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ELEMENTS

OF

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ELEMENTS
OF
FUEL OIL AND STEAM ENGINEERING

A PRACTICAL TREATISE DEALING WITH FUEL OIL, FOR THE CENTRAL STATION MAN, THE POWER PLANT OPERATOR, THE MECHANICAL ENGINEER AND THE STUDENT

BY

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DEDICATION

To the University of California and its splendid traditions whatever is good and helpful within these pages is affectionately dedicated by the authors.

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PREFACE TO SECOND EDITION

The rapidity with which the first edition of this book was exhausted has exceeded the fondest expectations of its writers. The field of fuel oil in its uses in steam electric generation, though an important one, is somewhat narrow in the number of people involved in its study. The early exhaustion of the first edition, however, has given the authors an opportunity to re-write the book and to add many new and interesting advances that have been made during the three year period since the first edition appeared upon the market.

We are particularly grateful to Prof. L. S. Marks of Harvard University, to Prof. W. F. Durand of Stanford University; Mr. C. R. Weymouth, chief engineer of Charles C. Moore & Co.; Messrs. R. J. C. Wood and H. L. Doolittle of the Southern California Edison Company; Mr. E. A. Rogers, chief engineer of steam electric generation for the New Cornelia Copper Co.; Mr. E. H. Peabody, consulting engineer; Prof. E. H. Lockwood of Yale; Dr. D. S. Jacobus, advisory engineer of the Babcock and Wilcox Company; and Mr. J. E. Woodbridge of Ford, Bacon and Davis, for valuable information, helpful suggestions, and a kindly and encouraging attitude for further efforts in the compilation of this work.

Many of the newer portions of this work have first appeared in print either in the columns of the Electrical World or the Journal of Electricity, and to these two publications we are grateful for the interest and kindly suggestions they have given to us.

Robert Sibley.
C. H. Delany.

San Francisco, Cal.,
January, 1921.

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PREFACE TO FIRST EDITION

Fuel oil in its power generating characteristics is a factor of prime importance on land and sea in these momentous times.

The clarion call to service is heard on all sides. And in answering this call, it must be remembered that to save is to serve.

Implicity hoping that this book may aid in establishing a fuller knowledge of the fundamental laws of fuel oil and steam engineering, and that a consequent saving in fuel will inevitably result where these laws are properly put into practice, no matter how small may be the resulting good, the authors offer to the engineering and industrial world at this time this work, which had its incipiency six years ago in certain power economy tests in Oakland, California, later to be used in lecture notes at the University of California, and finally to be rounded out by a study of power plant practice in California covering a period of several years.

The book has as its underlying theme a study of fuel oil power plant operation, and the use of evaporative tests in increasing the efficiency of oil fire plants. To accomplish this end the subject matter has been treated in three main subdivisions: First, an exposition of the elementary laws of steam engineering; second, the process involved in the utilization of fuel oil in the modern power plant; and, third, the testing of boilers when oil fired.

In treating the first subdivision, the elementary laws of steam engineering are set forth in a new manner, in that the viewpoint is taken from that of the oil-fired instead of the coal-fired power plant operator. In the second subdivision, the results of considerable labor and analysis are set forth from the collecting and collating of data involved in burner, furnace, and fuel oil tests, hitherto appearing in disconnected form and in widely varying sources. In the third subdivision the authors have given definite suggestions for fuel oil tests—largely suggestions recently presented personally by the authors at the invitation of the Power Test Committee of the American Society of Mechanical Engineers at a hearing of the Committee in New York City for the purpose of standardizing the rules for boiler tests where oil is used as a fuel.

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The many illustrative problems that have been worked out in the chapters on steam engineering and boiler economy are based upon the data obtained from the latest edition of Marks & Davis' "Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam," published by Longmans, Green & Company, which may be purchased through any reputable book dealer for the sum of one dollar. For a careful study of these illustrative examples the reader should provide himself with a copy of these steam tables, although this is not necessary for most of the discussions on fuel oil and furnace design as treated in the text.

The six beautiful views of the economy measuring apparatus installed at the Long Beach Plant of the Southern California Edison Company, featured in this book, are extended through the courtesy of R. J. C. Wood, superintendent of generation for the Southern Division of that company.

Throughout the work the authors have attempted to set forth standard practice in fuel oil and steam engineering. As a consequence they are indebted to a large group of manufacturers, engineers and power plant operators for their timely suggestions in pointing out and developing the fundamental laws of fuel oil and steam engineering practice that are dwelt upon in this work.

ROBERT SIBLEY.
C. H. DELANY.

SAN FRANCISCO, CAL.,
May 1, 1918.

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FUEL OIL AND STEAM ENGINEERING

CHAPTER I

THE MODERN POWER PLANT FOR FUEL OIL CONSUMPTION

FIG. 1.—A 20,000 h.p. Curtis turbine installed in San Francisco.

THE enormous growth of the electrical industry throughout the world during the past decade has entirely revolutionized methods of power development. Especially is this true west of the Rocky Mountains, where gigantic natural waterpowers have been put to a useful purpose. Owing to the fact, however, that most of the western streams show a great variation in flow in the different seasons of the year, it is not always possible to depend solely upon waterpower for the supply of electrical energy. In recent years the advent of crude petroleum upon the Pacific Coast, representing a total annual production of over one hundred million barrels, has made it possible when rainfall or water supply is lacking to economically supply the needed power. During certain hours of the day, too, when the so-called peak load conditions are to be met by a central station, additional electrical energy over that possible to supply from the hydro-electric station is found to be necessary. Hence, the steam power plant, consisting of large concentrated units, is now recognized as an indispensable auxiliary to continuity of service.

In order that there should be no excessive loss in distribution, these concentrated steam power units are usually found in the

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FIG. 2.—Exterior view Long Beach Plant, Southern California Edison Company. This plant is noted for its use of meters, for various sorts of economy studies and for records obtained in daily operating practice. Note the finish and æsthetic beauty of the exterior.

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heart of the great distribution centers. Especially is this true where abundance of circulating or cooling water may be obtained. Thus we find in Central California, Station A and Station C of the Pacific Gas & Electric Company, and the Fruitvale Station of the Southern Pacific Company, all situated in the distributing centers of San Francisco and its immediate vicinity. In the Los Angeles district we find that the Redondo and Long Beach plants of the Southern California Edison Company, owing to the lack of abundant cooling water near the distribution center are situated at a distance from it of some fifteen or twenty miles.

It will now be interesting and instructive to examine the details of a typical power installation of the sort just hinted at.

First, we shall describe the so-called steam cycle or the journey of the steam-making water from the time it enters the steam boiler until it has passed through the turbine, or power unit, and returned again to the boiler; secondly, we shall consider the circulating water which is necessary in large quantities to convert the exhaust steam back again into water; and thirdly, we shall also touch briefly upon the journey of the oil from the time it leaves the cars at the sidetrack until it disappears from the chimney as a flue gas. We shall also touch briefly upon the general size and functions of apparatus employed to accomplish these results.

THE STEAM CYCLE

The Storage Tank.—The supply of water for steaming purposes is usually brought to a make-up or storage tank from supply wells either on the immediate premises or nearby property. If these are not found, it is brought from rivers, lakes, or other bodies in the vicinity, or in many cases is purchased from some water company supplying the municipality. The storage tanks are varied in size, shape and capacity from a small tank, used as a receiver and hot-well, to a number of tanks, large and small, used for storage purpose only.

The use of storage tanks depends upon the source and quantity of water supplied and the load carried by the plant. Where there is a steady and positive source of supply, the tank may be of small capacity. In some cases where the supply is small and the storage at different periods is unfit for use, larger storage and settling tanks are required, and at times even filtration and cleaning tanks also are employed.

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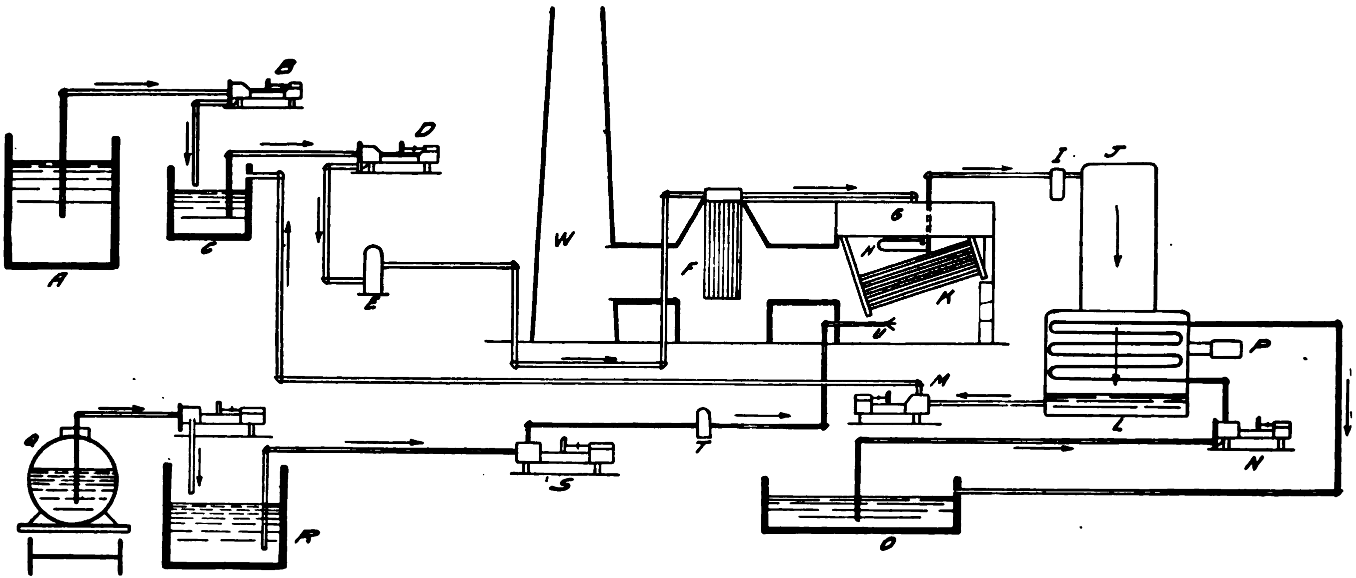

A = Supply water source.
B = Supply pump.
C = Hot well.
D = Feed-water pump.
E = Feed-water heater.
F = Economizer.
G = Boiler.
H = Superheater.

I = Separator.
J = Turbine.
L = Condenser.
M = Wet vacuum pump.
N = Circulating water pump.
O = Large water source.
P = Dry vacuum pump.

Q = Railroad oil tank.
R = Storage tank.
S = Oil feed pump.
T = Oil heater.
U = Oil burner or atomiser.
K = Furnace.
W = Smoke-stack.

Fig. 3.—A diagrammatic sketch showing the steam, circulating water and fuel oil cycles in the modern power plant.

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Pumps for Storage Supply.—The tanks above alluded to are filled either by pumps, gravitation flow, siphons, or piping from a water company's main. There seems to be no standard type of pump. Both reciprocating and rotary appear in standard practice. On the other hand, many plants have wells and water is lifted by air pressure. This, on account of the total absence of working parts, is particularly useful, where there are a number of scattered wells, and, also, when it becomes necessary to handle dirty water, that is, water containing sand, grit, and dirt in suspension. This air lift consists of a partially submerged water pipe and an air supply pipe. The casing of the well is driven below the lift pipe. The lift pipe is set in the well either with air surrounding it between the pipe and the casing, or with a cap over the casing, making the space in the casing air-tight. In some instances the well casing is used directly as the lift pipe.

FIG. 4.—Measuring and purifying tank for water supply at Redondo plant.

The Hot-Well.—The water from the storage tanks is either pumped or is caused to flow by gravity to a socalled hot-well. The hot-well is a tank which stores the water before it passes to the feed-water heater. It receives water from the condenser and admits an additional supply from the storage tanks to meet the needs of steam generation.

Feed-Water Heaters.—From the hot-well, water is taken into the feed-water heater. The function of a feed-water heater is to heat the entering water to a temperature approximating that of evaporating conditions in order to keep the boiler at as even a temperature as possible and at the same time to put the exhaust steam from the auxiliary apparatus to some useful purpose.

Feed-water heaters are divided into two general types: open and closed. In the open heater the steam comes in direct contact with the cooling water, and if there is a sufficient quantity of exhaust steam, it raises the water to 212°F., the excess steam

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passing to the atmosphere. In a closed heater, on the other hand, the water and steam do not come in contact with each other, the water usually passing through a set of tubes while the steam is brought in contact with the outside of the tubes.

Feed-Water Pumps.—The water is forced into the boiler by means of feed-water pumps, which may be either ordinary reciprocating or multistage centrifugal pumps, the latter being especially adapted to large power plants.

FIG. 5.—The Redondo Power Plant of the Southern California Edison Company with oil storage tanks.

In steam plants where closed feed-water heaters are used, the boiler feed pumps are placed before the heater, thus pumping through the heater to the boiler. When open feed-water heaters are used, however, the boiler feed pumps are placed between the heater and the boiler.

Economizers.—Economizers are sometimes installed as well as feed-water heaters. The economizer is a series of pipes through which the feed-water passes, placed in the path of escaping gases from the boiler.

The Boiler.—The boiler next receives the water from these heaters and converts it into steam at the desired pressure and

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temperature. The main types of boilers are water and fire tube. Modern practice indicates a decided preference for water tube boilers. A water tube boiler consists of steam and water drums placed on the top and a mud drum placed below, the two being connected by a series of tubes filled with water or steam. The fire is below these tubes, and the heat from the furnace is made to pass around them several times by means of baffles or partitions, thus supplying heat for steam generation.

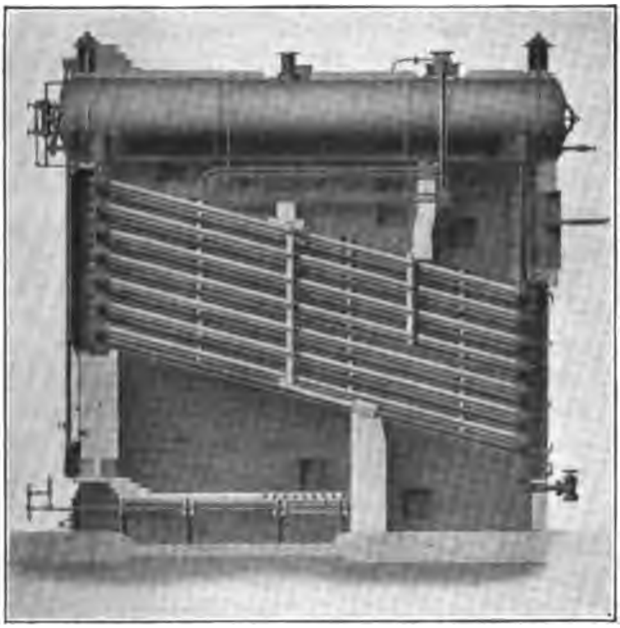

Fig. 6.—Cross-section of the Babcock & Wilcox Boiler—oil-fired—with Peabody furnace.

The Superheater.—The water being thus converted into saturated steam by heat from the furnace of the boiler, is next passed through a series of tubes known as a superheater. These tubes are exposed in a heated portion of the gas passage and thus the steam is raised to a point much higher than its saturated temperature.

The Separator.—From the superheater the steam passes through suitable piping to a separator, placed between the boiler and the engine, or between the boiler and the turbine. This separator is placed as near the power unit as possible in order to

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remove all condensed steam that may be found in the pipes. One form of separator performs its function by quickly reversing the direction of the flow of steam, thus depositing the water into a drip which is drained off into the condenser. Another form gives a rotary motion to the entering steam thus hurling particles of water off by centrifugal force and collecting it in proper receptacles. Again, baffle plates are at times employed wherein the flow is interrupted by corrugated or fluted plates, and the particles of water adhering to these are then drained off.

Reciprocating Engine or Steam Turbines.—The steam next goes from the separator to the main power generating units. The main units in earlier practice were reciprocating engines; in modern installations they are steam turbines.

Reciprocating engines may be divided into several classes, the details of which will not be outlined in this general discussion. Suffice it to say, however, that the main principle upon which reciprocating engines act is that steam enters a cylinder under pressure, thus forcing ahead a piston which is connected to a crank arm, thereby causing rotation and the consequent generation of power.

Steam turbines are divided into two general classes known as impulse turbines and reaction turbines. In the impulse turbine steam is allowed to expand in passing through a nozzle, thus causing the steam to travel at an enormous velocity. The steam, having acquired this velocity, by impinging against movable blades, causes rotation and the consequent generation of power. In the reaction turbine, however, the steam is allowed to enter the buckets or rotating vanes at a comparatively low velocity, These vanes are so designed that the steam may expand in this movable portion and by means of its expanding pressure cause rotation and hence the generation of power.

Turbines as a general rule have two other classifications, known as vertical and horizontal. The vertical turbine revolves upon a vertical shaft, which is supported at the bottom by a thin film of oil under the high pressure of about 900 to 1000 lb. per in. The horizontal turbine, however, as the name indicates, rotates on a horizontal axis, and is supported in the usual manner by means of suitable bearings.

Condenser.—From the steam turbine, the steam, having expanded to its useful limit, is dropped into an incasement through which cool water is being passed. Upon coming in contact with

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this cooling device the steam is again converted into water. The apparatus performing this function is known as a condenser, there being two general classes: surface condensers and jet condensers.

In the surface condenser the steam from the turbine and the cooling water from a nearby source of supply do not come into direct contact, but the cooling water is passed through inclosed tubes around which the steam from the turbine or power unit is made to pass. This type of condenser is used where large quantities of water are available for cooling purposes but not for steaming purposes. Thus the use of salt water, the only abundant supply available for ocean-going steamships and large power plants situated near the ocean, makes a condenser of the surface type imperative for such installation.

In the jet condenser the supply of cooling water is allowed to mingle with the steam as it drops from the turbine or power unit and thus the steam is at once condensed into water. The water from the jet, if the supply is pure, may be used in the hot-well for steam purposes.

Wet Vacuum Pumps.—The condensed steam, now in the form of water, is pumped from the condenser back again into the hot-well by means of what is known as the wet vacuum pump. This pump may be either a reciprocating or rotary pump, but in general the rotary type seems to have the preference. Thus the entire cycle for the water is traced from the make-up tank or hot-well through the boiler and power unit, and back again to the hot-well.

Dry Vacuum Pumps.—The condenser also has a dry vacuum pump in order to remove from the steam space within any air which may have been trapped from the steam generated in the boiler. This pump is nothing more nor less than an ordinary air compressor so designed that it will take air at a very low pressure and compress it up to atmospheric pressure, thus pumping, as it were, into the outer atmosphere such air as may have been entrapped in the condenser.

CIRCULATING WATER CYCLE

From the description of the working of the condenser it is seen that cooling water is necessary to convert the steam in the condenser back again into water. This cooling supply is known

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as circulating water, which is usually taken through pipes from some large natural lake or river or even the ocean and forced by means of reciprocating or centrifugal pumps through the condenser back again into the open. The water in its journey is raised in temperature in the surface condenser system from 15 to 20°F. above its entering condition.

THE OIL CYCLE

Of general interest to boiler testing and operation is the cycle employed in the utilization of crude petroleum as a fuel. Let us then briefly trace the journey the oil makes through the modern power plant.

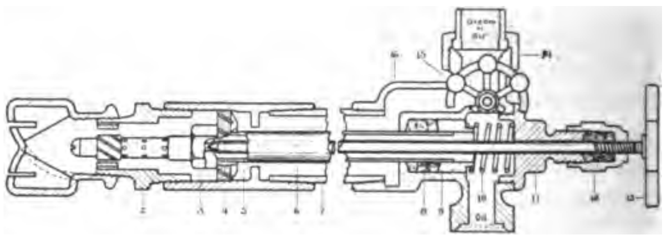

FIG. 7.—The Staples & Pfeifer fuel oil atomizer.

In the larger installations the oil is sidetracked from the main railway line in specially designed cars or barges for its easy conveyance and handling. An oil heater, consisting of a coil through which steam is passing, is lowered into the car in order to warm the oil as it is drawn, thus making its transfer considerably easier. By means of a pump this oil is then taken into a storage tank which may be of wood, steel, or concrete, depending upon the permanence of design thought necessary. From this storage tank the oil is pumped through oil heaters, the exhaust from the pumps in many cases being utilized in still further heating oil before it reaches the burner or atomizer.

An atomizer is a device used to vaporize or spray the oil into the furnace in fine globules or particles. This is accomplished by means of steam, air, or some mechanical contrivance. Immediately upon its being sprayed into the furnace carefully designed air regulating devices admit sufficient air from below to cause perfect combustion. The heat thus liberated from the oil due to its burning with the oxygen is caused to flow in and around

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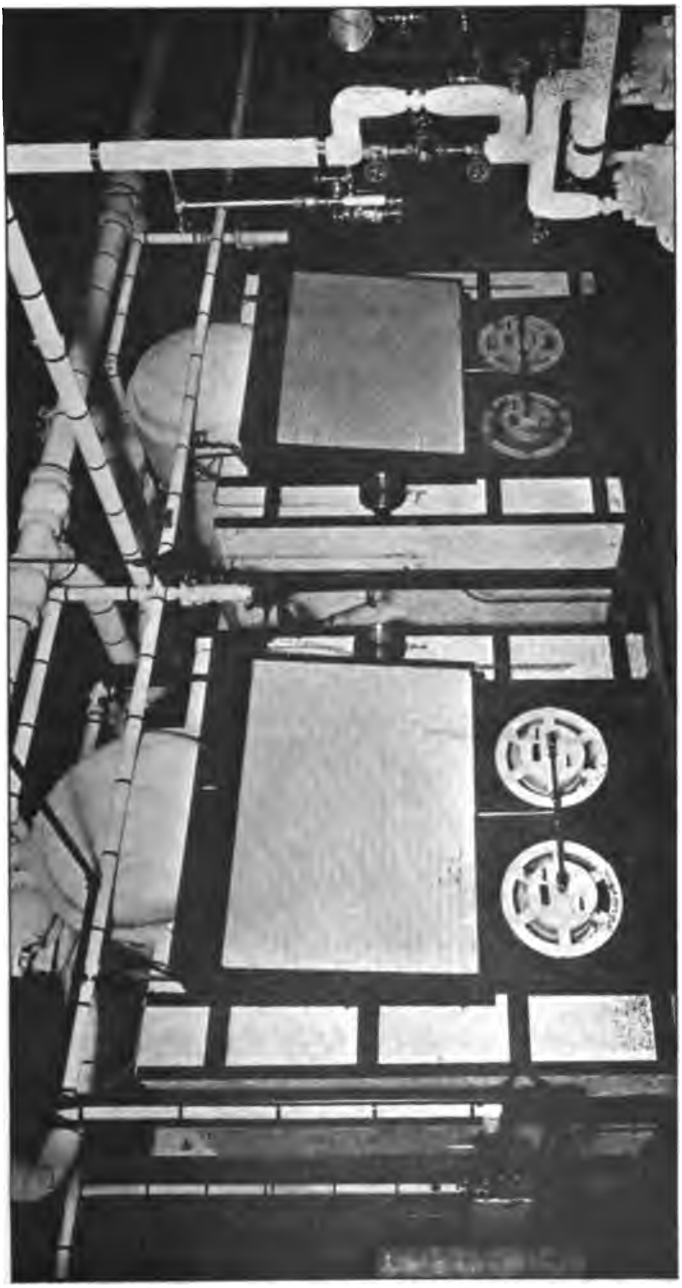

FIG. 8.—Mechanical atomizing oil burners applied to stationary boilers.

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numerous tubes through which water is passing and thus this water is converted into steam. After passing these tubes the heated flue gases brought to life by the burning of the oil with the entering air are then conducted through the chimney out into the atmosphere.

An interesting detail in the boiler plant is the automatic system of firing employed to minimize labor and improve efficiency in burning the oil. The Moore patent fuel oil regulating system which, from one central point, controls the oil supply, the atomizing steam and the amount of air to each furnace, is an interesting example.

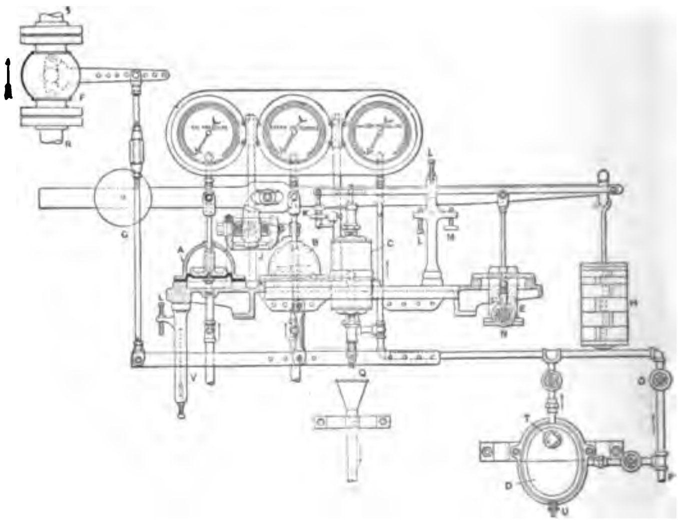

Fig. 9.—Moore steam to burner regulator.

This regulator is actuated by the pressure from the main steam header so that any variation in steam requirements will cause a corresponding change in the amount of oil fired, due to an increase or decrease in the steam supply to the oil pumps and atomizers. Any fluctuation in steam pressure operates a governor whose power arm controls a bleeder valve on the oil pump discharge line, thus cutting off the oil supply if the steam pressure is too high and increasing it if too low. Any change in pressure in the oil main, in turn, controls the amount of steam for atomizing and of air for burning the oil.

It is found that a simple straight line relationship exists between the amount of steam required for atomizing the oil and the

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amount of oil burned. Two diaphragms are employed to balance the pressures in the oil main and in the steam main connected to the burners. Any difference in oil pressure operates a rotary chronometer valve in the steam main through the medium of a fulcrum, water motor and lever connecting rod. Likewise the variance in oil pressure actuates a counterweighted rock shaft which moves the dampers so as to vary the amount of air admitted for combustion.

GENERAL SUMMARY

Thus it is seen in this brief description that by using crude oil as fuel three main cycles of operation are synchronously carried on in the modern power plant. Briefly summarizing, these are as follows:

Water is taken through the boiler, converted into steam and passed through a driving mechanism, after which the steam is reconverted into water and this water again passed through the boiler. Simultaneously with this action water is being pumped through the circulating system to bring about the conversion of the steam from the power unit into water. Again oil in a finely atomized or gaseous state is being fed through pipes into the furnace, where it immediately combines with the proper quantity of oxygen from the entering air, and thus sufficient heat is liberated from the oil to evaporate the water supply of the boiler into steam for power generation.

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CHAPTER II

FUNDAMENTAL LAWS INVOLVED IN STEAM ENGINEERING

Fig. 10.—Mechanical energy in reciprocating units at Redondo.

IN the awful throes of the French Revolution and the immediate years following, the old saying that "every cloud has its silver lining" proved true in certain lines of scientific advancement, for the metric system of units was conceived and put into practice at that period.

Our modern system of Arabic numerals, now practically universally adopted throughout the civilized world, required over five hundred years of human fumbling and competition with the old Roman method of numerical representation, before a complete replacement was accomplished, so intensely are we all creatures of habit and slaves to tradition.

And so it is that although a period of a century is now passed since the institution of the metric system, modern central station engineering practice is still entangled with Fahrenheit scales, boiler horsepowers, mechanical horsepowers, myriawatts, Baume scale readings for gravity, inches of mercury vacuum, pounds pressure per sq. in., feet and inches—all units related so unscientifically and empirically as to cause bewilderment in itself.

In the following discussion, however, the authors will endeavor to set forth the various units of expression in such simple language that it is hoped that even the beginner may have little difficulty in understanding their meaning. Let us first get some conception of the need for units of measurement and how such units are fundamentally conceived.

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Newton's Laws of Motion.—Fable has it that Sir Isaac Newton, when a boy in England lying one day under an apple tree and gazing upward, saw an apple fall to the ground. The contemplation of this phenomenon led Newton to give to the world three fundamental laws upon which modern engineering science is built. Briefly these laws are as follows:

Law 1. Every body continues in a state of rest or a state of uniform motion in a straight line except in so far as it may be compelled by force to change that state.

Law 2. Change of motion is proportional to impressed force and takes place in the direction of the straight line in which the force acts.

Law 3. To every action there is always an equal and contrary reaction; or the mutual actions of any two bodies are always equal and oppositely directed.

Hence a force is said to be acting according to Law 1 whenever the physical conditions are such that velocity is changed in magnitude or direction. Thus, when a train of cars is started or stopped, a force is necessary to cause this phenomenon, and this is evidently a change in the magnitude of the velocity. On the other hand, in the rotation of a fly wheel, the velocity may change solely in direction without a change in magnitude, and yet a force be necessary to maintain its parts in equilibrium. Hence a force may be considered as a push or a pull acting upon a definite portion of a body, but this tendency may be counteracted in whole or in part by the action of other forces. In the latter instance the force is usually denoted as pressure, and it is the consideration of this latter case, or the consideration of pressures, that will largely concern our attention in the generation of steam in a boiler.

Three Fundamental Units of Length, Mass and Time.—In considering Law 2, it is seen that there is some inherent property in matter that makes it difficult to set it in motion. Physicists have defined this quality of matter as being the inertia of a body. Inertia is expressed quantitatively in engineering practice in terms of its mass, which is measured in pounds. In order that these quantities, force and mass, now introduced may be quantitatively measured, it is necessary to have some fundamental units upon which to base our computations. Three units only are fundamentally required; namely, a unit of length, a unit of mass, and a unit of time. Scientific practice has deduced for these units the centimeter, the gram, and the second, which are

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well known and need no further illustration. In engineering practice, however, especially among English-speaking people, the foot, the pound, and the second seem to be in almost universal usage. We shall consequently largely express our deductions in terms of these latter units.

FIG. 11.—Electrical energy from steam turbine in San Francisco.

Velocity, Acceleration, and Force Defined.—Having now decided upon the three fundamental units of measurement, let us look into other fundamental definitions and secondary units to be employed.

Since engineering science must deal with motion and the change of motions per unit of time, it is necessary that we have units wherein to measure them. Change of distance per unit of time is known as velocity and is expressed in feet per second. A change in distance may, however, be undergoing a change, and this phenomenon is known as acceleration, which is measured by the change of velocity in feet per second.

Since a force P is fundamentally defined as being proportional to the change in motion of a body, it follows that a force is equal to a constant, M, multiplied by the change in motion, or, in other words, multiplied by the acceleration, a.

When M is in pounds mass and a is acceleration in ft. per sec. per sec., the force P is measured in **poundals.** The **pound force** is the unit, however, that has been universally adopted in engineering practice. The pound is such a force as will give to a pound mass the same change of motion per second as is acquired by a body falling freely to the earth's surface. A body falls to the earth's surface with an acceleration of g ft. per sec. per sec., wherein g has an average value of about 32.16. We have then the fundamental mathematical expression for force in pounds as follows:

$$P = Ma$$

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Whenever, however, it is necessary to ascertain the mass M in pounds from the known weight W of the body in lb., it is necessary of course to divide W by g in order to ascertain the mass. Thus we have in this case $P = \frac{W}{g}a$ (1a)

Thus, if an automobile weighing 3000 lb. accelerates from a stand-still to forty miles per hour in fifteen seconds, we compute the force required to accomplish this as follows:

$$P = \frac{3000}{32.16} \times \frac{40 \times 5280}{60 \times 60 \times 15} = 367 \text{ lb.}$$

Since this total force must be supplied from the engine cylinder this now gives us a preliminary clew as to how the total engine cylinder area is to be proportioned.

Engineers oftentimes find a more direct method of deriving the laws of motion by considering that the resistance which a body offers to its rate of change in velocity is called its inertia. The word "mass" has been invented as a quantitative unit by which inertia may be measured; hence we may forget any particular physical meaning of the word "mass" and consider it merely as a constant, the same as the Greek letter π enters into the area of a circle or its circumference, when speaking of them in reference to the diameter.

Mr. William Kent, the late author of Kent's "Mechanical Engineers Pocketbook," was the first to discuss this in scientific magazines, and it has later been used with much effectiveness and clearness by consulting engineers in establishing the fundamentals of mechanics.

Bodies acquire different changes of motion per second, or, in other words, different accelerations at different points on the earth's surface. A formula has been established by means of which proper corrections may be made. A concrete illustration of this will appear in the next chapter wherein a mercury column is used to measure atmospheric and vacuum pressures at different latitudes and altitudes.

It is unfortunate that mass and force have the same unit of expression, for they are definite distinct physical concepts and should be carefully distinguished in order to avoid confusion.

Conception of Work and Power.—In Law 2 we are informed that the change of motion takes place in the direction of the straight line in which the force acts. It is often convenient to note quantitatively the product of the force and the distance

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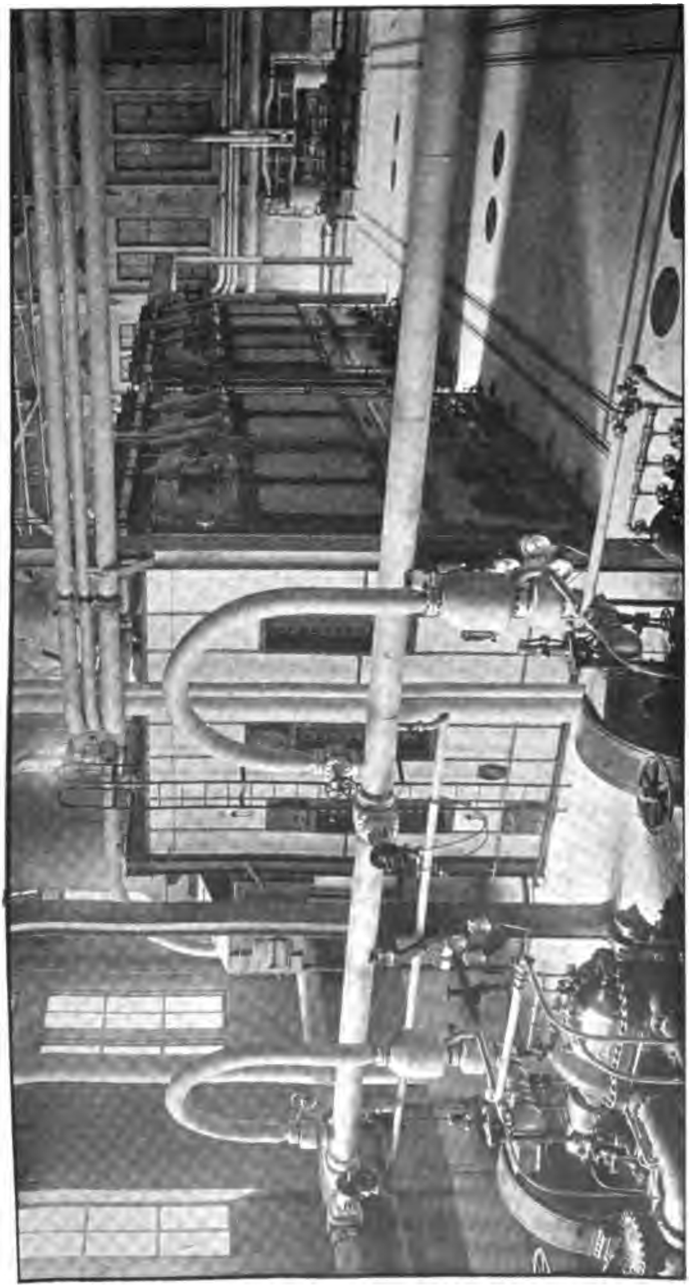

Fig. 12.—High pressure pumping system, San Francisco. One man feature of operation applicable in installations of this type. This plant is designed for stand by operation and kept in eternal readiness should a fire disaster similar to the one of 1906 ever again visit San Francisco.

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through which the force acts. This product is called "work" and is numerically computed by multiplying the force in pounds by the distance in feet through which the force acts. The resulting computations are then expressed in foot-pounds (ft.-lb.) Thus, if the mean effective pressure, P, in a cylinder is measured in pounds per sq. in. and the piston has an area of A sq. in., it follows that the total force or pressure acting in the direction of the motion of the piston is PA. When this force has pushed the piston the length of its stroke, L ft., the work accomplished is

FIG. 13.—The steam gage tester illustrates the application of a fundamental law, wherein a pressure is balanced against the force due to gravity.

PLA ft. lb., since this is the product of the force and the distance through which the force acts. If there are N working strokes per minute, the ft. lb. of work accomplished every minute are now seen to be $PLAN$.

The mention of the words "per minute" in the last statement now indicates to us that the time taken to perform a given quantity of work in engineering practice is of vast importance. Consequently this fact necessitates still another unit of measurement, namely that of power. Power is defined as the time rate of doing work. The horsepower is the basic unit. When 550 ft. lb. of work are performed per sec., or 33,000 ft. lb. per minute,

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a horsepower is said to be developed. Hence, since in the above engine cylinder $PLAN$ ft. lb. per min. are being developed, the horsepower is computed as follows:

$$H.P. = \frac{PLAN}{33{,}000} \tag{2}$$

Thus, in Alameda, California, a certain Diesel oil engine has a piston area of 113.15 sq. in., a stroke of 1.5 ft., a mean effective pressure of 77.3 lb. per sq. in., and each cylinder makes 125 working strokes per minute. Hence, each cylinder develops

$$H.P. = \frac{77.3 \times 1.5 \times 113.15 \times 125}{33{,}000} = 50.0$$

In a later discussion the particular power units employed in steam engineering practice will be considered in minute detail, such, for instance, as the horsepower, the boiler horsepower, and the myriawatt.

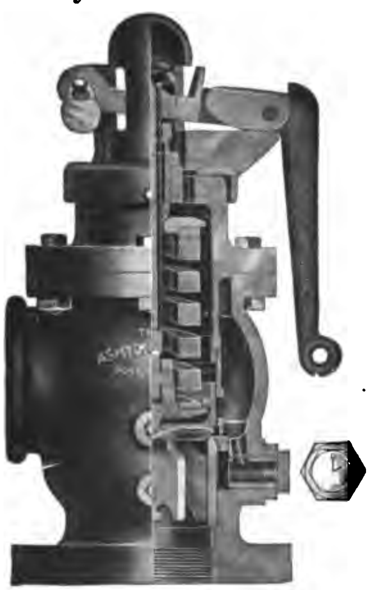

Fig. 14.—The safety valve shows the possibility of safety application, when pressures become unbalanced.

Various Types of Energy Employed for Useful Work.—Another important consideration is that of the physical characteristic of a body which enables it to perform work. This physical quality possessed by a body which enables it to perform a definite quantity of work is spoken of as its energy. Energy then is the capacity for work. In general we meet with two great classes of energy. One is that of **kinetic energy,** or energy of motion. According to Law 2, if the motion of a body be changed, a force is required. Hence a body actually in motion possesses kinetic energy. The other type of energy is known as **potential,** or energy of position. Thus steam moving with a high velocity, by the nature of its kinetic energy, is enabled to drive the wheels of an impulse turbine. On the other hand, crude petroleum when heated so that it will unite with the oxygen of the air gives out

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energy in the form of heat, which may be caused to do useful work. The energy inherently latent in the crude pertroleum is known then as potential energy. Engineering practice is largely concerned with the harnessing of various forms of energy. Looking about us in nature and in modern engineering accomplishment, we may see numerous instances of energy. The steam engine and steam turbine indicate a form of mechanical energy; the incandescent light, or the dynamo, that of electrical energy; the evolving of heat in the burning of crude oil, that of chemical energy; the human conducting of affairs, that of human energy; the rays of light from the sun, dissipating eternally 10,000 h.p. over each acre of the earth's surface, that of solar energy, and so on indefinitely. Modern investigation has conclusively established the fact that all types of energy are interchangeable, and though some types of energy are more readily convertible into other types, yet the basic law is true that no energy in sum total is ever destroyed, and on this basis, or law, known as conservation of energy, practically all of our engineering formulas and computations are evolved.

The conversion of the chemical energy of crude oil into heat energy of the furnace and thence into steam largely concerns our attention in this discussion. Thus each pound of California crude oil will be found in later articles to contain approximately 18,500 British thermal units of heat energy. This energy of one pound of oil when wholly converted into mechanical energy is sufficient to lift a person weighing 150 pounds through a vertical skyward journey of some 18 miles. Hence the study of the application of such enormous reservoirs of energy, latent in crude petroleum, will prove intensely interesting and instructive.

Bearing in mind these fundamental laws, we should now be able to see mentally the exact changes of energy that are going on in the modern power plant; first as chemical energy in oil, next as latent heat energy in furnace gases, then as latent heat energy in steam, next as energy of motion in the moving parts of the power generating apparatus, where the final transformation into electrical energy is brought about.

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CHAPTER III

THEORY OF PRESSURES

FIG. 15.—The thermometer suspension for barometer correction.

IN the preceding discussions we have seen that a force is said to be acting whenever the physical conditions are such that the velocity of a body tends to be changed in magnitude or direction. If two opposing forces are equally balanced, there is simply a tendency to change motion and such a force is known as a pressure. This opposing force in the case of a gas or vapor under pressure is supplied by the walls of the containing vessel. Pressures then constitute an important phase of steam engineering practice.

The Steam Gage.—In steam engineering practice heavy pressures, that is pressures above the atmosphere, are usually measured by means of an instrument known as a steam gage. This gage consists of a piece of hollow metal bent into a circular shape which, under pressure, tends to straighten out, see Fig. 16. This straightening effect is proportional to the pressure under which the boiler is working. A rack and pinion movement, placed on the end of this curved piece of metal in the steam gage, causes the needle of the gage to indicate pressure readings. By comparing this gage with a definite standard its accuracy is ascertained.

