clearance space of low-pressure cylinder $s'' =$ about 5 per cent.

Final pressure $w = w''$ from Table 142A, $w = 11{\cdot}76.$

Ideal clearance space $s_i = \frac{0{\cdot}05}{2{\cdot}35}$ . . $s_i = 0{\cdot}02.$

Ideal admission from Table 141 . . $h_i = 0{\cdot}1.$

Coefficient of expansion from Table 135 . $k = 0{\cdot}356.$

Back pressure and loss of work . $P_o + \sigma = 7.$

Then the mean pressure $P_m = kp - (P_o + \sigma) = {\cdot}356 \times 103 - 7 = 30{\cdot}7$ lbs. per square inch.

The piston speed in feet per minute, $c = 344.$

the effective area of the piston, $Q = 446$ square inches ;

the ideal indicated horse-power N_i,

$$= \frac{Qc\,P_m}{33000} = \frac{446 \times 344 \times 30{\cdot}7}{33000} = \text{approximately } 140.$$

The friction coefficient $\mu_z = 0{\cdot}16$ (Table 144).

The resistance of the engine when running empty (see Tables 144, 145). $l = l'_c = 2{\cdot}14 + 1{\cdot}56 = 3{\cdot}7$;

then the effective horse-power N_e

$$= \frac{Qc\,(P_m - l)}{33000\,(1 + \mu_s)} = \frac{446 \times 344\,(3017 - 3{\cdot}7)}{33000\,(1 + 0{\cdot}16)} = \text{nearly } 103.$$

The admission or cut-off in the high-pressure cylinder is determined as follows :—

The total expansion being $\frac{P}{w''} = \frac{103}{11{\cdot}76} = 8{\cdot}75.$

If now $s' = s'' = 0{\cdot}05$,

then $$\frac{V\,(1 + s'')}{v\,(h' + s')} = \frac{2{\cdot}35 \times 1{\cdot}05}{h' + 0{\cdot}05} = 8{\cdot}75\,;$$

therefore $$h' = \frac{2{\cdot}35 \times 1{\cdot}05}{8{\cdot}75} - 0{\cdot}05 = 0{\cdot}23.$$

The final pressure in the high-pressure cylinder will be

$$w' = \frac{(h' + s')\,P}{1 + s'} = \frac{(0{\cdot}23 + 0{\cdot}05) \times 103}{1{\cdot}05} = 27{\cdot}4$$

and we shall have the mean pressure in the low-pressure cylinder, taking the drop of pressure as 5·88,

$$p'' = 27{\cdot}4 - 5{\cdot}88 = 21{\cdot}52$$

then it follows that

$$h'' = \frac{w''\,(1 + s'')}{p''} - s'' = \frac{11{\cdot}76 \times 1{\cdot}05}{21{\cdot}52} - 0{\cdot}05 = {\cdot}525 \text{ sq. } {\cdot}55.$$

If the horse-power is given, and the diameter of the low-pressure cylinder is required, it can be approximated in the following manner:

From Table 142, p is selected, P_m from Table 146, and c from Table 147,

$$\text{then } Q = \frac{33000 \times N_i}{c\,P_m}.$$

TABLE 146.

	Initial pressure in lbs. per square inch.						
	59	74	88	103	118	132	147
$P_m =$	23·5	26·5	31	35·2	38·2	41	45·2

Indicator Diagrams for the Normal Horse-Power for the Compound Engines given in Table 147.

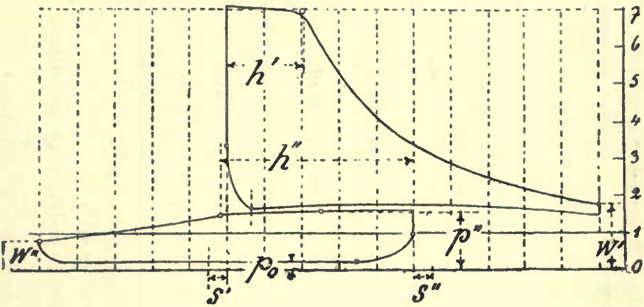


Fig. 1024.—Pressures in atmospheres absolute.

$$w' = \frac{(h' + s')\,p}{1 + s'}; \quad w'' = \frac{(h'' + s'')\,p''}{1 + s''};$$

$$h' = \frac{w'\,(1 + s')}{p} - s'; \quad h'' = \frac{w''\,(1 + s'')}{p''} - s'';$$

$$w' - p'' = \text{the drop in the pressure.}$$

The power in both cylinders is nearly equal.

Indicator Diagram for the Maximum Horse-Power for the Compound Engines given in Table 147.

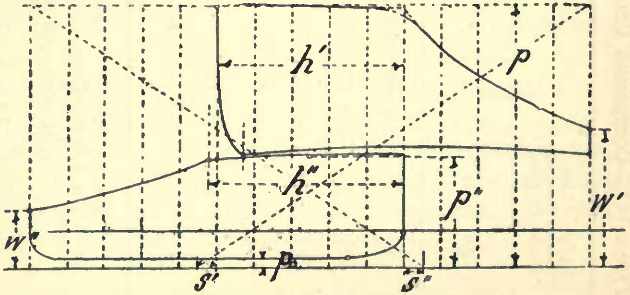


Fig. 1025.

Table 147.—**Compound Condensing Engines, 103 lbs. Absolute Pressure per square inch.**

Stroke.	High-Pressure Cylinder	Low-Pressure Cylinder														Per I.H.P. per hour.		Injection Water in gals. per hour.	Maximum power effective.	
H	d	D	$\frac{V}{v}$	n	c	Q	w	N_i	k	P_o	O	P_m	N	N_e		Water in lbs.	Coal in lbs.		N_e	Water per N_i per I.H.P. per hour in lbs.
20	11¾	18	2·3	90	300	254	11·76	·10	·356	3·1	2·7	30·7	65	47	45	18·66	2·86	2500	68	25·2
24	13¾	21	2·3	80	320	346	11·76	·10	·356	3·1	2·7	30·7	96	71	70	18·07	2·77	3525	107	23·8
28	15¾	24	2·3	75	350	452	11·76	·10	·356	3·1	2·7	30·7	140	106	100	17·62	2·64	5000	151	22·5
32	17½	27	2·35	70	370	572	11	·09	·33	3·1	2·7	28·1	174	132	130	16·95	2·40	6025	208	21·2
36	20	30½	2·35	67	400	730	11	·09	·33	3·1	2·7	28·1	235	179	175	16·29	2·20	7700	280	19·8
40	21¾	33½	2·37	65	430	881	11	·09	·33	3·1	2·7	28·1	308	237	230	15·63	2·09	9500	374	18·7

The high-pressure cylinder has variable cut-off from ·0 to ·5, compression 0′ = 1.
The low-pressure cylinder has fixed cut-off at ·5, compression 0″ = 2·56.

$\frac{V}{v}$ = ratio of the cylinder volumes ; Q = surface of low-pressure piston in square inches.

$s_i = \frac{s'}{\frac{V}{v}}$ = ideal clearance space ; h_i = ideal cut-off ; w = final pressure absolute.

SECTION XI.

THE EFFECT OF THE INERTIA OF THE RECIPROCATING PARTS OF A STEAM ENGINE.

THE effect of the inertia of the reciprocating parts of an engine especially with high speeds may have a great influence on the smoothness of the running.* The effect of inertia has been ignored by engine makers, it was pointed out in a paper on the Allen engine by Mr. C. T. Porter in 1868,† with special reference to the change in direction of the pressure on the crank-pins of steam engines at certain parts of the stroke.

Let P = the weight of the reciprocating mass in lbs.
f = the surface of the piston in square inches.

The approximate proportion of weight to piston surface will be $\frac{P}{f} = 3{\cdot}98$ for horizontal non-condensing engines, $\frac{P}{f} = 4{\cdot}27$ for horizontal condensing engines. In the Allen engine above alluded to, the weight of the reciprocating parts was 470 lbs., and the piston surface 113 square inches; this gives the proportion P to f as 4·16 to 1).

H = stroke in feet.
r = Crank radius in feet.
n = Revolutions per minute.

$v = \frac{2 r \pi n}{60}$ = the average speed of the crank pin in feet per second.

(a.) The Connecting-Rod of Infinite Length.

The pull on the connecting-rod necessary to overcome the inertia of the reciprocating parts is equal to the horizontal component of the acceleration of the crank pin multiplied by the mass moved; at the dead point this component is equal to the total acceleration of the crank pin towards the centre of the crank shaft, *i.e.*, to $\frac{v^2}{r}$.

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* Radinger Maschinen mit höher Kolbengeschwindigkeit.
† Institution of Mechanical Engineers' Proceedings, April, 1868.

The total drag on the connecting-rod at the beginning of the stroke is therefore $\frac{Pv^2}{gr}$, hence the pressure on unit surface of the face of the piston necessary to neutralize this $-\frac{Pv^2}{grf} = q1$. At the highest point of the crank $q = 0$, since at that point there will be no further pressure expended in producing acceleration of the reciprocating parts, as the piston has attained the same speed as the crank pin. At intermediate points $q = q_1 \cos. \omega$, where $\omega =$ the angle swept out by the crank pin from the dead point.

It is easy to represent graphically the retarding or accelerating forces acting on the piston due to the inertia of the reciprocating parts, as in this case the motion of the piston is simple harmonic, *i.e.*,

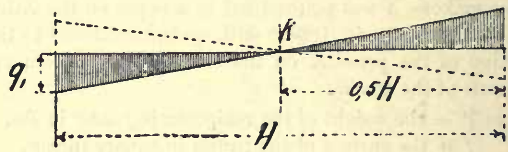


Fig. 1026.

the acceleration and therefore the accelerating force is proportional to the distance of the face of the piston from the centre of its path. If we plot out a curve where the ordinates are pressures and the abscissæ the corresponding position of the piston, we get a straight line which cuts the axis of abscissæ at the point k where $q = 0$.

Example. To determine the effect of the inertia of the moving parts of an engine where the diameter of the piston D = 16″ and the stroke H = 28″, $n = 100$.

$$\text{By our formula } \frac{P}{f} = 3{\cdot}98.$$

$$\therefore P = 3{\cdot}98 \times 16^2 \frac{\pi}{4}$$

$$= 798 \text{ lbs.}$$

$$v = \frac{2\pi \times 100 \times \frac{7}{6}}{60}$$

$$= 12{\cdot}2 \text{ feet per second.}$$

The pressure per square inch on the face of the piston necessary to move the reciprocating masses at the beginning of the stroke

$$= \frac{Pv^2}{grf} = \frac{798 \times (12{\cdot}2)^2}{32{\cdot}2 \times \frac{7}{6} \times 16^2 \times \frac{\pi}{4}} = 49{\cdot}4 \text{ lbs.}$$

The pressure diminishes towards the middle of the stroke, where it vanishes, the inertia of the reciprocating masses then assists the steam pressure, at the end of the stroke this pressure is 49·4 lbs. per square inch.

In order, therefore, that the pressure on the crank pin may be constant, we must have high pressure at the beginning of the stroke, and low pressure at the end of the stroke. The expansion of the steam does this, and compensates to a great extent for the forces introduced by the inertia of the reciprocating parts. If the compensation is not sufficient the momentum of the fly-wheel must help to drag the piston to its full speed at the beginning of the stroke, and oppose its motion at the end. Consequently, there is a change from extension to compression in the connecting-rod and a knock ensues which if neglected soon causes trouble. To remedy this, the steam must be cushioned at the end of the stroke by closing the exhaust early. The energy possessed by the moving reciprocating masses is thus employed in compressing the steam, and is restored at the beginning of the stroke where it is wanted.

(b.) With Connecting-Rod of Finite Length.

Let L = the length of the connecting-rod.