The Difference Between Absolute Pressure and Gage Pressure.—There is a point at which a gas is said to exert no pressure. This expanded condition of a gas has never been wholly realized in practice, yet this very beginning point or zero value is most convenient in expressing pressure valuations and such denotations are known as absolute pressure values. The steam gage attached to the boiler does not read absolute pressure values, but

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such pressure readings are known as pounds pressure per sq. in. (gage) which means that one must add the absolute pressure of the atmosphere, P_a, to the gage reading, P_g, in order to ascertain the true absolute pressure P under which the boiler in generating steam. Thus

$$P = P_a + P_g \tag{1}$$

Fig. 16.—Interior and exterior view of steam gage, showing principle of operation.

Thus, if a pressure gage of the steam boiler reads 186.4 per lb. sq. in. and the pressure of the atmosphere is found to be 14.6 lb. per sq. in., the absolute pressure under which the boiler is operating is

$$P = 186.4 + 14.6 = 201.0 \text{ lb. per sq. in.}$$

The Column of Mercury.—The most accurate method of measuring small pressures such as the pressure of the atmosphere and condenser vacuum pressure is by means of a vertical column of mercury. In its simplest form this consists of a long glass tube closed at one end and filled with mercury. The tube is then inverted and the open end placed in a vessel of mercury exposed to the atmosphere or condenser as the case may be, as shown in Fig. 18.

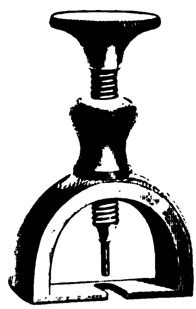

Fig. 17.—A hand adjuster for the steam gage.

In the case of atmospheric pressure determination the mercury will at once lower itself in the long tube until the height of enclosed mercury above that in the vessel is sufficient to balance the pressure from the

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atmosphere without. If the barometer be at sea-level and the temperature of the mercury column 32°F., the height of mercury

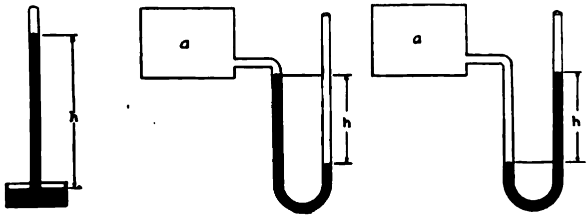

FIG. 18. -The principle of the atmospheric barometer, the condenser vacuum and the measurement of pressures above the atmosphere.

will now measure exactly 29.921 inches for such standard conditions.

Vacuum Pressures.—It has already been pointed out that measurement of pressure by means of the steam gage indicates a pressure over and above that exerted by the atmosphere and consequently to ascertain the true absolute pressure of the fluid under measurement we must add to the gage reading the atmospheric pressure of the day. And so in the measuring of the pressure of a condenser, unavoidably there has grown up a similar but opposite custom in which the pressure is measured down from the atmosphere. Such a reading is known as a vacuum pressure. In order then to ascertain the absolute pressure P under which a condenser is operating it is necessary to subtract the vacuum pressure reading P_v from the atmospheric pressure reading P_a. Thus

FIG. 19.—Typical condenser barometer for steam turbine operation.

$$P = P_a - P_v \tag{2}$$

Thus if a condenser is operating under 28.5 in. of vacuum and the atmospheric pressure is 29.92 in., we mean that the actual air and steam still undisposed of in the condenser exert an abso-

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lute pressure equivalent to the difference between 29.92 and 28.50 which is 1.42 in. of mercury.

Confusion in Pressure Units.—We now see that readings in inches of mercury for low pressure and pounds pressure per sq. in. for high pressure are expressions that are not at all comparable to each other and hence their interrelation becomes an endless source of confusion.

Relationship of Pressure Units.—By careful measurement of the atmosphere at sea-level, scientists have established that the height of a mercury column with the mercury at 32°F. in temperature is 29.921 in. Such a column of mercury one square inch in cross-section weighs 14.696 lb. This gives us at once a method by which we may transfer inches of mercury I_m into pounds of pressure per square inch P. Thus

$$\frac{I_m}{P} = \frac{29.921}{14.696} \tag{3}$$

Inches of Water and Pounds Pressure per Square Inch.—Very slight pressures are often measured in inches of water above or below atmospheric pressures. Thus, in determining the draft of a chimney, a "U" tube is inserted into the chimney, and the height of the unbalanced portion of the water column indicates the draft in the chimney in inches of water. Since a column of water 1728 in. high and one square inch in cross-section at 100°F. weighs exactly 62 lb., the inches of water I_w may be converted into lb. pressure per sq. in. P by the formula

$$\frac{I_w}{P} = \frac{1728}{62} \tag{4}$$

The Thirty Inch Vacuum.—In engineering practice a thirty inch mercury vacuum is considered to be the point of absolute zero in pressure. This is not strictly true, however, for we have just seen that such an absolute zero point is reached under a vacuum pressure of 29.921 in. of mercury. The reading of the column of mercury in this case is taken when the mercury is at a temperature of 32°F., which is the standard temperature for scientific measurement. If, however, we change our standard to that of 58.4°F. the same weight or pressure of mercury now measures just 30.0 in. This temperature is more nearly that of the condenser room where atmospheric pressures are read and since it makes a column of even thirty inches in height, we shall adopt such a reading at 58.4°F. as standard for absolute vacuum meas-

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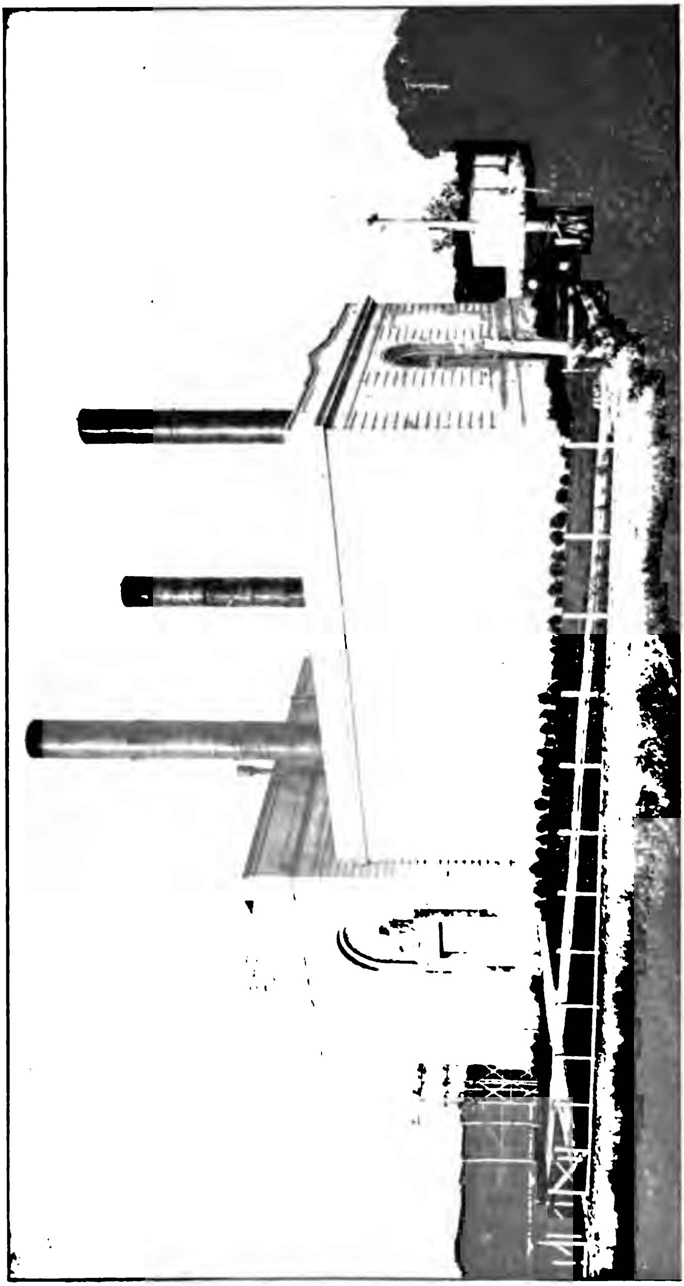

FIG. 20.—5,000 k.w. steam turbine station, oil burning, Pacific Gas and Electric Company, Sacramento, California. Standby plants of this nature viewed in connection with hydro-electric network afford unusual opportunities for æsthetic design in exterior, as shown in this view.

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urement. We shall, however, bear in mind that the same column at 32°F. would stand at 29.921 in.

The Practical Formula for Conversion of Pressures.—Since we have thus established an even unit for the standard vacuum, we may also consider 14.7 lb. pressure per sq. in. as its equivalent instead of the cumbersome figure of 14.696 as stated above. This involves an error of four points in fifteen thousand which is negligible. Our formula for reduction on the thirty inch vacuum becomes

$$\frac{I_m}{P} = \frac{30}{14.7} \tag{5}$$

To Reduce Barometer Readings to the Standard Thirty Inch Vacuum.—Although 58.4° is nearer the condenser room temperature than is the 32°F. basis, still for accurate measurement the actual temperature of the medium surrounding the mercury column should be ascertained and thus a correction must be made to reduce the height of the mercury column to what it would read if at a temperature of 58.4°F.

This is best illustrated by taking a concrete example. Let us suppose that the mercury column inserted into the condenser of a turbine reads 28.56 in. when the mercury temperature is 82°F. and that a barometer in the vicinity indicates the atmospheric pressure in the condenser room to be 30.08 in. of mercury when its mercury column is at 78°F.

The first thing to be done in the solution of this problem is to ascertain what the two mercury columns would have read had their respective mercury columns been at 58.4°F. Scientific investigation indicates that the expansion of mercury is according to the following equation in which I_t is the height in inches of mercury at t°F. and I_m at 58.4°F.

$$I_t = I_m\,[1.0026 + .000104\,(t - 58.4)] \tag{6}$$

Hence, to ascertain the true vacuum reading in inches of mercury we find by substitution

$$28.56 = I_m\,[1.0026 + .000104\,(82 - 58.4)]$$
$$\therefore I_m = 28.415.$$

Similarly to compute the corrected barometer reading of the day, we find by substitution that

$$30.08 = I_m\,[1.0026 + .000104\,(78 - 58.4)]$$
$$\therefore I_m = 29.942.$$

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The net absolute pressure will now be the difference between the corrected atmospheric barometer reading and the corrected vacuum reading for the condenser, which according to equation (2) is

$$I_p = 29.942 - 28.415 = 1.527 \text{ in. of mercury.}$$

Since all standard vacuums in engineering practice are now measured on a 30 in. vacuum basis, we find that the corrected vacuum reading I_{cv} for a condenser is

$$I_{cv} = 30 - I_p \qquad (7)$$

$$\therefore I_{cv} = 30 - 1.527 = 28.473 \text{ in. (vacuum).}$$

This corrected vacuum reading I_{cv} which in this case is 28.473 is commonly spoken of as the vacuum referred to a 30 in. barometer.

For delicate scientific work this reading should be carried to still further refinements by making a correction for the expansion of the brass on the barometer scale and also for a variation in gravity at the particular place of measurement. At high altitudes and extreme northern and southern latitudes such a correction is essential.

Corrections for the Brass Scale of a Barometer.—Professor Marks in his computation of steam tables for condenser work published by the Wheeler Condenser and Engineering Company has ably discussed the correction for relative expansion of mercury and the brass scale of the barometer as follows:

The linear expansion of brass is about one-tenth that of the apparent linear expansion of mercury exerting a constant pressure. Where a mercury column has a brass scale extending its whole height which is free to expand with changes in temperature, the readings on the brass scale of the height of the mercury column must be corrected for the relative expansion of the mercury and the brass scale. The following table is taken from table 99 of the Smithsonian physical tables and gives the constants for various barometer heights by which to multiply the temperature correction in order to obtain the corrections of the mercury column.

Example.—Reading of barometer 29.84, temperature of barometer 79°F. . In the foregoing table the nearest figure to 29.84 is 29.8 opposite which the correction factor is .0027. If it is desired to reduce the barometer to a 58.4°F. standard, the change in temperature is from 79° to 58.4° = 20.6° and multiplying .0027

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REDUCTION OF BAROMETRIC HEIGHT TO STANDARD TEMPERATURE CORRECTIONS FOR RELATIVE EXPANSION OF MERCURY AND BRASS SCALE

Height of Barometer in inches	Correction in inches per deg. F.	Height of Barometer in inches	Correction in inches per deg. F.
20.0	0.00181	28.0	0.00254
20.5	0.00185	28.5	0.00258
21.0	0.00190	29.0	0.00263
21.5	0.00194	29.2	0.00265
22.0	0.00199	29.4	0.00267
22.5	0.00203	29.6	0.00268
23.0	0.00208	29.8	0.00270
23.5	0.00212	30.0	0.00272
24.0	0.00217	30.2	0.00274
24.5	0.00221	30.4	0.00276
25.0	0.00226	30.6	0.00277
25.5	0.00231	30.8	0.00279
26.0	0.00236	31.0	0.00281
26.5	0.00240	31.2	0.00283
27.0	0.00245	31.4	0.00285
27.5	0.00249	31.6	0.00287

by 20.6 we get .056 in. as the barometer correction. Subtracting this from 29.84 in. we get the barometer reading for mercury at 58.4°F. as 29.84 in. − .056 in. = 29.784 in.

Corrections for Altitude and Latitude.—Since the height of a mercury column gives true pressure readings so long as it represents a definite force or weight and since the weight or force of gravity varies at different altitudes and latitudes over the earth's surface, it is necessary to enter such a correction when the extreme refinements of the work in hand demand it. The standard value of gravity is taken at 32.173. The following formula, in which I_{mg} is the correct reading, g is the gravity coefficient, λ the latitude, and h the altitude at the point of pressure measurement, may be applied for this correction.

$$I_{mg} = \left[\frac{32.173 - .082 \operatorname{Cos} 2\lambda - .000003h}{32.173}\right] I_m \qquad (8)$$

Thus in a certain engineering investigation in Berkeley, California, where the latitude is 38° and the elevation 50 ft., the condenser mercury column corrected for temperature read 28.473 in. What should its properly corrected reading be when gravity is taken into consideration?

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By substitution

$$I_{mg} = \frac{32.173 - .082 \times .2419 - .00015}{32.173} \times 28.743$$

$$= \frac{32.153}{32.173} \times 28.473 = 28.464.$$

Such refinements as the one for brass scale correction and especially for latitude and altitude readjustments are not necessary in most steam engineering tests. It is well, however, to bear in mind such computation in case investigations of extreme detail should arise.

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CHAPTER IV

MEASUREMENT OF TEMPERATURES

Fig. 21.—A thermocouple for high temperature measurement.

WHEN the finger is inserted into a cup of warm water and then again into water formed by the melting of ice a distinct sensation is felt in each case. Many years ago scientists and philosophers attempted to explain this sensation by assuming that a substance existed which they called "caloric" whose entrance into our bodies caused the sensation of warmth and whose egress therefrom gave the sensation of cold. But heat, if a substance at all, cannot be similar to those substances with which we are familiar, since a hot body weighs no more than one which is cold.

The discussion in this article is not concerned directly with heat but rather with one of its effects, namely, that of change in temperature. From the above it is readily seen that temperature is an indicator of the physical effect of heat rather than a quantitative means of heat measurement. This statement is easily proved, for when we place our fingers alternately upon a piece of cold and hot iron at the temperatures mentioned for water in the opening paragraph of this discussion, the same physical sensation is experienced. Yet to transform the iron from a temperature of freezing water to that of boiling water takes far less heat than for the transfer of water under similar conditions.

Fixed Points for Thermometer Calibration.—Since water is the most generally distributed substance through out nature and one of the most convenient for handling in the laboratory its

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freezing point and boiling point are used by common consent as two definite marks for temperature calibration. Thus in the Centigrade scale the freezing point of water is the zero point and the boiling point of water under standard conditions of atmospheric pressure is the one hundred unit point. Again, in the Fahrenheit scale the freezing point of water is the thirty-second division point and the boiling point of water the two hundred and twelfth division point. Similarly for the Reaumur scale, the freezing point of water is the zero division point and the boiling point of water, the eightieth division point.

The Various Temperature Scales Employed.—The Centigrade scale as described above has grown into rapid use in scientific investigation and now may be said to be universally adopted throughout the world for such practice. The Fahrenheit scale, on the other hand, has so ingrained itself into engineering practice that engineers are loath to part with it in spite of its cumbersome and unscientific divisions. In this work, then, we shall be compelled to express temperature measurement in the Fahrenheit scale. The Reaumer scale, mentioned above, finds slight application in this country and in such places where it is employed it is used for measurement in stills and breweries. All three of these scales are often for scientific purposes transformed to a socalled absolute zero which is 459.4°F. below the ordinary zero on the Fahrenheit scale. A free discussion of this absolute scale will be set forth in a discussion on thermodynamic laws of gases which will be found in another chapter.

Relationship of Fahrenheit and Centigrade Values.—In order that transfers from one thermometer scale to another may be conveniently and rapidly accomplished, it now becomes necessary to develop some simple mathematical relationships whereby this may be done. Since all of the scales are graduated uniformly between the freezing and boiling point of water, their relationship may be said to be linear. In the study of analytical geometry we find that such relationships may be expressed by the straight line formula:

$$\frac{x - x_1}{x_2 - x_1} = \frac{y - y_1}{y_2 - y_1} \tag{1}$$

wherein x and y represent any simultaneous temperatures expressed in different scale readings and the subscripts 1 and 2 represent definitely known points in correspondence. In order then to find a relationship between the Fahrenheit and Centi-

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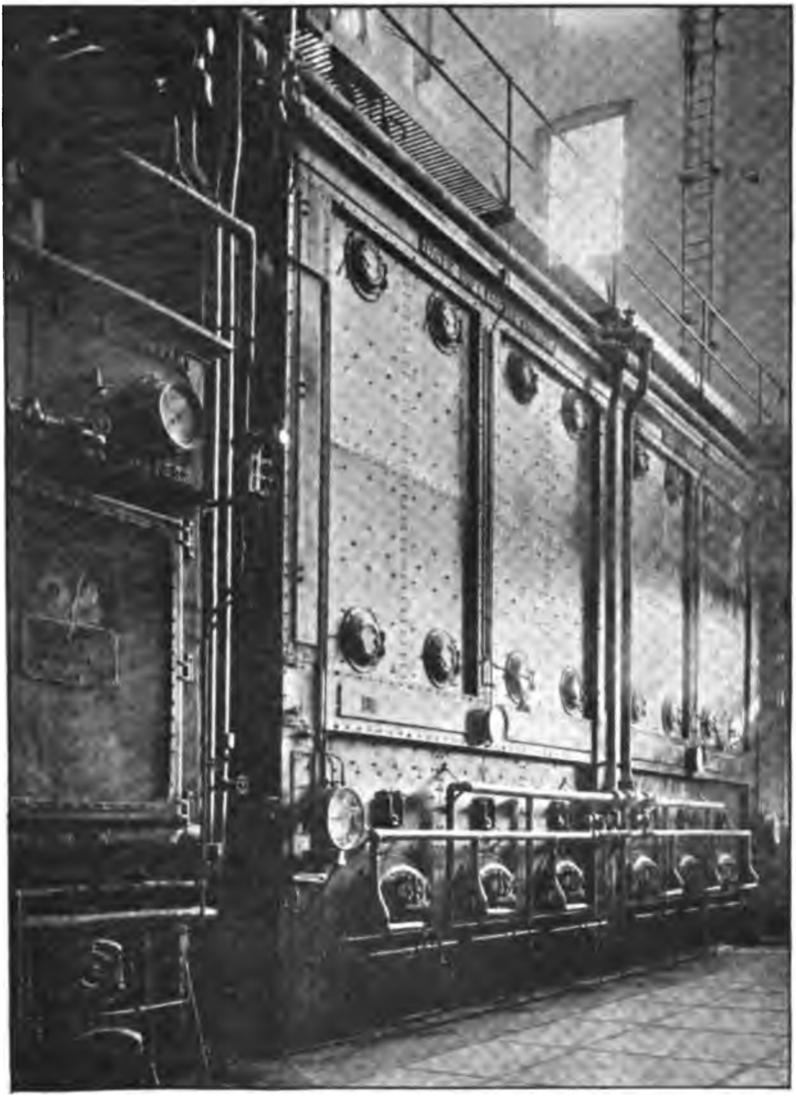

Fig. 22.—Oil-fired Stirling boilers at station *A*, Pacific Gas and Electric Company, San Francisco.

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grade scale, if x represents the Fahrenheit and y the Centigrade, we find that x_1 would be 32 when y_1 is 0, and x_2 would be 212 when y_2 is 100. Consequently we derive a relationship thus:

$$\frac{F - 32}{212 - 32} = \frac{C - 0}{100 - 0}$$

$$F - 32 = \frac{180}{100}C$$

$$\therefore F - 32 = \frac{9}{5}C \qquad (2)$$

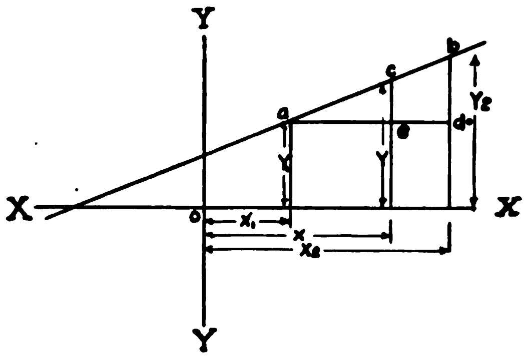

Fig. 23.—The linear relationship of temperature scales.

As an example, if the entering water in a boiler test is 84°F., this value is converted at once to the Centigrade scale by substituting in the formula

$$F - 32 = \frac{9}{5}C$$

$$\text{or } C = \frac{5}{9}(F - 32) = \frac{5}{9}(84 - 32) = 28.9°$$

Relationship of Fahrenheit and Reaumur Values.—A relationship between the Fahrenheit and Reaumur scales is similarly established.

$$\therefore F - 32 = \frac{9}{4}R \qquad (3)$$

Thus in order to illustrate the application of this formula a temperature of 84°F. reduces to the Reaumur scale as follows:

$$84 - 32 = \frac{9}{4}R$$

$$\therefore R = 23.1°$$

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Relationship of Centigrade and Reaumur Values.—To develop a relationship between the Centigrade and Reaumur scales the same reasoning is involved.

$$\therefore C = \frac{5}{4}R \tag{4}$$

Thus to convert a Centigrade reading of 28.9° into the Reaumur scale, we substitute directly

$$R = \frac{4}{5}C$$

$$\text{or } R = \frac{4}{5} \times 28.9 = 23.1°$$

In case that rapidity is necessary in the conversion of one scale to another and extreme accuracy is not required, a conversion chart is easily constructed whereby these three scales may be converted graphically from one to the other.

Methods of Temperature Measurement.—The ascertaining of correct temperatures is of extreme importance. Due to the wide range of temperatures that occur in practice, a number of different methods of temperature measurement are necessary. The method to be employed depends upon the range of temperature involved and often too upon the accessibility of the object whose temperature is desired. We shall describe first the approximate methods that are used in the ascertaining of temperatures.

Estimation by Flame Color.—A number of years ago in the steel industry, it was found that a flame emitted definite gradations of color depending upon its temperature. In 1905 the Bureau of Standards issued a bulletin covering this point and made a statement that one may ascertain temperatures with an accuracy of 100 to 150°F. by means of eye judgment. It is stated, however, that it is impossible to ascertain temperatures above 2200°F. As this is the upper limit of furnace heating in steam engineering, we need not then be concerned with exceeding the limit.

In a booklet published by the Halcomb Steel Company, 1908, the following tabulation is given to aid eye judgment in estimating temperatures:

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°C.	°F.	Colors	°C.	°F.	Colors
400	752	Red, visible in the dark......	1000	1832	Bright cherry-red
474	885	Red, visible in the twilight...	1100	2012	Orange-red.
525	975	Red, visible in the day-light..	1200	2192	Orange-yellow.
581	1077	Red, visible in the sunlight...	1300	2372	Yellow-white.
700	1292	Dark red....................	1400	2552	White welding heat.
800	1472	Dull cherry-red.............	1500	2732	Brilliant white.
900	1652	Cherry red..................	1600	2912	Dazzling white. (bluish white).

The Melting Point of Metals and Alloys.—Another method of approximately ascertaining the temperature is by means of the melting points of alloys and metals. A number of these alloys are on the market and are convenient in ascertaining the approximate temperature of furnaces and other heat generating apparatus.

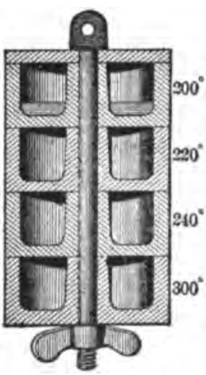

FIG. 24.—A cup for melting alloys.

The Method of Immersion.—A third method is by heating a piece of metal of known weight and specific heat to the temperature of the furnace and then immersing the heated body in water. By knowing the rise in temperature of the water, the temperature of the furnace may be approximately ascertained. The loss of heat in the hot substance is evidently equal to the heat gained by the water. Let, t_x be the unknown temperature of the hot substance, M_x the weight of the hot substance in lb., M_w the weight of the water in lb. t_2 the final temperature of the water, t_1 the initial temperature of the water and c_x the specific heat of the hot substance; then, if we assume that the specific heat of the water is 1, we may write at once

$$M_x(t_x - t_2)c_x = M_w(t_2 - t_1)$$

$$\text{Therefore, } t_x = \frac{M_w(t_2 - t_1)}{M_x c_x} + t_2 \qquad (5)$$

Mean specific heats of a number of metals which may be used for this purpose are as follows:

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MEAN SPECIFIC HEATS

Substance	Ordinary temperature	Mean for high temperature
Platinum	0.032	0.038
Iron (cast)	0.130	0.180
Nickel	0.109	0.136

As an example let us suppose that 4 lb. of cast iron heated to an unknown temperature is plunged into 20 lb. of water at 64°F., thereby raising the water to a temperature of 124°F. By substituting in the formula, we have

$$t_x = \frac{20(124 - 64)}{4 \times .180} + 124 = 1791°\text{F.}$$

The Alcohol and Mercurial Thermometers.—For accurate temperature readings up to 900°F. the expansion of liquids is made use of, for experiment shows that the expansion of a liquid is proportional to the rise of temperature. Since alcohol has a low freezing point, in fact so low that it cannot be reached by any natural temperature, the alcohol thermometer is usually made use of for low temperature readings. Since its boiling point is also comparatively low, it is impracticable for high temperatures. Mercury, on the other hand, is an excellent substance to use in thermometers as the variation in its expansion coefficient with rise of temperature is such that the deleterious effect of the expansion coefficient in the glass tube is very nearly offset by the compensating error introduced by assuming a constant expansion coefficient for the mercury. Mercury boils at 676°F. and for many degrees below this point gives off considerable vapor. As a consequence the ordinary mercurial thermometer cannot be depended upon for a higher temperature than 500°F. An ingenious device, however, enables us to make use of the mercurial thermometer up to 800 or 900°F. A small amount of nitrogen gas is put in the upper column of the thermometer tube and as the mercurial column expands it consequently compresses the nitrogen gas. The reactive pressure of the gas upon the mercury raises its boiling point so that the high temperatures above indicated can be accurately read.

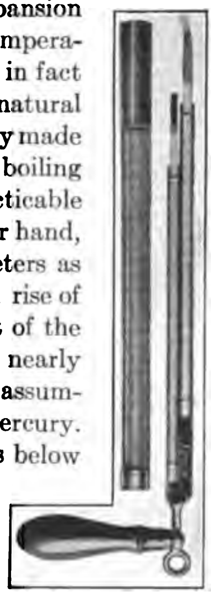

FIG. 25.—Hygrometer for boiler room humidity.

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The Expansion Pyrometer.—For the estimating of temperatures higher than 900° a number of types of instruments are employed. The expansion pyrometer, which acts upon the principle that the expansion of metals is proportional to the rise in temperature may be quite accurately used between the range of 1200 to 1500°F.

Electrical Thermometers.—Electrical thermometers are, however, the most satisfactory and accurate for steam engineering practice. Electrical thermometers act upon two distinct physical principles. One class operates upon the principle that the junction point of two metals when heated generates an electromotive force which is proportional to the temperature rise. Consequently if the readings are made by means of a delicate galvanometer, calibrated to read degrees Fahrenheit, an accurate type of instrument is at once evolved. The other principle is based upon the experimental fact that the resistance of a metal varies with the temperature rise. Hence, by measuring this rise in electrical resistance by delicately calibrated instruments, an accurate thermometer results. The thermo-couples made use of for the former type of electrical thermometer usually consist of platinum with platinum alloyed with 10 per cent. of rhodium. In the latter class the resistance element is enclosed in a highly refractory substance.

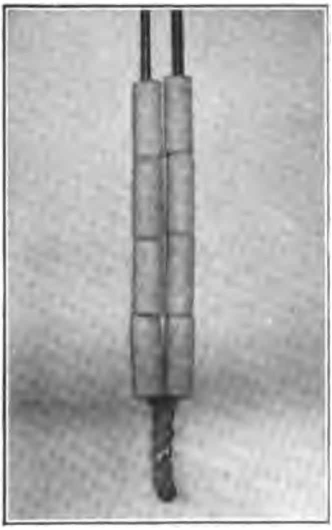

FIG. 26.—Thermo-couple ready for insertion in the furnace.

The Radiation Pyrometer.—Where temperatures are above the limit of measurement by rare metal pyrometers, or where the point of high temperature is inaccessible to the thermo-couple, the radiation pyrometer is employed. This instrument is focused on the hot body at a distance. The heat radiating from the object under investigation is reflected by a concave mirror in the back of the pyrometer telescope and concentrated at a focus point on a small thermo-couple. This thermo-couple gives off electrical energy proportional to the temperature of the radiat-

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ing body, and as a consequence the temperature is thus ascertained.

Thus, the whole range of temperatures met with in engineering practice is covered by some form of accurate temperature indicating device. The Bureau of Standards at Washington is ready to calibrate for a small fee any thermometer sent to them.

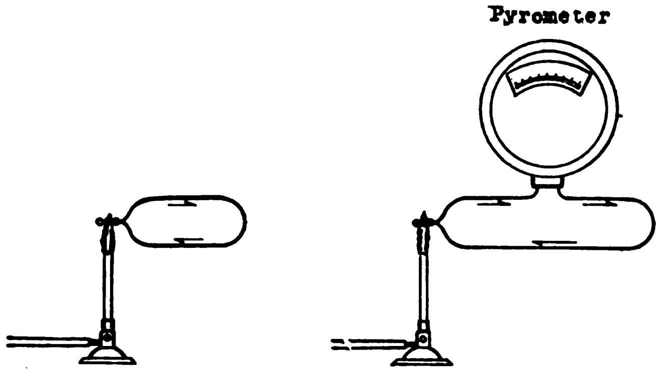

FIG. 27.—Principle of operation of the thermo-couple.

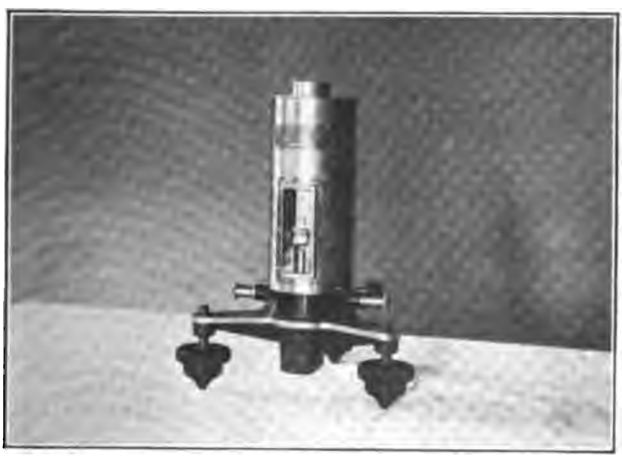

FIG. 28.—Galvanometer for delicate temperature measurement.

At least one carefully calibrated thermometer should be kept for reference or comparison in the laboratory of any one interested in steam engineering testing.

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Standardization and Testing of Thermometers.—The testing of thermometers is of utmost importance. All thermometers should be carefully calibrated for refined steam engineering tests. The Bureau of Standards has issued in its circular No. 8, an excellent guide for such work. All thermometers are calibrated when completely immersed in the substance whose temperature is being ascertained.

The Stem Correction.—In engineering practice temperatures of steam and water are usually ascertained by setting the thermometer into a well which is sunk into the pipe conveying the steam or water. This well is filled with mercury or oil and the heat transferred to the thermometer by conduction. As a consequence, however, a portion of the thermometer protrudes in the atmosphere above the well and is consequently at a lower temperature. A so-called stem correction is hence necessary to ascertain the correct reading of the thermometer.

FIG. 29.—Well for thermometer insertion.

This correction is large if the number of degrees emergent and the difference of temperature between the bath and the space above it are large. It may amount to more than 35°F. for measurements made with a mercurial thermometer at 750°F.

The stem correction may be computed from the following formula:

$$\text{Stem correction} = Kn(t_1 - t_2) \qquad (6)$$

K = factor for relative expansion of mercury in glass;
0.00015 to 0.00016 for Centigrade thermometers;
0.000083 to 0.000089 for Fahrenheit thermometers, at ordinary temperatures, depending upon the glass of which the stem is made.

n = number of degrees emergent from the bath.

t_1 = temperature of the bath.

t_2 = mean temperature of the emergent stem.

Thus suppose that the observed temperature was 100°C. and the thermometer was immersed to the 20° mark on the scale, so that 80° of the mercury column projected out into the air and the

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mean temperature of the emergent column was found to be 25°C., then

$$\text{Stem Correction} = 0.00015 \times 80 \times (100 - 25^\circ)$$
$$= 0.9^\circ.$$

As the stem was at a lower temperature than the bulb, the thermometer read too low, so that this correction must be added to the observed reading to find the reading corresponding to total immersion.

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CHAPTER V

THE ELEMENTARY LAWS OF THERMODYNAMICS

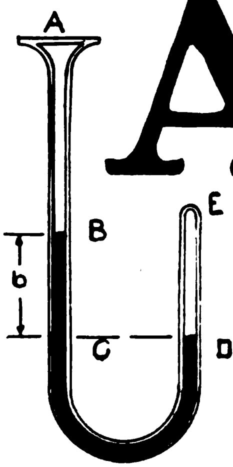

Fig. 30.—The establishment of Boyle's law.

AS pointed out in the discussion on temperatures, scientists in former times conceived that the phenomena accompanying the addition or subtraction of heat could only be explained by the existence of a fluid which they called "caloric."

But these scientists or calorists, as they were called, had to give a hitherto unknown property to their substance and maintained that "caloric" was a weightless fluid. This substance also had the property of filling the interstices of bodies and of passing between bodies over any intervening space. To illustrate, they said, "caloric" would fill the interstices of a body as water enters a sponge. Now, when we squeeze a sponge some of the water oozes out and wets our hands. The calorists assumed that the friction or rubbing of a body with the hand for instance, made the hand warm because friction was supposed to decrease the capacity of a body for holding "caloric," and as in the squeezing of the sponge, water oozes out, so caloric oozed out and made the hand feel warm.

The Irrefutable Experiments of Davy.—Davy, however, exploded this theory in 1799, when by rubbing two pieces of ice together, he actually caused the ice to melt. This evidently would be impossible under the caloric theory above stated, according to which friction caused capacity for caloric to be decreased. Yet here was evidenced the reverse. From time immemorial, men have considered that the force of truth is almighty, and yet how slow the human race is to overthrow an

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imperfect but well-established theory. For instance, so powerful was Sir Isaac Newton's grip on the scientific world that because he announced that no successful correction could ever be made for the uneven refraction of light rays in lenses, the whole world for fifty years thoroughly abandoned the idea of ever being able to use refractive telescopes, and consequently, during that period we find telescopic reflective mirrors used entirely.

FIG. 31.—The furnace gases and entering air obey rigid but simple thermodynamic laws. (Boiler fronts at Long Beach Plant of the Southern California Edison Company.)

Joule's Complete Demonstration of the Mechanical Equivalent of Heat.—And so it was in the case of the theory of heat. Notwithstanding the all-powerful demonstration of Davy in 1799, it remained for Joule, nearly fifty years later to finally put forth the finishing data to forever overthrow the caloric theory and introduce the modern idea of heat. This eminent scientist constructed a machine in many respects similar to an ice-cream freezer, the essential difference being, however, that the machine was used to increase the heat in the liquid instead of cooling the same. Joule conceived the idea that heat was one form of energy. Should this be true, it should be mutually convertible. One of the easiest methods of measuring energy is the well known pile driver. Energy is definitely computed by weighing the hammer in pounds and multiplying this weight by the distance in feet through which the weight falls. The result is foot-pounds energy. By a clever contrivance constructed somewhat on this principle, Joule measured the amount of energy absorbed in his

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machine and the consequent rise of temperature in the liquid. He soon established the fact that a definite number of foot-pounds of mechanical energy was equivalent to a definite number of heat units in the liquid. This experimental result is most important and is one of the basic principles of modern engineering. Careful scientific measurements have proved that one British thermal unit, or B.t.u., of heat energy is equivalent to 777.5 foot-pounds of mechanical energy.

The First Law of Thermodynamics.—From this discussion it follows that the first and greatest law of thermodynamics is the mathematical expression of the fact that heat energy and mechanical energy are mutually interchangeable. Thus if W represents energy in foot-pounds; H, energy in heat units; and J, this experimentally determined constant, we have the relationship

$$W = HJ \tag{1}$$

In steam engineering practice H is usually expressed in B.t.u. and the quantity J has a value of 777.5 as has been stated above. In other words, 1 B.t.u. of heat energy is equivalent to 777.5 ft. lb. of mechanical energy. In the chapter on units, we have defined the fundamental unit of energy, namely the foot-pound. This unit of heat energy now introduced, known as the British thermal unit, is the $1/180$th part of the heat necessary to raise one pound of water from 32°F. to 212°F. under standard atmospheric conditions of pressure. This is the unit which has been adopted by Marks and Davis, in their "Steam Tables and Diagrams" and although differing from other previously existing units is nevertheless practically universally adopted at this time.

Boyle's Law.—Early in the last century, Boyle established the fact that a perfect gas, such as air, follows very closely the law known as Boyle's law, that the product of its pressure and volume is always constant provided the temperature is kept constant. Expressing this in mathematical symbols, if p is the absolute pressure in lb. per sq. ft.; v, the volume in cu. ft. occupied by 1 lb., we have the relationship

$$pv = p_0v_0 \tag{2}$$

Steam is not a perfect gas and hence does not obey this law with exactness, still the formula may be used with a fair degree of accuracy when considering superheated steam. Accurate formulas will be given later for steam variation. As an instance, how-

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ever, of approximate computation, let us consider a boiler operating at 186.3 lb. gage or 201 lb. absolute pressure per sq. in., and producing superheated steam at 527°F. If we know the volume at one pressure we may ascertain approximately the volume at another pressure. In the steam tables the volume of steam at 201 lb. pressure per sq. in. is found to be 2.83 cu. ft. per lb. Hence at 250 lb. pressure the volume would become $v = \frac{200 \times 2.83}{250}$ = 2.26 cu. ft. per lb. The steam tables give this quantity by actual experiment as 2.31. Hence the formula is seen to work with superheated steam within 2.5 per cent. of accuracy. For chimney flue gases and air, however, Boyle's law is very exact.

FIG. 32.—Superheated steam approximately obeys simple thermodynamic laws. (Superheated steam ducts of station *C* of the Pacific Gas & Electric Company in Oakland.)

Charles' Law.—In 1806 another law was found connecting the variables of a perfect gas. This great law, known as Charles' Law, sets forth the fact that when the pressure is kept constant the volume of a gas increases proportionately to the increase in temperature. Thus if t is the temperature in degrees Fahrenheit this law states that

$$v = v_o \left(1 + \frac{t-32}{491.6}\right) \tag{3}$$

As an illustration, if we wish to compute the volume that 1 lb. of air would occupy in a furnace at 2100°F., knowing that v_0 has a value of 12.39 cu. ft. for air at 32°F. then

$$v = 12.39 \left(1 + \frac{2100 - 32}{491.6}\right) = 69.8 \text{ cu. ft.}$$

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The Absolute Scale.—The establishment of this law indicates the fact that all temperatures should naturally be measured from a point other than that of the freezing temperature of water. Thus it is seen from the above that a point of 459.6° below the ordinary Fahrenheit scale would be known as an absolute zero. Throughout this work then T will represent the absolute scale and t the ordinary scale. Thus

$$T = t + 459.6 \tag{4}$$

The Composite Law of Gases.—Since it is seen that the product of the pressure and volume is proportional to the change in absolute temperature we shall now write one of the most useful

Fig. 33.—Saturated steam obeys not at all the simple laws of thermodynamics. (Circulating water pumps operated by saturated steam at the Redondo plant of the Southern California Edison Company.)

formulas in the computation of gas constants, namely that

$$pv = RT \tag{5}$$

in which R is a constant.

In the case of air, let us see if we can compute this constant. Experimentally it is found that the volume of air in a boiler room temperature of 84°F. is 13.71 cu. ft. per lb. when the atmospheric pressure is 14.7 lb. per sq. in. Substituting in the above formula we have $14.7 \times 144 \times 13.71 = R\ (459.6 + 84)$. Therefore $R = 53.3$.

A Formula for Gas Density.—If we let γ be the density of a gas, it is evident that it has a value equal to the reciprocal of v

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in the above equation. In other words the density of a gas is the weight of 1 cu. ft. under standard conditions of pressure and temperature. We may then write without further proof the formula,

$$R\gamma = \frac{p}{T} \tag{6}$$

To Compute "R" for Any Gas.—In the measurement of gases there is a standard pressure and temperature to which all gas volumes and densities are reduced in order to have some basis of comparison. These standard conditions are the temperature of freezing water and the pressure of the atmosphere at sea-level.

From the equation last writen above, it is now evident that since p and T are constant for all gases under this standardized method of comparison, then the product of R and γ must also be constant. This gives us a method or rather formula by which we may obtain the value of R for any gas if we know its density. The molecular weights of all gases may be obtained by reference to any standard book on elementary chemistry.

Let us multiply both sides of the above equation by the molecular weight m and by rearranging the terms, we have

$$Rm = \frac{pm}{\gamma T}$$

For oxygen $\gamma = 0.089222$ lb. per cu. ft. at atmospheric pressure and 32°F. and $m = 32$.

$$\therefore Rm = \frac{14.7 \times 144 \times 32}{0.089222 \times 491.6} = 1544.$$

Since this product Rm is always a constant, we have for any perfect gas that

$$R = \frac{1544}{m} \tag{7}$$

This formula together with the preceding general formulas for pressures and volumes now enables us to ascertain practically all the constants for perfect gases.

As an example let us assume that the temperature of an escaping chimney gas is 400°F. What would be the density of the nitrogen content of the escaping flue gases? First find the value for R for nitrogen for which $m = 28$.

$$\therefore R = \frac{1544}{28} = 54.98$$

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Hence since

$$R\gamma = \frac{p}{T}$$

$$\text{We have } 54.98\gamma = \frac{14.7 \times 144}{459.6 + 400}$$

$$\therefore \gamma = .04475$$

It is always convenient to express volumes as the number of cu. ft. per lb. Hence when the symbol V is used it will mean the total volume content of the gas under consideration. If M represents the weight of this volume V we have the relationship

$$Mv = V \tag{8}$$

$$\text{or since } pv = R \times T, \text{ therefore}$$

$$pV = M \times RT \tag{9}$$

Thus if we have given 18.805 lb. of dry flue gas we can easily compute the volume it would occupy when leaving the chimney at 400°F., if it is known that the value of R for the chimney gas is 51.4. Thus

$$14.7 \times 144V = 18.805 \times 51.4 \times (400 + 459.6)$$

$$\therefore V = 393.5 \text{ cu. ft.}$$

Further Illustrative Examples.—In order to still further illustrate the wide uses to which the formulas above given may be applied in engineering practice, the following six problems are worked out in full:

1. Find the volume of one pound of air in a compressor at a pressure of 100 lbs. square inch, the temperature being 32°F.

From Boyle's Law:

$$pv = p_0 v_0$$

at 32°F., v_o for 1 lb. of air is 12.39 cu. ft. and $p_0 = 14.7 \times 144$

$$\therefore v = \frac{14.7 \times 144 \times 12.39}{100 \times 144} = 1.82 \text{ cu. ft.} \quad \textit{Ans.}$$

2. From Charles' Law find the volume of one lb. of air at atmospheric pressure and 72°F.

$$v = v_0\left(1 + \frac{t - 32}{491.6}\right) = 12.39\left(1 + \frac{72 - 32}{491.6}\right) = 13.4 \text{ cu. ft.} \quad \textit{Ans.}$$

3. Find the temperature of two ounces of hydrogen contained in one gallon flask and exerting a pressure of 10,000 lbs. per sq. in.

2 oz. = 1 gal.

16 oz. = 1 lb. or 8 gals. = 1.068 cu. ft.

$$\text{Then } T = \frac{pv}{R} = \frac{10{,}000 \times 144 \times 1.068}{765.86}$$

$$\therefore T = 2050°\text{F. (abs.)} \quad \textit{Ans.}$$

$$\text{or } t = 2050 - 459.6 = 1590.4°\text{F.} \quad \textit{Ans}$$

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4. How large a flask will contain 1 lb. of Nitrogen at 3200 lbs. per sq. in. pressure and 70°F.?

$$p = 3200 \times 144, \qquad T = 459.6 + 70 = 529.6, \; R = 54.98$$

$$pv = RT \qquad v = \frac{RT}{p}$$

$$\therefore v = \frac{54.98 \times 529.6}{3200 \times 144} = .0631 \text{ cu. ft.} \quad \textit{Ans.}$$

5. Ten lbs. of air at 200°F. occupy 120 cu. ft. What must be the pressure?

$$V = 120$$
$$M = 10$$
$$R = 53.33$$
$$T = 659.6$$

$$\therefore p = \frac{MRT}{V} = \frac{10 \times 53.3 \times 659.6}{120} = 2950 \text{ lb per sq. ft.} \quad \textit{Ans.}$$

6. How many lbs. of air does it take to fill 5600 cu. ft. at 15 lbs. per sq. in. pressure and 60°F.?

$$pV = MRT$$

$$V = 5600 \quad p = 15 \times 144 \quad R = 53.3 \quad T = 459.6 + 60 = 519.6$$

$$\therefore M = \frac{15 \times 144 \times 5600}{53.3 \times 519.6} = 437 \text{ lbs.} \quad \textit{Ans.}$$

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CHAPTER VI

WATER AND STEAM

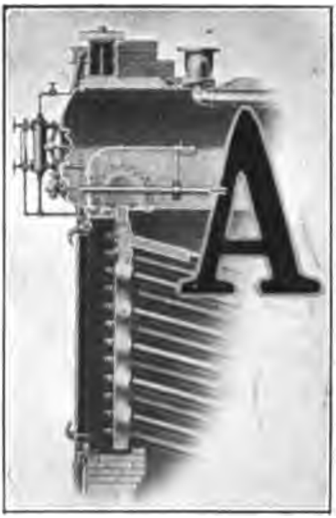

FIG. 34.—Water and steam space in water tube boiler.

AS we look about us in nature, we find that all inanimate creation presents itself to us in three distinct physical states. Certain bodies, for instance, of themselves readily maintain their shape while others, although non variant in density, nevertheless seem to have no particular physical configuration but seek, due to the force of gravitation, the lowest level attainable and consequently must as a rule be held in a containing vessel. On the other hand, a third class of bodies is found not only possessing no particular physical configuration, but which actually seem inherently desirous of expanding to such an extent that they must as a rule be completely housed, bottom and top, in a containing vessel.

In the class room or in the power plant, it is easy to find illustrations of these three general classifications. Thus, chalk, iron pipe, and coal are instances of the first division and are known as solids. Crude petroleum, water, and kerosene are instances of the second division, and are called liquids. Finally, air, steam, and producer gas illustrate the third division, and are called gases.

These States are Possible to all Bodies.—The most interesting thing about these socalled states of matter, and indeed the item of most importance to the engineer, is that by varying the pressure externally forcing itself against the sides of any one of these bodies and by adding or subtracting the heat that may be held in store within the body itself, any solid may be converted into a liquid and then into a gas, or any liquid may be converted

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into a solid or a gas, or any gas may be converted into a liquid then into a solid.

The Fundamental Principle in Steam Engineering.—It is this property of matter that makes the operation of the steam engine possible. For if we were not able to heat water and convert it into steam, it would be impossible to make use of this liquid for steam engineering purposes, although it is the most widely distributed in nature.

Again, since fuel oil must be converted into the gaseous state before it readily and efficiently burns beneath the boiler, it would certainly be cumbersome and impractical for its use in the great majority of central stations if it could not be conveyed through pipes or in oil tanks as a liquid from the oil fields to the place of consumption.

Steam Engineering Still Supreme.—Since water is so widely disseminated in nature and since it can be readily and efficiently changed from one state to another, it is the working substance that today still drives the vast majority of power developing mechanisms in the industries in spite of the rise of the gas engine and the great modern evolution in water power development.

Let us then trace the physical phenomena that accompany the transformation of water into the solid state which of course is necessary in the production of ice, and again from the liquid to the gaseous state which becomes necessary in the production of steam.

The Formation of Ice.—Let us first start with a pound of water at ordinary temperatures—say at 62°F. As we begin to lower the temperature, in other words to draw off heat, the volume slightly decreases. Thus the pound of water now occupies less space than formerly. Hence, if this water was on the surface of a mountain lake and the night was getting cooler, the surface water would sink to the lake bottom and allow warmer water from the bottom to rise only to be cooled at the surface to again drop to the bottom. This is what is known as water circulation and is very important in steam generation, as we shall see later.

When, however, the water under consideration lowers to a temperature of 39.4°F., a strange thing happens. Something develops in its internal structure that now makes the water expand as the temperature is further lowered. A unit volume of water now becoming lighter than formerly, no longer will it

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sink to the lake bottom but remains on the surface. Hence when a short time later the water on the surface is lowered to 32°F. or freezing point, ice is formed on the surface only, since water is a poor conductor of heat. Nature thus protects the fish in the waters below.

Coming back from the mountain lake, however, to the formation of ice in the ice plant when the temperature has reached 32°F., although heat be now driven off, the water does not lower itself in temperature but remains at this temperature until it has all been converted into ice.

Latent Heat of Fusion.—The quantity of heat necessary to form one pound of ice at 32°F. from one pound of water at 32°F. is a definite measurable quantity and is known as the latent heat of fusion. By careful measurement, the latent heat of fusion for water has been found to be 142 B.t.u. That is, to convert one pound of water at 32°F. into ice at 32°F. requires the drawing off of as much heat as would approximately be required to lower one pound of water one hundred forty-two degrees in temperature.

When this pound of water is converted into ice, its volume still further expands. Hence, one pound of ice will float in water. This accounts, of course, for the floating of icebergs on the water surface, and furthermore this sudden increase in volume accounts for the rutpure in pipes and other nuisances that occur in severely cold weather.

Going back to our pound of water now converted into a pound of ice, let us again proceed to draw off heat. It is now found that we may lower the temperature of the ice much more easily than when it existed as a liquid. Indeed only about one-half the heat is required to be drawn off per degree lowering in temperature while its volume practically remains constant.

The Formation of Steam.—Let us now proceed to a consideration of the physical changes and phenomena that occur when water passes into steam. Starting with water at say 62°F., as we add heat the temperature increases at the rate of about 1°F. for every unit of heat energy added to the water. At the same time the volume slightly increases. Hence, if our pound of water under consideration be situated at the bottom of the well-known tea-kettle, the observation of which led James Watt to the invention of the steam engine, this pound of water becoming now less dense will rise to the top and cooler water at the top will sink to the bottom which in turn is passed again to the top as it becomes

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heated to make way for more water from the top to be heated along the portion exposed to the heat application. Thus the water becomes warmer and warmer and the transference from bottom to top continues. The ease with which this transfer of heated bodies of water takes place has much to do with efficient operation of the steam boiler which may be likened to an enlarged

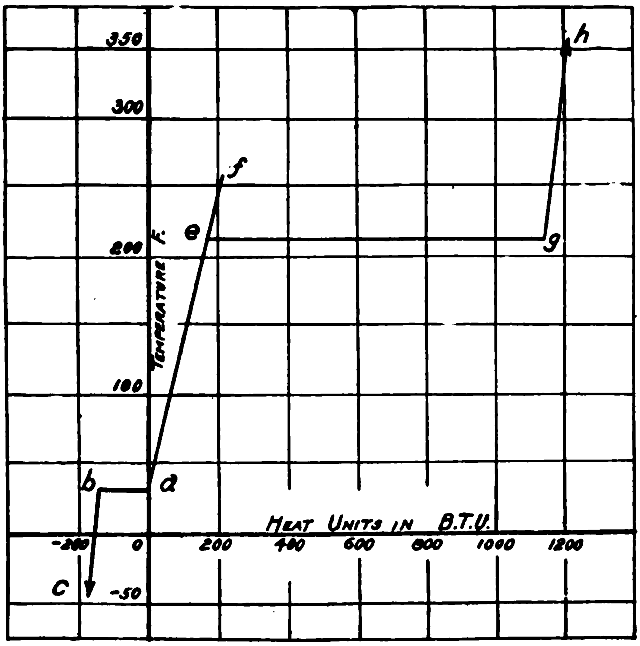

FIG. 35.—The temperature heat diagram.

Here is graphically indicated the history of a pound of water in its relationship with temperature and heat. Beginning at 32°F. and atmospheric pressure, by drawing off heat the horizontal line *ab* is traced, showing that the temperature remains constant until the water is completely converted into ice, after which the temperature rapidly falls at the rate of about one degree for every half unit of heat drawn away. By the addition of heat, however, at point *a*, the curve *ae* is traced, which indicates that the temperature rises with absorption of heat at the approximate rate of one degree for each unit of heat absorbed. At 212°F. and atmospheric pressure the horizontal line *eg* is traced until 970.4 B.t.u. are absorbed. After all the water is thus converted into steam the curve *gh* is traced for superheated steam, which rises at the rate of about one degree for every .47 of a B.t.u. absorbed.

tea-kettle with accessories and appurtenances to care for its increased responsibilities as compared to tea-kettle operation.

Latent Heat of Evaporation.—The water in this manner continues to absorb heat until if under atmospheric pressure, it reaches a temperature of 212°F. At this point, however, vast quantities of heat may be added and still the water will remain at this temperature although it may now be observed that steam

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is being formed which too, has the same temperature as the water. Not until 970.4 B.t.u. or sufficient heat units to raise ten pounds of water almost one hundred degrees in temperature have been added to the pound of water at 212°F. will the pound of water become entirely converted into steam at 212°F. This quantity of heat necessary is important in steam engineering and is known as the latent heat of evaporation for water under atmospheric pressure conditions. To be succinct, in steam engineering practice the quantity of heat necessary to convert one pound of water at a given temperature and pressure into dry steam at the same temperature and pressure is known as the latent heat of evaporation for that temperature and pressure and is usually expressed by the symbol L_t. Steam boilers seldom operate at a pressure so low as that of atmospheric conditions. Indeed, while such a pressure is but 14.7 lb. per sq. in., the modern boiler in the central station operates at something like ten to fifteen times this pressure. This fact materially complicates computation in steam engineering, for it is found that at pressures different than that of standard atmospheric conditions the latent heat of evaporation is wholly different. Indeed, so complex is this law of variation that no one as yet has been able to give an exact formula for its determination, although in subsequent chapters approximate equations will be set forth. Hence, it has become necessary to refer to carefully compiled steam tables for such information and a later chapter will set forth the manner of their use.

Other Variations Occur With Changes of Pressure.—When water passes into steam, the volume—say of one pound—vastly increases. At atmospheric pressure the volume of steam is about sixteen hundred times what it was when existing as water. At other pressures the volume relationships will of course be different. Again when the pressure increases at which steam is formed, the volume becomes less in proportion. No accurate mathematical formula has been found for this relationship hence once again must we appeal to the steam tables.

Data Easily Taken from Steam Tables.—By experiment it has been found that varying amounts of heat are required to raise water from a particular initial temperature to the boiling point, for the boiling point is not reached until a higher temperature is attained as the pressure is increased. On the other hand, less heat is required to convert a pound of water at these higher

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boiling points into steam. Since the volume and density too vary under varying pressures, the entire problem now becomes one of picking the proper constants for the particular temperature and pressure under discussion and when one by a little practice can use the steam tables with facility, it is surprising to see how simply and directly most problems in steam computation may be solved.

Total Heat of Steam.—Often in steam engineering practice problems arise in which we must express the total heat of steam quantitatively represented in each pound under consideration. It makes little difference at what point we begin to estimate such heat relationships, but by common consent the freezing point of water has been adopted. Hence, the total heat of steam is the heat required to raise one pound of water from 32°F. to the boiling point added to the heat required to convert this water into steam at that temperature. If the steam exist as superheated steam, there must also be added the heat required to raise dry saturated steam to the temperature of superheat. The various mathematical formulas for computing these numerical results will be taken up later in a chapter entitled Quality of Steam.

At this particular time we shall write down the simplest of these formulas as an illustration.

Total Heat of Dry Saturated Steam.—The total heat of dry saturated steam, written H_t for a given temperature t, is the sum of the heat of liquid and latent heat of evaporization for that temperature.

Hence we may write this important fundamental equation

$$H_t = h_t + L_t \tag{1}$$

Thus the total heat of steam at 212°F. is

$$H_t = 180 + 970.4 = 1150.4 \text{ B.t.u.}$$

Other Instances of Total Heats.—If, however, the steam is evaporated from the water and then superheated, that is, an additional quantity of heat is added after all the water has become steam, it will then begin to rise in temperature and the quantity of heat necessary for each degree rise in temperature is about one-half that required per degree rise when it existed as water. This exact ratio is however quite variable and ranges between .46 and .60 depending upon the pressure and degree of superheat attained. Hence once again appears the necessity of steam tables.

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It is now readily seen that in general three definite and distinct considerations present themselves in the solution of problems involving the computation of total heat. The first instance is one in which the steam exists in a dry state and at the temperature and pressure at which it is generated from the water. Such steam is known as dry saturated steam. The second instance is that in which the steam is not completely dry, but holds in suspension small particles or globules of water, and in this instance the mixture is known as wet steam. The third instance is of especial importance in modern central station practice and involves what is known as superheated steam. In this case the steam is first formed by evaporation from water into dry saturated steam, after which it is conveyed through pipes that are exposed to high temperatures, thus causing the temperature of the steam to be still further raised, although the pressure practically remains constant.

The complete solution of these three instances for computation of total heats will be found in the chapter on Quality of Steam as stated above. Meanwhile the thorough mastery of the fundamentals of the physical properties of water as herein set forth will be of vast assistance in a clear understanding of this later discussion.

Examples

1. The water entering a feed-water heater is at a temperature of 75°F. and leaves the heater at 190°F., what is the heat absorbed per lb. of water?

From the steam tables the heat of liquid at 75°F. is 43.05 B.t.u. and at 190°F. it is 157.91 B.t.u. Hence the heat absorbed per lb. of water is

$$157.91 - 43.05 = 114.86 \text{ B.t.u.}\text{—}\textit{Ans.}$$

2. Water enters a boiler at 160°F. and is converted into dry saturated steam at 200 lb. pres. per sq. in. abs., what is the total heat required to evaporate each lb. of steam?

The heat in the entering water at 160°F. is from the steam tables 127.86 B.t.u. The total heat of dry saturated steam at 200 lb. pres. abs., is 1198.1 B.t.u. Hence the actual heat necessary to evaporate each lb. of steam is

$$1198.10 - 127.86 = 1070.24 \text{ B.t.u.}\text{—}\textit{Ans.}$$

3. If the heat of liquid for boiling water at 212°F. is 180 B.t.u. and the latent heat of evaporation is 970.4 B.t.u., how much heat is required to evaporate a pound of water from an open water heater which is receiving its supply at 64°F.?

Each lb. of water entering at 64°F. has a heat of liquid of 32.07 B.t.u. Water evaporating into steam at 212°F. has a heat of liquid of 180 and a latent heat of evaporation of 970.4 B.t.u., making a total heat of evaporation of 1150.4 B.t.u. for every lb. of water so evaporated. Hence the net heat required is

$$1150.40 - 32.07 = 1128.33 \text{ B.t.u.}\text{—}\textit{Ans.}$$

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CHAPTER VII

THE STEAM TABLES

Fig. 36.—The book of steam tables.

IT has already been shown that since no simple mathematical laws have as yet been devised to express the temperature, pressure, latent heat, heat of liquid and other fundamental properties of steam and water that are absolutely necessary in the solution of steam engineering problems, we must resort to carefully compiled steam tables.

Practically all the research and scientific investigation along the lines of pure steam engineering of the last half century have been devoted to the more complete establishment of some of the fundamental constants involved in the steam tables.

The three most important of these are the zero point of the absolute temperature scale, the proper value for a constant employed in the conversion of mechanical energy into heat energy, and the exact determination of the heat required to evaporate one pound of water from 212°F. into dry saturated steam at 212°F.

Since these values are continually found by more careful and exacting experimental work to be slightly different than formerly held, we find that the steam tables of recent publication are different than those of former years.

The Steam Tables as Adopted in this Discussion.—The Steam Tables and Diagrams as computed by Marks and Davis and published by Longmans, Green & Company, are today universally recognized and are adopted as the standard compilation for the problems cited in this discussion.

In the rear of these steam tables an interesting discussion of the methods employed by these investigators in arriving at the

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three fundamental constants mentioned above is given. The result of these investigations shows that the absolute zero is to be taken at −459.6°F., the mechanical equivalent of heat at 777.5, and the latent heat of steam at 212°F. to be 970.4 B.t.u.

Recapitulation of Fundamental Evaluations.—These three constants are so important that they should be memorized and for emphasis let us recapitulate their exact interpretation.

The absolute zero is now found to be a point situated at 459.6° F. below the zero point on the Fahrenheit scale or 491.6°F. below the freezing point of water. At such a temperature it is supposed to be impossible to further draw off heat from any substance for at this temperature the heat storage is supposed to be absolutely exhausted.

The mechanical equivalent of heat as given above means that the energy represented by one B.t.u. or British thermal unit of heat is equivalent to 777.5 ft. lb. of mechanical energy. Or if one pound of crude petroleum contains 18,500 B.t.u., it possesses as stated in a previous chapter, sufficient energy to raise a human being weighing 175 lb. a vertical upward distance of over 18 miles.

The latent heat of steam at 212°F. and atmospheric pressure means that the quantity of heat necessary to evaporate one pound of water at 212°F. into dry saturated steam at 212°F. is found experimentally to be 970.4 B.t.u.

Analysis of a Typical Page of Steam Tables.—Let us now proceed to analyze a page of Marks & Davis' steam tables, column by column. The illustration as given is found on page 12 of this compilation and we shall follow across the page the line corresponding to a temperature of 231°F.

Temperatures in Fahrenheit Units.—Since all steam engineering computation is based on temperatures represented in the Fahrenheit scale instead of the Centigrade system, the temperatures are here listed in the Fahrenheit units.

Pressures in Absolute Notation.—This column means that the pressures here given represent the pressure in pounds per sq. in. at which water will boil when the temperature is that as listed in the first column. Further on in the steam tables an exactly similar table may be found to the one cited except in this latter instance the pressures are made to vary pound by pound and the corresponding boiling temperature of water given.

In this instance, then, we read that a pressure of 21.16 lb.

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per sq. in. will be produced before the water boils or the formation of steam begins at 231°F. This pressure, by the way, is in absolute units and would not be the pressure read on the steam gage of a boiler room. Since the steam gage indicates pressures above

Table 1: Temperatures

Temp. Fahr.	Pressure lbs.	Pressure Atmos*	Sp. Vol. cu. ft. per lb.	Density lbs. per cu. ft.	Heat of the liquid	Latent heat of evap.	Total heat of steam	Internal Energy B.t.u. Evap.	Internal Energy B.t.u. Steam	Entropy Water	Entropy Evap.	Entropy Steam	Temp. Fahr.
t	p	—	v or s	1/v	h or q	L or r	H	I or ρ	E	n or θ	L/T or r/T	N or φ	t
230°	20.77	1.413	19.39	0.0516	198.2	958.7	1156.9	884.3	1082.4	0.3384	1.3905	1.7289	230°
231	21.16	1.440	19.05	0.0525	199.2	958.1	1157.2	883.6	1082.7	0.3399	1.3875	1.7274	231
232	21.56	1.467	18.72	0.0534	200.2	957.4	1157.6	882.8	1083.0	0.3414	1.3844	1.7258	232
233	21.96	1.494	18.40	0.0543	201.2	956.7	1158.0	882.1	1083.2	0.3429	1.3814	1.7243	233
234	22.37	1.522	18.09	0.0553	202.2	956.1	1158.3	881.3	1083.5	0.3443	1.3784	1.7227	234
235°	22.79	1.550	17.78	0.0562	203.2	955.4	1158.7	880.6	1083.8	0.3458	1.3754	1.7212	235°
236	23.21	1.579	17.47	0.0572	204.2	954.8	1159.0	879.8	1084.0	0.3472	1.3725	1.7197	236
237	23.64	1.609	17.17	0.0582	205.3	954.1	1159.4	879.1	1084.3	0.3487	1.3695	1.7182	237
238	24.08	1.638	16.88	0.0592	206.3	953.4	1159.7	878.3	1084.5	0.3501	1.3666	1.7167	238
239	24.52	1.668	16.60	0.0602	207.3	952.8	1160.0	877.6	1084.8	0.3516	1.3636	1.7152	239
240°	24.97	1.699	16.32	0.0613	208.3	952.1	1160.4	876.8	1085.0	0.3531	1.3607	1.7138	240°
241	25.42	1.730	16.05	0.0623	209.3	951.4	1160.7	876.1	1085.3	0.3546	1.3578	1.7124	241
242	25.88	1.761	15.78	0.0634	210.3	950.7	1161.1	875.3	1085.6	0.3560	1.3550	1.7110	242
243	26.35	1.793	15.52	0.0644	211.4	950.1	1161.4	874.6	1085.8	0.3575	1.3521	1.7096	243
244	26.83	1.826	15.26	0.0655	212.4	949.4	1161.8	873.8	1086.1	0.3589	1.3493	1.7082	244

Fig. 37.—A typical page from the steam tables.

the atmosphere, one must subtract from this reading in the steam tables the atmospheric pressure of the day in order to find the proper gage pressure. Thus, in this instance, if the atmospheric

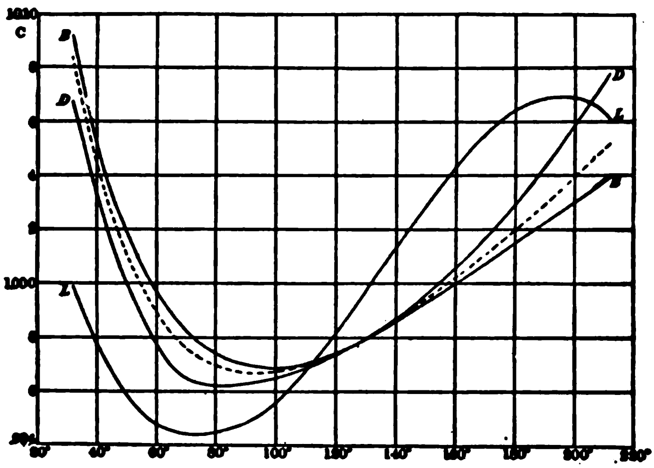

Fig. 38.—Marks & Davis method of collating data for specific heat of water from three noted investigators.

pressure of the day be 14.7 lb. per sq. in., a steam gage in a boiler room would read 6.46 lb. per sq. in., when the water in the boiler is 231°F.

This precaution is most important and the student should carefully reread the former chapter on pressures if he does not thoroughly understand the conversion of gage pressures, inches of vacuum, inches of mercury, etc., into standard absolute pressure units.

Pressures in Atmospheres.—In many engineering computations pressures are given as so many atmospheres instead of pounds per square inch. The pressure of the standard atmosphere is usually taken as 14.7 lb. per sq. in. but for very exact work it is more accurately 14.696 lb. per sq. in. Hence this column is computed by dividing each item in the preceding column by 14.696, which in this instance is found to be 1.440 atmospheres.

When, however, the reading is below that of ordinary atmospheric pressure, such values are often desired in inches of mercury since vacuum pressures for the condenser are given in such units. This particular column is therefore found by dividing the corresponding line in the preceding pressure column by the number of inches of mercury equivalent to one pound pressure per square inch. It is to be remembered that this does not even yet give the reading in inches of vacuum. Pressures in absolute inches of mercury and inches of vacuum cause seemingly endless confusion. A complete discussion of this feature was taken up under the chapter on pressures and its careful review is emphatically recommended if any unsettled question still exists in the mind of the reader.

Specific Volume.—The cubic feet occupied by one pound of dry saturated steam at a given temperature and pressure is known as the specific volume of the steam for that temperature and pressure.

This is a factor often necessary in steam engineering computations. Yet no known means has ever been invented whereby this factor can be accurately ascertained by experiment. The task is indeed one that involves such difficulties as to make its determination by experiment practically impossible. The science of higher mathematics has come to the rescue and here is indeed an instance where purely theoretical deductions have brought about a practical solution of an otherwise unsolvable problem in steam engineering.

This relationship involves the latent heat of evaporation L; the absolute temperature T at which the saturated steam is formed; the ratio of the increase in pressure Δp to the increase in

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temperature Δt of boiling points taken immediately below the temperature under consideration and immediately above it; the specific volume of the steam v that is found, which of course, is the unknown value we are desirous of computing; and the specific volume of a space occupied by one pound of water v_1 immediately before its conversion into steam. Algebraically the relationship is expressed thus:

$$L = T\left(\frac{\Delta p}{\Delta t}\right)(v - v_1) \qquad (1)$$

From the steam tables we will take our values for Δp and Δt immediately below corresponding to 230°F. and immediately above corresponding to 232°F. Hence

$$\begin{aligned}
\Delta t &= (232 - 230) = 2.\\
\Delta p &= (21.56 - 20.77)144 = 0.79 \times 144 = 114.\\
T &= 231 + 459.6 = 690.6.\\
L &= 958.1 \times 777.5.\\
v_1 &= .016 \text{ cu. ft.}
\end{aligned}$$

Substituting, we have

$$958.1 \times 777.5 = 690.6\left(\frac{114}{2}\right)(v - .016)$$

$$\therefore v = 18.98$$

The value in the table is 19.05 which is seen to be about one-third of one per cent. in error. This difference is probably due to the fact that decimals neglected in computation were made use of by the comp ler of the steam tables, and then too the small pressure and temperature variations were probably taken nearer together than is possible in the data actually set forth in the steam tables.

Specific Density.—The weight in fractions of a pound of one cubic foot of dry saturated steam is known as its specific density.

It is evident that if one pound of steam occupies 19.05 cu. ft. as taken from the previous column, then 1 cu. ft. of steam would weight 1/19.05 of a pound which is 0.0525 lb. Hence this column is computed in each case by taking the reciprocal of the data given in the preceding column.

The Heat of Liquid.—This is one of the most important columns necessary in steam engineering practice. Since the heat of liquid technically means the quantity of heat necessary to raise one pound of water from 32°F. to the temperature under

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consideration, it is evident that by experimental data as given in this column it has been found that to raise one pound of water from 32°F. to 231°F., 199.1 B.t.u. are necessary to be applied from an outside source.

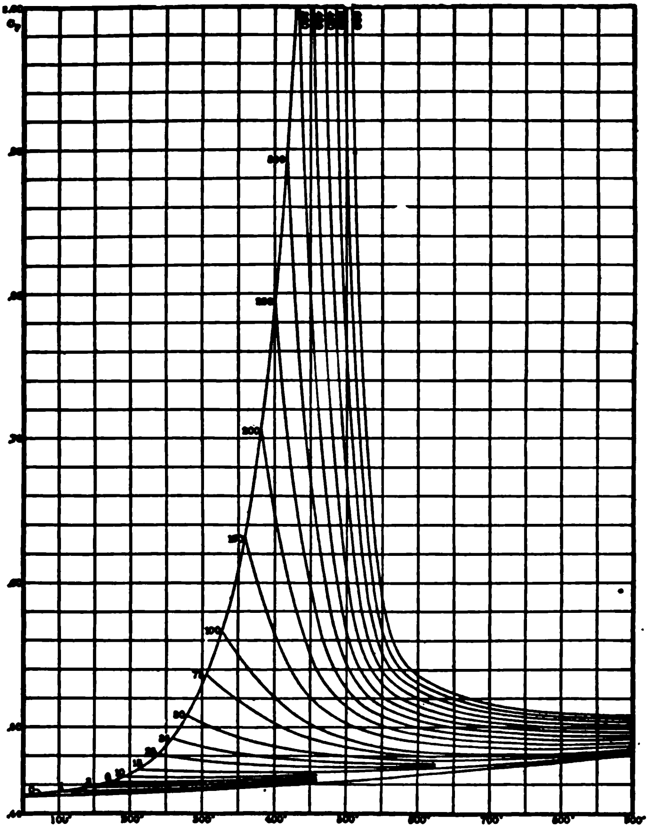

FIG. 39.—Determination of the specific heat of superheated steam from investigations of Knoblauch.

The Latent Heat of Evaporation.—Data for the latent heat of evaporation has been determined by careful experimental means. It is by definition the quantity of heat necessary to convert one pound of water at the temperature and pressure indicated into dry saturated steam at the same temperature and pressure. In this instance it is seen that to convert one pound of water at 231°F. into dry saturated steam at 231°F., 958.1 B.t.u. are necessary to be applied from an outside source.

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Total Heat of Dry Saturated Steam.—The total quantity of heat required to raise the temperature of one pound of water at 32°F. to the temperature at which dry saturated steam may exist under the pressure exerted in the particular instance, added to the quantity of heat then necessary to convert this water completely into dry saturated steam is known as the total heat of dry saturated steam. Numerically speaking, it is seen that this column is at once obtained by adding the heat of liquid and the latent heat of evaporation. In a word, this column is the sum of the two preceding columns. Thus

$$H_{231} = h_{231} + L_{231} \tag{2}$$

$$\therefore H_{231} = 199.1 + 958.1 = 1157.2.$$

Internal and External Work.—One wonders where the heat disappears when it is being continually applied to water at the boiling point and yet the temperature of the water or steam does not increase.

Upon careful investigation it is found that it disappears first in an internal absorption due to intermolecular rearrangement as water passes into steam which thereby stores up a considerable quantity of energy to be given out again when the steam is condensed back into water. The energy that disappears in this manner is known as energy necessary to perform internal work.

On the other hand in the generation of steam from water the volume is vastly increased. The pushing back against external pressure to make room for such an increased volume performs external work. So that the energy applied in steam generation which goes toward latent heat of evaporation may be divided into two classifications, known as external and internal work.

No one has as yet found a method of directly measuring internal work. We may, however, measure external work or even compute it and then by subtraction from total energy absorbed arrive at a value for internal work.

In a former chapter on gases it was shown that the external work accomplished by a gas expanding under constant temperature and pressure is computed universally by subtracting the initial volume from the final volume and then multiplying this result by the pressure. Thus

$$\text{External Work} = p\,(v - v_1)$$

To convert this into B.t.u., we have

$$\text{External Work} = \frac{p\,(v - v_1)}{777.5} \tag{3}$$

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From the tables it is seen that in this instance $p = 21.16 \times 144$, $v = 19.05, v_1 = .016$.

$$\therefore \text{External Work} = 21.16 \times 144\ (19.05 - .016) = 74.6 \text{ B.t.u.}$$
$$\therefore \text{Internal Work} = 958.1 - 74.6 = 883.5 \text{ B.t.u.}$$

Entropy of Water.—In certain advanced problems in steam engineering, engineers and physicists have found it convenient to invent fictitious qualitites of steam. While many have endeavored to give a physical interpretation of entropy, perhaps it is clearer for the student to consider it as merely a mathematical fiction which, however, often becomes extremely useful for the representation of steam engineering problems and indeed assists wonderfully in their solution.

On this assumption, entropy may be defined as such a quantity that when plotted against absolute temperatures the area under the curve connecting all such points will numerically represent the amount of heat supplied to one pound of matter in order to accomplish the indicated change in temperature. Thus in the instance at hand if one should plot a curve with ordinates representing absolute temperatures and with abscissas representing the entropy for each corresponding temperature, the area under this curve would be exactly 199.1 units. For it takes 199.1 units of heat energy to raise one pound of water from 32°F. to 231°F .or on the absolute scale from 491.6°F. to 690.6°F.

By analysis in higher mathematics it is found that entropy of water may be quite closely computed by the formula

$$\theta = \log_e \frac{T_2}{T_1} \tag{4}$$

Wherein θ is the entropy of water, T_2 the absolute temperature at the end of the heat application and T_1, the absolute temperature at the beginning which is usually taken at the melting point of ice or 491.6°F. on the absolute scale. Thus in this instance

$$\theta = \log_e \frac{T_2}{T_1} = \log_e \left(\frac{231 + 459.6}{32 + 459.6}\right)$$
$$= 2.306 \log_{10} \frac{(690.6)}{(491.6)} = .3399.$$

The values in the steam tables were arrived at by a slightly more accurate process than this by taking into account the fact that the specific heat of water is not constant as heat is added.

The Entropy of Evaporation.—Since the temperature remains constant during the evaporation of water into dry saturated

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steam, it is evident that the entropy curve in this case would simply be a rectangle as shown in the illustration wherein one dimension is of length T and the area swept off is of L units. Hence, the entropy for heat of evaporation is evidently

$$\text{Entropy of evaporation} = \frac{L}{T} \tag{5}$$

or in this instance,

$$\text{Entropy of evaporation} = \frac{958.1}{231 + 459.6} = 1.3875$$

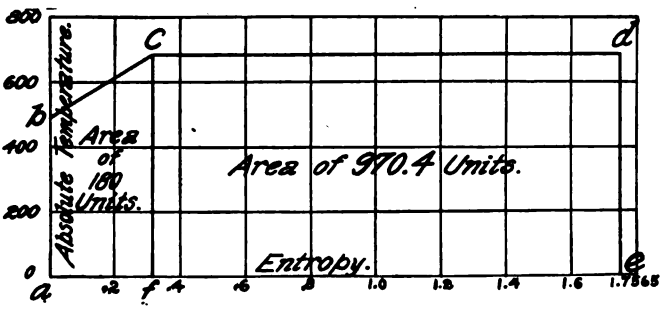

FIG. 40.—The temperature entropy diagram.

By the invention of a fictitious quality of water and steam, known as entropy, the plotting of a diagram is made possible, so that an area represents heat added. Thus, in the diagram above, the abscissas are entropy and the ordinates absolute temperatures. The area *abcf* is exactly 180 units, which is the heat required to raise water from 32°F. to 212°F. Similarly, the area *fcde* is 970.4 units, which is the heat required to evaporate one pound of water at 212°F. into steam at 212°F.

Total Entropy.—The sum of the entropy value for water and for heat of evaporation is called the total entropy of dry saturated steam. This is evidently arrived at numerically by adding together the two preceding columns. Thus, total entropy = entropy of water + entropy of evaporation

$$\therefore \text{Total entropy} = 0.3399 + 1.3875 = 1.7274 \tag{6}$$

Tables for Superheated Steam.—In later pages of the steam tables are to be found data relative to superheated steam. As a subsequent chapter will deal largely with superheated steam computations, we shall delay the consideration of superheated steam tables until the reader has been more thoroughly grounded in other fundamental computations of dry saturated steam.

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CHAPTER VIII

HOW TO COMPUTE BOILER HORSEPOWER

FIG. 41.—How James Watt would have standardized a mechanical horsepower at the Panama-Pacific Exposition.

THAT energy is never created or destroyed is a fundamental postulate of modern engineering practice. All of our machines and driving mechanisms are, then, simply devices by means of which we may convert one form of energy into another form to suit our convenience or meet the demands of industrial activity. Thus an electric generator does not create energy but is merely a device whereby energy existing in the waterfall or in the steam turbine may be converted into electrical energy. Neither does the energy exist inherently in the waterfall, but due to the emission of heat from the sun, this water has first been drawn from the ocean into the clouds to be later deposited on the lofty mountain peaks. Due to this superior position it is enabled to develop water power energy and thus transfer the energy of the sun's rays into more useful form to ease man's burdens. And so with the steam boiler, we have fundamentally a mechanism by which energy latent in fuel oil or other combustible is first given out as heat energy of combustion to be immediately converted into latent heat energy of steam.

The Meaning of the Word "Rating."—The rapidity with which this conversion of one form of energy into another form may be accomplished is known as the rating of the mechanism involved. Thus a small boy may by means of a block and tackle hoist a huge weight to the top of a modern sky-scraper and at a later observation one may see a team of horses straining to their

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utmost to accomplish the same task. By close inspection, however, it will be found that the small boy has by means of intervening pulleys been able to take from thirty to forty times longer to accomplish what the horses did in a comparatively short time. Hence power, the basis of comparative effort, is the time rate of doing work.

The Development of the Word "Horsepower."—After his invention of the steam engine, James Watt soon found that

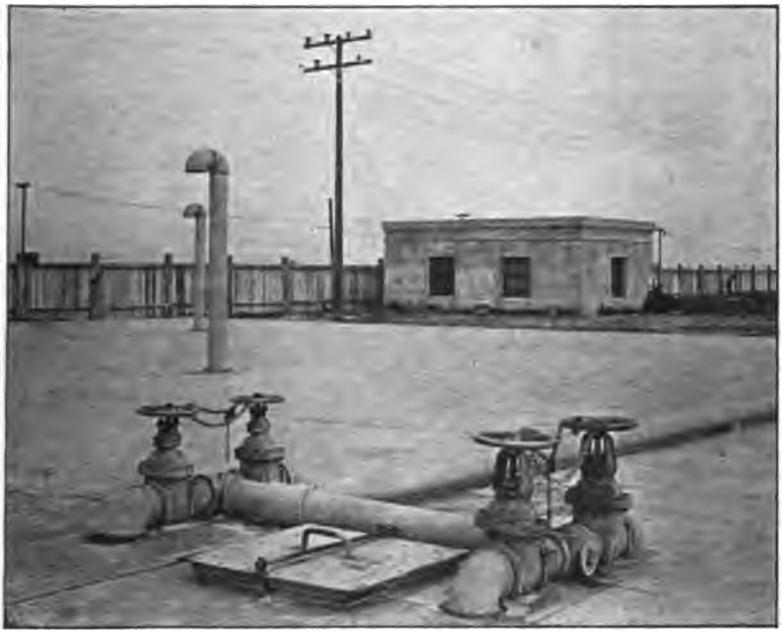

FIG. 42.—A close up view of the filling pipes for the oil storage reservoirs of the Southern California Edison Company's Long Beach Plant. These valves are under control of the oil company from whom the oil is purchased.

he must devise some unit or measuring stick, as it were, with which to measure the power of his mechanism. As he was a pioneer in the art, he had to cast about for some convenient unit to adopt. What more natural unit should he consider than that of the draft horse? After watching a horse drawing up large cakes of ice into an ice house by the use of a snatch block, it occurred to him that when the horse pulled up a fairly good load he must be doing a certain amount of work. After making several experiments he found that by adding more sheaves to

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the blocks the horse could raise a greater load but it took more time to do it. He found that the average dray horse was able to raise a load of 550 lbs. at the rate of 60 ft. per minute, or to do 33,000 ft. lbs. of work per minute. This unit Watt called a horsepower and applied it to the measurement of the power of his steam engines.