$\frac{r}{L}$ = ratio of crank radius to length of connecting-rod.

q = accelerating pressure in lbs. per square inch of the piston face necessary to overcome or retard the inertia of the moving parts.

ω = the angle swept out by the crank arm.

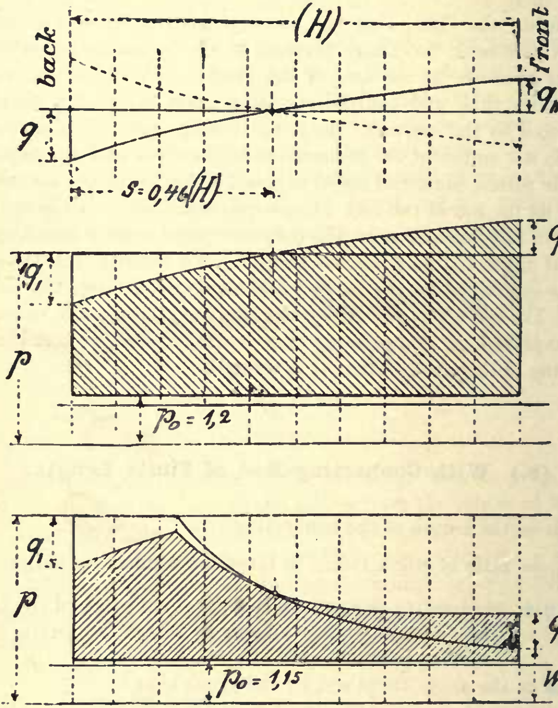
Then $q = \frac{Pv^2}{grf}(\cos.\ \omega + \frac{r}{L}\cos.\ 2\,\omega)$ approximately. At the dead point for the outstroke $q_1 = \frac{Pv^2}{grf}\left(1 + \frac{r}{L}\right)$ and opposes the steam pressure at the dead point at the end of the stroke $q_2 = \frac{Pv^2}{grf}\left(1 - \frac{r}{L}\right)$ and assists the steam pressure. The space S, travelled by the piston when the crank arm has turned through an angle ω is given by

$$S = r\left(1 - \cos.\ \omega + \frac{1}{2}\,\frac{r}{L}\sin.^2\omega\right) \text{ approximately.}$$

This distance S is best calculated by graphical construction.

The case of $\frac{r}{L} = \frac{1}{5}$ is shown in fig. 1027.

For all values of S the ordinates are plotted out.



Figs. 1027—1029.

Outstroke—

$$\omega = 0; \qquad q_1 = \frac{6}{5}\frac{Pv^2}{grf};$$

$$\omega = 180^\circ; \quad q_2 = -\frac{4}{5}\frac{Pv^2}{grf}.$$

Return stroke (steam pressure acting in opposite direction)—

$$\omega = 180^\circ; \qquad q_1 = \frac{4}{5}\frac{Pv^2}{grf};$$

$$\omega = 360^\circ; \quad q_2 = -\frac{6}{5}\frac{Pv^2}{grf};$$

when $\omega = 79°$; $s = 0{\cdot}46$ H; $q = 0$.

$\omega = 259°$; $s = 0{\cdot}54$ H; $q = 0$.

Example.—Take an engine where D = 16″, H = 28″, $n = 100$; $L = 5\,r$.

Then as in preceding example—

$$P = 798 \text{ lbs. } v = 12{\cdot}2 \text{ feet per sec.}$$

$$q_1 = \frac{6}{5} \times 49{\cdot}4 = 59{\cdot}3.$$

$$q_2 = \frac{4}{5} \times 49{\cdot}4 = 39{\cdot}5.$$

Suppose the steam pressure to be 140 lbs. per square inch at the beginning of the stroke and to be expanded five times; the pressure at the end of the stroke would be $\frac{140 + 15}{5} - 15 = 16$ lbs. per square inch approximately.

The pressure on the crank pin at the beginning of the stroke would be $= (140 - 59{\cdot}3)f = 80{\cdot}7f$, and at the end of the stroke it would be $= (16 + 39{\cdot}5)f = 55{\cdot}5f$. If the steam had been expanded 2·8 times, the pressure at the end of the stroke would equal the pressure at the beginning of the stroke. Had there been no expansion, the pressures would have been $80{\cdot}7f$ at the beginning and $179{\cdot}5f$ at the end of the stroke. The pressure on the crank pin at the end of the stroke would thus have been more than double the pressure at the beginning.

The values of q are plotted out in fig. 1027. In fig. 1028 the pressure on the crank pin is represented graphically when the steam pressure is maintained constant throughout the stroke. The shaded area above the line is exactly equal in area to the unshaded area below the line which could have been predicted since the energy communicated to the reciprocating masses at the beginning of the stroke is equal to the energy given up by them at the end of the stroke.

In fig. 1029, we see the influence of the reciprocating masses on the pressure on the crank pin for the same engine when the cut-off is at 0·4 of the stroke. From the diagram it is apparent that with quick-running engines working with late cut-off, the pressure on the crank pin due to the inertia of the reciprocating masses, and the pressure of the steam combined at the end of the stroke far exceeds the admission pressure. Cushioning the steam gets over this, and it seems at first sight to be most advantageous to have the compression at the dead point = the final steam pressure + the pressure due to the reciprocating masses; *i.e.*, $c = w + q_2$ (fig. 1030). Thus the admission pressure can be easily calculated.

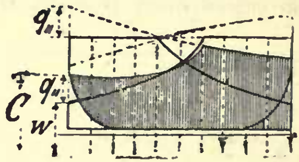


Fig. 1030.

In order to carry out a high degree of expansion, a high speed of piston is necessary, for the inertia of the reciprocating parts in this case compensates for the extreme variation in steam pressure consequent upon an early cut-off. If we consider smoothness of running only, the engine should be run so fast that the driving force produced by the highest pressure of the steam cannot exceed the inertia of the reciprocating parts, then a knock upon the centres becomes impossible.

The usual small compression given in slow-going engines causes nearly the whole of the initial pressure to act suddenly on the piston, and this makes the connecting-rod knock on the crank pin.

The author holds that it is of the very greatest importance to have smooth and noiseless running, and this is only possible with the help of compression. In engines with double valve gear and large clearance spaces, it is difficult to obtain sufficient compression, most makers avoiding the large valves and eccentrics necessarily required.

As a general rule, *always endeavour to have the compression in a non-condensing engine at least equal to half the admission pressure and in a condensing engine as much as possible.*

We can employ the standard proportions given on page 181. The valve gear, page 184, can only be used with divided valves (*i.e.*, in two short valves, one at each end of the cylinder for short ports) with small clearance space. The end pressure due to compression $C = \frac{(o + s)}{s} P_0$ can be determined by the use of the Tables 148 and 149, when the valve s of the clearance space, and the portion of the stroke during which compression takes place, are known. The pressures in the Tables are given in atmospheres. Non-condensing engines [$P_0 = 1{\cdot}15$], condensing engines [$P_0 = 0{\cdot}2$].

$$\text{Value of } q = \frac{Pv^2}{grf} \text{ (atmos.)}$$

TABLE 148.—**Pressure due to Compression in Absolute Atmospheres for Non-Condensing Engines, $P_0 = 1{\cdot}15$ At. Abs.**

Compression period.	Clearance space *s* in per cent.								
0	2	3	4	5	6	7	8	9	10
·00	1·15	1·15	1·15	1·15	1·15	1·15	1·15	1·15	1·15
·025	2·55	2·10	1·87	1·73	1·63	1·56	1·50	1·47	1·44
·050	4·00	3·06	2·59	2·30	2·11	1·97	1·87	1·78	1·72
·075	5·46	4·03	3·31	2·87	2·57	2·38	2·28	2·11	2·01
·100	6·90	4·90	4·03	3·45	3·07	2·79	2·59	2·42	2·30
·150	—	6·90	5·46	4·60	4·00	3·60	3·31	3·06	2·87
·200	—	—	6·90	5·75	4·98	4·43	4·03	3·70	3·45
·250	—	—	—	6·90	5·95	5·29	4·74	4·34	4·02
·300	—	—	—	—	6·90	6·07	5·46	4·98	4·60

TABLE 149.—**For Condensing Engines $P_0 = {\cdot}2$.**

·00	·20	·20	·20	·20	·20	·20	·20	·20	·20
·025	·45	·37	·33	·30	·28	·27	·26	·26	·25
·050	·70	·53	·45	·40	·37	·34	·33	·31	·30
·075	·95	·70	·57	·50	·45	·41	·38	·37	·35
·100	1·20	·86	·70	·60	·53	·48	·45	·42	·40
·150	1·70	1·20	·95	·80	·70	·63	·57	·53	·50
·200	2·20	1·53	1·20	1·00	·86	·77	·70	·64	·60
·250	2·70	1·86	1·45	1·20	1·03	·97	·82	·75	·70
·300	3·20	2·20	1·70	1·40	1·20	1·05	·95	·86	·80

The pressure of the compression C rises to a height which may be obtained by the formula

$$C = \frac{(o + s)}{s} P_0$$

and can be taken from the above tables for any given per-centage of clearance.

TABLE 150.—Values of $\frac{Pv^2}{rgf}$ in Atmospheres.

Stroke in inches. H	Diam. of piston in inches. D	Revolutions per minute n. 50	75	100	150	200	300	400	500
6	4	·04	·08	·15	·35	·60	1·40	2·50	4·00
12	8	·10	·21	·40	·90	1·60	3·50	—	—
20	12	·18	·42	·70	1·50	3·00	—	—	—
28	16	·27	·60	1·00	2·50	4·30	—	—	—
36	20	·34	·75	1·30	3·00	5·40	—	—	—
44	24	·40	·90	1·60	3·50	—	—	—	—
48	28	·45	1·05	1·80	4·00	—	—	—	—
56	32	·50	1·20	2·10	4·70	—	—	—	—
64	36	·60	1·35	2·40	5·40	—	—	—	—
72	40	·70	1·50	2·70	6·00	—	—	—	—

Unit to calculate with $\frac{Pv^2}{grf} \times \frac{1}{14{\cdot}7} = \frac{4\pi^2 n^2 H}{180 \times 32{\cdot}2 \times 14{\cdot}7} =$ $\cdot 0000463\, n^2$ H in atmospheres.

$\cdot 0000463 \times 14{\cdot}7\, n^2 H = \cdot 000681\, n^2$ H in lbs. per square inch.

Example.

$$\text{Let } H = 48'',$$
$$D = 28'',$$
$$n = 150, \quad \frac{r}{L} = \frac{1}{5}.$$

By the table $\dfrac{Pv^2}{grf} = 4{\cdot}15$ atmos.

Thus
$$q_1 = 4{\cdot}15 \times \frac{6}{5} = 4{\cdot}98,$$
$$q_2 = 4{\cdot}15 \times \frac{4}{5} = 3{\cdot}32.$$

Initial pressure $p = 7$ atmos.
$h =$ cut-off at $0{\cdot}2$.
$s = 6\%$ clearance.

This gives the pressure diagram, fig. 1031. The part shaded

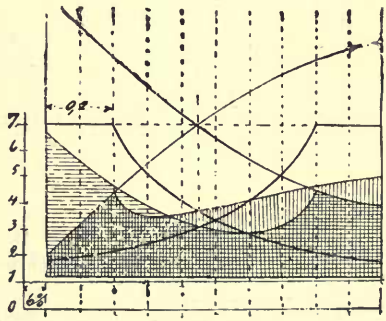


Fig. 1031.

vertically for the outstroke, and that shaded horizontally for the back-stroke.

SECTION XII.

FRICTION BRAKE, DYNAMOMETER.