The Boiler Horsepower.—In the early days of the steam engine the principle of the conservation of energy had not been firmly established. Indeed that heat was a form of energy at all was a debated question for many years after the steam engine became of vast practical importance.

FIG. 43.—Steam flow meter, recording pressure gage and indicating pressure gage, Station C, Pacific Gas and Electric Company, Oakland, California.

Hence, since the energy latent in steam was not then known to be the underlying reason for the power driving action of the steam engine, the first rating of the boiler was made on the basis of power development in the engine which received its supply of steam from the boiler in question. Thus a boiler that could supply steam to operate a steam engine developing 50 indicated h.p. was said to be a 50 h.p. boiler. Later it became evident, due to the rapidly increasing efficiencies of the steam engine that such a rating was wholly variable. It was found, however, that under ordinary working conditions a boiler which could evaporate

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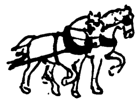
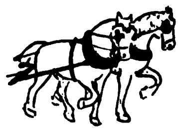


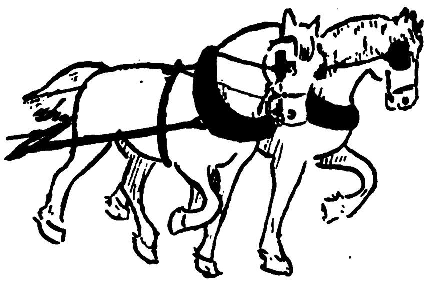
THE MECHANICAL HORSEPOWER

FIG. 44.—The unit of power in modern steam engineering.



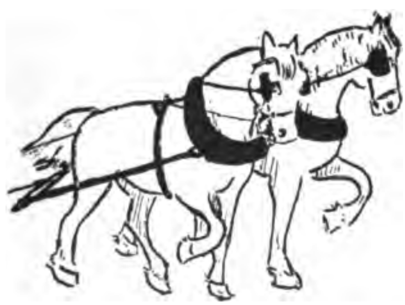
THE KILOWATT

FIG. 45.—The unit of power in electrical engineering, which is 1.34 times the mechanical horsepower.



THE BOILER HORSEPOWER

FIG. 46.—The unit of power in boiler practice, which is 13.14 times the mechanical horsepower.



THE MYRIAWATT

FIG. 47.—The unit of boiler rating proposed by certain national engineering societies, which is 13.4 times the mechanical horsepower.

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30 lb. of steam per hr. at 70 lb. pressure and taking feed water at 100°F. could usually operate a 1 h.p. engine, consequently this mode of boiler rating became popular.

In 1884, the American Society of Mechanical Engineers adopted the following definition for the boiler h.p.: That a boiler evaporating 34.5 lb. of water at 212°F. into steam at 212°F. per hr. should be known as a 1 h.p. boiler.

The Conversion of Boiler Horsepower to Mechanical Horsepower Units.—In later years the principle of the conservation of energy finally became well established and when engineers began to compute the actual energy represented in a mechanical horsepower as originally adopted by James Watt and then compare this to the energy represented in the steam generated by what was known as a one horsepower boiler, it was found that the boiler horsepower represented the conversion in unit time of over thirteen times the energy represented in the mechanical horsepower unit acting over the same unit of time.

It is instructive to follow this computation as it will familiarize the reader with these two distinct units. Let us then proceed to an analysis. The mechanical horsepower unit is defined as a performance of work or conversion of energy at the rate of 33,000 ft. lb. per minute. Since 1 B.t.u. of energy has been found to have its equivalent in 777.5 ft. lbs .of mechanical work, it is seen that 33,000 ft. lb. of work per minute, or 1,980,000 ft. lb. of work per hr. may be represented by 2547 B.t.u. per hr. From the definition of the boiler horsepower above mentioned, as that adopted by the American Society of Mechanical Engineers, it is seen that since it requires 970.4 B.t.u. to evaporate 1 lb. of water at 212°F. into steam at 212°F., one boiler horsepower represents 34.5 × 970.4 B.t.u. per hr. or 33,479 B.t.u. of heat energy per hr. Hence, when we compare the boiler horsepower with the ordinary horsepower it is seen that the boiler horsepower represents a unit which is 13.14 times larger than the ordinary horsepower.

The Myriawatt as a Basis of Boiler Performance.—In recent years, due to the tremendous growth in the electrical industry, engineers have recognized the inconsistencies of the boiler horsepower unit and an effort has been made by the national engineering societies to make a more rational standard of rating. As a consequence, the American Institute of Electrical Engineers has proposed that the Myriawatt be adopted as a standard of boiler

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rating instead of the Bl. h.p. A Myriawatt is the power equivalent of 10,000 watts or 10 kw. which converted into heat units become 34,150 B.t.u. per hr. Although it is still to be remembered that the Myriawatt does not yet make output and input of electrical units expressible in like quantities, since output is usually expressed in kilowatts, still the factor of 10 furnishes a basis readily convertible and makes possible a change in units without materially upsetting the old boiler h.p. range of capacity.

If, then, a boiler evaporates M pounds of steam per hour and the total heat of each pound of steam so evaporated be H and the heat of liquid represented in the feed water be h_f, then the rating of a boiler in Myriawatts is evidently

$$\text{Myriawatts} = \frac{M(H - h_f)}{34{,}150} \tag{1}$$

Relationship of Boiler Horsepower and Myriawatts.—Similarly, since one boiler horsepower is equivalent to heat absorption of 33,479 B.t.u. per hour and a myriawatt to 34,150 B.t.u. per hour, then we may convert a rating in Myriawatts to a rating in boiler horsepower or vice versa by the relationship:

$$\frac{\text{Rating in boiler horsepower}}{\text{Rating in Myriawatts}} = \frac{34{,}150}{33{,}479} \tag{2}$$

The Builder's Rating.—In the commercial evolution of the steam boiler there has grown up a method of rating boilers by "rule of thumb" process. It is evident that the area of the steam generating surface of the boiler actually exposed to the heated gases of the furnace has something to do with the capacity of the boiler. For different designs of boilers, however, the particular factor to be applied varies widely. It has become of common acceptance, however, that 10 sq. ft. of boiler surface exposed to the furnace heat shall be considered on this rule of thumb comparison as equivalent to one boiler horsepower. Hence to compute the builder's rating of a boiler we must compute the area in square feet of the surface exposed to the furnace. By dividing this area A by ten we arrive at the Builder's Rating:

$$\therefore \text{Bl. h.p. (Builder's rating)} = \frac{A}{10} \tag{3}$$

As a detailed illustration, let us take the case of a Parker boiler installed at the Fruitvale Power Station of the Southern Pacific Company in Oakland, California.

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This boiler is made up of three banks of tubes with two drums above, half exposed. In detail we compute as follows:

Tubes 4 in. diameter, circumference = 12.566 in. = 1.0472 ft.
Tubes 18 ft. long = 18 × 1.0472 = 18.85 sq. ft. of H. S.
Tubes 20 ft. long = 20 × 1.0472 = 20.94 sq. ft. of H. S.

		Heating area
Heating Surface, Bottom Row of Tubes:		
20 tubes with 18 ft. of length exposed to gases	= 18.85 × 20 sq. ft.	= 37.700
Heating Surface, First Pass:		
100 tubes with 20 ft. of length exposed to gases	= 20.94 × 100	= 2094.00 sq. ft.
Heating Surface, Second Pass:		
80 tubes with 20 ft. of length exposed to gases	= 20.94 × 80	= 1675.20 sq. ft.
Heating Surface, Third Pass:		
80 tubes with 20 ft. of length exposed to gases	= 20.94 × 80	= 1675.20 sq. ft.
Drums:		
2 drums 54 in. diameter, 18½ ft. of length exposed to gases: circumference = 14.1 ft.; ½ of circumference 7 ft.	= 7 × 18.5 × 2	= 259.00 sq. ft.
Total		6080.40 sq. ft.

Hence, we have that the builder's rating of this boiler should be

$$\text{Bl. h. p. (Builders rating)} = \frac{6{,}080.4}{10} = 608.04.$$

To Compute Actual Boiler Rating.—Since it is seen from the fundamental definition of the boiler horsepower that the standard reference boiler generates its steam from water at 212°F. into steam at 212°F., we must next develop a factor by which we can reduce ordinary boiler performances of high temperatures and pressures to this fictitious standard before we can proceed further. The next chapter will be devoted to this consideration.

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CHAPTER IX

EQUIVALENT EVAPORATION AND FACTOR OF EVAPORATION IN FUEL OIL PRACTICE

FIG. 48.—Piping in boiler setting where superheat temperatures are taken.

IN the previous chapter it was seen that as the fundamental definition of the boiler horsepower is based upon a fictitious boiler that receives its feed water at 212°F. and then evaporates it into dry saturated steam at 212°F. and atmospheric pressure, we must now develop some factor by which we can reduce boiler performances as actually met with in practice to this fictitious standard.

In order also to compare the steaming qualities of two different boilers or indeed to compare the same boiler under different conditions of water supply and steam generation, it is necessary that some standard of comparison be adopted. Thus a boiler under its normal condition of operation may be found to evaporate 13.61 lb. of water per lb. of oil fired per hour when taking its feed water at 169.1°F. and converting it into superheated steam at a temperature of 527°F. and a pressure of 185.3 gage. On the other hand, the identical boiler, when steaming under overload conditions of a feedwater temperature of 174.1°F., a superheat temperature of 536.9°F. and gage pressure of 194.1 lb. per sq. in. may be found to evaporate only 13.17 lb. of water per lb. of oil fired, even though the same quality of oil be used in each instance. It is evident then from sight that to compare these two evaporative quantities without taking account of the actual heat transferred from the fuel to the steam in the boiler would be a possible source of error.

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The Standard that Has Been Adopted.—To avoid inconsistencies and to develop some rational method of comparison, engineers have found it convenient and accurate to reduce all evaporative quantities of a boiler to a definite standard. In order to follow out this standardized comparison, all steam generating performances of boilers read as if the boiler took its feed water at 212°F. and atmospheric pressure, and converted it into dry saturated steam at 212°F. and atmospheric pressure, as set forth in the standard definition of the boiler horepower in the last chapter. It is clearly evident that no such theoretical

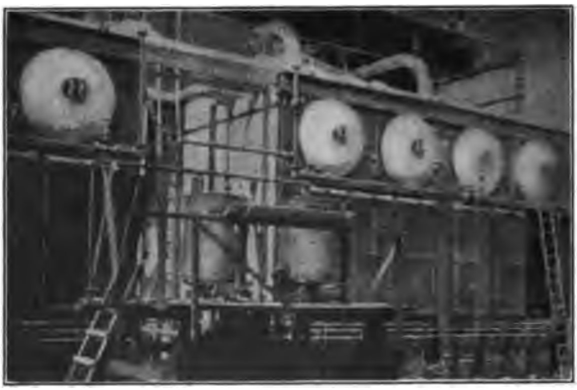

Fig. 49.—Platform scales and tanks for water measurement.

The boiler immediately to the right of the platform scales is under test. The tank below the platform scales into which the water is emptied after being weighed, is utilized to furnish all water for the boiler during the test. At the beginning of the test a hooked gage registers the height of the water in this tank, and at each hourly period thereafter sufficient water is weighed and emptied into it from the tanks above to maintain this exact level. By means of these data, properly taken, the factor of evaporation and the boiler horse-power are easily computed.

boiler has ever existed, yet this standard of comparison is found very convenient. Thus in any case of boiler performance, if M_e represents such an equivalent or comparative standardized evaporation in lbs. of water per lb. of fuel, and M_w the lb. of water actually evaporated in the boiler under conditions of test, we may now invent a factor to be known as the factor of evaporation, F_e, whereby such performances may be readily reduced:

$$M_e = M_w . F_e \qquad (1)$$

In the same way, the equivalent evaporation of water per hour may be computed from the formula

$$M_{eh} = M_{wh} . \quad F_e \qquad (2)$$

wherein M_{eh} and M_{wh} represent hourly conditions of evaporation.

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Let us next analyze the factor of evaporation and see how we may actually compute its value for any given case. We have previously found that in the operation of the boiler, steam appears in three different conditions or qualities, namely in what is known as dry saturated, wet steam, or superheated steam. Let us then consider the valuation of the factor of evaporation for these three distinct instances.

Dry Saturated Steam.—In the case of dry saturated steam, the water enters the boiler already possessing a heat of liquid h_f corresponding to its entrance temperature which may be readily found in the steam tables. This water is next converted into dry saturated steam which has a total heat (H_e) corresponding to the pressure at which the evaporation takes place. Consequently the actual heat which has been transferred from the boiler shell to the water is $(H_e - h_f)$ heat units. But to evaporate one pound of water at 212°F. into dry steam at 212°F. requires 970.4 heat units. Hence if M_w pounds of water are evaporated under test conditions, the number of pounds M_e under standardized conditions would evidently be $M_w \frac{(H_e - h_f)}{970.4}$. Therefore for dry saturated steam

$$F_e \text{ (dry saturated steam)} = \frac{(H_e - h_f)}{970.4} \tag{3}$$

Thus in the case of a boiler which takes its feed water at 101.8° F. and converts it into dry saturated steam at 180 lb. pressure per square inch, from the steam tables we find that H_e is 1196.4 and h_f is 69.8, hence the factor of evaporation is

$$F_e = \frac{1196.4 - 69.8}{970.4} = 1.16$$

Wet Steam.—In the case of wet steam all of the water entering the boiler is not converted into steam. As a consequence a certain portion of heat $(h_e - h_f)$ is required to raise the temperature of the water from entrance temperature t_f to the temperature of evaporation t_e and if only X_e parts of a lb. are then evaporated into steam, only X_eL_e B.t.u. are required to accomplish this result. Hence, the total heat required per lb. of water so evaporated is $(h_e + X_eL_e - h_f)$.

As a consequence the factor of evaporation in this case may from similar reasoning be expressed by the formula

$$F_e\text{(wet steam)} = \frac{(h_e + X_eL_e - h_f)}{970.4} \tag{4}$$

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As an instance showing the application of this formula let us assume that the boiler above mentioned did not evaporate the water into dry steam but that upon investigation it was found to contain 5 per cent. moisture. What now is its factor of evaporation? From the steam tables we find that h_e is 345.6, L_e is 850.8 and h_f is 69.8. Therefore the factor of evaporation is

$$F_e = \frac{345.6 + .95 \times 850.8 - 69.8}{970.4} = 1.117$$

Superheated Steam.—In the third instance steam is not only evaporated to a dry saturated condition, but is finally sent from the boiler in a superheated condition. The steam tables are so arranged that we may find the heat necessary to raise the total heat of superheated steam when its pressure and temperature are known. Considering that the water entered the boiler at 32°F. let us then call H_s the total heat of superheated steam. Since now the water entered the boiler with a heat of liquid equal to h_f the actual heat entering each lb. of steam evaporated in the boiler under these conditions is $(H_s - h_f)$ heat units. Hence in this instance the factor of evaporation is likewise from similar reasoning computed by the formula:

$$F_e(\text{superheated steam}) = \frac{H_s - h_f}{970.4} \tag{5}$$

To follow up the same example as set forth in the preceding illustration let us assume that the steam is evaporated under the conditions hitherto mentioned, but that it appears superheated to the extent of 100°. Looking in the steam tables we find that the total heat H_s of superheated steam at 180 lb. pressure and 100° superheat is 1254.3 and that the heat of liquid h_f is 69.8, consequently the factor of evaporation is

$$F_e = \frac{1254.3 - 69.8}{970.4} = 1.22$$

To Compute the Boiler Horsepower.—Since now by means of formula (2), we are enabled to compute the equivalent evaporation of M_{eh} in pounds of water per hour that the boiler under test would evaporate were it taking its feed water at 212°F. and converting it into dry saturated steam at the same temperature, we can at once compute the horsepower of the boiler. Under such conditions of operation for every 34.5 lb. of water evapo-

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rated per hour, the boiler is developing one boiler horsepower. Hence to compute the boiler horsepower, we write the formula:

$$\text{Bl. hp.} = \frac{M_{eh}}{34.5} \tag{6}$$

Thus if a boiler has an equivalent evaporation of 23,350 lb. of water per hour, its horsepower is found to be

$$\text{Bl. hp.} = \frac{23{,}350}{34.5} = 676.7$$

We could of course develop an expression for the computation of boiler horsepower by taking into consideration the heat absorbed by the generation of steam per hour. For in our discussion in the previous chapter it was shown that one boiler horsepower is equivalent to the absorption of 33,479 heat units per hour. Hence, by computing the heat absorbed by the total pounds of steam generated per hour and dividing this by 33,479, we can compute boiler horsepower and arrive at the same answer as given in the above formula. It is better, however, for the beginner to follow fundamental definitions rather than attempt too many short cuts to gain quick results.

In conclusion the important relationship to bear in mind is the vast difference between the socalled mechanical horsepower and the boiler horsepower which was brought out in the previous discussion. With this relationship firmly fixed it must be remembered that equivalent evaporation is such an evaporation as would be brought about by taking in water into the boilers at 212°F. and evaporating it into dry saturated steam at 212°F. and atmospheric pressure. The formulas deduced above for equivalent evaporation and factor of evaporation enable us to do this.

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CHAPTER X

HOW TO DETERMINE QUALITY OF STEAM

FIG. 50.—Thermometer inserted for superheat measurement.

STEAM as used in engineering practice is said to be wet, dry saturated or superheated, depending upon the degree to which heat has been applied in its generation.

Wet Steam.—As its name implies, wet steam is steam in which are suspended small globules or particles of water. Since such globules or particles of water indicate that insufficient heat has been applied, and consequently steam generation is imperfect, it is the function of all good boilers to generate steam as free from water as possible.

Although steam be generated dry or even superheated it may, however, after passing through conducting pipes appear at the power generating unit in a wet condition. Hence the determination of moisture content and the heat loss due to its presence is an important one in steam engineering.

Let us assume X to be the proportion by weight of dry steam that exists in wet steam. Then the total heat represented in every pound of such wet steam at temperature t is

$$H_t = h_t + XL_t \qquad (1)$$

This is evident at once when we consider that to raise each pound of original water from 32°F. to the temperature t, it required h_t, heat units. On the other hand since a proportion by weight

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equal to X has actually gone into steam, the heat required in the latent heat of evaporation is but XL_t.

Dry Saturated Steam.—As one may infer from the heading, saturated steam that contains no moisture is called dry saturated steam. In the chapter on properties of water the determination of its total heat was illustrated quite fully. We may, however, derive the equation for total heat of dry saturated steam from the equation above for wet steam. For in this latter instance since no water is present, evidently X becomes equal to unity. Hence for dry saturated steam

$$H_t = h_t + L_t \qquad (2)$$

Superheated Steam.—It has been hitherto pointed out that when water is being evaporated into steam the temperature remains constant until all the water disappears. So long, however, as steam remains in contact with the water from which it is being formed it is either dry or wet steam and its temperature cannot be raised above that which normally represents the boiling point of water for the pressure under which the steam is being generated.

It has, however, been found of immense economic value in steam engineering practice to actually use steam that is heated over a hundred degrees in excess of the temperature at which saturated steam may be generated under the existing pressure conditions. It is seen that such steam must, of course, first become absolutely dry and then any additional heat that may be added goes toward raising its temperature if the pressure be kept constant.

This is accomplished in the modern steam generating units by conducting the saturated steam from the main drums in which it is generated and passing it through tubes exposed to highly heated portions of the boiler furnace. Such a system of tubes is known as a superheater. The steam quickly absorbs sufficient heat to completely dry it and still further raise its temperature.

Computation of Total Heat of Superheated Steam.—If a definite constant quantity of heat were required to superheat a pound of steam one degree in temperature for all ranges of temperature and pressure, we could write down a comparatively simple formula for arriving at the total heat of superheated steam. Since, however, this specific heat constant has a wide range of .46 to .60 it is impossible to do so.

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Hence in each case of temperature, pressure and degree of superheat, we must refer to steam tables in order to find the proper value of total heat of superheated steam. And, indeed, this too is necessary to find all the other constants that relate to superheated steam.

The fundamental definition remains the same, however—namely that the quantity of steam required to raise one pound of water from 32°F. to the temperature t corresponding to the boiling point of water for the pressure at which the steam is generated, added to the latent heat of evaporation for this pressure, together with such additional heat as may be required to raise the one pound of now dry saturated steam to the degree of superheat given, is known as the total heat of superheated steam H_s. Expressing this algebraically we have

$$H_s = h_t + L_t + C_{pm} (t_s - t) \qquad (3)$$

As an example let us suppose that superheated steam is being generated at ordinary atmospheric pressure and delivered at a temperature of 312°F. We will suppose that the mean specific heat C_{pm} for the range of temperature and pressure under consideration is say 0.46. Then from the tables, we find

$$h_t = 180. \quad L_t = 970.4 \quad t_s = 312°.$$
$$t = 212°. \quad C_{pm} = 0.46$$
$$\therefore H_s = 180 + 970.4 + 0.46\,(312 - 212)$$
$$= 180 + 970.4 + 46 = 1196.4 \text{ B.t u.}$$

It is most important that the student should remember that although the value H_s may be taken directly from the steam tables, still it is based on the several steps above taken. In many steam engineering problems this separate analysis or dissecting must be done so it is well to clinch this matter without delay.

Steam Calorimeters.—The word calorimeter often causes considerable confusión because there are two entirely different and distinct types of mechanism that bear this name in engineering practice. Fundamentally it means "a measurer of heat." In order to determine the heat contained in fuel an instrument known as a calorimeter is employed which will be described in later pages. At this point, however, we shall now proceed to describe several types of an instrument that bears the same name and yet is entirely different both in design and in aim to be accomplished.

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The steam calorimeter is an instrument used in steam engineering practice to determine the exact quality of steam, whether it be wet, dry saturated, or superheated, and to what extent. Since the thermometer and the carefully calibrated pressure gage constitute the easiest and most direct method of ascertaining superheat, the uses of the steam calorimeter are usually limited to determination of moisture in wet steam.

The Determination of Superheat.—The method of ascertaining superheat will now be set forth.

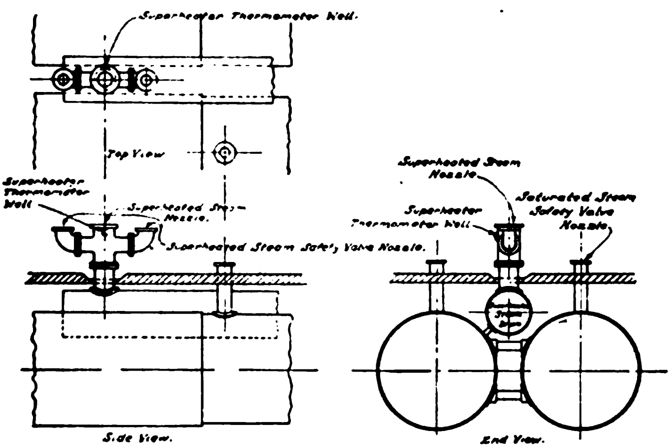

FIG. 51.—Temperature determination for superheated steam.

In taking the temperature for superheater steam a thermometer should be inserted as near the superheater drum as practicable. The thermometer has suspended at its side a second thermometer in order to ascertain the proper correction to be made for that portion emerging from the bath in which the main thermometer rests so that the stem correction may be made. In the illustration may be seen the point at which the thermometer well for ascertaining the superheated steam temperature was inserted in finding the superheat for an installation in Oakland, California.

A thermometer is inserted in the outlet of the superheater drum, and the temperature read, and at the same instant the pressure of the superheater drum is read on a steam gage attached to this drum. In order to correct for the exposed stem of the superheat thermometer a separate thermometer is attached to the stem of the former. Suppose the superheat thermometer reads 529°F., and 300° of its mercury column projects above the thermometer well; suppose also the attached thermometer reads 141°F. Then, by formula (6) on page 40, the stem correction equals

$$0.000086 \times 300 \times (529 - 141) = 10°\text{F.}$$

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The true temperature of the superheated steam is therefore 529 + 10 = 539°F. If now the steam gage reads 178.5 lb. per sq. in. and the atmospheric pressure is 14.7 lb. per sq. in. we proceed as follows:

The absolute pressure of the superheated steam is the sum of 178.5 and 14.7 which is 193.2 lb. per sq. in. Referring to steam tables, we find that water boils, or rather saturated steam is generated, at a temperature of 379°F. when under a pressure of 193.2 lb. per sq. in. Hence, the superheat of the steam under consideration is the difference of 539° and 379°, which is 160°F.

Determination of Moisture in Wet Steam.—There are many methods that may be used in determining the moisture content of wet steam. The particular method to be employed depends much upon the accuracy desired and the degree or intensity of the moisture content present.

The Barrel or Tank Calorimeter.—In this method, which should never be used except for approximate results, the steam is allowed to pass up through a barrel of water. Of course, the steam at once condenses into water and the resulting mixture with the water in the barrel raises the temperature. By taking the pressure of the steam and the two temperatures of the water—the one before applying the steam and the other after its application together with the weights of the water involved, we may at once write a mathematical relationship to determine the moisture content.

If we neglect radiation and other stray losses, the heat gained by the water in the barrel is equal to that lost by the steam under test.

In all the subsequent discussions in this chapter let us let 0 subscripts represent conditions of steam in the boiler; 1 subscripts, the initial conditions in the barrel; 2 subscripts, the final conditions in the barrel; and W will represent the weights involved.

The total heat of each pound of entering steam is by equation (1) found to be $(h_0 + X_0L_0)$ and since after this pound of condensed steam mixes with the water in the barrel it still has h_2 units of heat, there is then a net loss of $(h_0 + X_0L_0 - h_2)$ heat units. In the same way each pound of water in the barrel gains $(h_2 - h_1)$ heat units. If W_0 units of steam are involved and W_1 units of water are found in the barrel at the beginning of the test, we know then, since heat lost by the steam is equal

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to heat gained by the water, neglecting radiation and other losses, that

$$W_0 (h_0 + X_0L_0 - h_2) = W_1 (h_2 - h_1)$$

$$\therefore X_0 = \frac{W_1(h_2 - h_1) - W_0(h_0 - h_2)}{W_0L_0} \quad (4)$$

As an example, it was found in a test that a steam main under 90 lb. pressure (gage) deposited 3 lb. of condensed steam into a

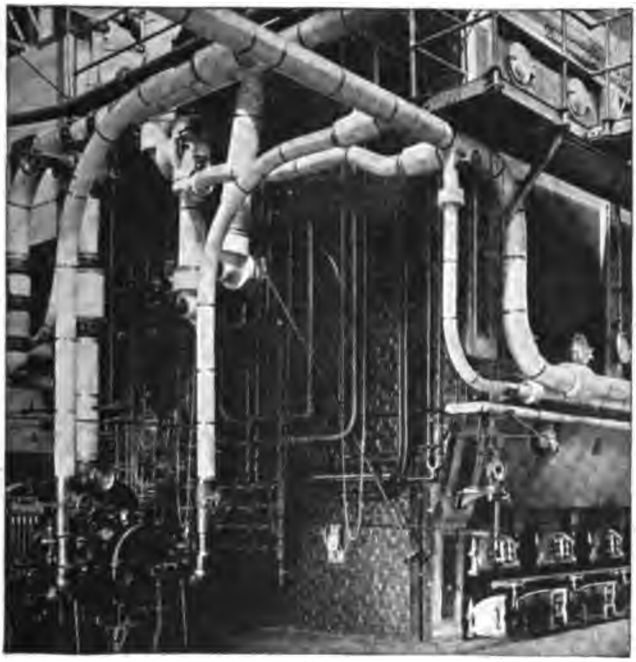

FIG. 52.—Auxiliary steam apparatus at left with boiler at right showing soot blower and auxiliary steam piping, Long Beach Plant, Southern California Edison Company.

vessel that contained 27 lb. of water at 62°F., thereby raising the temperature to 175°F. We compute the proportion of dry steam in the main as follows:

P_0 = 90 lb. per sq in. (gage) = 104.7 lb. per sq. in. abs,
W_1 = 27 lb., W_0 = 3 lb. t_1 = 62°F., t_2 = 175°F.

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Hence from steam tables—

$$L_0 = 885.4,\ h_0 = 301.8,\ h_1 = 30.1,\ h_2 = 142.9$$

$$\therefore X_0 = \frac{27(142.9 - 30.1) - 3(301.8 - 142.9)}{3 \times 885.4} = 0.968$$

$\therefore X_0 = 96.8$ per cent. dry steam in steam under test.

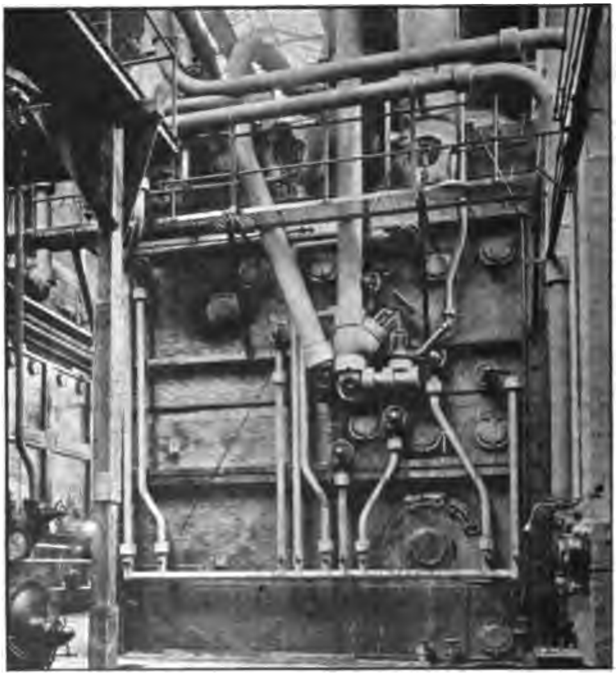

FIG. 53.—Side view of oil fired Stirling boilers showing the steam piping, soot blower, piping and explosion doors, station A, Pacific Gas and Electric Company, San Francisco.

Surface Condenser Tank Calorimeter.—This method varies from the one just set forth in that the condensed steam does not mingle with the water in the barrel. To accomplish this the steam is passed through a coil of piping which is inserted in the tank. As the steam comes in contact with the cooling surface of this pipe, it is condensed into water and of course the heat thus liberated or given out is absorbed by the water in the tank and its temperature correspondingly raised. Hence in this instance, it is necessary to weigh the water in the tank and the

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condensed steam discharged through the coil. It is also necessary to take the pressure of the steam under observation and to note the temperature of the tank of water before and after application as well as the temperature of the water discharged from the coils.

Proceeding by similar reasoning as set forth in the former instance, the heat lost by each pound of steam is sure to be $(h_0 + X_0L_0 - h_3)$, wherein the subscript 3 is to denote the condition of the steam condensed into water as it emerges from the coil. The heat gained by each pound of water in the tank is also seen to be $(h_2 - h_1)$ heat units. Hence if W_0 lb. of condensed steam are discharged and W_1 lb. of water are found in the tank, since the heat lost by the steam is equal to that gained by the water, neglecting radiation and other minor losses, we have

$$W_0(X_0L_0 + h_0 - h_3) = W_1(h_2 - h_1)$$

$$\therefore X_0 = \frac{W_1(h_2 - h_1) - W_0(h_0 - h_3)}{W_0L_0} \qquad (5)$$

To illustrate, let us assume that one pound of steam at a pressure of 100 lb. per sq. in. absolute is passed through coils immersed in a tank containing ten pounds of water at an initial temperature of 100°F. At the conclusion of the condensation the water in the tank is found to be at a temperature of 204.5°F., while that emerging from the coils is 210°F. The quality of the steam is at once found by substitution in the formula as follows:

From the test data we have $p_0 = 100$ lb., $W_1 = 10$ lb., $W_0 = 1$ lb., $t_1 = 100$°F., $t_2 = 204.5$°F., and $t_3 = 210$°F. From the steam tables we find $h_1 = 68$, $h_2 = 172.5$, $h_3 = 178$, $h_0 = 298.3$, $L_0 = 888.0$.

$$\therefore X_0 = \frac{10\ (172.5 - 68) - 1\ (298.3 - 178)}{1 \times 888} = 1.05$$

Since the quality of steam is greater than unity, it is evident that the steam in this instance is superheated.

The principle upon which the more accurate steam calorimeters operate is in general accomplished along similar lines. We shall, however, reserve further discussion on the subject until the next chapter wherein we shall deal at length with calorimeters.

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CHAPTER XI

THE STEAM CALORIMETER AND ITS USE

We come now to a consideration of the methods used in steam engineering practice to accurately determine the moisture content of saturated steam. In the preceding chapter certain approximate methods were set forth, but in the following discussion it will be seen that by care and patience the moisture content of saturated steam may be ascertained with a wonderful degree of accuracy.

The Chemical Calorimeter.—The Chemist has a method of determining the moisture content which finds little application in the steam engineering laboratory, but in the chemist's laboratory it is performed with a remarkable degree of accuracy. Certain salts absorb moisture held in a vapor. Hence by passing wet saturated steam over such salts, the moisture content is taken from the steam and by weighing the moisture so absorbed the degree of moisture held in suspension is ascertained.

The Throttling Calorimeter.—By reference to the steam tables it is seen that when saturated steam exists at say 200 lb. pressure per sq. in., each pound of steam represents a storage of heat equal to 1197.6 B.t.u. For it is seen from the steam tables that it took 354.4 B.t.u. to bring the original pound of water from 32°F. to its boiling point and then an additional 843.2 B.t.u. to evaporate this water into dry saturated steam.

Let us suppose for a minute this steam at 200 lb. per sq. in. were allowed to flow through an orifice and expand into a chamber which was at but 14.7 lb. per sq. in. From the steam tables it is seen that saturated steam existing under such a pressure holds in storage but 1150.4 B.t.u. What then becomes of the difference between 1197.6 B.t.u. and 1150 B.t.u. represented by the heat held in storage in the two instances? Evidently if the main at the lower temperature be well hooded so that no heat escapes, the heat given out must go toward superheating the steam at the lower pressure. Since the specific heat of superheated steam at the lower pressure is about 0.47, the 47.2 B.t.u. that are liberated

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would evidently superheat the steam about 100°. The actual measurement, then, of this superheat gives us at once a most accurate method of determining the quantity of moisture present in the steam at the original pressure. For if we find that the steam is superheated only 25°F., instead of 100°F., evidently some of the mixture must have been water, for otherwise its existence at the higher temperature as steam would aid in superheating still further the lower temperature.

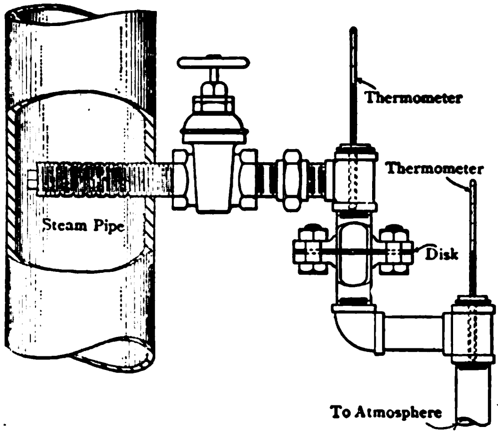

FIG. 54.—The throttling calorimeter and the sampling nozzle.

In the typical throttling calorimeter, steam is drawn from a vertical main through the sampling nipple, then passed around the first thermometer cup, then through a one-eighth inch orifice in a disk between two flanges, and lastly around the second thermometer cup and to the atmosphere. Thermometers are inserted in the wells, which should be filled with mercury or heavy cylinder oil. Due to the fact that the heat content in the steam under the expanded condition with which it reaches the second thermometer, is much less, the heat thus liberated superheats the steam at this point and thus a means is given for ascertaining the moisture originally in the steam sample.

A throttling calorimeter, then, is simply a contrivance by which we allow steam to pass from its high pressure through a small opening where its temperature and pressure are taken before it passes out into the atmosphere. Prior to its passage through the small opening, the temperature and pressure of the steam is noted. Let us denote by "s" subscripts the conditions of superheated steam in the low pressure chamber, "o" subscripts the steam in the steam main, and "3" subscripts saturated steam at the pressure of the low pressure chamber.

Each pound of wet steam in the steam main has X_o parts by weight existing as dry steam. Hence the total heat represented in each pound of this steam is evidently $(X_oL_o + h_o)$ heat units

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as seen from close inspection. In the same manner each pound of steam in the lower pressure chamber holds in storage $[H_3 + C_{pm}(t_s - t_3)]$ heat units as seen from previous reasoning. Since no heat is allowed to escape, evidently these

$$X_oL_o + h_o = H_3 + C_{pm}(t_s - t_3)$$

expressions are equal one to the other, or
C_{pm} for the low pressures has a value of 0.47, hence, we have

$$X_o = \frac{H_3 + 0.47\ (t_s - t_3) - h_c}{L_o} = \frac{H_s - h_o}{L_o} \tag{1}$$

in which H_s is the total heat of superheated steam in the low pressure chamber. Its numerical value may be taken directly from the steam tables when the pressure and degree of superheat are known.

As an illustration, let us assume that the pressure in the steam main is 153.6 lb. per sq. in. abs. and that its temperature is found to be 362.9°F., thus indicating at once that the steam is saturated and not superheated. After it has expanded into the low pressure chamber it is found to have a temperature of 261.3°F. and a pressure of 14.8 lb. per sq. in. absolute.

From the steam tables we find $L_o = 859.6$; $h_o = 334.8$; $H_3 = 1150.5$; $t_s = 261.3$; $t_3 = 212.4$°F.

$$\therefore X_o = \frac{1150.5 - 334.8 + 0.47\,(261.3 - 212.4)}{859.6} = 0.9758$$

Therefore the steam is evidently 97.58 per cent. dry.

Normal Reading of the Calorimeter.—For accurate work it is necessary to make a correction for the radiation loss from a calorimeter. This may be done by taking readings on the instrument when absolutely dry saturated steam is passing through it. This reading is called the Normal Reading of the calorimeter. A boiler that is standing by without a fire under it but with pressure up will deliver practically dry steam. It is, therefore, possible to secure the normal reading of the calorimeter right in place, by shutting off the oil burners and allowing the circulation within the boiler to come to rest. A more accurate method of obtaining the normal reading is shown in the illustration on page 89. In this arrangement the calorimeter is supplied from near the top of a horizontal pipe containing quiescent steam, a drain being provided to remove the moisture from the bottom.

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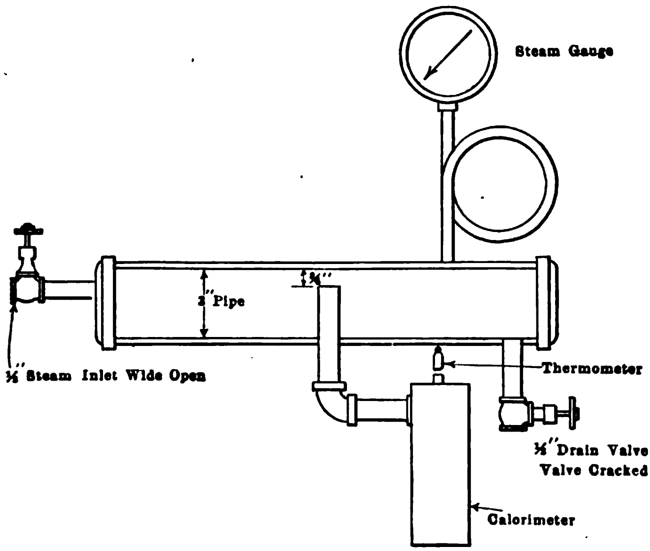
By using the normal reading (t_n) in equation (1) we have

$$X_n = \frac{H_3 + 0.47(t_n - t_3) - h_o}{L_o}$$

Here X_n is the quality of steam indicated by the calorimeter when in reality the steam is dry. X_o is the quality of the actual



A suggestion for a steam calorimeter attachment for determining normal reading of calorimeter. (See page 88.)

steam under test, as indicated by the same calorimeter. By subtracting one from the other we have the true moisture

$$\begin{aligned} x &= X_n - X_o \\ &= \frac{H_3 + 0.47(t_n - t_3) - h_o}{L_o} - \frac{H_3 + 0.47(t_s - t_3) - h_o}{L_o} \\ &= \frac{0.47(t_n - t_s)}{L_o} \end{aligned}$$

Thus in the foregoing example, let us suppose the normal reading of the calorimeter thermometer was 290°F. Then the true moisture in the steam becomes

$$x = \frac{0.47\ (290 - 261.3)}{859.6} = 0.0157$$

The steam therefore contains 1.57 per cent. moisture.

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The Limitations of the Throttling Calorimeter.—A little consideration of the underlying principle of the throttling calorimeter brings to light a definite range of limitation to its usefulness. It will be remembered that this fundamental principle consists in liberating sufficient heat at the lower pressure not only to evaporate any moisture that may exist but to actually superheat the entire mixture. If there is not sufficient heat liberated, that is if too much water is held in suspension in the saturated steam, the steam at the lower pressure fails to become superheated and hence we have no means of measurement.

Thus if steam pass from 100 lb. absolute pressure per sq. in. to 30 lb. absolute pressure per sq. in., the total heat at the upper pressure is $(X_oL_o + h_o)$ which from the steam tables becomes $(X_o888 + 298.3)$ heat units and that at the lower pressure is H_3, or 1163.9, if it be not at all superheated. Hence we have

$$X_o888 + 298.3 = 1163.9$$

$$\therefore X_o = \frac{865.6}{888} = 0.9748$$

This means that if there is a greater moisture content than 2.52 per cent. the steam calorimeter will fail to work because the mixture in the lower pressure space does not become superheated.

If instead of having the low pressure of 30 lb. absolute per square inch, the steam in the calorimeter had been throttled down to 14.7 lb. the value of H_3 would have been 1150.4 instead of 1163.9 so that X_o would become 0.9597 and the limit of the calorimeter in this case would be 4.03 per cent. of moisture.