In making trials of small engines, a friction brake is used to absorb the power during the trial. The original friction brake, fig. 1032, is due to Prony. A pair of wooden clamps, to one of which a lever l is fixed, are bolted together, fig. 1032, and nip the shaft to which they are applied with more or less pressure according to the tightness of the bolts; the shaft being run in the direction of the arrow, tends by friction between the clamps and shaft to raise the weight G; if the shaft runs uniformly and the belts are adjusted so as just to keep the weight balanced, the effect is very nearly the same as if the work done was that of raising the weight continuously at the speed G would move if allowed to run with the shaft.

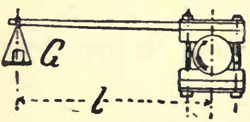


Fig. 1032.

Let G = the weight required in lbs.

l = the length of the lever in feet.

n = the revolutions per minute, so the "brake horse power" given out by the machine will be

$$\text{B HP} = \text{G} \times \frac{2\pi \times n \times l}{33000}$$

and G for any required load will $= \dfrac{33000\ (\text{B HP})}{2\pi \times n \times l}$.

This form of brake is still used in special cases, but the more modern form of rope brake, fig. 1033, is now much used. The rope R is spliced or seized together at one end, and passed through the "bight" as in the figure; to the upper a spring balance is hooked, the balance itself being hung from a firm support; to the other end are attached the weights necessary to give the required load. The rope is kept in its place by blocks of wood b,b; a small block of wood c, is inserted to keep the rope from nipping the other part which is on the wheel. If required for long runs, the wheel is made with deep flanges P, and water is run in to keep it cool. The calculation is the same as for the Prony brake, taking the length of l from the centre of the shaft S to the centre of the rope; but in measuring the

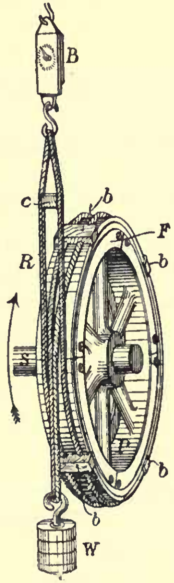


Fig 1033.

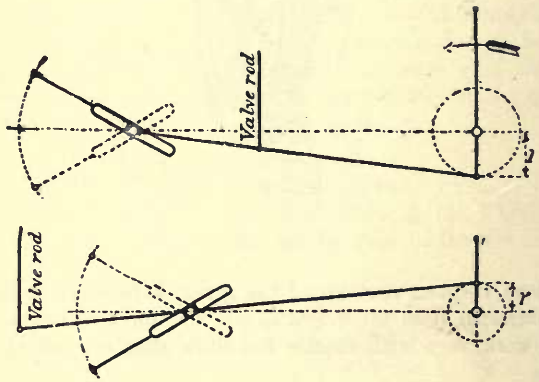
work done, frequent readings of the spring balance must be taken and subtracted from the weight to give the time load lifted. These brakes work very well, require but little attention and are fairly reliable.

SECTION XIII.

SUNDRY DETAILS.

Diagrams of Special Reversing Gears.

In addition to those already shown on page 224, the diagrams below show the principle of some special reversing gears that are sometimes used for marine and other engines.*



Figs. 1034, 1035.—Reversing gear by Hackworth.

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* "Zeitschrift der deutsch Ingenieur," 1885, page 949.

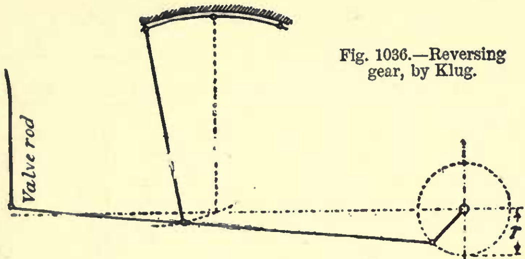


Fig. 1036.—Reversing gear, by Klug.

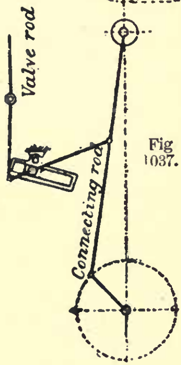
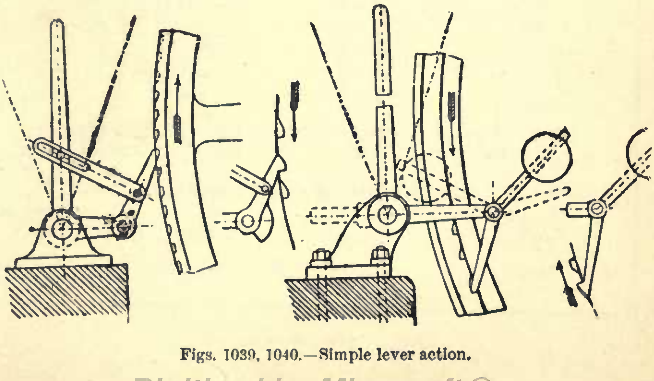


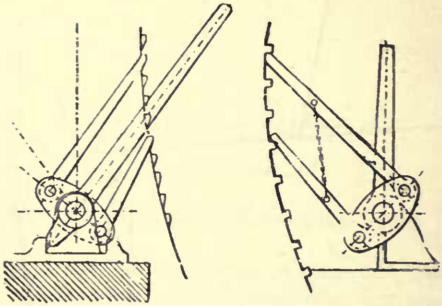
Fig 1037.

Fig. 1037.—Reversing gear, by Joy.

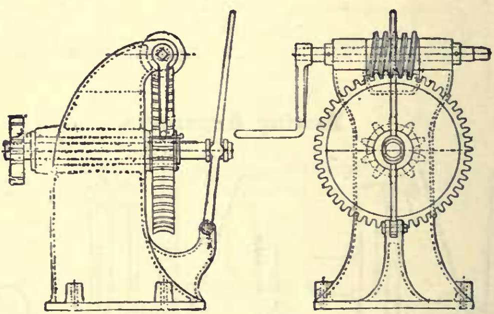
Barring Apparatus.



Figs. 1039, 1040.—Simple lever action.



Figs. 1041, 1042.—Double lever action.



Figs. 1043, 1044.—Worm and worm-wheel.

Table 151.—Particulars of Bolts.

Diameter of bolts in inches.	No. of threads per inch.	Diameter at bottom of thread. Inches.	Area at bottom of thread in sq. inches.
3/8	16	·295	·0683
7/16	14	·346	·0942
1/2	12	·393	·1213
5/8	11	·508	·2027
3/4	10	·622	·3039
7/8	9	·733	·4230
1	8	·840	·5542
1 1/8	7	·942	·6969
1 1/4	7	1·070	·8992
1 3/8	6	1·161	1·0569
1 1/2	6	1·287	1·2999
1 5/8	5	1·369	1·4210
1 3/4	5	1·494	1·7530
2	4 1/2	1·715	2·3087
2 1/4	4	1·930	2·9241
2 3/8	4	2·05	3·316
2 1/2	4	2·180	3·7311
2 3/4	3 1/2	2·384	4·464
3	3 1/2	2·634	5·450
3 1/4	3 1/4	2·855	6·402
3 1/2	3 1/4	3·105	7·563
3 3/4	3	3·323	8·673
4	3	3·573	10·027
4 1/4	2 7/8	3·804	11·368
4 1/2	2 7/8	4·054	12·908
4 3/4	2 3/4	4·284	14·404
5	2 3/4	4·534	16·146
5 1/4	2 5/8	4·762	17·810
5 1/2	2 5/8	5·012	19·72
5 3/4	2 1/2	5·239	21·57
6	2 1/2	5·489	23·64

All Dimensions are given in Inches.

SECTION XIV.

BOILERS.

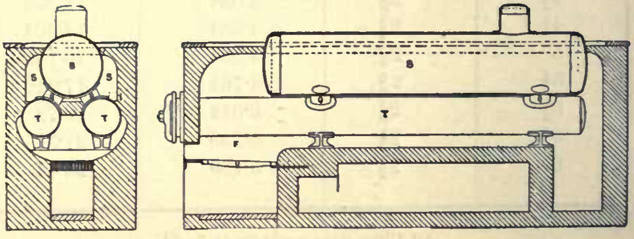
In the following pages (416 to 441), particulars of the most usual types of boilers will be found, together with figures showing their general construction.

Elephant or French Boiler.

Figs. 1045, 1046, show this type of boiler, which has been largely used in France and occasionally in England; the table gives the general proportions.

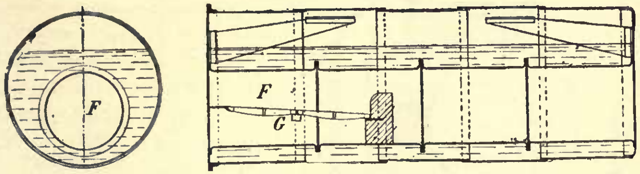
TABLE 152.—**Dimensions of Elephant Boilers.**

Nominal H.P.	10	12	15	20	25	30
Length of boiler in feet . . .	14	16	18	$23\frac{1}{2}$	28	30
Diameter of heaters, or "bouilleurs," in inches	18	19	$20\frac{1}{2}$	$21\frac{1}{2}$	23	24
Diameter of shell in inches . .	30	32	34	36	38	40
Approximate heating surface in sq. ft.	169	203	254	338	422	507



Figs. 1045, 1046.—Elephant Boiler.

Cornish Boilers.



Figs. 1047, 1048.

This type, together with the double flued or Lancashire boiler, fig. 1053, is perhaps more largely used on land than any other type for supplying steam for steam-engines, or general manufacturing purposes.

The Tables 153 to 166, with the exception of table 161, give the dimensions of these boilers by different English makers.

TABLE 153.—Dimensions of Cornish Boilers made by Messrs. Marshall, Sons & Co. Limited.

Nominal H.P.	4	6	8	10	12	14	16	20	25
Dia. of boiler .	3′ 10″	4′ 5″	4′ 5″	4′ 9″	5′ 3″	5′ 4″	5′ 9″	6′ 2″	6′ 2″
Length „ .	9′ 0″	10′ 1″	13′ 2″	14′ 6″	14′ 9″	17′ 6″	19′ 4″	21′ 0″	26′ 3″
Dia. of flue .	24	28	29	31	34	34	36	39	39
No. of Galloway tubes	—	—	—	—	2	3	3	4	4
Weight in cwts.	44	67	77	104	110	139	153	180	215

TABLE 154.—Cornish Boilers made by Messrs. Davey, Paxman & Co.

Nom. H.P.	4	6	8	10	12	14	16	20	25	30
Dia., boiler	3′ 6″	3′ 8″	4′ 6″	4′ 6″	4′ 9″	4′ 9″	5′ 0″	5′ 6″	5′ 6″	5′ 9″
Length „	9′ 0″	12′ 0″	12′ 0″	15′ 0″	15′ 0″	18′ 0″	20′ 0″	21′ 0″	24′ 0″	26′ 0″
Dia. of flue, ins.	24	24	30	30	32	32	33	36	38	38

Seating for Cornish Boilers (Figs. 1049—1051).

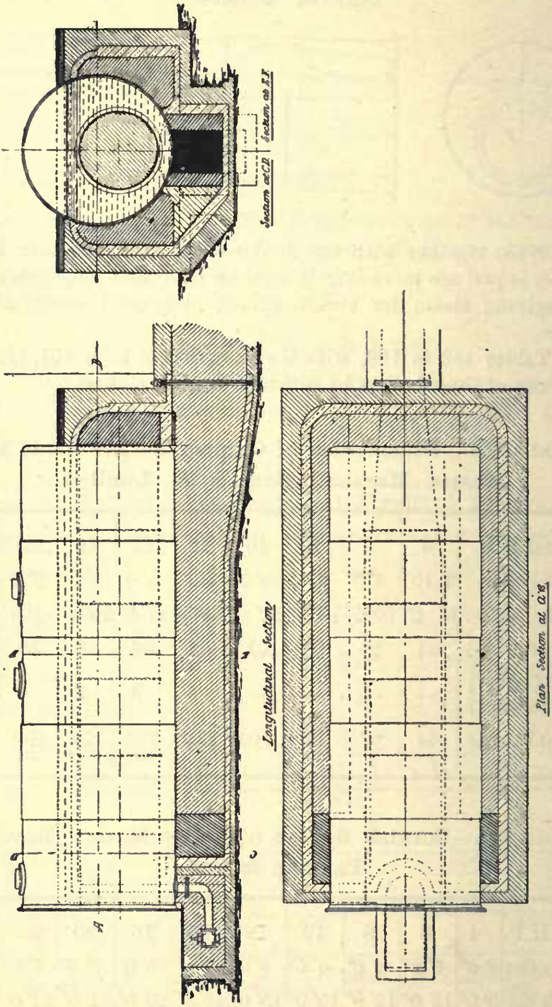


Fig. 1051.