Again if instead of steam at 100 lb. absolute pressure we had steam at 200 lb. and allowed the sample in the calorimeter to be throttled down to 14.7 lb. it may be found in the same way that the limit of the calorimeter is 5.66 per cent. of moisture. It is thus seen that the greater the difference in pressure between the high pressure and the low pressure in the calorimeter, the greater is the range of the calorimeter.

The Electric Calorimeter.—It is now evident that if a definitely measurable quantity of heat could be added to the steam before it was allowed to expand, even very wet steam might be accurately measured by the throttling calorimeter. This is seen at once when we analyze the total heats involved. If E_o be the heat units added to each pound of steam, then the total heat possessed by each pound of steam in the high pressure main

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is $(X_oL_o + h_o + E_o)$ heat units and since the heat in each pound of steam in the lower chamber is H_s, we have, since no heat escapes

$$X_oL_o + h_o + E_o = H_s$$

$$\therefore X_o = \frac{H_s - h_o - E_o}{L_o} \qquad (2)$$

In the Thomas electric meter an electrical mechanism has been invented whereby a series of small wires electrically heated impart a known quantity of electrical energy to the steam. This electrical energy dissipates itself as heat and since we can transfer electrical units into heat units and vice versa, a ready means is provided to assist the throttling calorimeter in doing its work by adding sufficient heat to widen the range of the throttling process.

Thus, although the throttling calorimeter was found definitely limited as set forth above, let us investigate a case where the electrical calorimeter may be used. Let us assume the upper pressure to be 200 lb. per sq. in. and the lower pressure 15.0 lb. per sq. in. In this case there were electrically added exactly 40 B.t.u. of energy and the temperature of superheat t_s was found to be 233.0°F., hence from the steam tables we find

$L^\circ = 843.2$; $h_o = 354.9$; $t_s = 233.0$; hence, $H_s = 1160.1$

$$\therefore X_o = \frac{1160.1 - 354.9 - 40}{843.2} = 0.908$$

The Separating Calorimeter.—In the separating calorimeter the moisture is mechanically separated from the steam. If we know the total amount of steam passing and also the weight of the water separated from the steam, it is of course an easy problem to compute the dryness of the steam. Thus, if W_1 is the weight of water separated per hour in the calorimeter and W_2 the weight of dry steam passing out of the calorimeter per hour, we have by inspection

$$X_o = \frac{W_2}{W_1 + W_2} \qquad (3)$$

Hence, if a separating calorimeter deposits 285 lb. of water per hour and if 10,000 lb. of dry saturated steam leave the calorimeter per hour, the dryness of the steam is

$$X_o = \frac{10{,}000}{10{,}000 + 285} = 0.972$$

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There are many principles upon which the separating calorimeter may operate. There are two forms, however, which are more usual than others. In one instance the steam mixture is given a rotary motion in its journey and consequently the water particles are thrown off by centrifugal force and collect in a drip below. In the other instance the stream flow receives a sudden reversal in direction. As dry steam easily performs this feat and water insists upon continuing its former direction of flow a separation is thus mechanically effected.

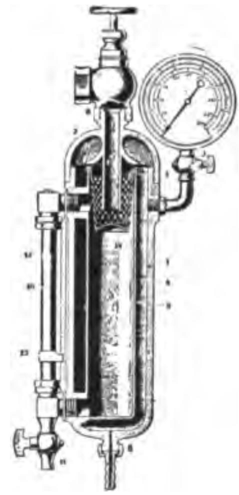

FIG. 55.—The separating calorimeter.

In this type of separating calorimeter the steam, with its moisture enters from the steam main at 6 and is forced to travel downward toward 3 at a high velocity. At 14, however, the direction is suddenly reversed upward toward 7 and later passed downward through 4 and out into the atmosphere at 8. When the sudden reversal takes place at 14, the moisture in the steam collects at 3 and its content is measured on the gage 12. The steam content, on the other hand, is calculated by means of Napier's formula as it passes through the orifice at 8 as illustrated in the text.

This type of instrument is not as accurate as the throttling type, as it does not get all the moisture out of the steam. When large quantities of moisture are present, however, it proves useful in taking out the bulk of the water or moisture while a throttling calorimeter connected in series later on accurately measures the remaining water content present. Thus by such a method of operation any degree of moisture present in steam is easily and accurately measured.

Correction for Steam Used by Calorimeter.—In a great many instances the total weight of steam passing per hour through the

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steam main under test is of prime importance. Since most forms of calorimeter operate by diverting a portion of this steam out into the atmosphere, it becomes necessary to have some quick and ready means of computing the quantity of steam so diverted.

Many years ago Napier deduced an approximate formula for the flow of steam into the atmosphere from a high pressure source. This formula is well within the degree of accuracy required for steam diverted through the calorimeter. If W is the pounds of steam flowing per second, p the pounds of pressure per square inch exerted by the steam in the main, and a the area of the orifice in square inches through which the steam passes, then

$$W = \frac{pa}{70} \qquad (4)$$

The Sampling Nipple.—The American Society of Mechanical Engineers recommends a sampling nipple made of one-half inch iron pipe closed at the inner end and the interior portion perforated with not less than twenty one-eighth inch holes equally distributed from end to end and preferably drilled in irregular or spiral rows with the first hole not less than one-half inch from the wall of the pipe. The failure to determine an average sample of the steam is the principal source of error in steam calorimeter determinations.

Conclusions on Moisture Measuring Apparatus.—Summing up the arguments of this chapter we see that for comparatively small quantities of moisture present in steam, the throttling calorimeter is the most accurate device for its quantitative determination. If, however, large quantities of moisture are present, two methods present themselves. Either we must first remove the major portion of the moisture by means of a separating calorimeter and later determine the remaining moisture content by means of the throttling calorimeter, or we must add a definite quantity of heat to the original steam supply by means of a device such as the Thomas Electric calorimeter and then determine with proper computation factors the moisture present by means of the throttling calorimeter.

As already shown the throttling calorimeter may be used up to moisture of 4 per cent. for steam at 100 lb. pressure and up to a little over 5 per cent. for steam at 200 lb. pressure. Most boilers deliver steam containing not more than 1½ per cent. or 2 per cent. of moisture so that for nearly all ordinary work the throt-

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tling calorimeter has sufficient range, and owing to its great simplicity and remarkable accuracy it is almost universally used. It is possible to make up a throttling calorimeter by means of pipe fittings by providing a disc within a pair of flanges having a small hole to act as the throttling agent, as shown in Fig. 54; or the throttling may be done merely by partially opening the valve on the sampling nipple close to the main steam pipe. An extremely convenient design of calorimeter and one that can be readily moved from place to place is shown in Fig. 56. A calorimeter of this type should have a deep thermometer well, *C*, so that the thermometer bulb will come well below the steam inlet *A*, thus giving the steam a chance to expand to the lower pressure before its temperature is taken. In this design a steam jacket is provided to prevent, as far as possible radiation losses from the calorimeter. For many further useful pointers and detail rules in ascertaining the moisture content of steam the reader is referred to the latest edition of "Steam" by the Babcock and Wilcox Company, and to the report of the Power Test Committee of the American Society of Mechanical Engineers which is to be found in Vol. 37, transactions A. S. M. E. for 1915, to which publications we are indebted for much of the information contained in this discussion.

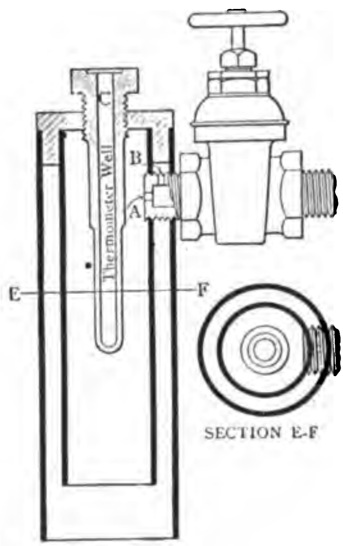

Fig. 56.—A suggestion for a convenient and compact type of throttling calorimeter.

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CHAPTER XII

RATIONAL AND EMPIRICAL FORMULAS FOR STEAM CONSTANTS

It has hitherto been pointed out that the relationships of temperature, latent heat and other steam properties are so complicated with varying pressures that no one as yet has been able to set forth simple mathematical equations for their representation.

There exist, however, a vast number of more or less complicated formulas that express with some degree of accuracy a relationship between these various factors. When such a relationship is deduced from some process of reasoning based upon known laws the equation is said to be rational. If, on the other hand, some one by sifting through the sands of time, as it were, has happened upon an equation with no rational backing the formula is said to be empirical.

Most of the equations used to set forth steam variables are partly rational and partly empirical.

Any equation, unless it be comparatively simple, is of little practical use to the steam engineer, for he may pick the values desired from the modern steam tables with such facility that it is really burdensome to try and remember any formulas connecting these properties.

The Value of Formulas in Steam Engineering.—In certain theoretical reasoning, however, a formula setting forth these relationships becomes often of inestimable value and indeed at times leads one to attain data otherwise impossible to compute. Such is the case of the formula from which the specific volume of saturated steam is obtained by computation and set forth in Chapter VII. Here it is found impossible to obtain by experiment that which is easily computed by application of this formula.

We shall next set forth some of the comparatively simpler relationships or equations that have been devised or annunciated by various authors. These will serve to give the student an insight only into such complicated formulas that arise in attempting a mathematical expression for these data.

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Unless one desires to go deep into the theoretical discussions of vapors and superheated gases such a brief introduction is nevertheless fully sufficient for the mastering of most problems in steam engineering computation.

Relation Between Temperature and Pressure of Saturated Steam.—It has already been set forth that water boils or that saturated steam begins to be formed from water at different temperatures for each variation in pressure. No one as yet has set forth a simple rational formula connecting this relationship. In the issue of Power of March 18, 1910, is to be found a formula which is the simplest and yet one of the most accurate empirical relations yet established. This formula connects the temperature in Fahrenheit degrees with the pressure in pounds per sq. in. at which water boils, and is as follows:

$$t = 200p^{\frac{1}{6}} - 101 \tag{1}$$

For a pressure of 10 lb. per sq. in. the error is but 0.28 per cent., while for 300 lb. per sq. in., it becomes but 0.32 per cent. The intermediate values are far less in error, so that this formula has, indeed, a wide range of usefulness.

The Total Heat of Saturated Steam.—Almost a century ago Regnault gave to the world his celebrated data on steam engineering. So accurately and so carefully did he perform his work that even today his experimental results are used in steam engineering computation, although of course corrections are applied where certain constants involved in computation are now known to have different values.

Regnault's Formula.—Regnault's formula for the total heat H_t of saturated steam at temperature t is one of the simplest ever invented and is as follows:

$$H_t = 1091.7 + 0.305(t - 32) \tag{2}$$

Let us test this formula by comparing its results with those set forth in the steam tables for 235°F.

Regnault's Formula:

$$H_{235} = 1091.7 + 0.305(235 - 32) = 1153.6 \text{ B.t.u.}$$

From Steam tables:

$$H_{235} = 1158.7 \text{ B.t.u.}$$

$$\therefore \text{Error} = \frac{1158.7 - 1153.6}{1158.7} \times 100 = 0.44\%$$

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Hence we see that for low temperatures the error involved by using the classic equation of Regnault is less than one-half of one per cent.

Henning's Formula.—Marks and Davis have in the rear of their steam tables set forth a formula of Henning, which though somewhat more complicated than Regnault's is, however, very accurate. This formula may be expressed as follows:

$$H_t = 1150.3 + 0.3745\ (t - 212) - 0.000550\ (t - 212)^2 \tag{3}$$

Let us now test the accuracy of this formula by substituting the same temperature of 235°F. as used in Regnault's formula.
By Henning's formula:

$$H_{235} = 1150.3 + 0.3745\,(235 - 212) - 0.000550\ (235 - 212)^2 = 1158.64 \text{ B.t.u.}$$

From steam tables:

$$H_{235} = 1158.7.$$

$$\therefore \text{Error} = \frac{1158.7 - 1158.64}{1158.7} \times 100 = 0.0052\%$$

Hence the error involved in the use of this formula is seen to be extremely slight.

Latent Heat of Evaporation.—Thiesen, after observing certain limits toward which the latent heat of evaporation seemed to tend, suggested the following formula for the latent heat of evaporation of water:

$$Lt = 138.81\ \ (689 - t)^{0.315} \tag{4}$$

Let us compare this with the steam table data for a temperature of 235°F.
Thiesen's formula:

$$L_{235} = 138.81(689 - 235)^{0.315} = 953.7.$$

Steam tables:

$$L_{235} = 955.4$$

$$\therefore \text{Error} = \frac{955.4 - 953.7}{955.4} \times 100 = 0.178\%$$

While the error here found is comparatively small, at higher temperatures this error becomes excessive. Hence we should apply this formula with due regard to its limitations.

A Second Formula for Heat of Evaporation.—Students in the classes of mechanical engineering at the University of California

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have established a relationship for latent heat and temperature as follows:

$$L^2 = 1209423 - 1289.5t \quad (5)$$

This formula is simple and yet accurate to within one-third of one per cent. for a wide range of temperatures of from 100°F. to 350°F. and within three-fourths of one per cent. for practically the entire range involved in steam engineering practice. The constants set forth were obtained by the method of Least Squares.

Relationship of Specific Volume for Superheated Steam.—In the chapter on The Elementary Laws of Thermodynamics, it was shown that the pressure, volume, and absolute temperature of a perfect gas are connected by a very simple relationship as set forth in the composite formula given in equation (5) on page 46. Indeed, it was shown that while superheated steam is not a perfect gas, still for approximate results this equation may be used.

For accurate work, however, the equation of Linde is found quite satisfactory although exceedingly cumbersome in its application. This equation connects the pressure p in pounds per sq. in. and specific volume v in cu. ft. per pound with the absolute temperature T in the following relationship:

$$pv = 0.5962T - p(1 + 0.0014p) \left[\frac{150300000}{T^3} - 0.0833\right] \quad (6)$$

To illustrate the application of this formula, let us endeavor to find the specific volume of superheated steam at 526.8°F. used in a turbine test when the steam was under a pressure of 187.2 lb. per sq. in. absolute.

It is seen that the absolute temperature of the superheated steam was

$$T = 526.8 + 459.6 = 986.4$$

and since the absolute pressure p was 187.2 lb. per sq. in., we have by substitution in the formula

$$v = 3.05$$

From steam tables:

$$v = 3.05$$

Hence in this instance the formula appears to be absolutely accurate for the range of units involved in the steam tables.

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A Simplified but Limited Formula.—A convenient formula for a pressure of 175 lb. per sq. in., the approximate pressure involved in steam turbine operations, has been worked out by the mechanical engineering students at the University of California for superheat between fifty and six hundred degrees and is as follows:

$$v = 2.67 + 0.00377t_s \tag{7}$$

Wherein t_s is the number of degrees superheat. The accuracy of this formula is within one-half of one per cent. for the range of superheat above set forth.

Other Relationships Exist.—By making use of certain theoretic considerations in thermodynamics many other equations might be written setting forth still other relationships involved in the determination of steam constants, but sufficient illustrations have now been given the reader for a thorough introduction to such formulas. Perhaps after all the most important lesson one derives from their use is that their application is often so tedious and their range of accuracy often so questionable, that one had better stay on well trodden paths and master to their fullest extent the application of the steam tables and diagrams in the solution of all steam engineering problems.

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CHAPTER XIII

THE FUNDAMENTALS OF FURNACE OPERATION IN FUEL OIL PRACTICE

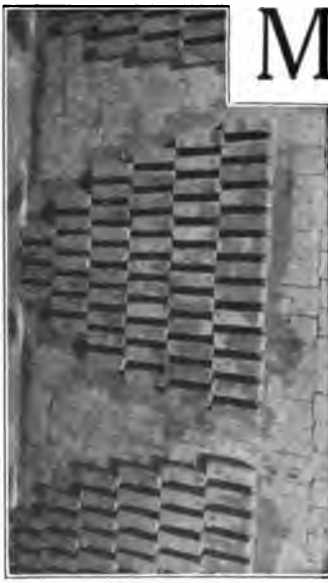

FIG. 57.—Air ducts for furnace floor.

MANY of us are familiar with the famous painting that pictures James Watt as a boy gazing in wide-eyed amazement at the homely tea-kettle spouting forth its hitherto unharnessed power generating vapors. The eyes of the youth are illuminated with that strange and wonderful light that set forth in a measure some of the dreams of constructive imagination which must have been filling his consciousness at that time.

The great inventor of the steam engine undoubtedly saw in the tea-kettle before him, not the homely object of the kitchen, but in its expanded form one of the most necessary mechanisms for modern industrial development—namely, the steam boiler.

Let us then examine the fundamental operation and construction of the steam boiler, and consider this great giant of modern industrial aggrandizement to see wherein it varies from its progenitor—the homely tea-kettle of Watt's boyhood dream.

The Fundamentals of the Tea-Kettle and the Boiler are the Same.—The tea-kettle in its construction and operation may be considered under three separate discussions. First, there must be some means of generating and imparting heat; secondly, a container for the water and steam must be constructed with physical characteristics to meet the stresses and strains involved;

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and, thirdly, the cycle of physical operations through which the water and steam pass in the generation of steam is of vast importance.

The tea-kettle operation in its simplest analysis consists of a flame placed beneath a metal container. This metal container absorbs the heat from the flame and transmits it to the water within the container. When sufficient heat has been absorbed by the water within the container to raise its temperature to the boiling point corresponding to the external pressure of the atmosphere, the tea-kettle boils or in the language of the steam engineer the tea-kettle generates steam.

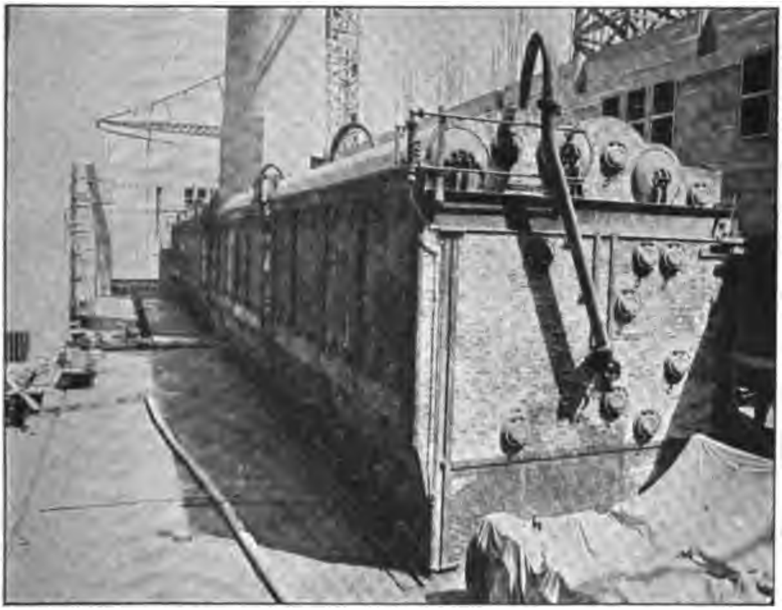

Fig. 58.—Boiler installation at the Long Beach Plant of the Southern California Edison Company under construction.

In its fundamental makeup, the boiler too, quite closely follows this familiar and homely object—the tea-kettle. For in the modern boiler heat is first generated in a furnace. This heat is then imparted to a metallic drum or tubes through which water is passed. When sufficient heat is thus imparted to raise the temperature of the water to the boiling point for the pressure involved, steam generation takes place.

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Inefficiency of Tea Kettle Operation.—In modern kitchen economics but little attention is paid to the manner in which the heat is imparted to the tea-kettle. Usually the stove lid is taken off and the kettle placed over the fire space thus created. Some minutes later, the house-wife, ignorant of the vast heat losses that have taken place, returns to draw off the hot water thus inefficiently obtained as convenience may require. As a matter of fact, the slightest and most casual investigation shows that in the United States millions of dollars are wasted every year for

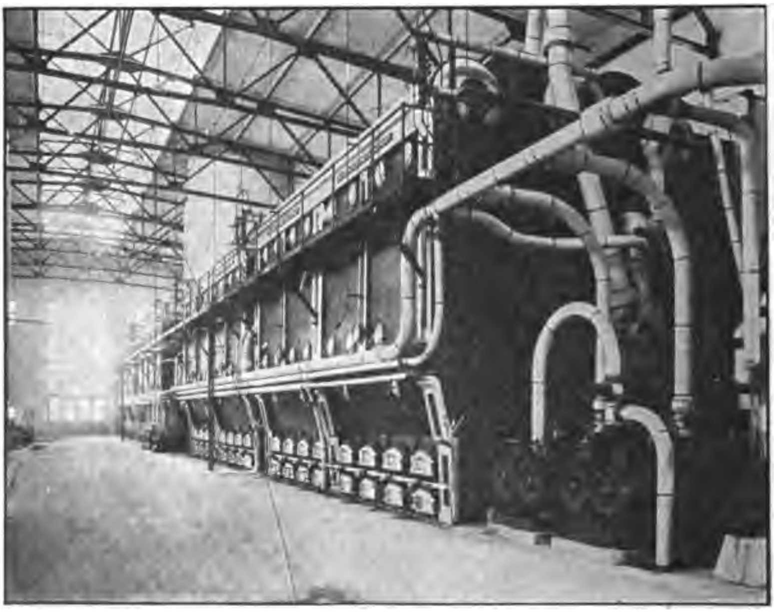

Fig. 59.—Same view of boiler plant when completed, showing auxiliary apparatus and steam piping.

lack of reasonable care in the kettle operation. This loss is, however, so widely distributed over thousands of homes that it is not felt in any concentrated form.

Efficiency in the Modern Steam Boiler a Necessity.—In the case of the modern central station, however, efficiency is the cry of the day. For with competition on all sides and regulating commissions to limit the prices charged for the power supply, the utmost in economic steam generation is essential.

Hence, in modern steam boiler operation, especially in its

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heat generating properties, a wide variation from tea-kettle operation is in vogue, not so much in fundamental principles involved as in efficiency of methods employed in the heat generating mechanisms

Efficient Furnace Construction of Utmost Importance.—To accomplish this efficiency an enclosed compartment beneath the boiler proper is built. This is known as the furnace. In this furnace heat generating substances such as coal, wood, and crude petroleum are burned. In the study of chemistry it has been found that certain primary elements, notably carbon, hydrogen,

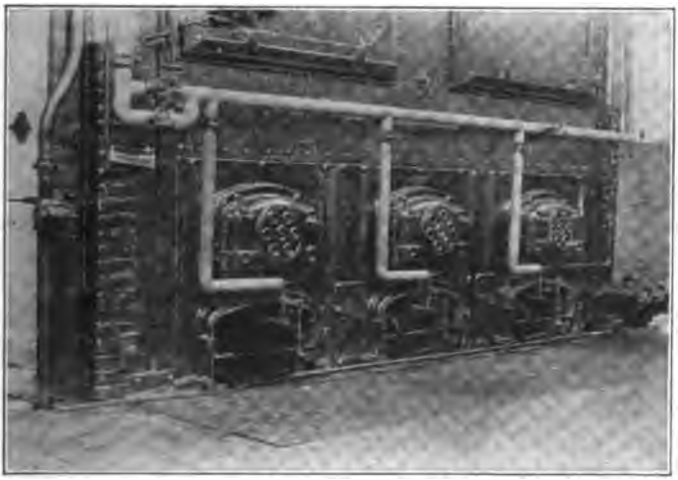

FIG. 60.—Typical boiler front in fuel oil practice.

In this illustration may be seen the fuel oil atomizer in the ash pit entrance, the covered steam pipes for supplying steam used in atomization, the fuel oil supply pipes, the damper control, the draft gage and other accessories for fuel oil operation.

and sulphur, upon coming in contact with heated oxygen undergo a chemical reaction and in doing so give out enormous quantities of heat. It is the generation of this heat and its ultimate absorption by the water in the boiler that makes the modern steam engine and steam turbine the giants in commercial enterprise that today they represent.

Fuels Defined.—In nature, substances such as coal, wood and crude petroleum are found in vast quantities and since these contain large amounts of free carbon and hydrogen, they make excellent articles for heat generation and are called fuels.

An Air Supply Essential.—It has been mentioned that a supply of oxygen is absolutely necessary so that a chemical reaction

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may take place and thus liberate the heat held in suspense in the fuel. The air about us is made up of about 20 per cent. oxygen and 80 per cent. nitrogen. The nitrogen is an inert, valueless ingredient that must pass into the furnace, absorb some of its heat and go out through the chimney, thus conducting away into the outer atmosphere some of the heat generated. The oxygen, however, upon coming in contact with the heated carbon, hydrogen and sulphur of the fuel, readily chemically reacts with them.

Enormous quantities of heat are thus liberated, later to be absorbed by the water of the boiler, eventually to produce the steam delivered for the driving of the steam engine or the steam turbine.

Furnace Operation.—Since this series of articles is largely concerned with fuel oil practice, let us briefly outline the furnace operation for such practice. In a later chapter this will be taken up in more detail.

The Fuel Oil Burner and Its Function.—The fuel oil is sprayed into the furnace by means of an atomizer or burner which pulverizes the oil and delivers it in a gaseous vapor or in small globules at the hottest place in the furnace. Air is admitted from below and as soon as the temperature is raised to the ignition point chemical reaction takes place with the atomized fuel oil, and thus heat is generated. This heat is absorbed by the gases of the furnace and consequently their temperature is at once raised often times to 2300° or 2500°F. These furnace gases consist of the inert nitrogen that partly constituted the entering air, the carbon dioxide or carbon monoxide formed by the burning of the carbon, water vapor formed by the burning of the hydrogen, sulphur dioxide formed by the burning of the sulphur content, which latter ingredient is always small, and a considerable quantity of free oxygen depending on the amount of excess air admitted to the furnace.

The Path of the Furnace Gases.—In their expanded condition, due to the absorption of such huge quantities of heat, the gases now travel upward. As they come in contact with the boiler drums or tubes through which water is circulating, the gases are, of course, cooled and the temperature of the water raised. In this maner the gases, having been chilled or lowered in temperature to 500 deg. or 600 deg. F., are finally passed up through the chimney, and steam generation within the boiler is accomplished.

The Economizer and Its Economic Value.—In some boiler installations a series of tubes through which cold water is passing,

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is placed between the boiler and the chimney. The chimney gases are thus forced to give up still more of their heat. These outgoing chimney gases are consequently reduced still further in temperature.

Such a device as cited above, is known as an economizer. This reduction in the temperature of the out-going chimney gases reduces the draft of the chimney. Hence, the economizer is an economic success so long as the saving in feed-water heating is greater than the interest on the cost of the economizer installation and other apparatus necessary to produce artificial draft, plus the cost of maintenance of this additional apparatus.

Quantity of Air Required.—It has been observed that the entrance of air into the furnace is absolutely essential for furnace operations. Too much air, however, is detrimental, for more oxygen may be admitted than can be economically used by the fuel. Hence, too great an excess of air simply means the passage up through the chimney of excess gases which absorb heat only to convey it out into the atmosphere without performing a useful function. In successful boiler operation, therefore, some means must be provided, first to measure the draft; second, to test the ingredients of the outgoing gases; and third, to regulate the entrance of air into the furnace.

The Draft Gage and Its Principle of Operation.—A draft gage usually consists of a column of water placed in a U-tube. The

FIG. 61.—The differential draft gage.

In order to exaggerate the readings of the draft in inches of water, the measuring tube rests on a slope of ten to one in this type of instrument, and thus readings to another decimal point are ascertained which would otherwise be impossible.

pressure in the chimney is less than the atmosphere without. Therefore, if one end of this tube is inserted into the chimney and the other rests under the atmospheric pressure without, the difference of water level thus obtained in the U-tube indicates the draft in inches of water. This may be converted into pounds

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pressure (absolute) per square inch by applying the formulas previously set forth in the chapter on pressures.

Apparatus for Determining Ingredients of Out-Going Chimney Gases.—For economic boiler operation the steam engineer should know the exact composition of the outgoing chimney gases. Since this is a matter of vast importance a later chapter will be given in which detailed discussions of methods involved and apparatus employed will be given. Suffice it to say at this point, however, that by means of such apparatus the engineer may determine whether the fuel is being properly consumed in the furnace and whether too little or too much air is being admitted into the furnace.

Draft Regulating Devices.—In fuel oil practice the proper supply of air may be determined to a nicety. Hence some means must be provided to regulate the air supply with the same precision. This is done by varying the amount of opening of either the ash pit doors or the boiler damper or both. If the air is regulated by partly closing the ash pit doors and leaving the damper wide open a strong draft may occur inside the boiler setting which tends to draw air in through the brick walls. As this is a detriment it is preferable to regulate the air by means of the damper.

The Chimney.—After the gases have passed through and around the various heat absorbing tubes and drums employed in the modern steam boiler and economizer, they are shot up into the atmosphere through a long vertical passage. The structure housing this passage is known as a chimney. The height of the chimney and its area of cross-section through which the flue gases pass have an important bearing on the economic boiler or rather furnace operation.

In a general way, the reader now has a grasp of the fundamentals involved in modern furnace operation for the steam boiler. We shall next consider the container or shell for steam generation and its accessories.

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CHAPTER XIV

THE BOILER SHELL AND ITS ACCESSORIES FOR STEAM GENERATION IN FUEL OIL PRACTICE

FIG. 62.—The clean, clear cut appearance of the oil fired boiler room.

LET us now consider some of the fundamental laws involved in heat transference, and then discuss the container or shell employed in steam generation together with the accessories that must accompany any high pressure steam generating unit to accomplish safe and efficient operation.

Going back once again to the homely tea-kettle for a simple illustration, we find that the container for the water and steam usually consists of a flat bottomed metallic vessel with free opening to allow the steam generated to escape to the atmosphere. There is also usually to be found an opening with a lid covering at the top where water may be passed in or the vessel cleaned at more or less irregular periods of operation in household economics.

In the case of the steam boiler, however, vast improvements in physical configuration and construction become a necessity. Let us then examine some of these differences.

The Laws of Heat Involved in Steam Generation.—The transference of heat is found by experimental observation to take place in three separate and distinct ways—namely, by conduction, by radiation, and by convection.

On a wintry night if one stands in front of a blazing fireplace it is easy to find illustrations of these three methods of heat transference. Thus standing, one feels the heat radiating to his face in outward projections from the fire, for if an article such

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as a solid screen, opaque to heat radiation, be placed between the face and the fire the sensation of heat on the face immediately disappears. If now from behind the screen one holds a metallic poker in the hot fire, it will not be long before the poker even at

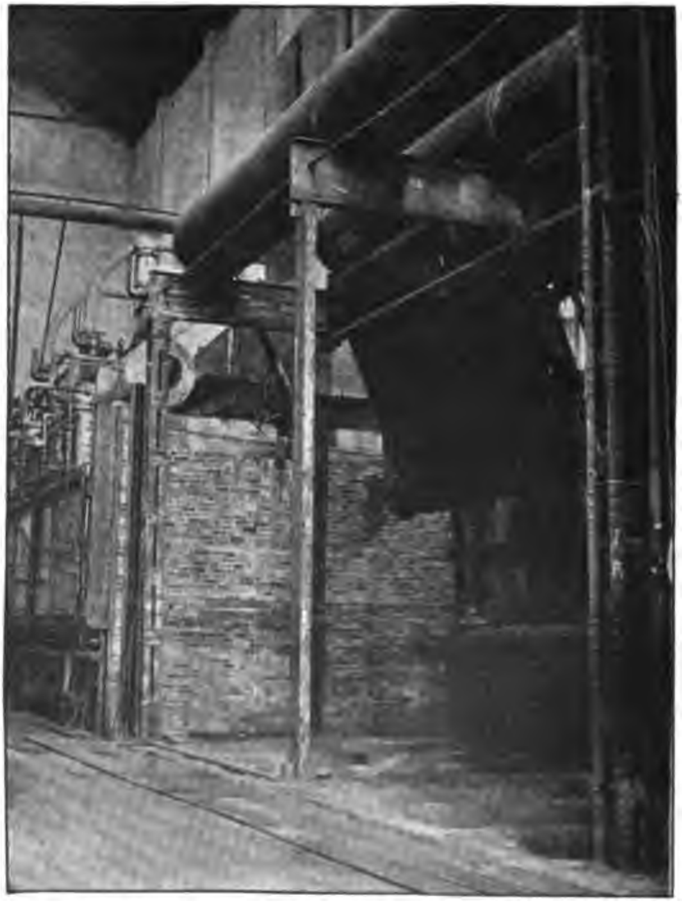

FIG. 63.—Front view of new boiler installation.

In this view may be seen the pit and foundation setting for oil furnace in the new additional installation recently put in by the Pacific Gas and Electric Company, San Francisco. Note the clean trim appearance extending to the older fuel oil installation on the extreme left, which is quite characteristic of boiler rooms where oil is used as fuel.

the point behind the screen becomes so hot by conduction that it cannot comfortably be held in the hand. And finally should a sudden gust of wind blow down the chimney a hot gust of air

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may be driven out into the room and around the screen to the observer's face, thus illustrating the transference of heat by convection.

The Principle of Operation of the Steam Boiler.—Let us then see how these three methods of heat transfer are utilized in modern boiler operation.

As has been previously noted, the burning of the fuel in the furnace causes enormous quantities of heat to be given out in the furnace space. This heat is immediately absorbed by the furnace gases, thereby raising them to a high temperature. By convection currents, and also by radiation, this heat is now transferred to the outer surface of the boiler shell and tubes containing the water that it is desired to convert into steam. The metallic shell and water tubes having now absorbed the heat, convey it to their inner surface by conduction where it is transferred to the water in the boiler. This water, becoming heated, expands, and due to its lighter density thus created is forced to go to the top of the water surface to make way for cooler, heavier water which in turn absorbs heat and disappears to make way for other water. This last activity is evidently again transference by convection currents and such a movement of water is called circulation. The efficient manner in which this circulation takes place has much to do with the economic operation of the boiler.

Mathematical Equation for Heat Transfer.—In 1909 Dr. Wilhelm Nusselt of Germany devised a formula whereby the factors involved in the rate of transference of heat are set forth quantitatively. This formula reduced to English units by the Babcock and Wilcox Company in their book on Steam is as follows:

$$a = 0.0255 \frac{\lambda_w}{(d)^{.214}} \quad \frac{(Wc_p)^{.786}}{(A\lambda)} \tag{1}$$

Wherein a is the transfer rate in B.t.u. per square foot of surface per degree difference in temperature; W is the weight of pounds of the gas flowing through the tubes per hour; A is the area of the tube in square feet, d is the diameter of the tube in feet; c_p is the specific heat of the gas at constant pressure; λ is the conductivity of the gas at the mean temperature and pressure in B.t.u. per hour per square foot of surface per degree Fahrenheit drop in temperature per foot; and λ_w is the conductivity of the steam at the temperature of the wall of the tube.

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Mathematical Law for Total Heat Absorption.—The application of this formula is cumbersome and indeed upon careful analysis it is seen to be largely empirical in its nature. Let us then cast about for another equation.

Stefan's law sets forth that the heat absorbed per hour by radiation is proportional to the difference of the fourth powers of the absolute temperature of the furnace gases T and the absolute temperature of the tube surface t of the boiler. In addition to this if we add the loss of heat given up by the outgoing gases due to their cooling from the absolute temperature T_1 to the absolute temperature T_2 on the assumption that the boiler tubes have absorbed all this heat, we have for the total heat absorption:

$$E = 1600 \left[\left(\frac{T}{1000}\right)^4 - \left(\frac{t}{1000}\right)^4 \right] S^1 + WC(T_1 - T_2) \qquad (2)$$

In which E is the total evaporation of a boiler measured in B.t.u. per hour, S^1 is the area of boiler surface, W is the weight of gas leaving the furnace and passing through the setting per hour, and C is the specific heat of the gas.

Relationship of Rate of Heat Transfer.—By means of the integral calculus it may now be found from the above equation that the rate of heat transfer R may be expressed by the equation

$$R = \frac{WC}{S} \log_e \frac{(T_1 - t)}{(T_2 - t)} \qquad (3)$$

This law shows an important relationship of temperatures whereby we may design condenser shells as well as boiler shells to accomplish a maximum rate of heat transfer.

Fig. 64.—A safety water column.

In the Babcock and Wilcox type of boiler the constants involved in heat transference have been quite accurately ascertained. By substituting these constants the above equation is found to reduce to the simple relationship:

$$R = 2.00 + 0.0014 \frac{W}{A} \qquad (4)$$

Necessity for Boiler Accessories.—Since the modern boiler operates under pressures and temperatures far in excess of the tea-kettle and since the quantities of water involved are far beyond hand operation, the necessity for the creation of acces-

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sories to properly care for these increased responsibilities early became apparent in the evolution of steam engineering.

Injector or Pump for Feed Water Supply.—In order to supply the boiler with the necessary water involved in steam generation the injector has made its appearance in some instances, while feed-water pumps are used in other instances.

Since the modern boiler operates at from 100 to 275 lb. pressure per square inch, it is evident that the water must be forced into the boiler, for no ordinary water supply is obtainable to meet such adverse pressures.

FIG. 65.—Stop, check and blow-off valves.

The type of pump most frequently met with for boiler feed purposes is the ordinary duplex double acting pump in which the steam cylinder is made larger than the water cylinder to enable the water to be forced into the boiler at a pressure greater than that of the steam itself. Pumps of this type are very reliable and if chosen of sufficient size so they can be operated at slow speed give excellent satisfaction.

For large power plants the centrifugal pump is coming into favor owing to the small space it occupies and the small attendance required. It is built in four, five or six stages, depending on the water pressure required, and may be driven by either an electric motor or a small steam turbine.

The operation of the injector is accomplished by drawing a certain amount of steam from the boiler and allowing it to attain an enormous velocity. This steam then comes in contact with the feed water supply which at once converts this impinging steam into water. The immense impetus of the outflowing steam and the conversion of the latent energy of this steam into kinetic energy of motion causes the feed water to be sucked in and driven against the check valves of the boiler with such force

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as to overcome the opposing pressure and allow such water to enter as may be needed.

The injector is limited in its field of operation by the fact that the water must be cold enough to condense the injected steam—in other words the injector cannot pump hot water. As the hotter the feed water the more economical the plant the injector is only suitable in plants where there is no hot water available. This condition exists on the locomotive where the injector finds its greatest usefulness.

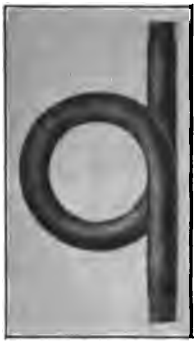

FIG. 66.—Siphon for keeping steam gage cool.

Check and Non-Return Valves.—In order that no water should flow back out through the entrance valve, some means must be provided. Many types of valve are used in practice to perform this function. An illustration of a typical type of check valve is shown in the picture exhibited herewith. Fig. 65.

The Steam Gage and the Water Gage.—In the operation of the tea-kettle the escaping of the steam into the atmosphere readily prevents the possibility of explosion, and the ever watchful eye of the housewife is utilized to see to it that the water supply is sufficient for safe operation. The use of high pressures and inclosed boiler shells makes it imperative in steam engineering to have some means of ascertaining the pressure under which the boiler is operating and to determine the height of the water in the boiler shell. The steam gage meets the former requirement. This type of instrument was described in the chapter on pressures.

To ascertain the water level in the boiler shell, the installation of water columns enclosed in glass tubes makes visible the height of the water in the boiler. The water column is located so that its center is at about the proper height of water in the boiler. The upper end of the column is connected to the steam space of the boiler and the lower end to the water space, so that the water in the column always rises to the same height as the water in the boiler. The bottom of the glass must be a little higher than the lowest level at which it is safe to carry the water to prevent damage by overheating the sheets or tubes, and the top of the glass must be a little lower than the level at which water would begin to be lifted and carried out with the steam. Pet-cocks are provided so that the water column may be cleaned

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of sediment at frequent intervals to insure its safe and accurate operation. Since the ascertaining of the exact water height in the boiler is of such vast importance, three additional pet-cocks called Gage Cocks are usually installed near the water glass. One of these is located above the proper water level, the second at about the water level, and the third below it. Hence, upon trial if the boiler is properly operating, the first should emit colorless dry saturated steam, the second water vapor and the third hot water.

Manholes.—To clean and examine the boiler interior some means must be provided by which access may be had to its interior. On all modern types of boilers will be found man-holes and hand-holes whereby this access may be obtained when occasion arises.

Provision for Expansion.—The excessive temperatures under which a boiler operates and the sudden change from one temperature to another make it absolutely imperative that some means be provided to take care of uneven expansion in its parts. Most boilers on the market do not for this reason allow the boiler shell to rest upon the furnace structure, but on the other hand the boiler is suspended from above and all suspended parts are allowed to swing free with ample clearance between them and the brick-work. The care with which uneven expansion and its disastrous results are provided for makes much for efficient boiler design.

The Mud Drum.—Since all water contains a certain amount of impurities, some space must be set aside for the collection and segregation of these impurities. Such a compartment is known as the mud drum. This is cleansed at definite periods by blowing down the boiler, that is by opening the blow-off valve at the bottom of the mud drum and allowing some of the water to escape from the boiler into the atmosphere.

FIG. 67.—Pop safety valve.

Safety Valve.—All boilers are definitely standardized so that steam generation must not exceed a certain pressure development. To prevent this excessive generation of pressure a safety valve is always installed. These are in general of two types, the one having its outlet to the outside air controlled by a spring set for the pressure desired, the other controlled by a weight and

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lever arm set for the blow-off pressure desired. Since the total pressure required to open the valve equals its area in square inches times the pressure in pounds per square inch the compression in the spring or the weight on the lever may be determined in advance for any desired pressure in the boiler.

Having now a general ground work of boiler shell characteristics in their relation to heat transfer, and bearing in mind the accessories that must accompany the modern boiler, we shall next consider the commercial type of boiler and its classification.

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CHAPTER XV

BOILER CLASSIFICATION

FIG. 68.—A battery of fifteen boilers oil-fired, requiring but two men for their operation.