Fig. 1049.

Fig. 1050.

Figs. 1049—1051.—Seating for Cornish Boilers.

The ordinary arrangement of flues for Cornish boilers is shown in the above figures, 1049, 1051; in all cases where possible, the flues should be made large enough to admit of thorough inspection being made.

TABLE 155.—**Cornish Boilers made by Messrs. Ruston, Proctor, Limited.**

Nominal H.P. .	3	4	6	8	10	12	14	16	20
Diam. of boiler.	3′ 0″	3′ 6″	4′ 0″	4′ 6″	4′ 6″	5′ 0″	5′ 0″	5′ 6″	6′ 0″
Length of boiler	8′ 0″	9′ 0″	10′ 0″	12′ 0″	15′ 0″	16′ 0″	18′ 0″	19′ 0″	20′ 0″
Diam. of flue .	20	22	24	28	28	32	32	38	40
No. Galloway cross tubes .	1	2	2	2	3	3	3	4	4

TABLE 156.—**Cornish Boilers made by Messrs. Ransomes, Sims & Jefferies, Limited.**

Nom. H.P.	4	6	8	10	12	14	16	20	25	30
Dia., boiler	3′ 6″	4′ 0″	4′ 6″	4′ 6″	5′ 0″	5′ 0″	5′ 6″	5′ 6″	6′ 0″	6′ 0″
Length „	9′ 0″	10′ 0″	12′ 0″	15′ 0″	16′ 0″	19′ 0″	20′ 0″	23′ 0″	24′ 0″	27′ 0″
Dia. of flue	24	24	28	28	32	32	36	36	40	40
No. Gallo. cross-tubes	1	1	2	2	2	3	3	4	4	6

Double-Flued or Lancashire Boiler (Figs. 1052, 1053).

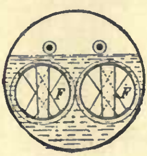


Fig. 1052.

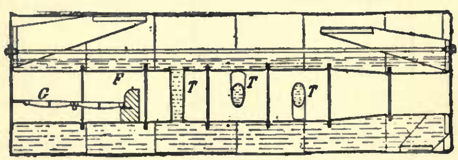


Fig. 1053.

This type is generally used for large boilers on land, and is very efficient and durable. Below are tables of dimensions of these boilers by the same English makers as those given of Cornish boilers, and in the same order.

TABLE 157.—**Dimensions of Lancashire Boilers.**

Nominal H.P. . . .	20	25	30	35	40
Diameter of boiler . .	6′ 2″	6′ 6″	6′ 8″	6′ 8″	7′ 0″
Length	19′ 0″	22′ 0″	24′ 0″	27′ 0″	28′ 0″
Diameter of flues . .	28″	30″	30″	30″	32″
Weight in cwts. . . .	200	235	250	276	300

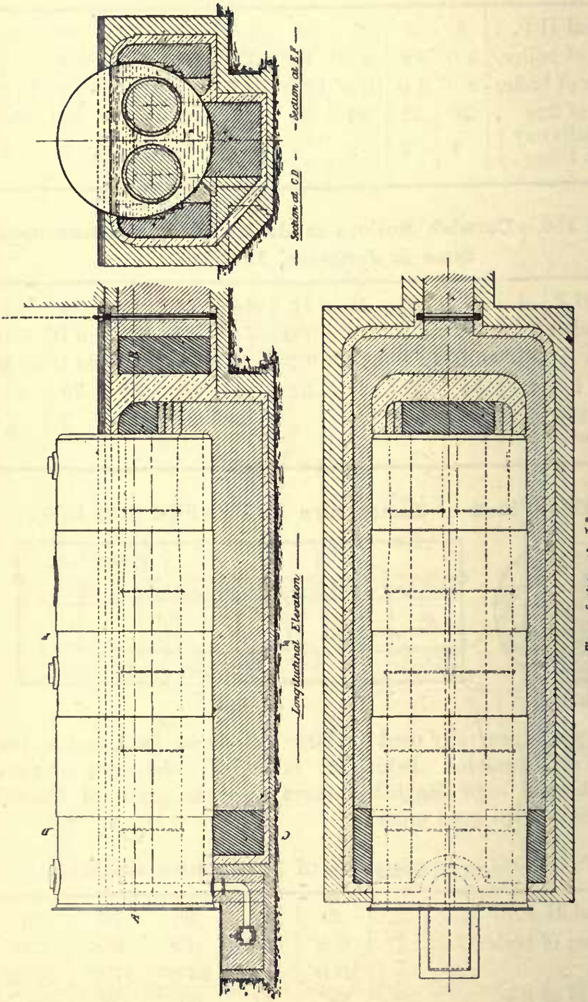


Fig. 1054.

Fig. 1055.

Figs. 1054—1056.—Seating for Lancashire Boilers.

TABLE 158.

Nominal H.P. .	20	25	30	35	40	45	50	55
Diameter of boiler .	5′ 6″	5′ 6″	6′ 0″	6′ 3″	6′ 6″	6′ 9″	7′ 0″	7′ 6″
Length of boiler .	19′ 0″	21′ 0″	22′ 6″	24′ 0″	26′ 0″	28′ 0″	30′ 0″	30′ 0″
Diameter of flues .	24	24	27	28	30	$31\frac{1}{2}$	33	36

TABLE 159.

Nominal H.P. .	18	22	25	28	30	35	40	50
Diameter of boiler .	5′ 6″	6′ 0″	6′ 0″	6′ 6″	6′ 6″	6′ 6″	7′ 0″	7′ 0″
Length of boiler .	19′ 0″	20′ 0″	24′ 0″	24′ 0″	26′ 0″	28′ 0″	28′ 0″	30′ 0″
Diameter of flues .	22	26	26	29	29	29	31	31
No. of Galloway cross-tubes .	6	6	6	6	6	8	8	10

TABLE 160.

Nominal H.P. .	16	20	25	30	35	40	45	50
Diameter of boiler .	6′ 0″	6′ 0″	6′ 6″	6′ 6″	7′ 0″	7′ 0″	7′ 6″	7′ 6″
Length of boiler .	19′ 0″	22′ 0″	23′ 0″	26′ 0″	28′ 0″	30′ 0″	28′ 0″	30′ 0″
Diameter of flues .	28	28	28	28	32	32	36	36
No. of Galloway cross-tubes .	4	4	6	6	8	8	8	8

Seating for Lancashire Boilers (Figs. 1054—1056).

The flues in the above figs. (1054—1056), are shown arranged for the hot gases to pass through the flues, then down underneath the boiler, then round the sides, and finally to the chimney. With Cornish boilers the flues are usually arranged to take the hot gases round the sides first, and then along underneath the boiler. Each method has its advocates, but it is generally held that in Lancashire boilers the hottest possible gases should pass underneath, in order to act on the body of water between and below the flues; in Cornish boilers there is less water under the flue.

Cornish Boilers of German Type, with Flue on one side.

Fig. 1057. Fig. 1059.

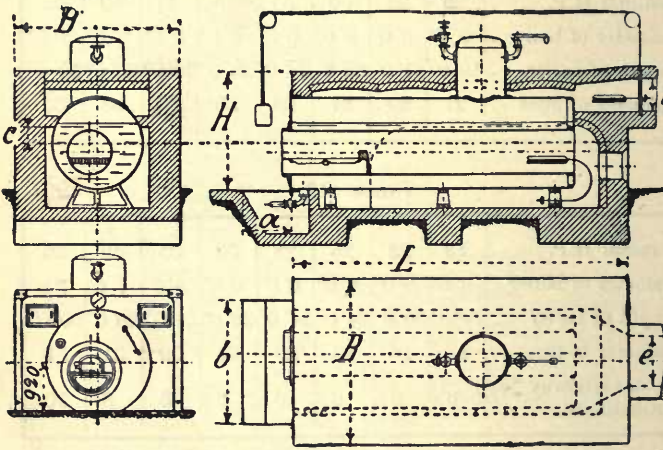


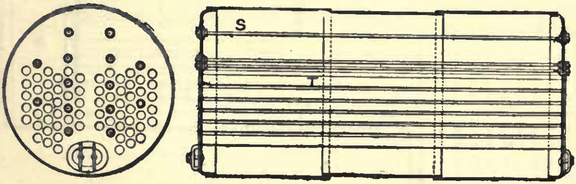
Fig. 1058. Fig. 1060.

TABLE 161.—**Dimensions of German Cornish Boiler Seating** (Figs. 1057—1060).

Horse-power.	Heating-surface sq. feet.	L	B	H	*a*	*b*	*c*	*e*	*f*
2	54	95	$77\frac{1}{2}$	70	18	32	6	20	20
4	97	136	82	75	20	36	7	20	20
6	140	173	85	77	22	39	$7\frac{3}{4}$	20	20
8	172	198	87	78	24	43	8	24	24
10	215	220	92	81	26	47	9	24	24
15	300	228	98	90	28	51	10	24	24
25	430	285	110	94	30	55	11	28	28
40	690	340	118	94	32	59	12	28	28
50	860	427	138	96	34	63	$13\frac{1}{2}$	28	28

All dimensions in Inches except heating-surface, which is given in square feet.

Cylindrical Multitubular Boilers.



Figs. 1061, 1062.

Table 162.—Dimensions of Multitubular Cylindrical Boiler (Figs. 1061, 1062).

Nominal H.P. .	6	8	10	12	14	16	20	25	30	35	40	50
Dia. of boiler .	$3\frac{1}{2}$	$3\frac{1}{2}$	4	4	$4\frac{1}{2}$	$4\frac{1}{2}$	5	5	6	6	$6\frac{1}{2}$	$6\frac{1}{2}$
Length ,,	6	$7\frac{1}{2}$	8	9	$9\frac{1}{2}$	$10\frac{1}{2}$	$11\frac{1}{2}$	$12\frac{1}{2}$	$12\frac{1}{2}$	$13\frac{1}{2}$	$13\frac{1}{2}$	1
No. of tubes .	22	22	28	28	32	32	38	44	40	52	60	60
Dia. of tubes .	—	—	—	—	—	—	—	—	—	—	—	—
Heating-surface	—	—	—	—	—	—	—	—	—	—	—	—

This type of boiler is set in brickwork, the furnace and flue doors being also built in.

The boiler is fired from below, and the gases pass along underneath the boiler and through the tubes, returning back by the sides to the chimney. The tubes are easily got at for cleaning, and almost any description of fuel may be used. These boilers are suitable for export, owing to their small size in proportion to their heating surface.

Multitubular Loco-Type Boilers.

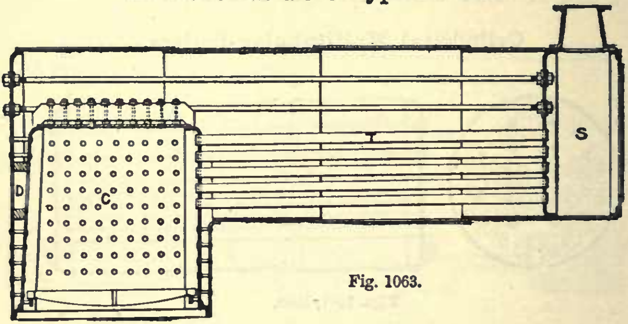


Fig. 1063.