IN the generation of steam by the tea-kettle the cycle of operations through which the water and steam pass is quite simple. The heat applied at the bottom of the tea-kettle is absorbed by the water along its surface exposed to the heat application. As this heat is absorbed the water is raised in temperature and due to its immediate expansion becomes lighter than the water above it and consequently passes to the top to allow cooler water to descend, which in turn becomes heated and passes to the top to make way for still other water to become heated. This cycle of operations continues and finally evaporation takes place. The steam thus generated passes to the atmosphere without.

In the modern high pressure steam boiler the operation is somewhat more complicated. The water circulation proceeds on the same general principle but since steam generation is the important function and not merely the supplying of hot water as in the tea-kettle, some space must be provided wherein to store the steam that is generated. This is usually accomplished in the space above the water level in the main boiler shell or drum. If superheated steam is to be produced, the saturated steam is conveyed from this space into tubes known as a superheater. These tubes are exposed to the hot furnace gases and the steam passing through them readily absorbs heat, thus super-heating the saturated steam to any temperature determined upon.

The Boiler Drum and Tubes.—It has been mentioned that the tea-kettle is a most inefficient boiler, and so it is. While mechanical stresses and strains involved necessitate the employment of cylindrical shells for boilers, still the boiler itself resem-

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bled in the early days of the steam engine but slight variations from the tea-kettle.

It soon became apparent, however, that the actual surface exposed to the heated gases of the furnace has much to do with efficient steam generation. Hence while the first type of boiler was made in a solid shell, variations from this standard soon made their appearance. Let us now examine some of these types.

Internally and Externally Fired Boilers.—In the earlier type of boiler the fire was kindled beneath the solid cylindrical boiler shell. Such a type became known as an externally fired boiler.

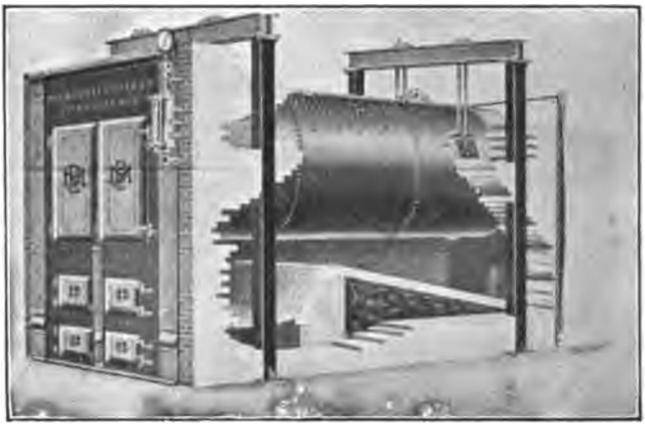

Fig. 69.—A Milwaukee high pressure horizontal tubular boiler with full front and suspension setting.

Later the boiler compartment was hollowed out and the fire kindled inside this hollow space, thus introducing the internally fired type. The locomotive boiler is today an illustration of this type of boiler.

The Return Tubular Boiler.—Another type soon developed wherein the fire was kindled beneath and the flue gases returned to the front part of the boiler through a series of flutings or tubes passing through the main part of the boiler shell. Such a boiler became known as a return flue boiler or return tubular boiler depending upon whether or not the tubes or flutings exceeded six inches in diameter—the flue being the larger diameter and the tube the smaller.

The Fire Tube and the Water Tube Boiler.—A great many types of boilers finally made their appearance on the market in

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some of which the fire passed through the tubes which were surrounded by water in the boiler shell, and in other instances the water passed through the tubes around the external surface of which the heated gases were made to pass. The former became known as fire tube while the latter were called water tube boilers. It is generally conceded that where rapid steaming is required the latter type is far preferable.

It is now universally the custom to use water tube boilers in large stationary power plants. The principal reasons are the following:

1. Small floor space required.
2. Greater safety at high pressures due to the small diameter of the drums and tubes.
3. Greater flexibility so that expansion strains are not injurious.
4. Rapid steaming and sudden change of load more easily accommodated.

Tubular boilers still have their field in small one man plants, where owing to the large quantity of water contained in the shell they maintain a uniform pressure with but little attention.

Tubular boilers of the Scotch marine type are still extensively employed in marine work where owing to the steadiness of the load they have met with great success.

Vertical and Horizontal Types.—Still other classifications are made based upon whether the tubes and boiler shell be in a horizontal or vertical position, the former being called, as one would presume, the horizontal and the latter the vertical type of boiler. As time went on still other boilers appeared which could neither be called horizontal nor vertical but an intermediate classification became necessary.

Let us now examine two types of boiler used in the modern central station in order the more clearly to grasp the fundamentals of boiler design and principles of operation.

Illustrations of Principles of Construction and Operation.—Before proceeding to the brief description of these two types of boiler, the reader must bear in mind that these particular two are picked as best setting forth principles of construction and operation, and not necessarily as a preference for commercial installation. Many types of boiler are today upon the market and in their separate and distinctive features such possess characteristics that must be carefully considered in making a commercial choice. With this understanding let us then proceed to examine these two boiler types of commercial practice.

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The Babcock and Wilcox Boiler.—By a close examination of the illustration shown on page 7, the Babcock and Wilcox boiler is seen to be composed of one or more horizontal shells or drums from which are suspended a series of inclined tubes.

In this type of boiler installation the oil burner is located in the rear of the furnace and the fuel oil is shot forward toward the front. The heated gases then pass upward and around the tubes through what is known as the first pass. At the top of this pass the heated gases envelop the lower half of the boiler shell and are then diverted downward through and around the superheater tubes shown immediately below the drum until the journey through the second pass is completed. At the rear of the furnace wall they are once again diverted upward through the third pass and then after contact with the boiler shell, they are conveyed out through the breeching up the stack or chimney. In this manner the heated gases are brought into intimate contact with the water tubes and efficient steam generation accomplished.

Water Circulation.—The water for steam generating purposes is introduced through the front drumhead of the boiler. It then passes to the rear of the drum, downward through the rear circulating tubes to the sections. Then it courses upward through the tubes of the sections to the front headers and through these headers and front circulating tubes again to the drum where such water as has not been formed into steam retraces its course. The steam formed in the passage through the tubes is liberated as the water reaches the front of the drum. The steam so formed is stored in the steam space above the water line.

The Parker Boiler.—Another type of boiler which is exceedingly interesting, as its operating principles are almost diametrically opposite to the foregoing is that of the Parker boiler.

As seen from the illustration on page 119, the fuel oil is shot from the front of the furnace to the back. The heated gases in their journey toward the rear come in contact with the lower set of tubes and at the rear they pass up through the superheater. They are then deflected back horizontally toward the front, passing parallel along the water tubes. At the front they return again to the rear along the third set of tubes and also along the lower half of the boiler drum above.

Water Circulation.—Water enters the upper set of horizontal tubes from the front without passing first into the boiler drum above. At the rear it is conveyed upward into the drum which

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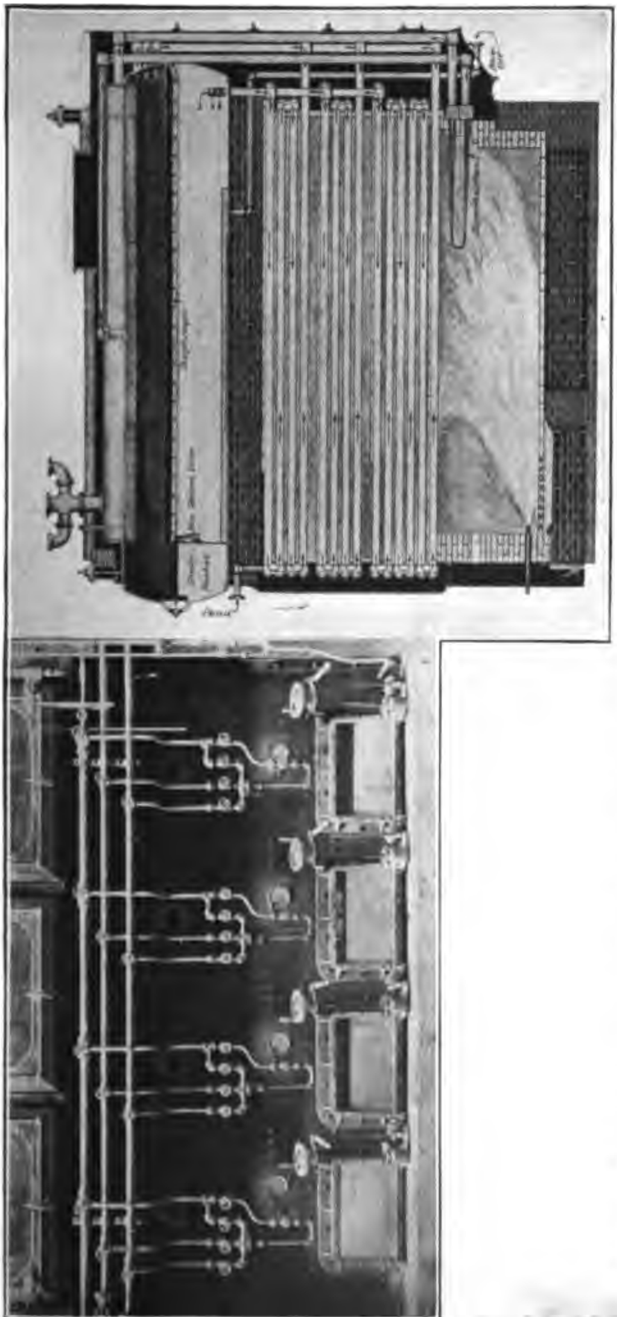

FIG. 70.—Parker boiler at station C in Oakland showing method of fuel oil application and formation of gases in furnace.

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has a longitudinal diaphragm separating the steam section above from the water section beneath. This water having emptied upon the diaphragm in the upper compartment flows down along the diaphragm to the front. At this point it is dropped down into the next section of tubes to be again discharged upward into the upper rear section to flow again down along the diaphragm to the front and again to be lowered into the lowest section of horizontal tubes to return into the diaphragm section above as saturated steam.

It is seen by comparing those two types of steam generation that contrary and opposite theories are used. The first fires the oil flame from the back toward the front, while the latter applies the opposite process. The first admits the water into the drum and then produces a water circulation from the lower sections upward; the latter takes the water first through the top sections and winds up at the lower. The first sets forth the theory of right angle impingement of heated gases against the water tube surface while the latter takes the paralleling flow theory. The remarkable thing about the whole comparison is that both have produced wonderfully efficient steam generating achievements in carefully conducted fuel oil tests on the Pacific Coast.

The Stirling Type.—The Stirling boiler consists of three steam drums connected to one mud drum by means of bent tubes. The bending of the tubes does away with the necessity of using headers and furthermore provides for expansion of the tubes due to change in temperature. As a result this boiler is not only simple in design but very flexible and capable of withstanding a good deal of abuse. The baffles are arranged in such a manner that the gases of combustion travel up the front bank of tubes, down the middle bank and up the rear bank. Stirling boilers are also constructed with an extra baffle wall so as to give the gases a longer travel before leaving the boiler. With this arrangement which is known as a four pass Stirling boiler, the front two or three rows of tubes are exposed to the furnace, the gases passing down the remaining tubes of the front bank, up the midde bank, down a portion of the rear bank and up the remaining rear tubes to the stack. Both the standard three pass and the four pass Stirling Boilers have proved very successful for oil burning.

Other boilers of the general Stirling type are the Badenhausen, the Rust boiler and the Erie City vertical boiler.

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The Heine Type.—The Heine boiler is a horizontal water tube boiler similar to the B. & W. boiler except that instead of having separate headers the tubes are all expanded into a single water leg at the rear and another at the front. These water legs have large flat surfaces which have to be strengthened by stay bolts. Owing to the fact that all of the tubes are connected to the same water legs, this boiler is not as flexible as the other two types described above. The Heine boiler is usually provided with

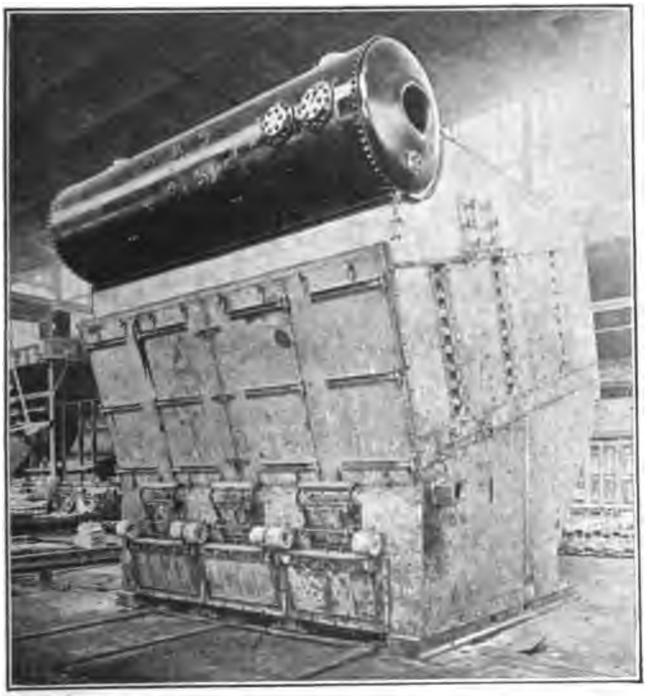

Fig. 71.—The B. & W. marine type of boiler—front view.

Water tube boilers for marine service are built as modifications of both the B. & W. and Heine type, the tubes being shorter and smaller in diameter than is the case in stationary boilers. These boilers are encased in steel, lined with light insulating material inside, instead of being set in brick. They are constructed of the highest grade of forged steel. Marine boilers frequently carry pressures as high as 300 lb. per sq. in.

horizontal baffles so that the gases of combustion pass first to the rear of the boiler and then forward among the tubes and then back again. With this arrangement of baffling the oil burner introduced through the front wall is very successful. Other boilers of the Heine type are the Keeler and Edgemoor boilers.

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Marine Boilers.—For mercantile marine service the standard boiler for many years has been the Scotch marine boiler which is a fire tube boiler consisting of a large shell within which are placed corrugated furnaces, a combustion chamber at the rear and tubes running forward from the combustion chamber to the front of the boiler, whence the gases pass through the uptakes to the smokestacks. Owing to the large size of the shell, Scotch boilers are made of excessively thick steel and consequently are

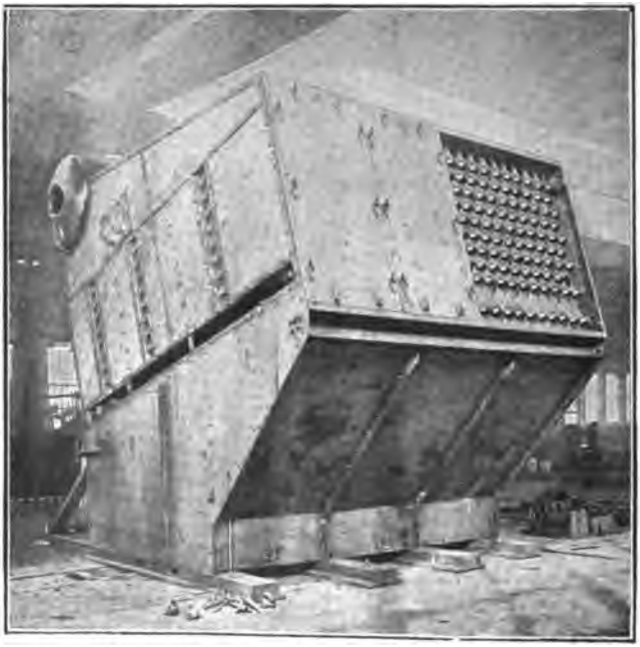

FIG. 72.—The B. & W. marine type of boiler—rear view.

entirely lacking in flexibility. They are, therefore, liable to give trouble due to expansion strains from change in temperature and are successful only where the load is absolutely steady as it is on ordinary merchant ships.

In the navy water tube boilers are used exclusively and these are coming into use more or less in the mercantile marine as well.

Water tube marine boilers are built as modifications of both the B. & W. and the Heine types, the tubes being shorter and smaller in diameter than is the case in stationary boilers and the

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boilers being encased in steel, lined with light insulating material inside, instead of being set in brick.

For torpedo boat destroyers and other small high speed craft boilers of the Thornycroft type are used, which consist of a large number of very small diameter tubes expanded into upper and lower drums somewhat similar in general type to the Stirling stationary boiler. These boilers are extremely light and are rapid steamers which are necessary characteristics of boilers for high speed boats.

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CHAPTER XVI

FUEL OIL AND SPECIFICATIONS FOR PURCHASE

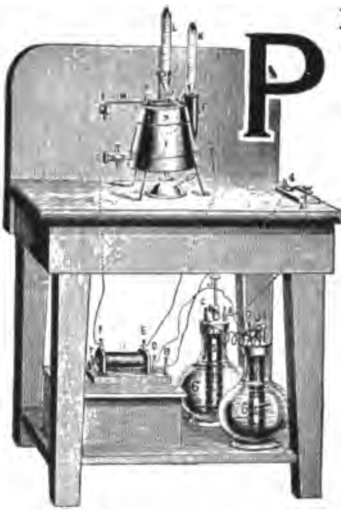

FIG. 73.—The Saybolt electrical equipment for flash and fire tests.

PETROLEUM has been known in the United States from prehistoric times. It is certain that the mound builders had wells from which petroleum was obtained. These are still in existence along with the most modern of our own times.

Petroleum was used as a medicine by many tribes of Indians. It was supposed to have many magical as well as medicinal properties. Its inflammable nature seems also to have been known.

No use was discovered for petroleum other than as a medicine until in 1852, when a chemist, by the name of Kier, bethought himself of distilling it and extracting from it the more volatile portions. The American people took readily to the use of these oils as illuminating agents from the fact that for some time previously the mineral oils, extracted from lignites and anthracites, according to the process of Sellegries, the Swiss chemist, were in current use.

Enormous Consumption of Fuel Oil in the Industries.—The use of crude petroleum as a fuel for steam generation and power production has now an established position in all parts of the industrial world. Especially is this true of the Pacific Coast and in the southwestern section of the United States where the enormous yield of this product in Oklahoma, Texas and California now constitutes an ever-increasing factor in the total production of the world. Indeed, California alone with her yield of over one hundred million barrels in 1919 produced over 25 per cent. of the world's output.

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At its first incipiency it was thought that the probable production of crude petroleum would be limited to but a few years. Due to this factor many power plants on the Pacific Coast were constructed so that an easy change over to operation by coal could be made should this time ever arrive. It is now recognized by many that the probable yield of oil will last as long as the coal fields of the world. Hence this uncertainty is largely dispelled in the industrial production of power.

Advantages of Crude Petroleum as a Fuel.—Oil has many distinct advantages over coal. Due to the simple mechanisms that are involved the cost for handling fuel oil is far less than for coal. By the elimination of stokers an important labor item is found unnecessary. Again for equal heat value oil occupies much less space than coal. Hence for ocean-going vessels it is especially applicable. Combustion too is more perfect so that the quantity of excess air required is reduced to a minimum. The furnace temperature may be kept practically constant as the furnace doors need not be opened for cleaning or working the fires. Smoke may to a large measure be eliminated with the consequent cleanliness of heating surfaces. Again, the intensity of the fire is subject to delicate regulation and sudden load fluctuations are easily handled. Oil does not disintegrate or lose its calorific value when stored. In the boiler room the cleanliness and freedom from dust and ashes results in a saving in wear and tear in machinery. Hence it is clearly evident that the efficiency and the steaming capacity of a boiler, oil fired, is increased in a marked manner.

The disadvantages of fuel oil are of comparatively small moment. For this reason wherever oil can be obtained at a reasonable figure as compared to the prevailing market price of coal it has attained a marked popularity in steam generation and in the industries.

Let us then look into some of the physical properties of this new and important source of heat generation.

Liquid Fuels Classified.—Petroleum is practically the only liquid fuel sufficiently abundant and cheap to be used for the generation of steam. There are three kinds of petroleum in use, namely, those yielding on distillation paraffin, asphalt and olefine. To the first group belong the oils of the Appalachian Range and the Middle West of the United States. These are a dark brown in color with a greenish tinge. Upon their distillation such a variety

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of valuable light oils are obtained that their use as a fuel is prohibitive because of price. To the second group belong the oils found in Texas and California. These vary in color from reddish brown to a jet black. Since they are used extensively as a fuel in the United States, our discussion in this chapter shall largely be concerned with this class of oils. The third group compr ses the oils from Russia, which like the second group are used largely for fuel purposes.

Physical and Chemical Properties of Oil.—Mineral oils as found in nature, are a mixture in indefinite proportions of several combinations of hydrogen and carbon designated as hydrocarbons. Oxygen and sulphur are found in very small amounts. Nitrogen is found in a smaller proportion than the latter.

On account of the complexity of their composition, mineral oils differ considerably both physically and chemically.

Odor and Color.—Oil is generally found in a very fluid condition in North and South America, while in Russia and East India it is found in a very dense and syrupy condition. They all possess a characteristic odor while their color varies from amber or greenish yellow to dark brown. By reflection they are all greenish.

Effect of Heat.—Heat will separate the different hydrocarbons successively according to their volatility and cause them to dissociate at higher temperatures. Low temperatures will solidify these products, the highest freezing at a lower temperature.

Density of Various Oils.—The density varies from 0.765 to 0.970 compared with water at 4°C., as found in nature (crude). Distillates will be much lighter.

DENSITIES OF OILS

Origin of crude	Specific gravity
Persia	0.777
East Indies	0.821
Kyouk-Phyon (Burma)	0.818
California	0.960
Pennsylvania	0.850
South America	0.852
Russia	0.836
India	0.955
Terra-di-Lavors (Italy)	0.970

Physical Properties of California Oils.—We shall now consider as a typical example a sample of California crude petroleum taken from an average of forty samples drawn from the Kern River oil field by representatives of the U. S. Bureau of Mines.

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The specific gravity or density of fuel oil is an important factor to be known and is the ratio of the weight of an oil sample as compared with the weight of an equal volume of water. The average oil sample is found to have a specific gravity of 0.9645, which on the Baume scale at 60°F. is 15.16°. Hence, the average gallon of fuel oil weighs 8.03 lbs.

The determination of the gravity of fuel oil and the relationship of specific gravity with gravities expressed on the Baume scale are of such importance that a subsequent chapter has been set aside for detailed discussion and analysis.

The Calorific Value of Fuel Oil.—In steam boiler economy the heat producing value of the fuel per pound consumed in the

Fig. 74.—Laboratory equipment for fuel oil testing.

In the gathering of fuel oil data for boiler tests the three things to be ascertained accurately are the specific gravity, the moisture content, and the calorific value of the oil sample. The principal pieces of apparatus necessary are the Westphal Balance, the chemist's scales, a Parr calorimeter, and a still with their accessories as shown. In the text of this article these physical characteristics of fuel oil are set forth. In later discussions the laboratory procedure in order to ascertain each of these points will be discussed in separate chapters.

furnace is of utmost importance. The average sample of Kern River oil generates or gives out 10,307 calories per gram, which transferred to steam engineering units is found to be 18,553 B.t.u. per pound or 148,980 B.t.u. per gallon of oil.

Oil, like water, requires the actual absorption of a large quantity of heat in its conversion into the gaseous state. Indeed the latent heat of evaporation for fuel oil is approximately

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130–150 B.t.u. per pound under atmospheric pressure, as compared with 970.4 for the latent heat of evaporation of water as set forth in previous discussions. Hence, the actual heat given out by the average sample above referred to is approximately 18,700 B.t.u. per pound, but since we must gasify the oil to make use of its heat generating characteristics in the furnace the net value of 18,553 is solely of commercial importance.

The determination of the calorific value of fuel oil and the many computations involved are of such vast importance that several chapters have been set aside for future discussions of these various factors.

The Flash Test and the Burning Point of Oil.—The **flash test** of an oil is the temperature at which it gives off inflammable vapors. For the purpose of safety in handling, fuel oils should not give off inflammable vapors below 150°F. The flash point of an oil is determined by heating the oil in a vessel adjacent to which is a small flame. When the oil has been heated to a point where vapor rises and ignites from the flame, this temperature is called the flash point. The flash point of the average California oils is 108°C. or 226.4°F.

The burning point of oil is the temperature at which its ingredients will permanently ignite. This is determined by continuing the heating of the oil after the flash point has been ascertained until the "flash" becomes permanent, that is, until the oil ignites and continues to burn quietly. For the average Kern River oil sample the burning point is found by the open cup test to be 130°C. or 266°F.

Viscosity.—Some oils are more fluid or mobile than others. All are familiar with the difference between "cold molasses" and "hot molasses." And so in oil flow we have a similar phenomenon. This tendency for the particles of oil to cohere to one another is known as viscosity. Viscosity is determined by measuring the time it takes oil to flow through a standard sized tube under standard conditions. On the so-called Engler scale the average viscosity of Kern River oil at 20°C. is found to be 915.6. The viscosity is very materially lessened as the temperature is increased. Hence at once is seen the advantages of oil heating both for efficiency in transmission through long pipe lines, and for feeding the oil to the burners. In power plants the oil is heated to a temperature of 160°F. before reaching the burners.

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Moisture.—All oils have a certain quantity of moisture present either in a free state or in the form of an emulsion, and its presence is always a hindrance to the full development of the heat producing qualities of the oil. Since this is a matter of great importance, the methods used in the quantitative determination of moisture present in fuel oil will be set forth in a subsequent chapter. The average Kern River sample contains about 0.5 per cent. moisture. Hence the actual fuel oil ingredient is 99.5 per cent.

Sulphur, Gas and Other Ingredients.—All oils have a certain quantity of sulphur present. This sulphur has a heat producing quality, yet its deleterious effect in producing obnoxious gases and the corroding effect it has on the boiler tubes and other metallic parts makes a certain excess of sulphur most undesirable in the use of fuel oil. Sulphur in burning forms sulphur dioxide, SO_2, which, on combining with water forms sulphurous acid. Ordinarily corrosion does not occur as long as the temperature is above 212°F. Hence corrosion of the boiler tubes is negligible as long as the boiler is kept in operation, but if a boiler is used for standby purposes and is kept shut down a good deal of the time there may be active corrosion of the tubes, and drum shells. Steel stacks are subject to corrosion from this cause, especially the upper portions exposed to the weather, where the cooling effect of the metal causes condensation of the moisture in the gases. The average Kern River oil sample contains 0.83 per cent. sulphur. There is no gasoline ingredient found in this oil sample. On the other hand, refined lamp oil appears to the extent of 6.6 per cent. and refined lubricants to the extent of 39.2 per cent. The refining losses are 5.9 per cent. and distilling losses, 0.5 per cent. The commercial asphaltum present is 47.3 per cent., thus indicating why California oils are known as possessing an asphaltum base.

Specifications for the Purchase of Oil.—In the above discussion of the physical properties of fuel oil it is seen that the flash point, burning point, viscosity, heating value, moisture content, sulphur content, and other characteristics are fundamentally concerned in the commercial evaluation of crude petroleum. "Below are given "Notes on Specifications for the Purchase of Fuel Oil" by J. M. Wadsworth, published by the U. S. Bureau of Mines in the handbook entitled, "Efficiency in the Use of Oil Fuel." The points set forth are of fundamental importance for

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the economic use of fuel oil in all steam boiler practice and the reader should carefully bear them in mind.

NOTES ON SPECIFICATIONS FOR THE PURCHASE OF FUEL OIL

1. In the purchase of fuel oil by large users, all buying should be done under competitive bids. In determining the award of a contract consideration should be given to the quality of the fuel offered by the bidders, as well as the price, and should it appear to the best interest of the purchaser to award a contract at a higher price than that named in the bid or bids received the contract should be so awarded.

2. Each bidder should be required to submit an accurate statement regarding the fuel oil he proposes to furnish. This statement should show:

(*a*) The commercial name of the oil.

(*b*) The name or designation of the field from which the oil is obtained.

(*c*) Whether the oil is crude oil, a refinery residuum, a distillate, or a blend.

(*d*) The name and location of the refinery, if the oil has been refined at all.

3. The fuel oil should be delivered f.o.b. cars, vessels, tanks, or tank wagons, according to the manner of shipment or delivery at such places, at such times, and in such quantities as may be required during the fiscal year ending ——.

3a. Minimum and maximum weekly or monthly deliveries should be specified.

4. Should the contractor, for any reason, fail to comply with the writtten order to make delivery, the purchaser is to be at liberty to buy oil in the open market and charge against the contractor any excessive price, above the contract price, of the fuel oil so purchased.

5. It should be understood that the fuel oil delivered during the term of the contract shall be of the quality specified. The frequent or continued failure of the contractor to deliver oil of the specified quality should be considered sufficient cause for the cancellation of the contract.

ESSENTIAL PROPERTIES OF THE OIL

6. Viscosity.—Fuel oil, as regards viscosity, may be divided into two general classes, namely:

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Class 1. Asphaltic base crudes, residuums, or other oils which require heating facilities to reduce the viscosity in order that the oil may be handled by the storage and burning equipment.

Class 2. Oils of a sufficiently low viscosity to make heating equipment unnecessary.

In general, an oil of Class 1 should not have a viscosity above 2000° Engler at 60°F. Oils of a higher viscosity than this can be used at plants provided with special equipment. It is imperative that oils of this class be heated to a temperature at which they have a viscosity of 12° Engler or lower before they reach the burner, in order to obtain proper atomization. It is desirable that this viscosity be obtained at a temperature below the flash point of the oil, in order to minimize fire hazards and to insure uniform feed to the burner.

For an oil of Class 2, 12° Engler at 60°F. is the approximate maximum viscosity permissible.

Method of determination: Viscosity should be determined with a standard Engler instrument according to the recognized method of manipulating this viscosimeter. Other standard viscosimeters may be used in special cases and their readings converted to Engler degrees by means of recognized tables or formulas.

7. Flash Point.—In general it is desirable that the flash point of Class 1 oils should not be below 140°F., and that of Class 2 oils not below 120°F. It should be noted that for Class 1 oils, specifications for flash point are contingent upon viscosity requirements as well as upon general considerations for safety requirements and evaporation losses.

Method of determination: Pensky-Martens closed-cut tester manipulated according to standard procedure.

8. Specific Gravity.—Specifications for specific gravity are superfluous. In case oil is purchased by weight and measured by volume an accurate determination of its specific gravity is essential.

Method of determination: By specific gravity balance, pycnometer, or hydrometer. If conversion of Baume readings to specific gravity is necessary it is essential that the Baume hydrometer be accurate and that the proper modulus for this instrument be used. Specific gravities should be reported at 60°F. compared to water at 60°F. If they are determined at other temperatures the temperature corrections given in Bureau of Standards Circular 57 should be used.

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9. Impurities.—The oil should not contain more than 2 per cent. by volume of moisture and sediment. Proper deductions should be made from all oil deliveries for the impurities contained therein so that the oil purchased shall be pure oil.

Method of determination: A definite volume of the oil sample should be thoroughly shaken or "cut" with an equal volume of gasoline of a specific gravity not greater than 0.74, and centrifuged. An appropriate tube that goes with a special machine is commonly used for this purpose. Centrifuging should be continued until there is a clear line of demarcation between the water and sediment and oil in the bottom of the tube, and until a constant reading of water and sediment is obtained. From this reading the percentage by volume of water and sediment is computed. If the oil under consideration has a specific gravity greater than 0.96, one volume of oil to three volumes of gasoline should be used rather than equal volumes. When there is a question that the gasoline used for thinning the oil in making this determination renders insoluble certain of its fuel constituents, then mixtures of gasoline and carbon disulphide, or of gasoline and benzol may be used for "cutting," providing the specific gravity of such mixtures is not greater than 0.74. If, after continued centrifuging, a clear line of demarcation between the impurities and the oil is not obtainable, the uppermost line should be read. If this procedure proves unsatisfactory, 100 cc. of the sample may be distilled with an excess of hydrocarbons saturated with water and having boiling points slightly above and below that of water. Distillation is continued until a volume equal to the volume of hydrocarbons added has been distilled over into a graduated tube. The water in the oil is thus distilled over and readily collects at the bottom of this tube, where the percentage may be read off. The percentage of sediment in the oil may then be determined on the sample remaining in the distilling flask by "cutting" it with gasoline and centrifuging. The percentage of water obtained in the tube added to the percentage of sediment gives a total percentage to be deducted for moisture and impurities.

10. Sulphur Content.—Appreciable sulphur content in a fuel oil is objectionable. However, a content of 4 per cent. or less is not sufficiently objectionable to cause the rejection of a fuel oil for general purposes. (In general, experiments in burning fuel oils of various sulphur content have shown that the corrosive

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effects on the boiler tubes or heating surfaces are negligible. However, with steel stacks and low stack-gas temperatures, considerable corrosion in the stack has been noted. In handling these oils, prior to burning, the corrosive action of the sulphur on steel storage tanks, piping, etc., is quite apparent and should be considered. If the oil is to be used for special metallurgical or other purposes where sulphur fumes are decidedly objectionable, it is necessary to specify a limiting figure for the sulphur content of the oil.)

Method of determination: Complete combustion in a bomb by means of oxygen or sodium peroxide, the sulphur being weighed as barium sulphate.

11. Calorific Value.—A standard of 18,500 B.t.u. to the pound of pure fuel oil is a good figure to be taken as the basis, if the fuel oil is to be purchased on calorific determinations. A bonus may be paid for calorific value in excess of this figure and deductions made if the heating value of the fuel is below 18,500 B.t.u. per pound.

Method of determination: Any bomb calorimeter of recognized accuracy.

12. Methods of Sampling.—The accuracy of these different tests depends upon the care with which an average representative sample of the fuel oil delivery has been taken, and the importance of obtaining such a sample can not be overestimated. Top, middle, and bottom samples should be taken with a standard "car thief" and these samples should be combined and thoroughly mixed to form one sample for car deliveries. Where oil is received in tanks or reservoirs the swing pipe should first be locked at a position well above the level of the water and sediment usually found in the bottom of such tanks. Tanks should be sampled every foot for the first 5 feet above the bottom of the swing pipe, and at 5-ft. intervals from there to the surface of the oil. This sampling should be done with a standard tank thief, the samples "cut" individually, and deductions for impurities made on the separate volumes which these samples represent. If the tank is a large one, it should be sampled through at least two hatches. In receiving large deliveries of the more viscous oils it is necessary to take many samples in order to insure fair and average impurity (M. and B. S.) deductions. This is because water and sediment do not readily settle out of such oils.

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13. General Specifications can not be Drawn to Advantage for Fuel Oils.—Individual conditions and requirements at the points of consumption influence to a large degree the specifications for viscosity, flash point, and sulphur content. Definite specifications can be drawn for a fuel oil which will meet practically all requirements, but it can readily be seen that such specifications will exclude much of the fuel oil now available, and for most purposes the requirements need not be severe. Hence, it is advised that in purchasing fuel oil the individual requirements be studied and that as lenient specifications as possible be written which will insure an oil that will be satisfactory for the conditions for which it is intended.

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CHAPTER XVII

FUEL OIL PRICES AND OIL PRODUCTION

By October 1920, as this Second Edition of "Fuel Oil and Steam Engineering" is about to go to press, the authors have come to the realization of the fact that the price of California fuel oil has more than doubled since the first edition of this book appeared just two years ago. Since, then, the constantly depleting storage of oil by the large production companies is making marked inroads into the possible supply of fuel oil in the near future, a discussion of this is of vital interest to the subject matter of this

FIG. 75.—Water softening plant at the Sierra and San Francisco Power Company's 27,000 kw. oil burning station. The use of pure water free from scale forming matter is the first requisite toward keeping boilers clean.

book in that economy production in the power plant depends directly upon the prices of this commodity prevailing in the open market and the ability of the operator to obtain fuel supply.

For the following able discussion of this matter we are indebted to J. E. Woodbridge, formerly chief engineer of the Sierra and San Francisco Power Company.

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The exercise of prophecy in the field of prices is beset with pitfalls, as has been shown by the totally unexpected price revolution of the last five years. However, as all business must be planned in part on future price probabilities, we are obliged to make the best attempt we can at an estimate of their trend.

In the matter of fuel in the state of California, the most pertinent data for prediction as to future prices are the trend of the last few years, including the recent history of production and consumption, with other relevant information, such as the influence of the war, probable future additional sources and markets.

PRICE FLUCTUATION

Except for the continuity of the thought, we need hardly inform readers of this book that the price of delivered fuel oil in California has advanced during the last five years from approximately 60¢ to approximately \$1.85 per barrel, the two approximations covering slight variations with locality. The figure which directly influences the production of oil in California, and constitutes the basic element of all prices to consumers is the market price offered by the large distributing organizations to the producers in the oil fields. The price offered by the Standard Oil Company of California for fuel oil, that is oil having a specific gravity between 14° and 17.9° on the Baume scale, has gone through the following changes during the last six years.

On October 3, 1914, this price dropped from 40¢. per barrel to 37½¢.

June 7, 1915	\$0.32½
Oct. 26, 1915	0.37½
Nov. 20, 1915	0.40
Dec. 28, 1915	0.43
Feb. 2, 1916	0.48
Feb. 16, 1916	0.53
April 1, 1916	0.58
July 7, 1916	0.63
Sept. 20, 1916	0.68
Nov. 21, 1916	0.73
May 11, 1917	0.78
June 7, 1917	0.88
June 28, 1917	0.98
May 1, 1918	1.23
Mar. 17, 1920	1.48

The price doldrums of 1915 were due to the large number of gushers brought in during the years 1913 and 1914, which resulted in a production so much in excess of consumption that a stock of approximately 60,000,000 barrels of oil (nearly a year's consumption) was accumulated early in 1915. The price advances during the subsequent three years were largely due to war demands and the depreciation of our currency,

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which made oil field development difficult and expensive. In particular, the large jump of May 1, 1918, was made at the request of the National Fuel Administrator. The price advance and other factors so stimulated production and controlled consumption that stock increased during a portion of 1918 and 1919 by 3,000,000 barrels. Many who have not looked further into the fuel-oil situation have reasoned, from the above, that we may rest easy as to the price of this raw material in the immediate future. However, it does not behoove power companies to "sleep at the switch." Let us, therefore, look a little deeper into the prospect.

DECREASING SUPPLY

On the occasion of the last price change of March, 1920, the California Railroad Commission criticized the Standard Oil Company of California, in response to which President Kingsbury of that company answered in part as follows:

"The Pacific Coast supply of fuel oil and of petroleum products is rapidly approaching exhaustion. Since May 1, 1915, crude oil stocks in California have decreased from over 60,000,000 barrels to 28,738,921 barrels on March 1, 1920.

"The available supply of crude oil in stock is today less than 13,000,000 barrels.

"The balance of the stocks are taken up in the factor of safety of 10,000,000 barrrels, which the Petroleum Committee of the State Council of Defense found essential to the safety of Pacific Coast industries, and in the oil in pipe lines and tank bottoms which the same committee estimated at 6,000,000 barrels not available for use.

"At the present rate of consumption and of production the available stocks will be exhausted in about twelve months, at which time consumers of California fuel oil will be cut off from between 25,000 to 30,000 barrels per day."

"In 1918 the average daily consumption was 279,576 barrels; in the last half of 1919 it was 292,278 barrels; in January, 1920, 301,000 barrels and in February, 1920, 304,120 barrels.

"Superimposed on these figures is the fact that 7,000 barrels of fuel oil a day which formerly went to Arizona from California are now supplied from Texas and Mexico. The existing shortage, therefore, has developed in face of the fact that 2,500,000 barrels of fuel oil a year have been restored to the California supply.

"Added to these considerations are the demands of the navy and the United States Shipping Board.

"The former estimates its 1920 requirements on the Pacific Coast at 2,950,800 barrels, against 1,532,650 barrels in 1919. The Shipping Board has invited bids for 1920 for 4,000,000 barrels.

"The United States Shipping Board is establishing oil fueling stations

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throughout the world, and will supply these points from the cheapest markets. Thus California will be drawn upon or spared in the relation that California prices bear to prices elsewhere. Even at the new price this oil market is lower than many other points with which this market is in competition.

"The inevitable result will be that the Pacific Coast will be further drained of its supply by buyers who seek the cheapest market.

"Shipping Board vessels already have sought cargoes of fuel oil which formerly were obtained in Mexico, so that the Mexican cargoes could be released for the Atlantic coast.

"There is further the fact that improved refining processes will increase the volume of refined products extracted from crude oil and thus reduce the resulting residuum for fuel oil.

"The company is now installing at a cost of $10,000,000 new processes by which it is estimated that more refined products, incuding gasoline, will be recovered from crude oil in such quantities that the company's production of fuel oil within a year will be necessarily lessened about 30 per cent. or 20,000 barrels a day."

Further light on the probable price trend is obtainable from the following table of production and consumption of crude oil of all gravities in and from the California fields during the last seven years.

Year	No. producing wells, Dec. 31	Production	Average daily production per well	Consumption *	Total stocks Dec. 31
1912	5,626	90,074,000	44	86,500,000	46,698,000
1913	5,870	97,867,000	46	96,695,000	47,870,000
1914	6,106	103,624,000	46	92,968,000	58,526,000
1915	6,532	89,567,000	38	90,946,000	57,147,000
1916	7,333	91,822,000	34	104,933,000	44,036,000
1917	8,053	97,268,000	33	108,854,000	32,450,000
1918	8,606	101,638,000	32	102,045,000	32,043,000
1919	9,127	101,222,000	30	102,785,000	30,480,000

* Consumption is here taken as production plus withdrawals from storage or less additions to storage, as the case may be.

Examining the column headed "production," the gusher peak of 1914, amounting to over 103,000,000 barrels from approximately 6000 wells will be noted. A significant fact is that less oil was produced in 1919, in spite of a price three times as high as that of 1914. In the meantime the number of wells has in-

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creased 50 per cent. The average production per well, of today, has gradually fallen from that of the gusher period to 30 barrels in 1919. This means that a large number of wells are approaching the marginal production below which it will not be profitable to pump them, unless the price advances. The average rate at which the production of California wells declines is fairly well established for the first few years of their life. The following table gives the average production of California oil wells by years of life in percentages of production of the first year.