Table 163.—Multitubular Loco-Type Boilers for 140 lbs. working pressure.

Nominal Horse-power.	8	10	12	16	20	25	30	40
Diameter of boiler .	$31\frac{1}{2}$	$33\frac{1}{2}$	$35\frac{1}{2}$	39	$41\frac{1}{2}$	$47\frac{1}{2}$	50	52
Length of boiler . .	135	146	148	162	172	$187\frac{1}{2}$	$189\frac{1}{2}$	$235\frac{1}{2}$
Number of tubes .	31	38	34	36	43	58	64	73
Length of tubes . .	$77\frac{1}{4}$	81	81	96	102	108	108	123
Diameter of tubes .	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$
Heating surface . .	144·2	158·3	189·7	230·2	285·5	398·0	439·5	618·4
Length of fire-box .	31	35	35	37	38	$43\frac{1}{2}$	$45\frac{3}{4}$	$56\frac{3}{4}$
Grate area, sq. ft. .	5	6·5	7·3	7·8	9·0	12·1	13·6	17·7
Thickness of plates, tube . . .	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
Thickness of fire-box, top and sides	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
Thickness of front plate, fire-box .	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
Thickness of front tubeplate, fire-box	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
Thickness of barrel .	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$
Diameter of chimney	12	15	15	15	18	18	20	26
Length of smoke-box	$24\frac{1}{2}$	25	25	25	$26\frac{1}{2}$	$31\frac{1}{2}$	$31\frac{1}{2}$	$40\frac{1}{2}$

Dimensions in Inches, except heating surface which is in sq. feet.

This is a well-tried type of boiler, and is universally used for locomotives, and portable and traction engines.

Multitubular Loco-Type Boilers.

These boilers are similar to those shown, fig. 1063, Table 163, but for smaller powers; and are for 80 to 90 lbs. working pressure.

TABLE 164.—**Dimensions of Loco-Type Boilers.**

Horse-power.	3	4	5	6	7	8	9	10	12
Dia. of boiler .	$28\frac{1}{2}$	$31\frac{1}{2}$	$33\frac{1}{2}$	$35\frac{1}{2}$	$37\frac{1}{2}$	39	39	$41\frac{1}{2}$	$43\frac{1}{2}$
Length " . .	$102\frac{1}{2}$	115	$115\frac{1}{2}$	117	125	129	133	139	$146\frac{1}{2}$
No. of tubes .	15	19	21	25	27	31	33	36	39
Length " . .	57	66	67	67	$72\frac{1}{2}$	76	78	$78\frac{3}{4}$	$87\frac{3}{4}$
Dia. " .	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$
Total heating-surface of boiler	66·9	94·5	105·5	122·0	140·7	165·5	179·5	197·8	234·9
Length of fire-box	$22\frac{1}{2}$	$24\frac{1}{2}$	25	$25\frac{1}{4}$	$26\frac{1}{4}$	$27\frac{1}{4}$	$29\frac{1}{2}$	$32\frac{1}{4}$	$32\frac{1}{4}$
Grate area, sq. ft.	3·37	4·59	4·67	5·07	5·51	6·16	6·59	7·83	8·28
Thickness of front tube	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
" fire-box side & top plates	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
Thickness of fire box front plate	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
Thickness of fire box tube plate	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
Thickness of barrel plates	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$
Dia. of chimney .	8	9	10	12	12	12	12	15	15
Length of smoke-box .	17	19	19	$19\frac{3}{4}$	$20\frac{3}{4}$	$20\frac{3}{4}$	20	20	$21\frac{1}{2}$

Dimensions in Inches, except heating surface which is given in sq. feet.

Vertical Boilers.

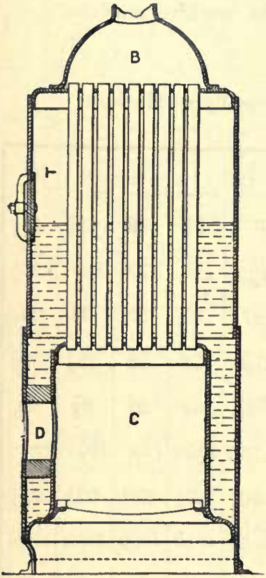


Fig. 1064.

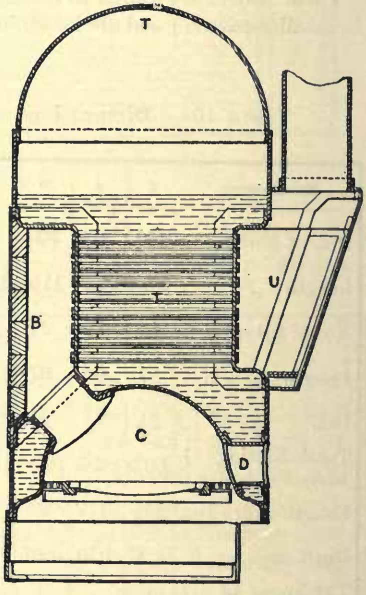


Fig. 1065.

Owing to the comparatively small space which vertical boilers occupy, and their convenient shape, they are largely used, notwithstanding the fact that their efficiency does not approach to either Cornish, Lancashire, or multitubular boilers. The form shown on fig. 1067, is in most general use, owing to its simplicity and cheapness ; it is at the same time perhaps the most wasteful of fuel, to do away with which defect many other forms of vertical boilers have been constructed, to give increased heating surface and greater economy. Fig. 1064 shows one of these arrangements, in which the uptake is dispensed with, the gases, &c., from the fire-box passing through vertical tubes to the chimney. Fig. 1065 shows a vertical boiler constructed by Messrs. Cochrane, of Birkenhead. In this case the gases are conducted up one side of the boiler, and then pass through a number of horizontal tubes to the uptake; the tubes being

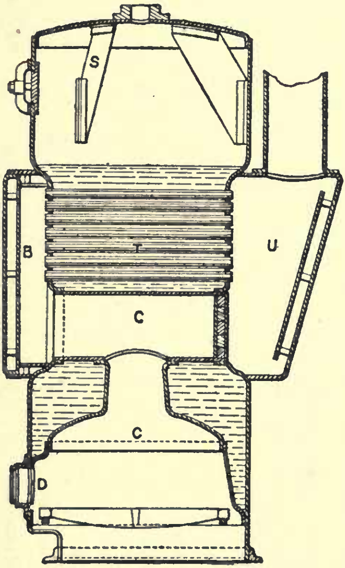


Fig. 1066.

arranged to give easy access for cleaning. Fig. 1066 shows a good form of vertical boiler by Messrs. Tinker, Shenton & Co.

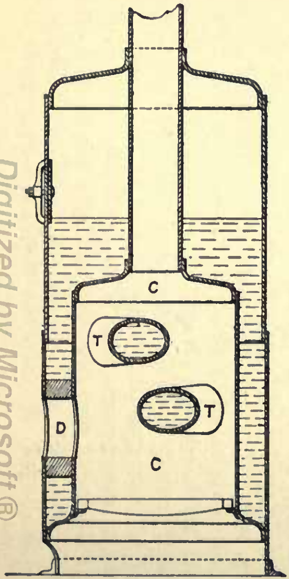


Fig. 1067.—Vertical Cross-Tube Boiler.

TABLE 165.—**Dimensions of Vertical Cross-Tube Boilers** (Fig. 1067).

Nominal horse-power	3	4	5	6	7	8	10	12	14	16
Diameter of boiler	30	33	36	39	39	42	48	48	51	57
Height of boiler	72	78	84	90	102	102	114	144	144	150
Approx. heating surface in sq. ft.	$28\frac{1}{2}$	$38\frac{1}{2}$	50	60	69	80	100	120	130	150
Grate area, sq. ft.	$3\frac{1}{2}$	$4\frac{1}{4}$	$5\frac{1}{4}$	$6\frac{1}{4}$	$6\frac{1}{4}$	$7\frac{1}{2}$	$10\frac{1}{4}$	$10\frac{1}{4}$	$11\frac{1}{2}$	15
Number of cross-tubes	2	2	2	3	3	3	3	4	4	4
Shell plates, thickness	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
Fire-box, thickness	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$
Cross-tube, thickness	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$
Uptake, thickness	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
Crown, thickness	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
Diameter of cross-tubes	7	8	8	8	8	9	9	9	9	10
Height to top of fire-box	41	44	48	51	57	57	64	81	81	84
Diameter of fire-box	26	—	—	—	—	—	—	—	—	—

Dimensions in Inches, except heating surface.

These vertical boilers are extensively used on board ship as donkey boilers, and also for working steam cranes.

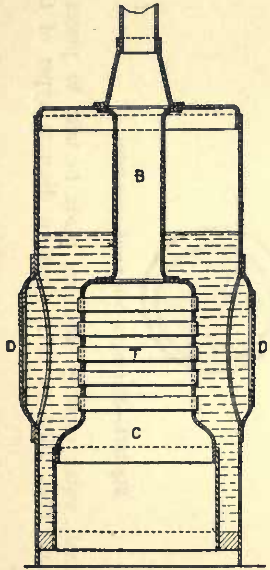


Fig. 1068.

TABLE 166.—**Dimensions of Hopwood's Vertical Boilers** (Fig. 1068).

Nominal horse-power	2	3	4	5	6	8	10	12
Diameter of boiler	$27\frac{1}{2}$	33	36	$38\frac{1}{2}$	41	43	$48\frac{3}{4}$	52
Height of boiler	66	77	81	90	91	96	111	117
Height to uptake (B)	$41\frac{3}{4}$	$47\frac{1}{2}$	52	$54\frac{3}{4}$	60	64	76	79
No. of tubes	12	20	28	24	28	26	36	36
Diameter of tubes	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$
Length of tubes	16	$20\frac{3}{4}$	$20\frac{3}{4}$	$26\frac{1}{2}$	27	$28\frac{1}{4}$	$34\frac{1}{2}$	$34\frac{1}{2}$
Heating-surface (total) sq. feet	25	47·7	55·1	64·6	75·8	90·1	121·2	141·8
Grate area	2·94	4·31	5·24	5·62	6 89	7·56	9·05	11·7

These boilers have manholes at D for access to tubes. The tubes are higher at one end than at the other. Vertical stays are used in the larger sizes, placed round the uptake, and connecting top of fire-box to crown of boiler.

Small vertical boilers are used for steam fire-engines where steam has to be raised in the very shortest possible time. They have a

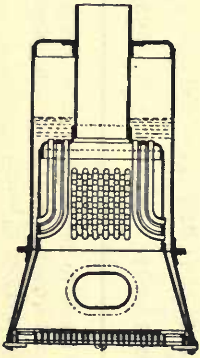


Fig. 1069.—Fire-engine boiler with cross and vertical tubes.

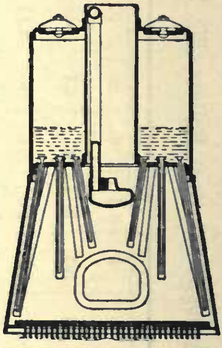


Fig. 1070.—Fire-engine boiler with Field tubes.

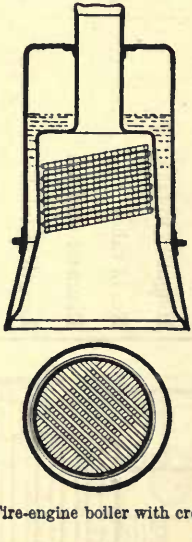


Fig. 1071.—Fire-engine boiler with cross-tubes.

very small water capacity, and are made to take to pieces for cleaning. Figs. 1069 to 1071, show in diagram types of these boilers.

Marine Boilers.

The type of marine boiler has gradually settled down, after many modifications, into an internally fired flue and tube boiler, and has

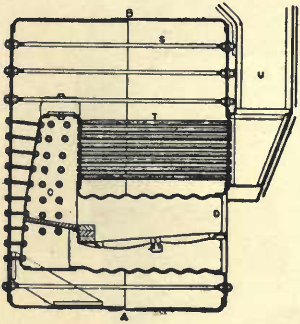


Fig. 1072.

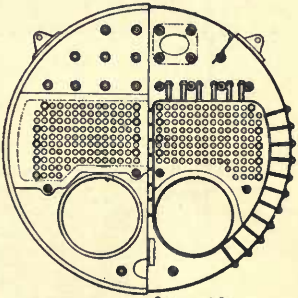


Fig. 1073.

now been successfully used on board ship for pressures up to 170 lbs.; 160 being, however, the usual limit. Figs. 1072, 1073, show in diagram the general construction of the modern marine boiler.

In torpedo boats the locomotive type of boiler, as well as the above, was largely used, but now they are entirely replaced by water-tube boilers.

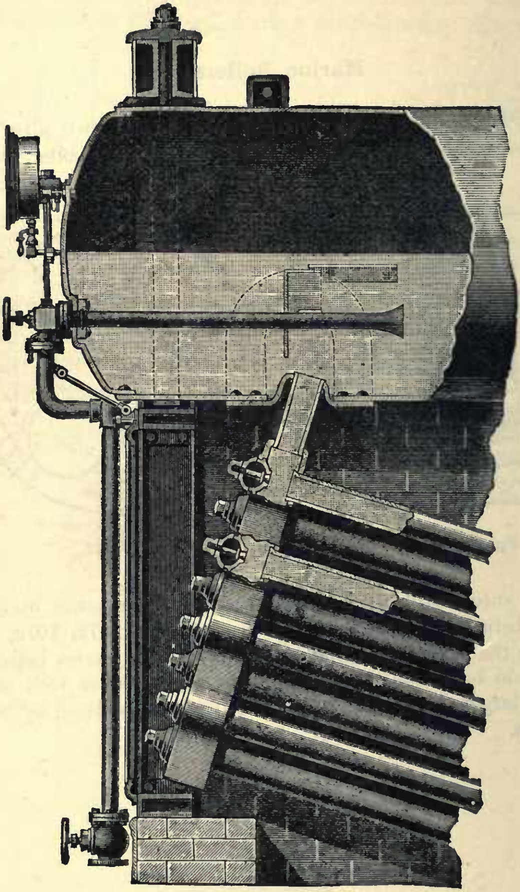


Fig. 1074.

Fig. 1075.

Water-Tube Boilers.

A very great advance has been made in the construction of this most important type of boiler in the last few years, and now numbers of them are in use both on land and sea. For land use, a feature important in towns liable to sudden fogs is that steam can be very

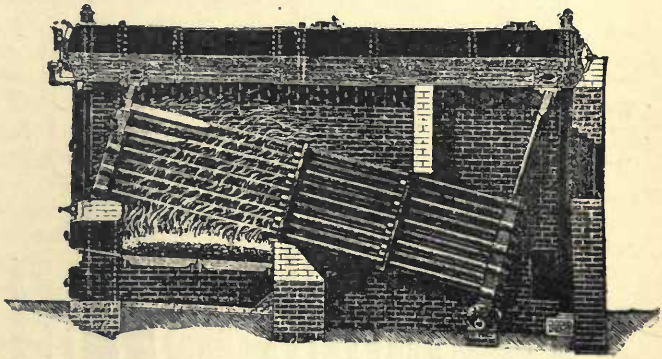


Fig. 1076.

quickly raised. This is of special advantage in electric light stations, where a sudden fog or darkness causes an immediate demand for more boiler power and quick steam raising is imperative. Another advantage is that the boilers can be taken apart and carried into works where it would be impossible to pass a large boiler in whole. At sea their lightness is perhaps one of the most important points in their favour. Fig. 1076 shews the well-known type of Messrs. Babcock and Willcox. The tubes are expanded into connecting boxes in zigzag, figs. 1074, 1075.

Fig. 1076 shews the arrangement of flues, and fig. 1074 a partial longitudinal section taken at the front end of the boiler. A mud drum is provided, to form a receptacle for the deposit as it falls down into the more quiescent parts of the boiler. Messrs. Babcock's boiler is largely used in electric light stations, with working pressures of 120 to 180 lbs. per square inch. The steam separates from the water in the large receiver forming the upper part of the boiler.

The Belleville water-tube boiler was invented by M. Belleville as long ago as 1849, but has only of late years been brought to perfection. In the French navy it has been adopted for many years, and is now received with favour in this country, and is being supplied to some of the newest cruisers, also to several mail steamers.

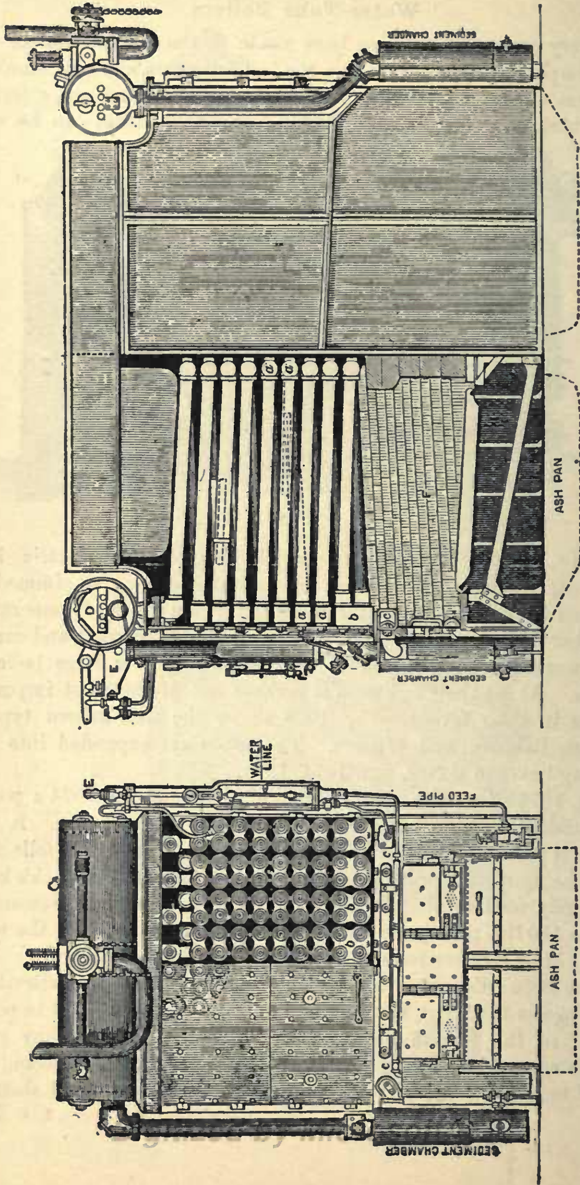


Fig. 1077.—Belleville Boiler. Marine Type.

The general construction of these boilers is that of a number of flattened spirals, formed by screwing the ends of straight tubes into junction caps of malleable cast-iron or steel. These caps are placed vertically above one another, the upper end of one tube being on the same level as the lower end of the one above it.

The tubes are slightly inclined to the horizontal, and the lower cap of each flattened spiral or element is connected to a horizontal cross tube at the front of the boiler, called the feed collector.

The upper tube is connected to the lower part of a cylindrical steam receiver placed outside the boiler casing. A vertical circulating pipe, also placed outside the casing, conveys the down current to a mud drum, placed at the base of the boiler, the upper part of which is connected to the feed collector.

The feed water is delivered into the steam receiver at the end remote from the inlet of the down pipe, passes along the receiver bottom down the external pipe, through the mud drum into the feed collector, and thus into the several elements, to be treated by the action of the fire on its way upwards through the tubes, from which it emerges into the receiver a mixture of water and steam. The latter is separated by baffle and dash plates from the former, which passes along its course with fresh feed-water. The admission of the feed-water is regulated by a self-acting arrangement. The position of the water level is at the fourth row of tubes from the top.

Fig. 1077 gives two views of the marine type of Belleville boiler. In both figures a part of the casing is removed to shew the arrangement of the tubes.

It is an essential feature in M. Belleville's system that the boiler pressure should be considerably higher than the working pressure of the engines. A reducing valve is therefore introduced between the boiler and engine. The effect of thus wire-drawing the steam slightly superheats it and delivers it dry to the engine. The tubes in these boilers are from 3 to 5 inches diameter. At the front end doors are provided which can be readily removed and replaced when steam is down for cleaning out the tubes. As the tubes are comparatively short for their diameter the cleaning out can be done with ease.

Steam can be got up quickly, in from 10 minutes in the smallest to one hour in large steamers.

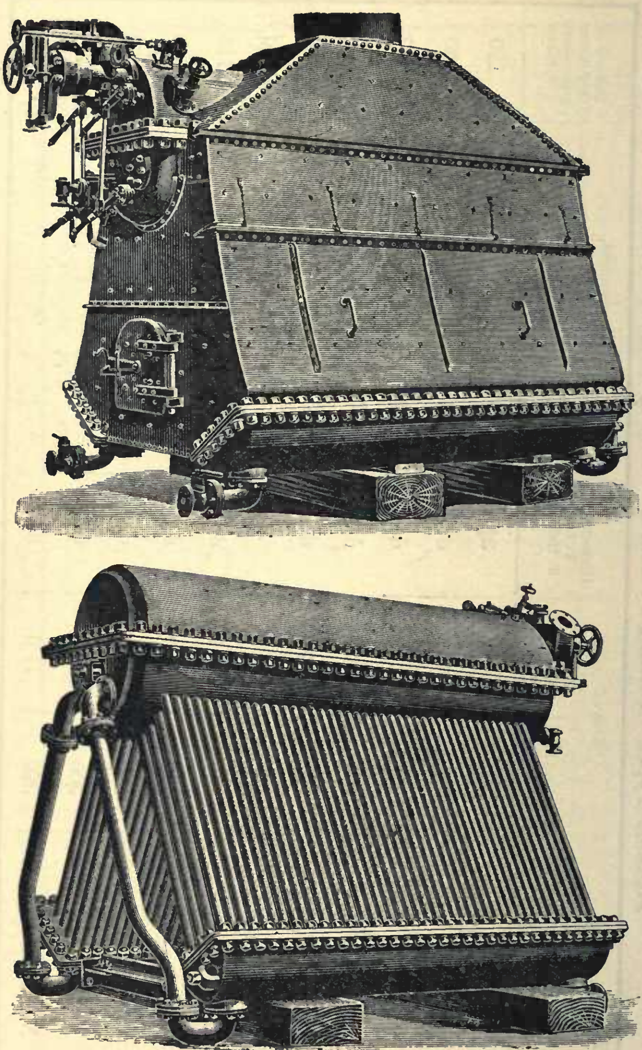
Compared with cylindrical marine boilers the head room required is small, 12 feet 6 inches for the largest size, and they are considerably lighter in weight than most other boilers. Taking an example, a range of twelve cylindrical boilers, with a grate area of 494 square

TABLE 167.—**Particulars of Belleville Boilers made by Messrs. Maudslay, Sons & Field, Ltd. Stationary Type. Working Pressure, 170 lbs. per square inch.**