1st year	100
2nd year	68
3rd year	51
4th year	41
5th year	33

Obviously, as the fields grow older, and the gushers become only a recollection, the wells become more crowded, the gas pressure goes down, and initial production of the new wells drilled becomes less and less year by year, a continuously expanding drilling program must be carried out to keep up a constant supply. If, on top of that, the demand increases, conditions are still more strained for the old law of supply and demand.

The future price trend would, of course, be checked in its upward career, by the development of new oil fields within reasonable shipping distances. While no one can predict positively that new oil fields will not be developed, it is highly improbable that the production of any such fields, on the Pacific Coast, will be sufficient materially to lower the California price. Freight rates from the mid-continent fields prohibit importation from that source, even if mid-continent prices were lower than our own. Mexican oil is still further out of reach on account of freight rates by any route. The only other source in sight is Colombia and Venezuela, but the probable tanker rate of a dollar or more for the 3000-mi. haul will prevent this source from ever giving us cheap oil, even if the price at the source is not held high, as it undoubtedly will be by East-coast demands and by the large volume of shipping through the Canal which will in future utilize oil fuel for propulsion.

A NECESSARY DEVELOPMENT

The point we wish to make from the above data is that power development, as in the ordinary steam station, at an average

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efficiency of 185 to 200 kw-hr. per barrel, will cost for fuel alone at least one cent per kw-hr. with fuel prices of $1.85 upwards. This figure, added to other steam operating costs and all fixed charges, makes the total cost of power so developed much higher than that of hydro-electric power, even at present high costs of money and construction. This being the case for steam-turbine plants of reasonably fair to good efficiencies, it is much more true of the steam locomotive with its wasteful boiler and non-condensing reciprocating engine.

In other words, the cost of the fuel oil alone used in the steam locomotives in California and neighboring states, based on its value at the market (amounting to approximately $50,000,000 per annum which will increase year by year until this fuel can with difficulty be obtained at all) will in time compel the steam railroads to seek other methods of operation, chief and most promising of which is electrical. Disregarding all other reasons and arguments for electrification, the cost of fuel is going to force it.

Coal is, of course, a possible substitute. On a thermal-unit-cost basis, Utah coal is now on a par with oil on the Sierra grades. Much as the dyed-in-the-wool steam railroad man mistrusts electrification, we believe he would have a greater dislike for the use of coal where he has been accustomed to oil, on account of its inconvenience in firing large furnaces. Pulverization offers a possible means of overcoming these difficulties but involves many complications over the use of oil.

In view of the inability of the power companies to meet their present commercial demands, and the apparently insuperable financial program involved in meeting future demands, a large railroad load cannot be anticipated with any great degree of satisfaction. The solution, however, is obvious—namely, a railroad power rate which will finance the necessary development.

The hope for the situation seems, then, to be in the electrification of the steam railroads, and thus the conservation of fuel oil which will retrieve the depleting of the stocks now on hand will be brought about. The entire matter is of the utmost concern and its varying characteristics from month to month will be followed by the public and particularly by the steam power plant engineer with the keenest interest.

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CHAPTER XVIII

THE SAFE OPERATION OF STEAM BOILERS

Many fatal accidents both to life and property have happened due to foolhardy methods in design and operation of the steam boiler. This early became so apparent that rigid governmental inspection of boiler operation was insisted upon. To aid in systematic inspection the Department of Commerce and Labor at Washington has issued general rules and regulations for such supervision under Form 801 entitled Steamboat Inspection Service. Many insurance companies have, too, put into force rigid rules of inspection to safeguard their interests in assuming risks. The most complete publication on the subject, however, is to be found in the recently published report of the Boiler Code Committee of the American Society of Mechanical Engineers, entitled: "Rules for the Construction of Stationary Boilers and for Allowable Working Pressure." These rules have been adopted by law in a number of States, including California where they have been incorporated in the Safety Orders of the State Accident Commission.

In the discussion taken up in this chapter only fundamentals will be considered. The thorough mastering of these fundamentals will, however, enable the reader to understandingly read the deeper discussions alluded to above.

The Inspection Tests Involved.—The testing of the water and steam gages, the checking of fittings and appliances, and the trying out of the safety valves and other accessories constitute, of course, important details of boiler inspection. The most important feature, however, is to ascertain by computation the maximum allowable working pressure that may be safely put upon the boiler. After this maximum allowable pressure is ascertained the boiler is subjected to a hydrostatic pressure test by filling the boiler completely with water and then pumping enough additional water into it to raise the pressure to the desired point. This apparatus is held under proper control and the total pressure put upon the boiler is one and one-half times the maximum allowable working pressure.

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Thus if the maximum allowable working pressure on a boiler is 160 lbs. per sq. in. above the atmosphere, the test pressure to be applied should be 240 lb. per sq. in.

Many carefully compiled instructions have from time to time been issued by various boiler makers, inspectors, and others interested in economic and safe operation. The instructions compiled by J. B. Warner chief inspector of the San Francisco

FIG. 76.—An inspector's testing and proving outfit.

Here is a typical outfit for boiler and power plant inspectors. It consists of a standard test gage, a screw test pump, a gage hand puller, a hand set and other useful conveniences.

department of the Hartford Steam Boiler Inspection and Insurance Company are especially good, and largely the ideas appearing in the following lines come from this source:

Preliminary Precautions.—Whenever going on duty in the boiler room, find out, first of all, where the water level is in the boilers. Never lower nor replenish the fires until this is done.

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Make sure that the gage glass and gage cocks, and all the connections thereto, are free and in good working order. Do not rely upon the glass altogether, but use the gage cocks also, and try them all, several times a day.

Before starting up the fires, open each door about the setting and look carefully for leaks. If leaks are discovered, either then or at any other time, they should be located and repaired; but cool the boiler off first. If leaking occurs at the fore and aft joints, the inspecting company should be notified at once. This is important, whether the attendant considers the leakage serious or not; and it is especially important when the boiler has a single bottom sheet, or is of the two-sheet type.

Fig. 77.—A portable boiler test pump.

After the maximum allowable working steam pressure for the boiler has been computed, the boiler is then submitted to a hydrostatic test of one and one-half times this allowable pressure. The above apparatus is especially adapted for those having frequent occasion to make hydrostatic tests of boilers.

When a boiler has been emptied of water, do not fill it again until it has become cold.

In preparing to get up steam after the boiler has been out of service, be sure that the manhole and hand-hole joints are tight. Do not use gaskets that are thin and hard.

Vent the boiler in some way, first, to permit the escape of air. Then fill the boiler to the proper level, open the dampers, and start the fires. Start them early so as to have the pressure up the required hour, without forcing.

Ventilate the setting thoroughly before lighting the fire. Never turn on the fuel supply when starting up without first

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placing in the furnace a lighted torch or a piece of burning waste to ignite the fuel instantly.

Connecting up Boiler Units.—In firing up a boiler that is to be connected with others that are already in service, keep its stop-valve closed until the pressure within the boiler has become exactly equal to that in the steam main. Then open the stop valve a bare crack, and slowly increase the opening until the valve is wide open. The complete operation should occupy two minutes or more. Close the valve at once if there is the slightest evidence of any unusual jar or disturbance about the boiler. See that the steam main to which the boiler is to be connected is thoroughly drained before the valve is opened.

Low Water Encountered.—In case of low water, immediately shut off the oil supply at the burners. Do not turn on the feed under any circumstances, and do not open the safety-valve nor tamper with it in any way. Let the steam outlets remain as they are. Get your boiler cool before you do anything else.

Avoid Making Repairs Under Pressure.—No repairs of any kind should be made, either to boilers or to piping, while the part upon which the work is to be done is under pressure. This applies to calking, to tightening up bolts under pressure, and to repairs of any kind whatsoever.

The safety-valve must not be set at a pressure higher than that permitted by the insurance company's policy. Try all safety-valves cautiously, every day. If the actual blowing pressure, as shown by the gage, exceeds the pressure at which the valve is supposed to blow, inform the office immediately, so that prompt notice may be sent to the company. The safety-valve pipe should never have a stop-valve upon it.

Removal of Sediment.—To remove sediment from the bottom of the boiler, open the blowoff valve in the morning, or before the circulation has started up. The valve should be opened wide for a few moments, but it should be opened and closed slowly, so as to avoid shocks from water-hammer action. When surface blowoffs are used, they should be opened frequently for a few minutes at a time.

In case of foaming, check the draft and shut off the burners. Shut the stop-valve long enough to find the true level of the water. If this is sufficiently high, blow down some of the water in the boiler, and feed in some fresh. Repeat this several times

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if necessary. If the foaming does not stop, cool the boiler off, empty it, and find out the cause of the trouble.

Keep Out Cylinder Oil.—Cylinder oil must be kept out of the boilers, because it is likely to cause overheating of the plates. Oily deposits may be removed, in large measure, by scraping and scrubbing, although more efficient methods of treatment may be required in bad cases. If kerosene is used in a boiler, keep all open lights away from the manholes and handholes, both when applying the kerosene, and upon opening the boiler up afterwards; and ventilate the inside of the boiler thoroughly, after oil has been used in it.

Cooling and Cleaning the Boiler.—In cooling a boiler before emptying it, first let the fire die out, and then close all doors and leave the damper open until the pressure falls to the point at which it is desired to blow. Clean the furnace and let the brick-work cool for at least two hours before opening the blowoff valve. If it is desired to cool the boiler further, after it has been emptied open the manhole and leave everything else as in full actual service—the fire doors, front connection doors, and cleaning doors being closed, and the damper and ash-pit doors open.

First cool the boiler as explained in the last paragraph. Never blow out under a pressure exceeding ten or (at most) fifteen pounds by the gage. If time will permit, the boiler should be left full of water for two or three days after shutting down. This prevents the scale baking to the tubes so it remains softer and is more easily removed.

The engineer must find out for himself how often his boilers need to be opened and cleaned. In many plants it is necessary to clean every week, while in some favored few it is sufficient to clean every three months. When using kerosene or large amounts of scale solvent, or when (as in the spring-time) the water becomes unusually soft, the boilers must be opened oftener than usual. In washing out a boiler, wash the tubes from above, as well as from below.

Never touch any valve whatsoever, in any part of the room, while a man is inside of a boiler, nor even after he has come out again, until he has reported that his work is finished and that he will not enter the boiler again. It is well to lock the stop-valve and blowoff valve upon every boiler in which a man is working, while other boilers are under steam. Padlocks and chains may be used for this purpose.

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In water-tube boilers the covers opposite the three rows of tubes nearest the fire should be taken off once a month, and the tubes thoroughly scraped and washed out; and all the tubes should be thoroughly scraped and washed out at least once in four months. This is for water of average quality. If the water is bad, clean the tubes oftener.

When mechanical hammers or cleaners are employed for removing scale from tubes, the pressure used to operate them should be as low as will suffice to do the work. Do not allow the cleaner to operate for more than a few seconds upon any one spot, and see that it goes entirely through the tube. Avoid high temperatures in the steam or water used to operate the cleaner.

Putting Boiler Out of Service.—In putting a boiler out of service, it should be cooled, emptied, and thoroughly cleaned, both inside and outside. The setting should likewise be cleaned in all its parts. Leave the handhole covers and manhole plates off. After washing the interior of the boiler, let it drain well. Then see that no moisture can collect anywhere about the boiler, nor drip upon it either internally or externally. Empty the siphon below the steam gage if the boiler room is likely to be cold, or take the gage off and store it safely away.

Do not allow moisture to come in contact with the outside of the boiler at any time, either from leaky joints or otherwise. Keep the mud drums and nipples, and the rear ends of horizontal and inclined tubes in water-tube boilers, free from sooty matter. If internal corrosion is discovered, notify your employers at once.

Examine your boilers carefully in all their parts, whenever they are laid off, and keep them as clean as possible, both inside and outside. See that all necessary repairs are made promptly and thoroughly. Keep the water glass and pressure gage clean and well lighted. If any contingency arises that you do not understand, report the matter to your employers at once; and if you think it possible that serious trouble may be impending at any time, shut down the boiler immediately.

Inform yourself respecting any local laws or ordinances relating to the duties of engineers and firemen, or to the plant in which you work. If there be any such, attend to them faithfully.

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CHAPTER XIX

HOW TO COMPUTE STRENGTH OF BOILER SHELLS

In order to ascertain by computation the maximum allowable pressure we must first compute the bursting strength of the solid boiler shell, then find the weakest part of this shell, which, of course, will give us the point where the shell would really give way. We next compute the steam gage pressure that would cause the boiler to rupture at its weakest point. This is known as the bursting pressure. It is important to note here the difference between the bursting pressure of the boiler and the bursting strength of the boiler shell. The former indicates the reading of

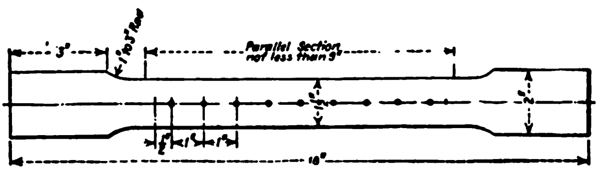

Fig. 78.—Standard form of test specimen.

In order to thoroughly test out plate material for boilers, a form of standard specimen has been established by the Boiler Code Committee of the American Society of Mechanical Engineers. The above illustration shows the standard form for the tension, cold-bend, and quench-bend test to be made from each boiler plate as rolled.

the steam gage at which the bursting will take place while the latter indicates the unit internal pressure in the boiler material when rupture occurs.

As a working gage pressure for boiler operation a factor of safety of 5 is often used—that is, a gage pressure $\frac{1}{5}$ that of the bursting pressure is considered as the largest gage pressure that may be safely put upon the boiler. It should be noted that when considering the safety of a boiler we always deal with gage pressure and not absolute pressure. The bursting pressure of a boiler is the difference between the pressure inside the boiler and the pressure outside, when rupture would occur, and as the latter is always the pressure of the atmosphere the bursting pressure must be the amount the inside pressure would be above the atmospheric pressure, which is the same thing as gage pressure.

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In order to ascertain the breaking strength of boiler material, a sample known as a standard form is put to test. Experimentally it has been found that whether a piece of material is subjected to rupture by tension, compression, or shear, the unit force required to rupture a square inch section, is equal to the total force observed in rupturing the specimen in each particular case divided by the cross-sectional area. This fundamental law enters largely in computation of boiler strength. Let us then proceed to this analysis.

The Strength of the Solid Plate.—In the study of gases and vapors it has been experimentally established that the pressures exerted by such substances are felt equally in all directions at any given point under consideration. Let us then consider the most

FIG. 79.—A diagrammatic representation of internal boiler pressure.

Since the pressure of a vapor is exerted equally in all directions we should consider that direction which would produce the most active results in tearing apart a boiler when deeducing expressions for the safe working pressure. In order to ascertain the total pressur-tending to burst the riveted section shown in the middle figure above, the pressure should be taken with the direction as shown by the arrows in this figure.

disastrous direction for pressure action. This evidently would be in such a direction as would tend to tear the boiler shell apart. If the length of shell considered be of length p equal to the distance from center to center of the riveted section or what is known as the pitch of the rivets, we have for a boiler of thickness t a resisting area of pt sq. in. If the solid shell will not burst until each square inch of area has upon it a unit force of S_t pounds, the total resistive force according to the experimental law stated in the previous paragraphs evidently ptS_t. Hence if A is the strength of solid plate, we have

$$A = tpS_t \tag{1}$$

Rule I. Multiply the thickness of the plate by the pitch of the rivets and by the tensile strength of the plate. The result is equal to the strength of solid plate.

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As an illustration let us compute the strength of the solid plate for a boiler whose thickness of shell is $\frac{1}{4}$ in., whose spacing of rivets is $1\frac{5}{8}$ in., and whose tensile strength stamped upon the boiler plate is found to read 55,000 lb. per square inch.

Applying Rule I, we have that the strength of solid plate is

$$A = tpS_t = 0.25 \times 1.625 \times 55{,}000 = 22{,}343 \text{ lb.}$$

The Strength of the Net Section.—As in the case of the weakest link determining the strength of the chain, so the strength of the boiler shell is determined by its weakest section. This will evidently be at the point where the shell has been perforated for the insertion of rivets. The actual area that will resist rupture is now no longer pt but since it has been weakened by an area dt wherein d represents the diameter of the rivet hole, B, the net resistive force now becomes

$$B = (pt - dt)S_t = (p - d)tS_t \tag{2}$$

Rule II (a). From the pitch of the rivet subtract the diameter of the rivet hole, then multiply by the thickness of the plate and again by the tensile strength of the plate. This result is equal to the strength of the plate between rivet holes—in other words to the strength of the net section.

Taking as an illustration the same boiler mentioned in Rule I, we have, if the diameter of the rivet hole is $\frac{11}{16}$ in., that the strength of the plate B between rivet holes is

$$B = (p - d)tS_t = (1.625 - 0.6875)\ 0.25 \times 55{,}000 = 12{,}890 \text{ lb.}$$

Resistance to Shear.—A boiler may not only fail by bursting apart the actual shell material but the rivet itself may give way. Under pressure the riveted boiler seam may pull apart and cut or shear off the rivet similar to the action that would take place by using a huge pair of shears. The area of cross-section of the rivet is evidently the only opposition that such an action would receive over the distance between one set of rivets in case of a single row of rivets, or if there be n rows of rivets, the area resisting shear is n times that for a single row. Hence, the force that would oppose rupture due to shear is evidently $n\ (.7854d^2)\ S_s$, where S_s is the pounds pressure exerted over each square inch of cross-section under shear. From results shown by tests, average iron rivets will shear at 38,000 lb. per sq. in. in single shear and 76,000 lbs. in double shear; steel rivets at 44,000 lbs. in single

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shear and 88,000 lb. in double shear. Hence we have that the resistance to shear C for a riveted section is

$$C = .7854d^2nS_s \qquad (3)$$

Rule II (*b*). Multiply the area of the rivet ($.7854d^2$) by the shearing resistance as follows. If iron rivets in single shear, allow 38,000 lb. per sq. in. of section, or if of steel allow 44,000 lb. per sq. in. If the resistance is in double shear add 100 per cent. to the above. The result is the bursting pressure for shear.

Continuing the example above cited, we have that the shearing strength C of one rivet in single shear is

$$C = n \times .7854d^2S_s = 1 \times .7854 \times .6875^2 \times 44,000 = 16,332 \text{ lb.}$$

Resistance to Compression.—Again the rivet may be forced to give way by having its longitudinal section (dt) actually crushed if the total crushing force of the steam pressure exceed dtS_c, where S_c is the crushing pressure in lb. per sq. in. over each unit area of the rivet. Hence the resistance to compression D is

$$D = dtS_c \qquad (4)$$

Rule II (*c*). Multiply the diameter of the rivet by the thickness of the boiler plate and then multiply by the unit bursting stress for compression for the rivet, which is taken at 95,000 lb. per sq. in. The result is equal to the strength of the rivet section for compression.

The resistance to compression D for the example above cited is then

$$D = dtS_c = 0.6875 \times 0.25 \times 95,000 = 16,328 \text{ lb.}$$

The Efficiency of the Riveted Section.—We now see that the riveted section weakens the solid plate in three ways. In the

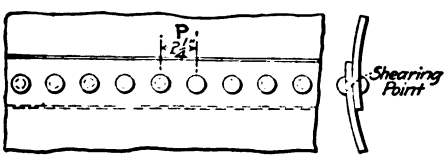

FIG. 80.—A single riveted lap joint for boiler plates.

By taking into consideration the stresses involved in a sectional distance equal to the pitch of the rivets, P, as shown, we are enabled to deduce the safe working gage pressure for boiler operation.

first place, the boiler may give way more easily because a section equal to the rivet hole has been cut from the solid plate. In the second place, the rivet may be actually sheared in two, and in the third place, it may be crushed longitudinally. The next

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thing to do then is to determine the ratio that each one of these factors bears to the strength of the solid plate and adopt the weakest or smallest ratio as the possible point where rupture will take place. Compute these three efficiency ratios for the joint E_j as follows:

$$E_j = \frac{B}{A}, = \frac{C}{A}, = \frac{D}{A} \tag{5}$$

Rule III. Divide the strength of the weakest section by the strength of the solid plate. (See Rule I.) The result is the efficiency of the riveted section.

Thus in the example cited we have seen that the strength of the solid plate is 22,343 lb., that its strength between rivet holes is 12,890 lb., that the shearing strength is 16,332 lb. and that the crushing strength of the plate in front of one rivet is 16,328 lb. Hence, the weakest place is in the strength between rivet holes and consequently the efficiency of point E_j is

$$E_j = \frac{12{,}890}{22{,}343} = .578.$$

Gage Pressure Necessary to Burst the Solid Boiler Plate.—We come now to the most interesting point of our analysis, namely to compute the bursting pressure of the solid plate.

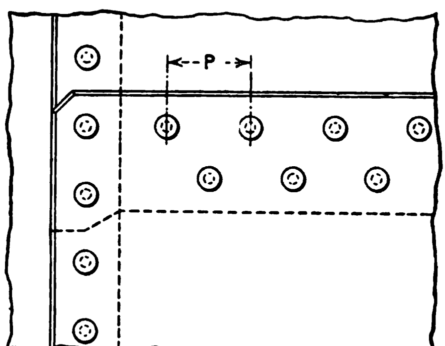

FIG. 81.—A double riveted lap joint.

By introducing a number of rows of rivets for riveted lap joints the shearing strength and the crushing strength of the riveted section are proportionately increased, while the tensile strength of the net section remains the same.

In the discussion of the strength of the solid boiler plate we found that the force of steam pressure acting so as to tear the boiler plate apart longitudinally would evidently prove most disastrous in bursting the solid boiler plate. Since the pressure

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of steam exerts itself equally in all directions, we shall compute the total pressure available in this particular direction as this would give us the critical pressure for our present consideration.

If the boiler is of length 1 in. and inner diameter D in. the area of steam pressure is Dl. Since now the boiler gage pressure is P_s lb. per sq. in., the total pressure of the steam would evidently be $P_s\,Dl$ lb. To resist the boiler tearing apart there is a strip of boiler metal on each side of length l and thickness t. Hence the total metallic area of resistance is $2lt$. If now the force of resistance offered by the metal is S_t lb. per sq. in., we have, when an explosion or bursting apart is about to take place, that this resistive pressure is $2ltS_t$.

Equating these two pressures, we have

$$P_sDl = 2ltS_t$$

$$\text{or } P_s = \frac{2tS_t}{D} = \frac{tS_t}{D/2} \qquad (6)$$

Thus we formulate.

Rule IV. Multiply the thickness of the plate by the tensile strength of the plate and divide by the radius (one-half of the diameter). The result is equal to the bursting pressure of the solid plate.

In the example previously cited we now compute the bursting pressure of the solid shell of the boiler under consideration for a boiler diameter of 36 in. as follows:

$$P_s = \frac{tS_t}{D/2} = \frac{0.25 \times 55{,}000}{36/2} = 764 \text{ lb.}$$

This means that a gage pressure of 764 lb. per sq. in. would rupture the given boiler if it existed without a riveted seam.

Bursting Pressure of the Seam.—But our boiler under consideration would evidently burst before the bursting pressure of the solid plate were reached for the riveted section has weakened its total strength. In Rule IV we found that the efficiency of the riveted joint is the ratio of the strength of the weakest point to the strength of the solid plate. Hence we have that the gage pressure P at which the boiler will probably rupture at the riveted joint is

$$P = P_sE_j \qquad (7)$$

Rule V. Multiply the bursting pressure of the solid plate by the efficiency of the joint. This result is equal to the bursting pressure of the seam.

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Thus since the efficiency of the joint E_j is found to be .578 and the bursting pressure P_s of the solid plate to be 764 lb., we have that the bursting pressure P of the joint which is the weakest part of the boiler construction is

$$P = P_s E_j = 764 \times .578 = 442 \text{ lb.}$$

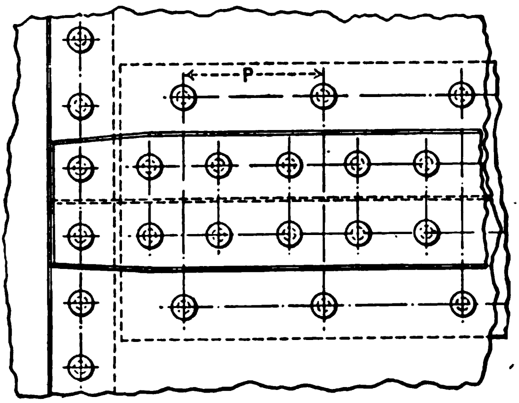

FIG. 82.—A double riveted butt and double strap joint.

In general the butt joint doubles the shearing strength of the joint while the net tensile strength and the crushing strength of the joint remain the same as in the lap joint discussion.

The Safe Working Pressure.—Of course the boiler is never allowed to operate anywhere near this bursting pressure. A factor of safety is insisted upon. The U. S. tables are based upon a factor of safety of 3.5 for drilled holes and 4.20 for punched holes, which are the lowest factors allowed in any civilized country. The factor in most European countries is either 5 or 6. In any case, if factor of safety f is used, we have that the working pressure P_w is found from the formula

$$P_w = \frac{P}{f} \tag{8}$$

The rule advised by the Hartford Insurance Company's inspectors is as follows:

Rule VI. Divide the bursting pressure of the seam by the following safety factors: 0 to 125 pounds, 4.2; from 125 to 150 pounds, 4.5; 150 pounds or over, 5. The result is the safe working pressure under which the boiler is to operate. The American Society of Mechanical Engineers in their Boiler Code require a factor of safety of 5 for all new boilers.

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Thus in the case at issue the safe working pressure P_w becomes

$$P_w = \frac{P}{f} = \frac{442}{4.2} = 105 \text{ lb.}$$

Recapitulating the discussion of the six rules, we now see in its completeness the method involved in computing the safe working pressure of a boiler. In this particular instance we find that a boiler of 36 in. diameter, with 1/4 in. plates and a single row of rivets spaced 1 5/8 in. apart may safely operate under 105 lb. pressure (gage).

Example of a Lap Joint, Longitudinal or Circumferential, Double-Riveted.—By similar reasoning we may now compute the efficiency of a lap joint which is double riveted whether longitudinal or circumferential. Thus, if the tensile strength of a boiler is stamped 55,000 lb. per sq. in. with thickness of plate 5/16 in., pitch of rivets 2 7/8 in. diameter of rivet hole 3/4 in., we have by applying our rules:

$$A = 2.875 \times 0.3125 \times 55{,}000 = 49{,}414.$$
$$B = (2.875 - 0.75)\, 0.3125 \times 55{,}000 = 36{,}523.$$
$$C = 2 \times 44{,}000 \times 0.4418 = 38{,}878.$$
$$D = 2 \times 0.75 \times 0.3125 \times 95{,}000 = 44{,}531.$$
$$\therefore E_j = \frac{36{,}523}{49{,}414} = 0.739$$

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CHAPTER XX

FURNACES IN FUEL OIL PRACTICE

FIG. 83.—Interior of a furnace, showing brickwork and air-spacing.

LET us now set forth the cycle of operations necessary in the utilization of crude petroleum as an economic factor in the production of steam. The oil in a heated state and under pressure must be sprayed into a heated compartment or furnace so that its particles are in fine globules or even in a gaseous state. Such an operation is known as atomization and this must be accomplished in an efficient and thorough manner. Three methods are utilized in practice to accomplish this. In the first instance steam under pressure is mixed with the oil and the ingredients thus shot into the furnace. In the second instance compressed air is used to accomplish this result, and in the third instance, some mechanical device or physical characteristic of the oil is made use of to whirl or thrust the oil into the furnace in a pulverized or atomized state. Literally hundreds of inventions have been made to effect the atomization of oil. It is to be remembered, however, that in the consideration of fuel oil economy, the furnace and its efficient construction are after all the real factors that go toward economic fuel consumption.

Fuel Oil Furnace Operation.—When the oil is atomized, it must be brought into contact with the requisite quantity of air for its combustion, and this quantity of air must be at the same time a minimum to avoid undue heat losses that may be carried away in the outgoing flue gases. To accomplish this result the checker work under the burners that control the admission of air must be properly designed. The proper quantity of air admission as a whole is controlled by means of draft regulation.

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An illustration of how this may be sensitively controlled was shown in the chapter on the fundamentals of furnace operation.

To accomplish the even admission of air into the furnace the arrangement of the check-board of brick-work below the flame is of utmost importance, otherwise unequal heating and imperfect combustion is sure to follow. Let us then examine a chart formulated by E. N. Percy of the Standard Oil Company's technical staff, shown in Fig. 84. In Fig. 1 of this chart we have a fan-shaped flame with openings between all the bricks. The flame does not cover all of the bricks, hence, no matter what the

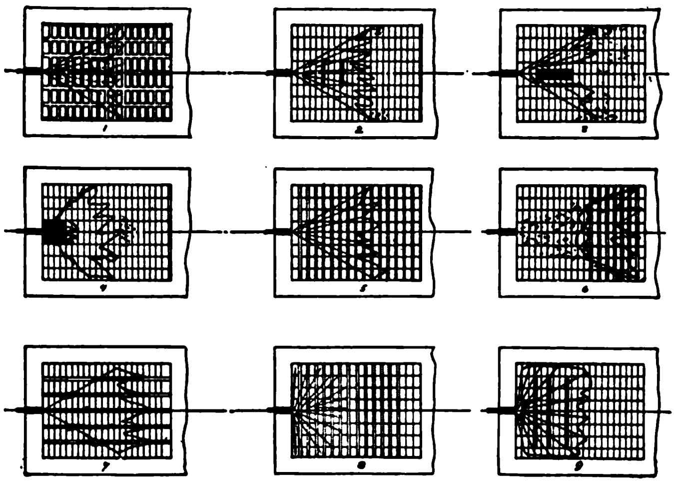

FIG. 84.—Theoretical display for brickwork and air-spacings.

In the nine illustrations shown above are graphically displayed the behavior of the furnace flame and the formation of carbon for various arrangements of air spacings below the flame. In the ninth instance a theoretically perfect flame is obtained.

conditions are there will be an excess of air and the boiler cannot work economically since it costs as much to heat air as it does to heat water. Figure 2 shows two large openings under the middle of the flame; such a flame will burn hot in the center and deposit carbon in the corners as shown. In Fig. 3 we have a large opening under the flame flow; this arrangement will cause the flame to tear and burn intensely at the center while depositing carbon around the corners, as well as allowing cold air to rise and strike the boiler directly. The large opening in Fig. 4 allows quantities of oil to escape over the flame; intense combustion will take place close to the burner, thereby over-heating it, and at the same time the flame will be irregular and ragged. It will smoke and deposit carbon at

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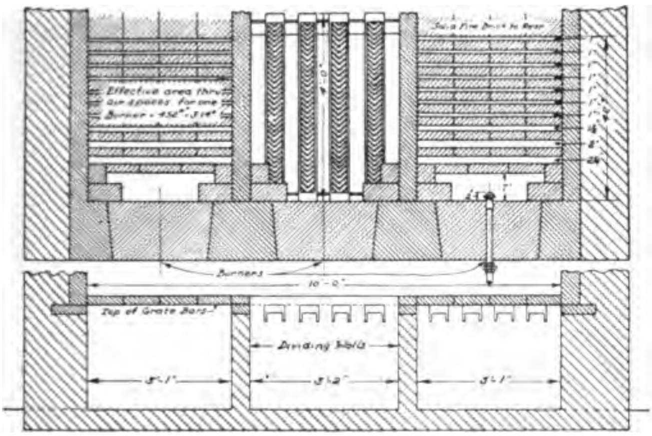

Fig. 85.—Arrangement of air-spaces and grate bars for fuel oil practice.

The details of furnace construction have more to do with efficient operation in the burning of fuel oil than anything else. In each particular installation this matter should receive careful attention. In the illustrations are shown the plan and elevation of the air spaces and grate bars for the Parker boiler installation for the Fruitvale Station of the Southern Pacific Co. This boiler developed an evaporative efficiency of 83.69 per cent. under trial test.

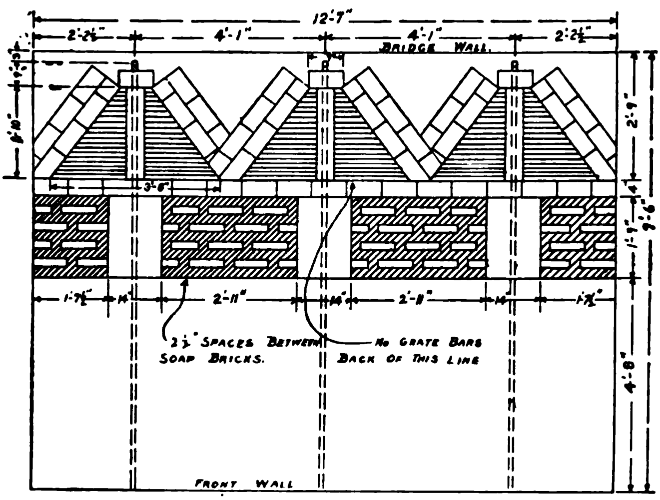

Fig. 86.—A former type of furnace.

In this view the floor plan of a back shot furnace arrangement is shown. The burner is set in a recess in the bridge wall. This design has proven of high order in central station installations of the West, but has now been replaced by the more recent type shown on the following page.

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the tips. The transverse openings between all the bricks as shown in Fig. 5 allows at all times a great excess of air and hence are not economic. Figure 6 shows draft orifices in the neighborhood of the burner; such a flame will burn clear at the tips, but it will smoke and deposit carbon near the burner. The longitudinal slots in Fig. 7 tend to tear the flame. In Fig. 8, the arrangement gives a broader and more correctly shaped flame, still an excess of air is admitted and cold air allowed to pass up against the boiler because the draft slots extend beyond the end of the flame. Figure 9

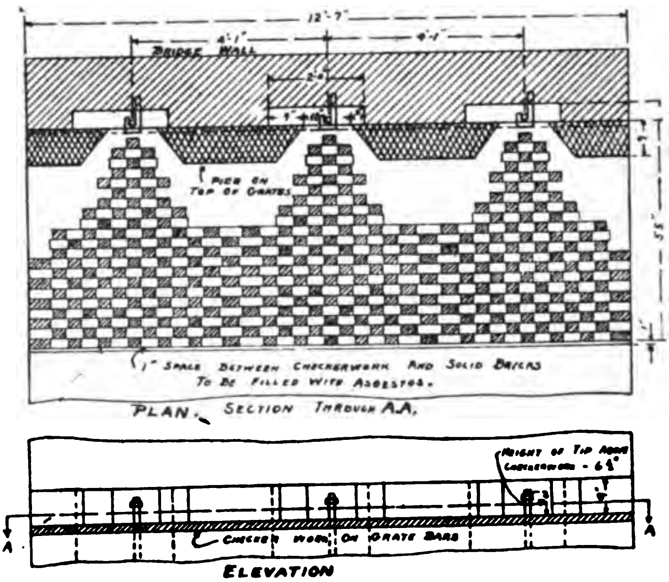

FIG. 87.—An excellent furnace arrangement.

Here is an excellent furnace arrangement designed for a 524 hp. boiler with standard low setting. The checker work on the grate bars shown in shaded area represents openings 2½ by 3 in. through the brickwork. The free area through the checker work is 2.44 sq. in. per hp., around the burner 0.62 sq. in. per hp., making a total free area of 3.06 sq. in. per hp.

approaches more nearly to the correct arrangement of bricks and the correct shape of flame for a flat flame furnace.

An excellent furnace is shown in Fig. 87, which sets forth the floor plan of a back shot furnace arrangement, the burner being set in a recess in the bridge wall. The recess is made large enough for the removal of the burner and piers of fire brick are built on the furnace floor in front of the recess so that there is an opening about 12 in. by 9 in. through which the mixture of oil and steam enters the furnace from the burner. A certain

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quantity of air enters through the same opening, being drawn in by the force of the oil and steam. A bracket is provided to hold the burner at the center of the opening.

Air openings through the checker work on the grates commence some 8 or 10 in. from the burner, the number of openings and the width increasing gradually until about 2 ft. from the burner the openings extend across the full width of the furnace. There are no openings between the burners near the bridge wall so that no air can enter except where it comes in contact with the atomized oil. The fire brick piers between the burners become hot and assist in the ignition of the oil.

The distance the air openings are extended from the burners and the total area of air openings depends on the draft available and the capacity required from the boiler. With a draft of 0.1 of an inch in the furnace a free area of 2½ sq. in. per rated boiler horsepower through the checker work and ½ sq. in. per horsepower around the burner, making a total of 3 sq. in. per horsepower, is sufficient to operate the boiler from its rated capacity up to 50 per cent. overload. If more capacity than this is required either a greater furnace draft must be provided or more openings through the checker work must be installed so as to increase the area. The amount of stack draft necessary to maintain 0.1 of an inch furnace draft depends upon the type of boiler and the capacity at which it is operated as this will determine the draft loss through the boiler. The loss of draft between the breeching and the furnace usually runs from about 0.15 of an inch at the boilers' rating up to 0.8 or 1 in. at double rating.

The location of the flame can be varied by changing the height of the burner above the checker work, this height usually varying from 4 in. to 8 in. or 9 in. The character of the flame can also be varied by changing the distance the air openings extend from the burners. It is customary to have the furthermost air openings about 4 or 5 ft. distant from the burner, the furnace floor beyond this point being covered with solid brick. By bringing the air openings somewhat farther out than this the flame can be made to turn up or by having the air openings extended out a shorter distance the flame can be made to hug closely to the floor of the furnace.

The Commercial Furnace.—Illustrations are shown in this article that set forth the check-board of brick work for air admission in the commercial practice of boiler economy. Let us now

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consider all the principal factors that must be considered in picking an efficient type of commercial furnace.

The furnace must be constructed of such heat tested brickwork that it will stand up under the high temperatures developed and the refractory material of which it is composed must be so installed as to radiate heat to assist the combustion of the heated ingredients of the fuel.

This combustion must be entirely completed before the gases come in contact with the heating surfaces of the boiler. Otherwise, the flame will be extinguished, possibly to unite later in the flue connection or in the stack. This means that ample space must be provided in the volumetric proportions of the

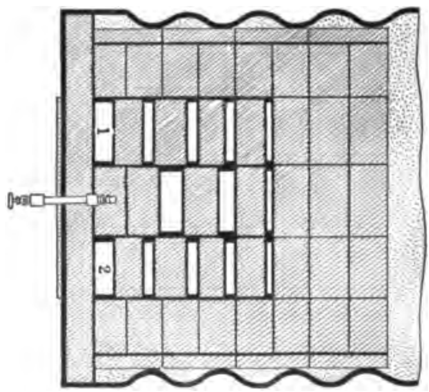

FIG. 88.—Plan of brickwork and air-spacings in marine practice.

In the practical application of the theoretical deductions for proper air spacings, commercial designers differ somewhat from the theoretical reasoning involved. In this illustration is shown the brickwork and air-spacings for Scotch marine boilers recommended by a prominent company.

furnace to insure this combustion before the gases begin to travel upward against the boiler surfaces.

Finally, there must be no localization of the heat on certain proportions of the heating surfaces or trouble will result from overheating and blistering. This is one of the more serious defects that had to be overcome in the earlier days of fuel oil practice. The burner has much to do with the avoidance of this localization activity.

The area of air openings through the checker-work should be made of sufficient size to operate the boiler at the maximum capacity required and then when operating at lighter loads the air supply should be very carefully regulated.

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Location of Burners.—The simplest form of oil burning furnace has the burners entering through the boiler front, the flame shooting back towards the rear of the furnace. This is the most

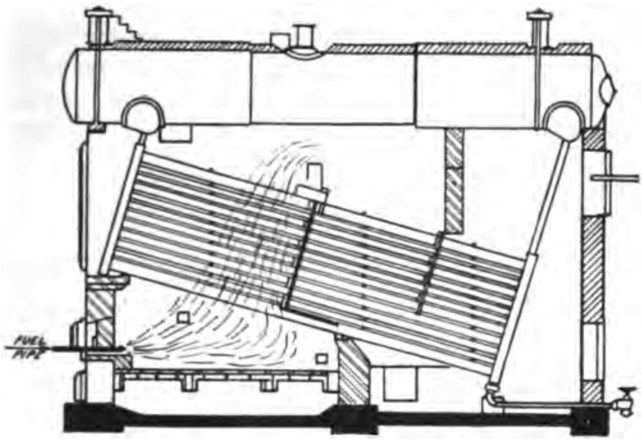

Fig. 89.—B. & W. boiler with oil burner in front.

With this furnace arrangement the flame does not fill out the first pass, so the front end of the tubes do not do their share of the work.

suitable type of furnace for return tubular boilers or for water tube boilers having horizontal baffles such as Heine or Parker boilers.

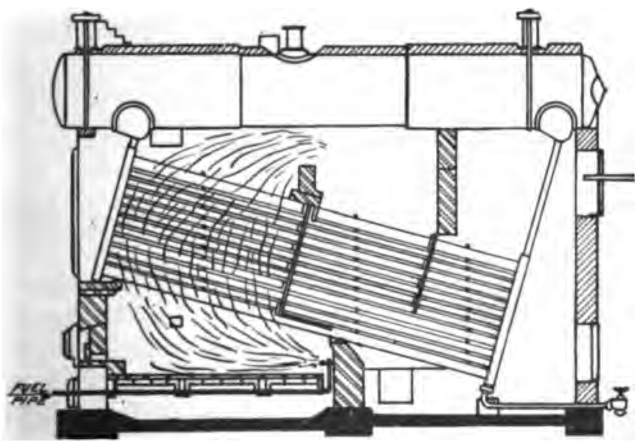

Fig. 90.—B. & W. boiler with Peabody furnace.