MARK.	Area of.		Diameter of Tubes in Inches.	Space Required.			Pounds of Dry Steam guaranteed per Hour.
	Heating Surface, in sq. ft	Fire Grate, in sq. ft.		Depth.	Width.	Height.	
A. No. 3	290	9	3¼	6′ 9¼″	4′ 11″	10 6′	1,000
A. No. 4	380	12	do.	do.	5′ 7″	do.	1,350
A. No. 5	470	15	do.	do.	6′ 4″	do.	1,700
A. No. 6	560	18	do.	do.	7′ 0″	do.	2,050
A. No. 7	650	21	do.	do.	7′ 9″	do.	2,400
A. No. 8	740	24	do.	do.	8′ 5″	do.	2,750
A. No. 9	830	27	do.	do.	9′ 2″	do.	3,100
A. No. 10	920	30	do.	do.	9′ 10″	do.	3,450
A. No. 11	1,010	33	do.	do.	10′ 7″	do.	3,800
A. No. 12	1,100	36	do.	do.	11′ 4″	do.	4,150
B. No. 4	520	17·00	4	7′ 11¼″	6′ 10″	12′ 3″	2,000
B. No. 5	640	21·25	do.	do.	7′ 9″	do.	2,500
B. No. 6	760	25·50	do.	do.	8′ 7″	do.	3,000
B. No. 7	880	29·75	do.	do.	9′ 5″	do.	3,500
B. No. 8	1,000	34·00	do.	do.	10′ 3″	do.	4,000
B. No. 9	1,120	38·25	do.	do.	11′ 2″	do.	4,500
B. No. 10	1,240	42·50	do.	do.	12′ 2″	do.	5,000
C. No. 4	790	26	5	9′ 44¼″	9′ 3″	13′ 9″	3,200
C. No. 5	980	32	do.	do.	9′ 6″	do.	3,950
C. No. 6	1,170	39	do.	do.	10′ 6″	do.	4,700
C. No. 7	1,360	46	do.	do.	11′ 7″	do.	5,450
C. No. 8	1,550	53	do.	do.	12′ 7″	do.	6,200
C. No. 9	1,740	60	do.	do.	13′ 8″	do.	6,950
C. No. 10	1,930	67	do.	do.	14′ 9″	do.	7,700

TABLE 168.—Particulars of Belleville Boilers made by Messrs. Maudslay, Sons & Field, Ltd. Portable Type. Working Pressure, 170 lbs. per square inch.

Mark	Area of.		Diameter of Tubes in Inches.	Space Required.			Pounds of Dry Steam guaranteed per Hour.
	Heating Surface, in sq. ft.	Fire Grate, in sq. ft.		Depth.	Width.	Height.	
T. A. No. 4	50	1·6	$3\frac{1}{4}$	2′ 7″	2′ 10″	7′ 5″	220
T. A. No. 5	70	2·5	do.	2′ 11″	3′ 2″	do.	330
T. A. No. 6	95	3·7	do.	3′ 3″	3′ 6″	do.	490
T. A. No. 7	120	5·1	do.	3′ 7″	3′ 10″	do.	660
T. A. No. 8	150	6·8	do.	3′ 11″	4′ 2″	do.	820
T. B. No. 5	120	4·4	4	3′ 7″	3′ 6″	8′ 1″	550
T. B. No. 6	160	6·1	do.	3′ 11″	3′ 11″	do.	710
T. B. No. 7	200	8·0	do.	4′ 3″	4′ 4″	do.	930
T. B. No. 8	250	10·3	do.	4′ 7″	4′ 8″	do.	1,200
T. B. No. 9	300	12·9	do.	4′ 11″	5′ 1″	do.	1,450
T. B. No. 10	360	15·6	do.	5′ 3″	5′ 6″	do.	1,750



Figs. 1078, 1079.—Water Tube Boiler by Messrs. Yarrow & Co.

feet and 14,480 square feet of heating surface weighs, including water and all fittings, 440 tons; the Belleville boilers supplying the place of the above grate area 674 square feet, heating surface 21,100 square feet, weight, including water and all fittings, 249·5 tons. The boilers are made of different types suitable for stationary, portable or marine work. The stationary type is set in brick work, the portable type is built light and is used with portable engines, fire engines, steam cranes, &c., an auxiliary type specially strong for supplying steam to winches and for use with salt water.

Messrs. Yarrow, of Poplar, have brought out a simple and highly efficient water-tube boiler, the general features of which are shown in figs. 1078, 1079. In this type of boiler all the tubes are straight, and the upper ends are secured into the steam drum, which is made in halves and bolted together to facilitate cleaning and repairs, and the lower ends are secured into semicircular vessels, of which the top flat plates into which the tubes enter are bolted to the semicircular base.

The two groups of tubes are arranged in an inclined direction, one on each side of the furnace. In order to study the circulation in water-tube boilers, Messrs. Yarrow carried out a series of experiments, and arrived at the conclusion that the circulation in the tubes nearest the fire was established at once, and that any additional heating of the tubes clearly increased the circulation, so that no external pipes, called "down-comers," were necessary—the outer tubes, those furthest from the fire, acting as down-comers themselves.

These boilers are being used on the fastest type of vessels afloat, the Torpedo Boat Destroyers, and Messrs. Yarrow have built a number of them for Dutch cruisers. The boilers for these cruisers have steel tubes 1⅛″ diameter and 5′.0″ long, arranged in ten rows on each side of the furnace; the area of grate surface is 40·25 square feet, the heating surface 2,017 square feet, their weight is about 0·24 cwt. per indicated horsepower, whilst that of return-tube marine boilers is about 1·07 cwt. per indicated horsepower. A water-tube boiler of this type, capable of supplying steam to an engine indicating 3,000 HP., occupies about the same space as an ordinary return-tube marine boiler supplying steam to an engine indicating 2,250 HP.

The water-tube boiler by Messrs. Thornycroft, of Chiswick, differs in many respects from that of Messrs. Yarrow in construction, but in efficiency but little difference seems to exist. From figs. 1080, 1081, it will be seen that the tubes are curved, and are inserted into the upper or steam-drum above the water line; and in the special type (fig. 1081) there is one lower drum and one upper one, the furnace being divided and placed outside the group of tubes. In another

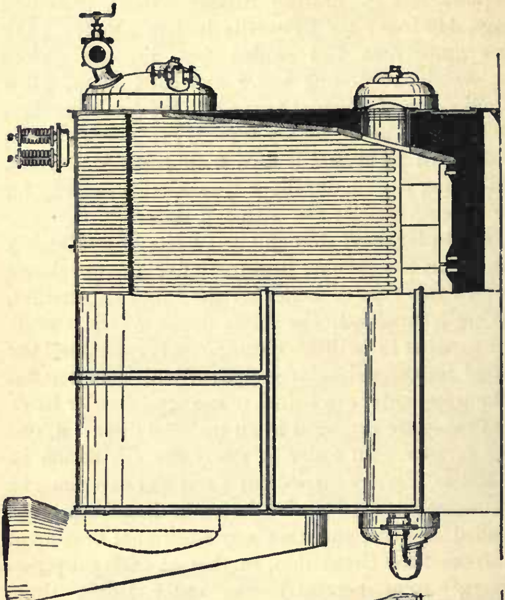


Fig. 1081.

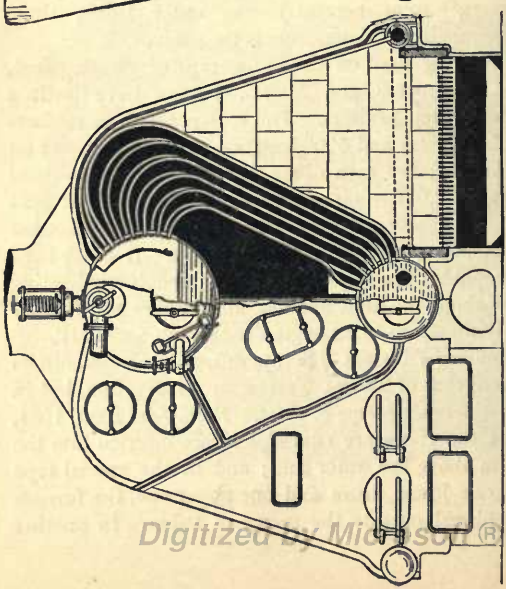


Fig. 1080.

Water Tube Boiler by Messrs. Thornycroft & Co.

type of boiler by the same firm, there are two lower drums, and the furnace is arranged between the two groups of tubes. The former is named the "Daring" type and the latter the "Speedy" type, after the torpedo boats to which they have been applied. The bends in the tubes are said to give more heating surface, and to give an amount of elasticity, to prevent undue strain by difference of temperature. In the "Speedy" there were eight Thornycroft boilers, supplying steam to engines indicating about 4,500 HP.; the area of the grate in each boiler was 25·5 square feet, the heating surface 1,840 square feet, the weight of each boiler empty about 8·94 tons.

Many other water-tube boilers are at present in use, and new varieties are still being brought out. Amongst those which are coming into use are the "Normand" boiler of the Thornycroft type; the "Clyde" boiler by Fleming and Ferguson; the "Stirling" boiler; the "Niclausse" boiler: in this boiler the tubes are placed as in the Babcock boiler, that is, nearly horizontal, but they are connected by headers at the front end only, thus all the joints are readily accessible. The header is divided vertically into two parts by a division plate, the tubes are also double; the outer tube is fixed into the back plate of the header, and the closed end rests on support at the back of the furnace; the inner tube is fixed in the division plate of the header and is open at both ends, the open back end being a few inches short of the closed end of the outer tube; by this means circulation is kept up briskly when the boiler is under steam. The headers are connected to a drum over the front part of the boiler. It is very probable that this boiler will be more generally used as it becomes more known. Other varieties of water-tube boilers are too numerous to mention, but most of them are designs based on those mentioned above.*

* See "Bertin's Marine Boilers," translated by L. S. Robertson. London: John Murray.

Lever Safety Valve.

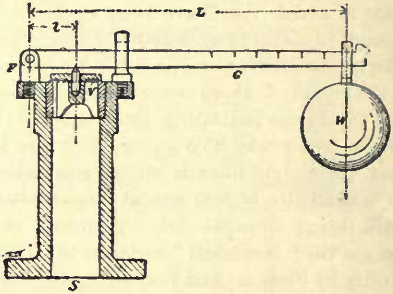


Fig. 1082.

Lever safety valves of the type shewn in Fig. 1082 are largely used but are rather giving way to the more reliable type of Dead-Weight Safety Valves, fig. 1084.

The weight for any point of blow-off pressure may be calculated by the formula below. In all but very small valves the weight of the valve itself and of the lever must be taken into account :—

A = area of valve in square inches.
L = length of lever in inches from fulcrum to centre of weight W.
l = length of the short arm of the lever in inches from centre of valve to fulcrum.
w = weight of lever.
G = distance of centre of gravity of lever from fulcrum in inches.
V = weight of valve in lbs.
W = „ in lbs. to balance blow-off pressure.
P = blow-off pressure in lbs. per square inch.

$P = \frac{W\,L}{A\,l}$ = blow-off pressure in lbs. per square inch due to weight W.

$W = \frac{A\,P\,l}{L}$ = Weight for given blow-off pressure P, without accounting for weights of valve and lever.

$\frac{G\,w}{l\,A}$ = Pressure in lbs. per square inch due to weight of lever.

$\frac{V}{A}$ = Pressure in lbs. per square inch due to weight of valve.

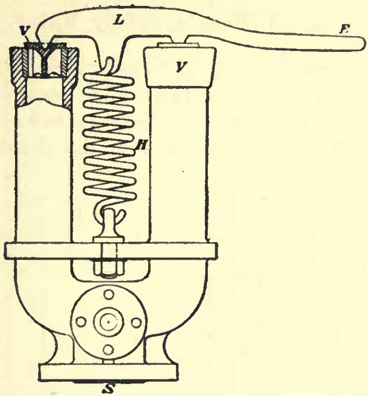


Fig. 1083.

Table 169.—Dimensions of Ramsbottom's Safety Valves (Fig. 1083).

Diameter of valves	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{1}{2}$	3	$3\frac{1}{2}$
Diameter of flange	$7\frac{1}{4}$	9	9	10	$11\frac{1}{2}$	12
Between centres of valves V	$6\frac{1}{2}$	7	$7\frac{1}{2}$	$8\frac{1}{2}$	$9\frac{1}{2}$	$9\frac{3}{4}$
Height from flange to top of casting	16	$17\frac{1}{2}$	$18\frac{3}{4}$	$21\frac{1}{2}$	$23\frac{1}{4}$	$25\frac{1}{2}$
Thickness of lever	$\frac{9}{16}-\frac{5}{16}$	$\frac{5}{8}-\frac{3}{8}$	$\frac{3}{4}-\frac{7}{16}$	$\frac{7}{8}-\frac{1}{2}$	$\frac{7}{8}-\frac{1}{2}$	$\frac{7}{8}-\frac{1}{2}$
Total length of lever	$16\frac{1}{4}$	$17\frac{3}{4}$	$18\frac{1}{2}$	26	$26\frac{3}{8}$	$26\frac{3}{4}$
Length of spring	$6\frac{1}{8}$	$6\frac{7}{8}$	$6\frac{11}{16}$	$8\frac{3}{8}$	$9\frac{5}{16}$	$10\frac{7}{16}$
Diameter of spring	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{3}{4}$
Pull on spring for 140 lbs. pres.	494·7	473·4	992·8	1374·2	1979·0	2693·8
Pull on spring for 90 lbs. pres.	318·0	432·9	638·2	883·4	1272·2	1731·7

Dimensions are given in Inches.

A loose link (not shown in the figure) is provided, to connect the lever L with the eye to which the spring is attached, so that, if the spring should break, the lever would be held by the link, and thus the valve would only blow off, but not leave the seat.

Dead-Weight Safety Valve.

Fig. 1084 represents a group of 3 dead-weight safety valves, one of the group being shown in section. This class of valve is made single, or with several valves in a group attached to the same base. The valve V is fixed in the interior of the circular casting C, on

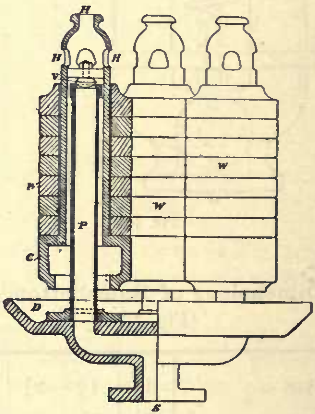


Fig. 1084.

which are placed the weights *w*, *w*. The seating for the valve is fixed on the top of the pipe P. The pipe in fig. 1084 is shown of wrought iron, although often made of cast iron. This type of safety valve requires no guides, as all the weights are below the valve, and therefore it returns by its own gravity to its proper place.

It is also very difficult to tamper with, owing to the large increase of weight required to make any appreciable difference to the blowing off pressure.

TABLE 170.—**Hyperbolic Logarithms.**

No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.
1	0·00000	26	3·25810	51	3·93183	76	4·33073
2	0·69315	27	3·29584	52	3·95124	77	4·34381
3	1·09861	28	3·33220	53	3·97029	78	4·35671
4	1·38629	29	3·36730	54	3·98898	79	4·36945
5	1·60944	30	3·40120	55	4·00733	80	4·38203
6	1·79176	31	3·43399	56	4·02535	81	4·39445
7	1·94591	32	3·46574	57	4·04305	82	4·40672
8	2·07944	33	3·49651	58	4·06044	83	4·41884
9	2·19722	34	3·52636	59	4·07754	84	4·43082
10	2·30259	35	3·55535	60	4·09434	85	4·44265
11	2·39790	36	3·58352	61	4·11087	86	4·45435
12	2·48491	37	3·61092	62	4·12713	87	4·46591
13	2·56495	38	3·63759	63	4·14313	88	4·47734
14	2·63906	39	3·66356	64	4·15888	89	4·48864
15	2·70805	40	3·68888	65	4·17439	90	4·49981
16	2·77259	41	3·71357	66	4·18965	91	4·51086
17	2·83321	42	3·73767	67	4·20469	92	4·52179
18	2·89037	43	3·76120	68	4·21951	93	4·53260
19	2·94444	44	3·78419	69	4·23411	94	4·54329
20	2·99573	45	3·80666	70	4·24850	95	4·55388
21	3·04452	46	3·82864	71	4·26268	96	4·56435
22	3·09104	47	3·85015	72	4·27667	97	4·57471
23	3·13549	48	3·87120	73	4·29046	98	4·58497
24	3·17805	49	3·89182	74	4·30407	99	4·59512
25	3·21888	50	3·91202	75	4·31749	100	4·60517

Hyp. log. 10, correct to eight places of decimals, = 2·30258509.

Common logs. multiplied by 2·30258 give hyperbolic logs.

Hyp. log. 75 = 4·31749.

Hyp. log. 7·5 = 4·31749 − hyp. log. 10.

·31749 − 2·30258 = 2·01491 the hyp. log. of 7·5.

TABLE 171.—**Temperatures, Volumes, and Pressures of a Pound of Steam.**

Temperatures.		Absolute Pressures in lbs. per square inch.	Volume occupied by a pound of Steam in cubic feet.
Fahr.	Centi.		
32	0	0·085	3390
68	20	0·333	934·6
95	35	0·806	404·8
122	50	1·78	192·0
149	65	3·62	98·45
176	80	6·86	53·92
203	95	12·26	31·26
212	100	14·70	26·36
230	110	20·80	19·03
275	135	45·49	9·124
302	150	69·21	6·153
329	165	101·9	4·280
356	180	145·8	3·057
365	185	163·3	2·748
383	195	203·3	2·236

Formulæ for Steam Pressure and Volume Curves.

V = volume in cubic feet to the pound.
P = pressure in lbs. per square inch.

$$V = 26{\cdot}36 \left(\frac{14{\cdot}7}{P}\right)^{\frac{n}{m}}$$

$$P = 14{\cdot}7 \left(\frac{26{\cdot}36}{V}\right)^{\frac{m}{n}}$$

$\frac{m}{n} = \frac{17}{16}$ for Rankine's curve,

$\frac{m}{n} = \frac{10}{9}$ for the so-called $\frac{10}{9}$ curve,

$\frac{m}{n} = \frac{14}{10}$ for the approximate adiabatic curve.

Feed Pumps.

Considerable attention has of late years been given to the design of feed pumps. In early steam engines the feed pump was practically a part of the engine, but with the necessity of using large installations of engines and boilers it became necessary to separate the feed pump from the engine and place it near to the boilers.

A large number of feed pumps have been made with steam cylinders working the pump through the medium of crank shaft and connecting rods; of late years the demand has been for direct-acting pumps working without the intervention of connecting rods or cranks.

A great many varieties of direct-acting pumps are in use, and one very successful one may be taken as a type of the best practice in feed pumps.

Messrs. Weir's feed pump is one that forces the water into the boiler slowly and steadily, and is much in favour at sea and on land. Fig. 1085 shows a section through the feed pump; the steam is admitted to the main valve by means of a small auxiliary valve at the back, and the admission is so managed that the steam valve is always in a position ready to start the pump when steam is turned on. The arrangement of the steam and pump valves is clearly shown in the figure.

The valve gear consists of a main and auxiliary valve. The main valve is for distributing steam to the cylinders; the auxiliary for distributing steam to work the main valve. The main valve moves horizontally from side to side, being driven by steam admitted and exhausted from each end alternately. The auxiliary valve is actuated by lever gear from the rod of the pump, and moves on a face on the back of the main valve, and in a direction at right angles to the main valve. By this arrangement there is no dead centre, the action being absolutely positive, because the only possible position in which the main valve can rest is at full travel—either for an up or down stroke of the piston.

Both the main and auxiliary valves are simply slide valves, but the former is half round, the round side working on the cylinder port face, which is bored out on one side to fit the valve. On the back of this main valve a flat face is formed for the auxiliary valve to work upon. Both ends of the main valve are lengthened so as to project beyond the port face, and are turned cylindrical with flat ends. Caps are fitted on each of these ends forming cylinders, which

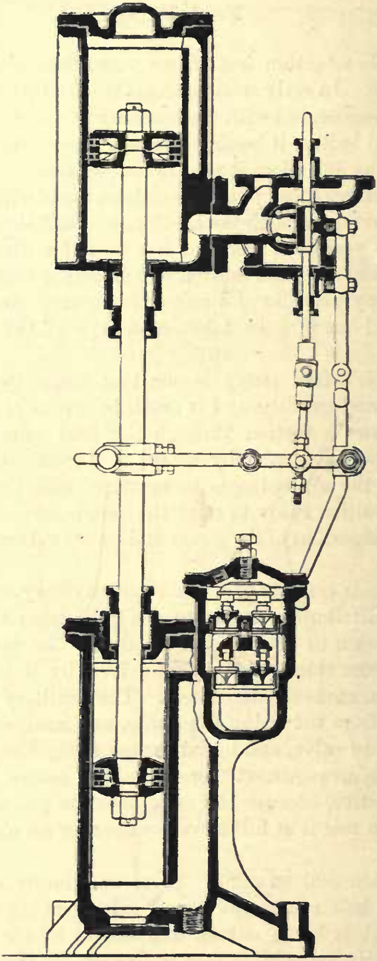


Fig. 1085.

are closed at the mouths by the flat ends of the main valve, which act as pistons.

The function of the auxiliary valve is to admit steam through the ports on the back of the main valve to move the main valve from side to side. The ports for admitting steam to the top and bottom of the cylinder are arranged to cut off before the end of the stroke, and so slow down the pump, thus permitting the water valves to settle quietly and relieve the connections from any shock. On the last quarter of the stroke the steam is thus used expansively, so effecting a considerable economy in steam consumption. Provision is made, however, by turning round the caps covering the end of the main valve for admitting live steam during the entire stroke, as, when the pumps are starting and the metal is cold, the steam condenses and it is necessary to clear out the chambers of water. These caps are turned by means of the gunmetal spindles with indicating pointers at each side of the steam valve chest. When the pump is fairly started, these bye-passes—one for the up-stroke and one for the down—are closed till the pump is working silently. The main and auxiliary valves are practically the only two moving parts in the valve chest. The stroke can be adjusted while the pump is working by the nuts on the valve spindle in accordance with the centre punch marks on the front stay, and is constant.

These pumps are sometimes arranged in pairs to work compound side by side. Table 172 gives sizes and capacities of the standard pumps made for land installations.

TABLE 172.—Standard List of Sizes and Dimensions of the Weir Direct-Acting Pump for Land Installations.

Diameter of pump.	Cylinder of pump.	Length of stroke.	Gallons discharged per double stroke.	* Gallons discharged per hour at 12 double strokes per minute.	Floor space.	Height.
5½″	7½″	15″	2·49	1792·8	1′ 5½″ × 1′ 7″	6′ 4½″
6″	8″	15″	2·94	2206·8	1′ 9½″ × 1′ 10″	6′ 10″
6″	8″	18″	3·53	2541·6	1′ 9½″ × 1′ 10″	7′ 7″
7″	9½″	18″	4·86	3499·2	1′ 10″ × 2′ 0″	7′ 7″
7″	9½″	21″	5·67	4082·4	1′ 10″ × 2′ 0″	8′ 4″
8″	10½″	18″	6·352	4573·4	2′ 2″ × 2′ 3″	8′ 0″
8″	10½″	21″	7·43	5349·6	2′ 2″ × 2′ 3″	8′ 9″
8″	10½″	24″	8·49	6113·8	2′ 2″ × 2′ 3″	9′ 6″
9″	12″	21″	9·34	6724·8	2′ 4″ × 2′ 6″	8′ 9″
9″	12″	24″	10·67	7682·4	2′ 4″ × 2′ 6″	9′ 6″
9½″	12½″	24″	11·83	8521·2	2′ 4″ × 2′ 6″	9′ 6″

* This is the best speed for boiler feeding.

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