With this furnace arrangement the gases have ample volume in which to burn, and they distribute themselves over the entire first pass, resulting in efficient operation.

For water tube boilers in which the tubes incline downward toward the rear and having vertical gas passages, such as the

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Babcock and Wilcox Boiler, a furnace has been developed in which the burner head is placed at the bridge wall and the flame shoots forward from there. This furnace, which is known as the Peabody furnace, was the result of an extensive series of tests made by Mr. E. H. Peabody in California some years ago.

Owing to the fact that the tubes are inclined downwards toward the rear, the Peabody Furnace gives a larger furnace volume at the end of the furnace farthest from the burner. This is of

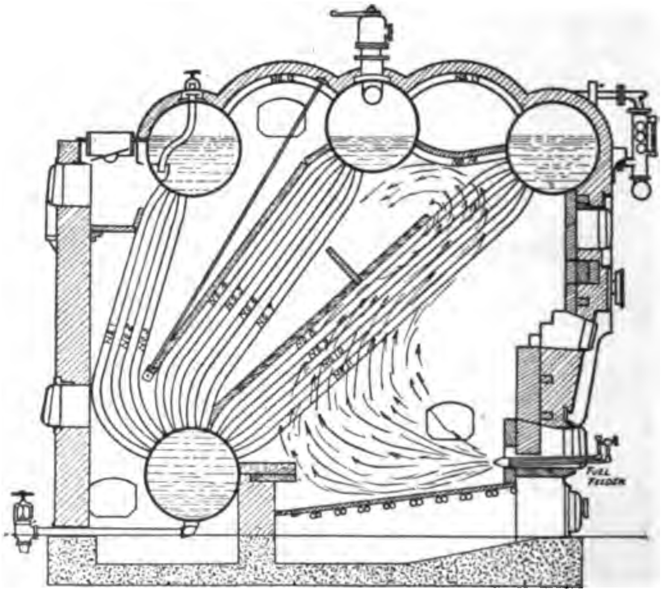

FIG. 91.—Stirling boiler with oil burner in front.

With this furnace arrangement the tubes are swept by the hot gases for their full length, but this advantage is gained at the expense of furnace efficiency owing to the smaller volume available as combustion chamber.

considerable advantage in permitting the gases to expand and cause perfect combustion. Another advantage of this type of furnace with the B. & W. boiler is that the flame is shot forward and comes in contact with the front end of the tubes, whereas with the burner in the front wall the gases are forced back close to the front baffle and do not have any tendency to fill the front pass of the boiler. This condition is illustrated in the adjoining Figs. 89 and 90. The result is that with the front burner arrangement a considerable portion of the heating surface is by-passed by the gases and is therefore, non-effective.

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With the Stirling boiler excellent results are obtained either with the front burner arrangement, or with the burners located at the bridgewall. With the former arrangement the gases come into intimate contact with the boiler heating surface. With the latter arrangement a very large furnace volume is made available, so that perfect combustion is obtained even at very high capacities, and as the front tubes absorb a large amount of radiant heat excellent efficiencies are obtained. With a four pass Stirling

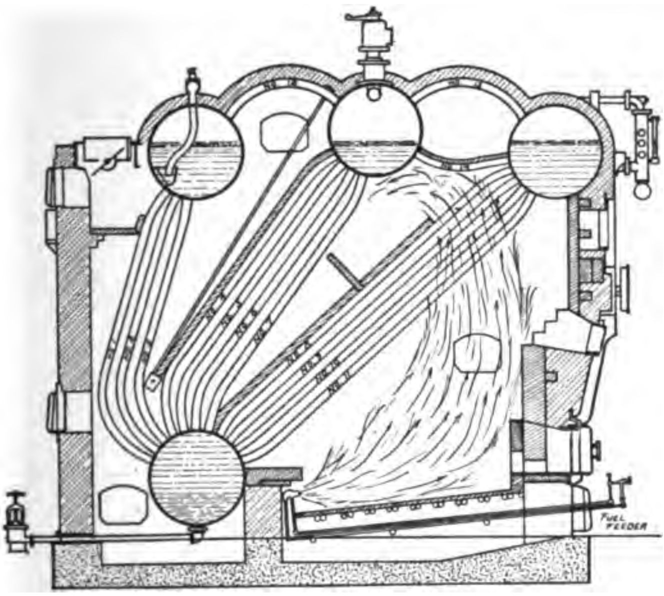

FIG. 92.—Stirling boiler with Peabody-Hammel furnace.

This furnace arrangement gives a splendid furnace with large volume. The gases come in contact with about one-third of the tubes in the front bank, the remainder absorbing radiant heat direct from the furnace.

boiler and a Peabody furnace still better results are obtained, for in this type of boiler there are fewer tubes exposed direct to the fire, and the gases are guided down among the remaining tubes after passing the first baffle. This arrangement therefore gives a combination of a large combustion chamber and efficient heat absorption.

Service For One Burner Only.—Where boilers having more than one burner are operated at very light loads it is necessary at times to have only one burner in operation, the other burner being shut off. For such service as this it is very desirable to

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have the ash pit divided into as many sections as there are burners so that when one burner is shut off the ash pit door opposite that burner can be closed tight and no air from the other ash pit doors will enter the furnace opposite that particular burner. With this arrangement it is possible to operate a large boiler at fractional loads and still maintain fairly good economy. This divided ash pit is one of the patented features of the Hammel

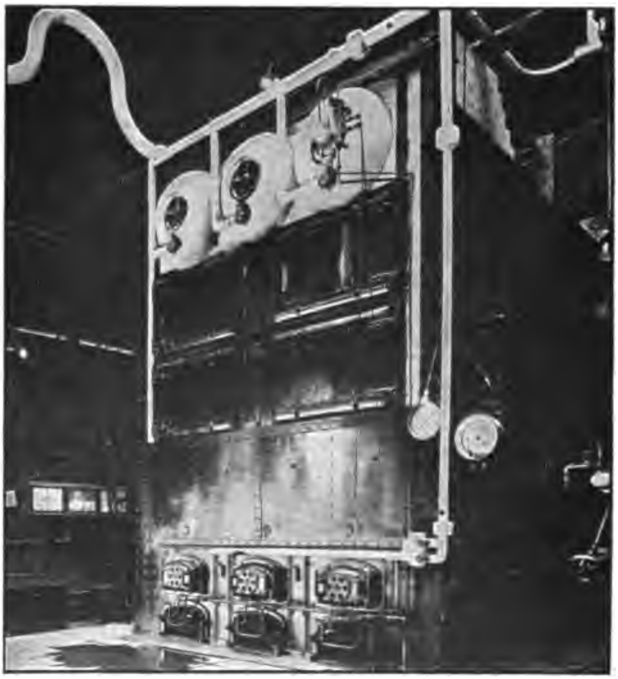

Fig. 93.—Babcock & Wilcox boilers, station C, Pacific Gas and Electric Company, Oakland. Boiler is set three feet higher than the standard height so as to provide large combustion chambers for the oil furnace.

Furnace, which is similar in many respects to the Peabody Furnace.

Large Furnaces.—In order to operate oil burning boilers at high capacity it is necessary to provided ample furnace volume. With the usual type of steam atomizing burner it is possible to burn as high as 6 or 7 lb. of oil per hour per cu. ft. of furnace volume, but for the best efficiency the quantity should not exceed 3 or 4 lb. per hour per cu. ft.

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There are two ways by which the furnace volume can be increased—one by lengthening the furnace and the other by raising the boilers. In Babcock and Wilcox boilers, a larger furnace can often be obtained by moving the bridgewall back to the mud drum, and placing the burners there. This gives a furnace the

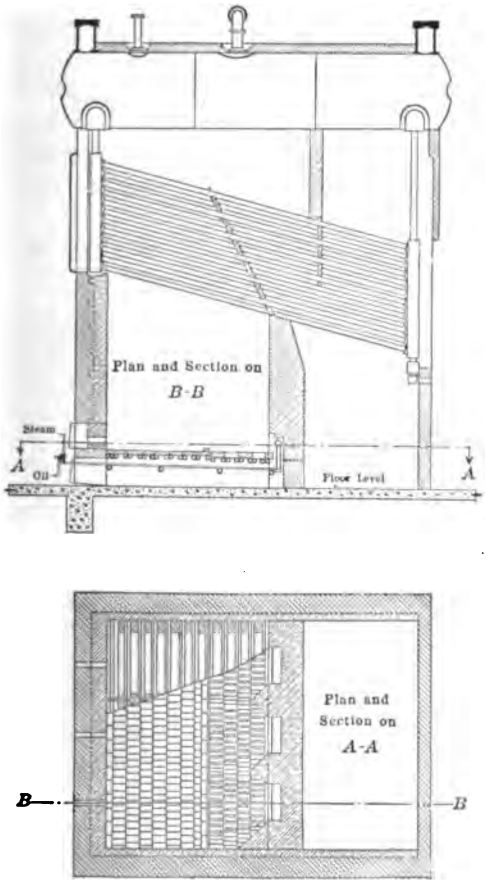

FIG. 94.—The Hammel patent oil burning furnace with boiler raised to give large combustion chamber.

full length of the tubes, the bottom tubes being covered with tile from the rear headers to the front baffle to prevent the gases by passing direct to the third pass of the boiler.

A large volume Hammel furnace, obtained by raising the boiler a few feet higher than the standard height, is shown in Fig. 94.

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CHAPTER XXI

BURNER CLASSIFICATION IN FUEL OIL PRACTICE

In 1902 and 1903 the U. S. Naval Fuel Oil Board made an exhaustive inquiry into burners of various types. In their report a classification of burners was set forth which comprehensively details the fundamentals of various types of burners known as the drooling, the atomizer, the chamber, the injector, and the projector types.

In the drooling type the burner allows the oil to drool from an upper opening down to a lower opening from which the steam is issuing. An atomizer burner allows the oil to drop directly on the steam. The chamber or inside mixer atomizes the oil within the burner after which it issues from an orifice of the desired form. An injector burner is designed primarily to operate without a pump as it is presumed that the oil will be sucked from the reservoir by the siphoning or injector-like action of the steam jet inside. In the projector burners the steam blows the oil from the tip of the burner.

Two other general classifications prevail depending upon the character of the flame emitted—namely, the fan tail and the rose. In the former type the burner produces a flat flame while in the latter a circular flame is sent forth.

The three principal types of burner that are encountered in central station practice are, however, known as the **inside mixer,** the **outside mixer,** and the **mechanical atomizer.**

The Inside Mixer.—In burners of this class, the steam and oil come into contact, and the oil is atomized inside of the burner itself, and the mixture issues from the burner tip ready for combustion at once. The Hammel burner is of this type.

The accompanying Figure 95 illustrates the construction of this burner. Oil enters at A, flows through D into the mixing and atomizing chamber C; steam enters at B, passes through F, E, and then through three small slots, G, H and I, into mixing chamber C where it meets the oil, and as these small steam jets cut across the oil steam at an angle, the energy of the steam is utilized.

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The burner requires for its operation about 2 per cent. of the steam generated by the boiler. The heavy hydrocarbons of the

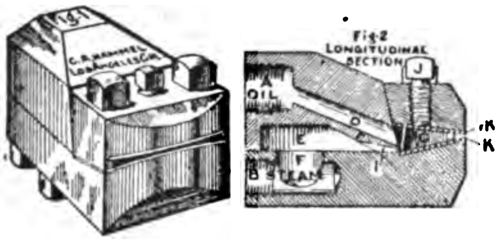

FIG. 95.—The inside mixer type of burner.

In burners of this type the steam and oil come into contact and the oil is atomized inside the burner itself. The mixture then issues from the burner tip ready for combustion. The Hammel burner shown in the illustration above is of this type.

oil are atomized, the light hydrocarbons are vaporized, and the completed mixture issues from the burner and ignites like a gas

FIG. 96.—Typical burners and control pipes ready for installation.

flame. In normal service there is no tendency to carbonize, and the only way in which carbonizing can be caused is by turning off

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the steam and leaving the burner filled with oil instead of blowing it out before shutting down.

All oil is usually more or less gritty and will cause wear of some part of the burner. This is provided for in the Hammel burner—the removable plates *KK* can be quickly replaced.

The Outside Mixer.—In the outside mixing class the steam flows through a narrow slot or horizontal row of small holes in the burner nozzle; the oil flows through a similar slot or hole above the steam orifice, and is picked up by the steam outside of the burner and atomization thus accomplished. The Peabody burner is typical of this class. It will be noted that the portions of the burner forming the orifice may be readily replaced in case of wear or if it is desired to alter the form of the flame.

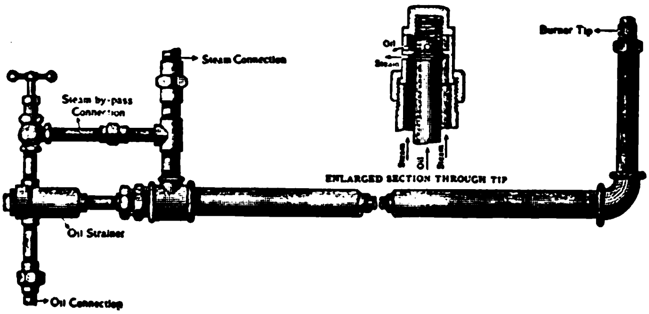

FIG. 97.—The outside mixer type of burner.

In this type of burner the steam flows through a narrow slot or horizontal row of small holes in the burner nozzle. The oil flows through a similar slot or hole above the steam orifice and is picked up by the steam outside of the burner and thus atomized. The Peabody burner which is shown in this illustration is a typical burner of this type.

There are many other makes of oil burners on the market, which operate on the same principle as the Hammel or the Peabody burner. A few of these are illustrated on the following pages, Figures 98 to 106 inclusive.

An Example of the Mechanical Atomizer.—As an illustration of one of the many interesting types of burners that produce atomization by the mechanical process, let us consider for the moment the rotary burner of the Fess System Company. The mechanism that accomplishes the atomization is operated by a small electric motor as shown of ¼ to ⅓ hp. The motor operates a rotary pump through a worm gear. This pump brings the crude oil from the storage tank and applies it to the burner, which is placed in the center of the fire box. The burner rotates

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at a sufficient speed to thoroughly atomize the oil by centrifugal force and by the proper admission of air a smokeless flame is produced equally distributed throughout the fire box.

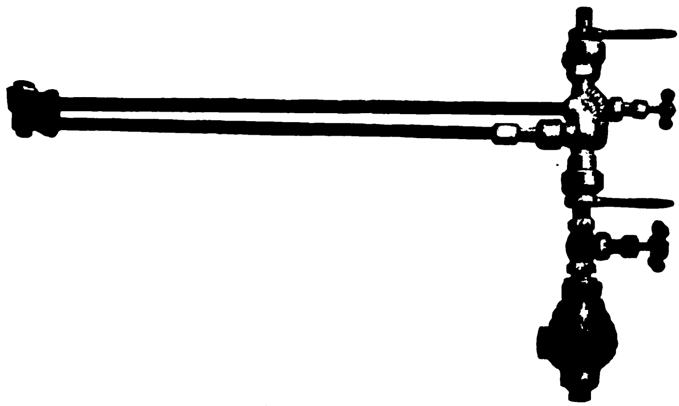

FIG. 98.—The Leahy oil burner.

The Leahy burner is an outside mixer and is made either for the back firing (Fig. 99) or front firing (Fig. 98). The burner equipment includes a bypass valve, two quick action unions, special oil regulation valve and oil strainer.

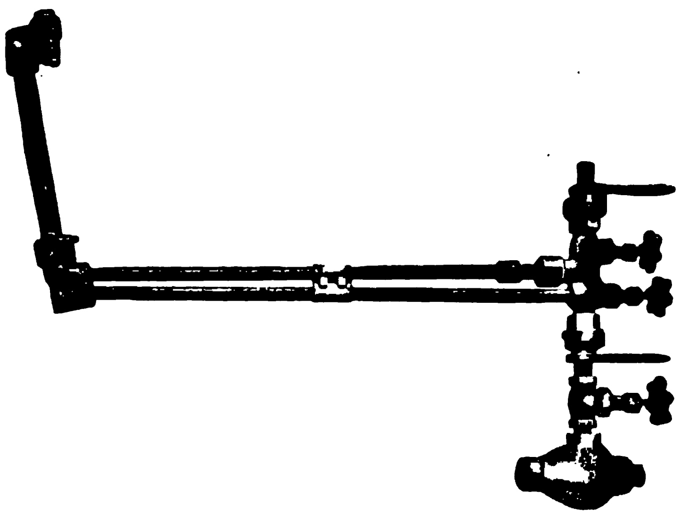

FIG. 99.

The Home-Made Type of Burner.—Patented oil burners are practically unknown in the oil fields. Every operator makes his own burner out of ordinary fittings. The construction varies

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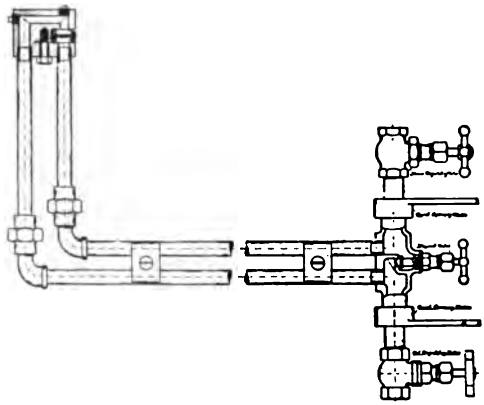

Fig. 100.—The Witt burner.

This is an outside mixing burner with removable steel tip. The above illustration shows a backshot burner suitable for a Peabody furnace.

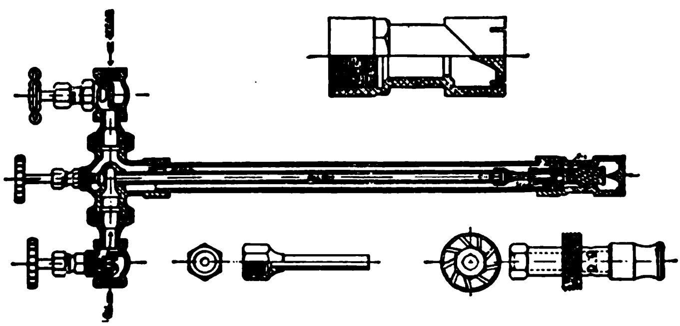

Fig. 101.—Here is shown a Witt inside mixing burner for front firing.

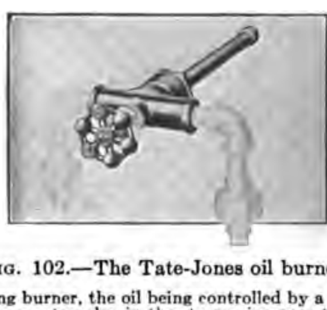

Fig. 102.—The Tate-Jones oil burner.

This is a steam atomizing burner, the oil being controlled by a needle valve in the burner head and the steam by a separate valve in the steam pipe near the burner. The length of burner and size of opening are arranged to suit each particular installation.

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somewhat depending upon the ideas of the maker and the quality of oil burned. The general principle of the burner is illustrated in Fig. 107. No oil pumps are used, the oil being supplied by gravity from a tank set from 6 to 10 ft. above the ground.

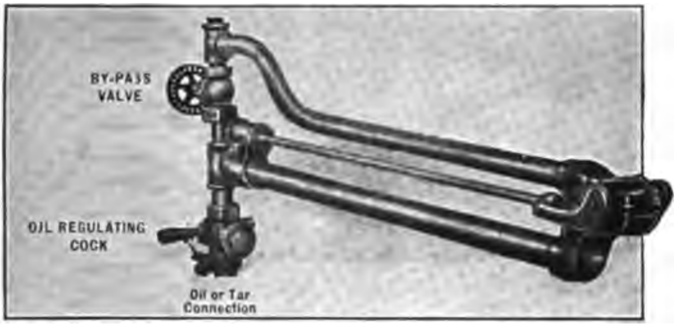

FIG. 103.—The W. N. Best oil burner.

This is an outside mixing burner, the oil and steam leaving the burner in directions at right angles to one another.

An important peculiarity of the burner is that it is self-regulating to a great extent. The impact of the jet of steam issuing from the inner pipe produces a back pressure on the oil issuing from the annular space between the pipes. If the steam valve

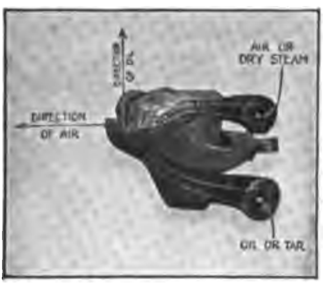

FIG. 104.—W. N. best burner.

This burner is provided with a hinged lip which may be raised while the burner is operating, opening up the steam slot so that it can be blown out, thus preventing clogging. This burner can be used with either steam or air as the atomizing medium.

is adjusted for good atomization any increase of the steam pressure will cause more steam to flow through the inner pipe. This will increase the back pressure at the tip and choke back the oil coming from the annular space, thus decreasing the fire.

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If, on the other hand, the steam pressure drops, the back pressure at the tip is decreased, more oil will flow and the fire will be increased.

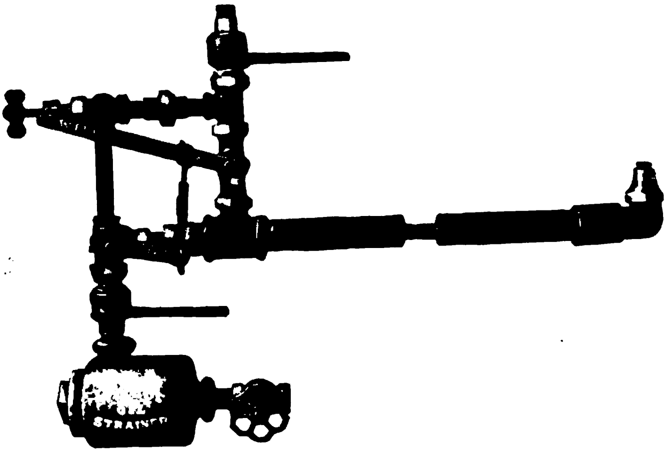

Fig. 105.—The Wilgus oil burner.

This is an outside mixing burner with removable tip which can be readily changed if a different shape of flame is desired. The burner is provided with oil and steam regulating valves interconnected so they can both be operated by one lever. This insures the steam being cut down to suit the quantity of oil used at light loads.

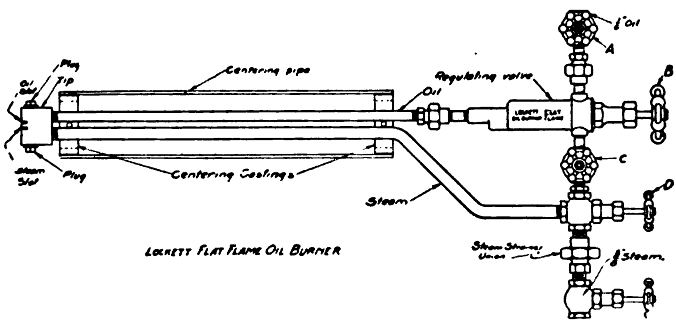

Fig. 106.—The Lockett flat flame oil burner.

This is an outside mixing burner with renewable tip. A feature of this burner is the combined regulating valve and strainer through which the oil must pass to enter the burner. This regulating valve lets the oil out through a slot instead of the annular ring that occurs in an ordinary globe valve when partly open. The slot gives an opening larger than the holes in the strainer, so clogging of the oil burner is prevented. The strainer can be readily removed for cleaning.

This type of burner is sensitive to variations in steam pressure. As the steam pressure goes up, the fire is cut down until a point is reached at which the fire becomes spasmodic or "bucks."

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While this self regulating feature helps to maintain constant pressure on the boiler, it is not economical because as the steam pressure increases, thus diminishing the quantity of oil, the quantity of steam increases with the pressure. Thus, the less oil is burned the more steam is used for atomizing, which is just the opposite of what it should be.

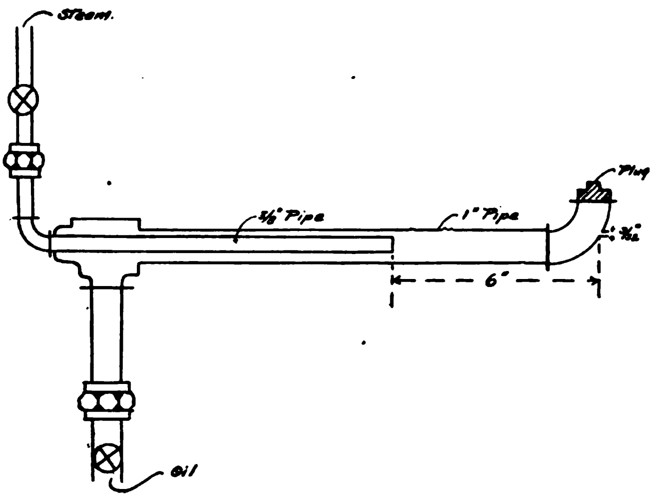

FIG. 107.—The homemade burner.

This ingenious type of homemade burner is a product of the oil fields. The impact of the jet of steam which issues from the inner pipe produces a back pressure on the oil issuing from the annular space between the pipes, thus making the burner self-regulating to a great extent.

Another peculiarity of the burner is that it will begin to atomize when the steam pressure is less than a pound above atmosphere. As soon as a sizzle is heard issuing from the steam pipe, the burner will make a fairly good fire.

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CHAPTER XXII

MECHANICAL ATOMIZING OIL BURNERS

While steam atomizing oil burners are used almost exclusively in stationary power plant work at the present time, the mechanical atomizing oil burner has found great favor in marine work, and is the standard method of firing oil burning marine boilers.

There is a good deal of interest shown at present in the use of the mechanical atomizing burner in stationary plants, and it is possible that it may eventually entirely displace the steam atomizing burner. The mechanical atomizing burner, sometimes called

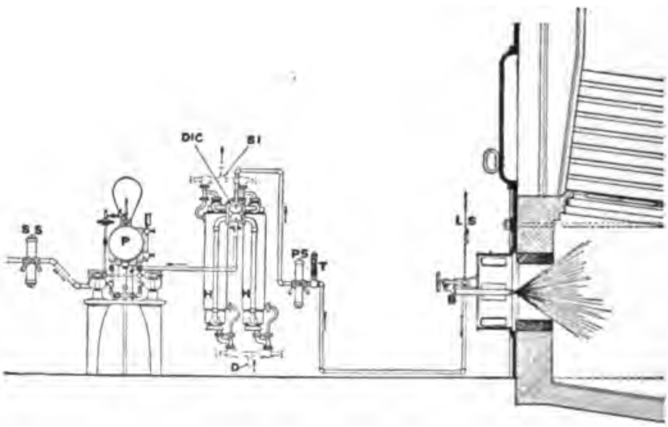

Fig. 108.—General arrangement Koerting mechanical oil burning system on a stationary boiler.

the pressure jet burner, is made in a number of different forms all of which operate on the same principle. It consists essentially of a nozzle containing a small conical shaped orifice through which the oil is forced at high pressure, the oil being first heated to a temperature approaching its flash point. Inside the nozzle means are provided to give the oil a whirling motion of sufficient intensity to make it fly into a spray on account of the centrifugal force as soon as it leaves the nozzle. The burners are placed in the boiler front and each burner is provided with an air regulating device which admits the air around the burner, at the same time

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regulating the quantity of air to suit the combustion requirements. In some makes of mechanical burners the air is given a twisting motion as well as the oil, which adds to the effectiveness of the mixing of air and oil for combustion.

There are so many different makes of mechanical atomizing burners that no attempt will be made to describe them all. A brief description of a few of the most prominent will, however, be of interest.

Koerting Burner.—In this burner, which is illustrated on page 174 the oil is given a rotary motion by being forced through the passages of a helical screw into a small conical chamber containing the outlet orifice. The air enters through a cylindrical chamber having openings parallel to the axis of the burner, and is controlled by an adjustable cover which is rotated over these openings.

Dahl Burner.—In this burner, the oil is given its rotary motion by passing through small channels formed between two parts of the burner. These channels deliver the oil tangentially at the periphery of the conical chamber in the tip, through which it passes in the form of a vortex to the orifice outlet. The air is controlled by the furnace doors, and the mixture adjusted by a conical deflector surrounding the burner.

Peabody Mechanical Burner.—In this burner the rotary motion is secured by means of a flat disc having a ¼ inch hole, and four slots which lead the oil tangentially toward this central hole. This slotted disc fits into the burner tip which contains the conical chamber and orifice outlet, the diameter of the conical chamber at its base being the same as that of the central hole in the disc.

With this burner the air is given a rotary motion as well as the oil. This is done by means of a cast iron truncated cone provided with blades, in the center of which is placed a so-called impeller plate, also bladed. The impeller plate gives a rotary motion to the air entering close to the burner, and the cone gives a rotary motion to the air entering around the edge of the impeller plate. In operation, the impeller plate and the burner may be moved in and out together, being fixed in relation to each other but adjustable in relation to the truncated cone.

Moore Shipbuilding Company Burner.—In this burner the oil is forced through slots cut longitudinally on the outer surface of

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a plug, the slots being curved at the end to give the necessary rotary motion to the oil as it enters the conical chamber in the burner tip. The position of the slotted plug in relation to the burner tip may be adjusted while the burner is in operation by means of a rod passing through the burner.

Coen Burner.—In this burner, which is illustrated in Fig. 109 the oil is delivered to the central chamber by means of small tangential channels, whose area can be altered by means of a rod inside the burner running its full length, operated by the hand-wheel of a special angle valve. It is thus possible to regulate the fire from individual burners without altering the oil pressure.

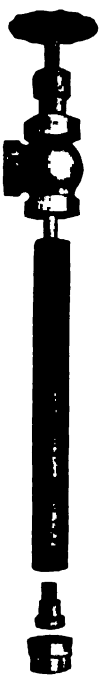

Fig. 109.—The Coen mechanical atomizing burner.

The main air supply is controlled by a sliding plate in front of the cylindrical burner chamber, and a sliding cylinder surrounding this chamber controls a secondary air supply.

Draft.—Mechanical atomizing burners operate satisfactorily with natural draft at limited capacities, but if the boilers are to be forced much above their rated capacity it is necessary to equip them with forced draft fans, delivering the air under a slight pressure to the burner air chambers.

Pressure and Temperature of Oil.—The operation of mechanical atomizing burners varies with both the pressure and temperature of the oil. The pressure must be at least 25 lb. in order to produce good atomization and it may be increased up to 200 lb. Pressures higher than 200 lb. are not required and only cause unnecessary stress of the oil piping. The quantity of oil burned is regulated by varying the pressure from 25 lb. up to 200 lb. If more oil is required than can be obtained at the higher pressure it is necessary in most makes of burner to change the nozzles, using tips with larger orifices. If less oil is needed than passes through the burners at 25 lb. pressure it is necessary to shut off some of the burners.

A temperature of at least 150°F. is required to properly atomize the oil. The atomization is improved by increasing the temperature up to about 200°. Above this temperature no change is made in the atomization but the flame becomes shorter and the

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combustion occurs closer to the burner as the oil is heated to a higher temperature. As the oil is heated the first effect is to reduce its viscosity resulting in a greater quantity of oil passing through the burner. At the higher temperatures, however, the increased temperature has little effect on the viscosity, but owing to the increased volume of the oil the capacity of the burner is reduced.

The following table gives the quantity of oil that will flow through a burner having a 1/16 in. diameter orifice at a pressure of 200 lb., the oil having a gravity of 20° Baumé and a flash point of 220°F.:

Temp. °F.	Pounds oil per hour	Remarks
60	340	No atomization
110	440	Bad sparking
153	375	Atomization good
210	330	Flame white and short
260	300	Flame shorter
304	273	Flame 2 ft. long

Advantages of Mechanical Atomizing.—The principal advantage of the mechanical atomizing system is that a large quantity of oil can be burned in a furnace of a given volume.

The steam atomizing burner as applied to stationary boilers produces a flat flame and the combustion occurs on the upper and lower surfaces of this flame, with the result that a considerable proportion of the volume of the furnace is not made use of.

With the mechanical atomizing burner, on the other hand, there is what may be called volume combustion, the mixture of air and gases occurring throughout the entire furnace. It is therefore, possible to completely burn more oil in a given size of furnace by this method than by the steam atomizing method, or if the same quantity of oil is burned in both cases the combustion with the mechanical atomizing system will be more complete in the furnace proper, so that the flame will not travel so far among the boiler tubes and the entire efficiency of the boiler will be greater. The cooling of the gases by their coming in contact with the boiler tubes before the combustion is completed is one of the principal causes of low efficiency in boilers that are forced

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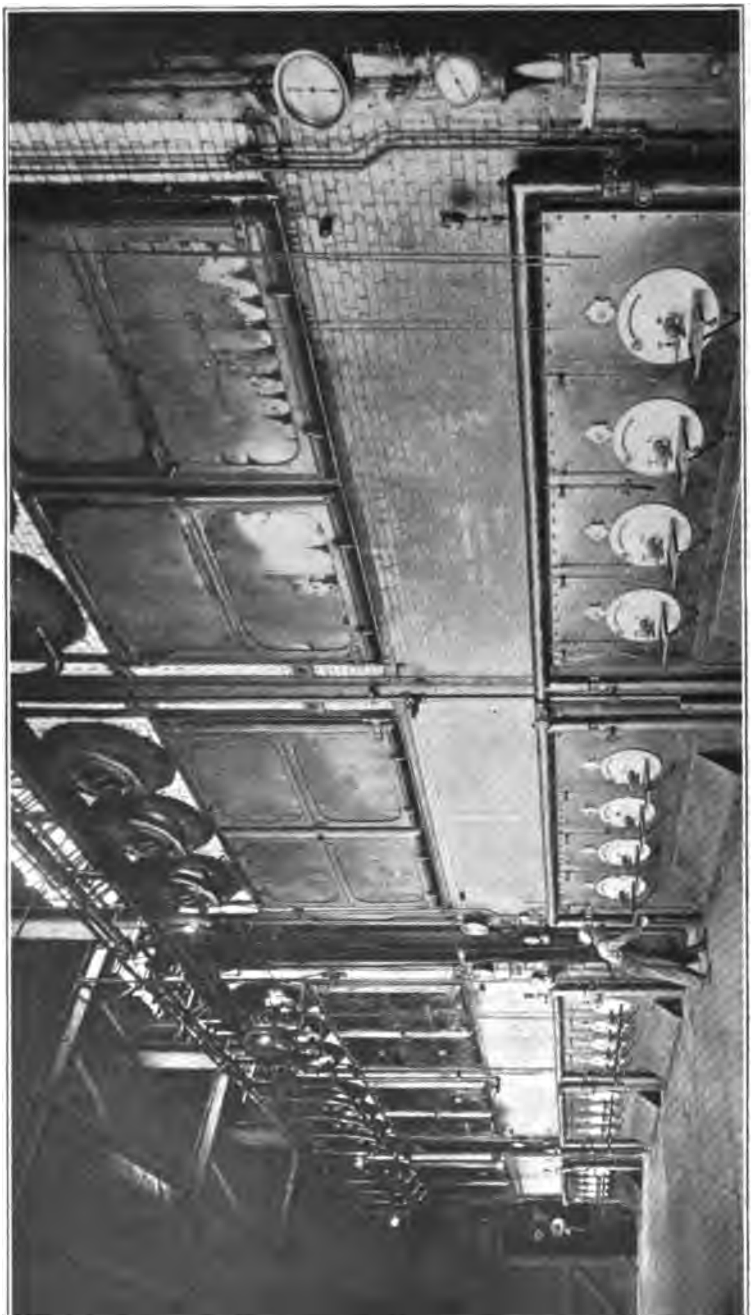

FIG. 110.—The mechanical fuel oil atomizers installed under two 250 h.p. Heine boilers. The view is the standard equipment at eleven pumping stations of the Shell Company of California oil pipe line, installed by Sanderson and Porter. The operation is entirely automatic, oil being supplied to the burners under the control of the pump governors so as to maintain constant pressure.

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up to high capacities, and any method of combustion that prevents this occurring tends to improve the efficiency of the boiler.

The maximum quantity of oil burned by steam atomizing burners amounts to 6 or 7 lb. of oil per hour per cu. ft. of furnace volume, whereas with mechanical atomizing burners as much as 11 or 12 lb. per hour per cu. ft. has been burned.

The mechanical atomizing system also saves the steam that is used in the steam atomizing system, which often amounts to 3 per cent. or 4 per cent. of the total steam generated. While some additional steam is required in the mechanical system for heating the oil up to the higher temperature and for pumping it to the higher pressure, the condensation or exhaust from this steam returns to the feed water heater so that the quantity utilized is not great. The loss of the steam used by the steam atomizing burner is of much greater importance for marine boilers than for stationary boilers, because it means a loss of fresh water as well as a loss of steam. It is for this reason that mechanical atomizing burners have already largely displaced steam atomizing burners on board ship.

Disadvantages of Mechanical Atomizing.—The principal disadvantage of the mechanical atomizing burner is that there is aways a tendency for the small orifice in the burner to choke up. This is sometimes due to grit and other solids in the oil which pass through the strainers and sometimes due to carbonization on account of the flame occurring close to the nozzle. To overcome this difficulty it is necessary in some cases to merely slacken back the feed screw of the burner and then readjust it, and in some cases to turn off the oil altogether and blow steam through the burner. Whenever the fire is burning one sided, sparking on one side or showing black streaks it is a sign that the burner needs cleaning. As the number of burners required for mechanical atomization is greater than for steam atomization this difficulty means a good deal of careful attention on the part of the operator. With steam atomizing, three burners are sufficient to operate a large 800 h.p. boiler, whereas the same size boiler would require 6 or 8 mechanical atomizing burners.

Another objection to mechanical atomizing burners for stationary power plant work is the fact that the air supply for each burner must be adjusted separately every time there is a change in the quantity of oil burned. This is not objectionable where there

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is a steady load as occurs on board ship, but in a stationary power plant where the load is changing up and down continuously a continual adjustment of the air supply means a large amount of attention on the part of the fireman. This difficulty, however, may be overcome by the use of automatic regulation of the air, applying to mechanical atomizers the same principles that have already proved successful in the automatic regulation of steam atomizing burners.

Until recently the use of mechanical atomizing burners in stationary plants has been confined to small stations operating at a fairly steady load, such as the pumping stations on oil pipe lines. Recent installations have proved that the system can be applied to large boilers, and is desirable especially where high capacities are required, capacities as high as 300 per cent. of the boiler rating having been obtained. It must be realized, however, that the use of this system for variable load boilers is still in the experimental stage, at the time of writing. (April, 1920).

Recent research along the lines of mechanical atomization by D. S. Jacobus and N. E. Lewis is of great importance in future development of steam electric generation, where oil is used as fuel. As a consequence, in the following pages we shall set forth in full the data presented by these gentlemen before the Pasadena Convention of the National Electric Light Association in May, 1920, on this timely subject:

The practice in central stations of running boilers up to 300 per cent. of rating in carrying peak loads has made it necessary to consider some other means of burning the oil than with steam atomizing burners, as a boiler that would normally be operated over peak load intervals at 300 per cent. of rating or thereabouts with coal could not be operated at over about 200 per cent. of rating with steam atomizing burners.

The limitation of capacity with steam atomizing burners led the Babcock & Wilcox Company to make a series of tests to develop mechanical atomizing burners of the type used in marine practice that would be especially adapted to stationary boilers.

The tests indicated that the economy obtainable with steam atomizing and with mechanical atomizing oil burners at the lower capacities was about the same. At the higher capacities the efficiency with the mechanical atomizing burners was higher than with the steam atomizing burners. Considerably higher capacities could be obtained with the mechanical atomizing burn-

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ers than with the steam atomizing burners. The results of comparative tests on Babcock & Wilcox boilers are shown by the curves in Fig. 111, where the lower curve represents the net efficiencies obtainable at different ratings with steam atomizing burners and the upper curve the corresponding efficiencies with mechanical atomizing burners. By net efficiency of the steam atomizing burners is meant the efficiency based on the total steam evaporated less the steam blown to waste at the burners in atomizing the oil. In the case of the mechanical atomizing burners no deduction is made for the relatively small amount of steam required for heating and pumping for the mechanical

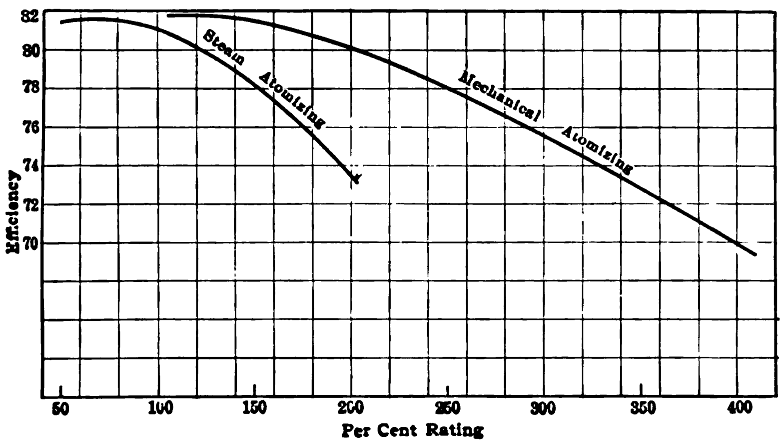

FIG. 111.—A relationship showing a comparison between steam atomization and mechanical atomization of fuel oil under varying boiler efficiencies and percentages of rating.

atomization of the oil, as the heat in this steam is usually returned to the system by employing it for heating the oil or by passing the exhaust steam from the oil pump to a feed water heater.

An examination of the curves will show that with the boilers operated at rating, the efficiency in each case is in the neighborhood of 82 per cent. At 200 per cent. of rating, which for the usual type of boiler employed in large boiler plant practice, represents the approximate limit of capacity for steam atomizing burners, the difference in efficiency in favor of the mechanical atomizing burner is in the neighborhood of 6 per cent. At high rates of driving, feed-water conditions, as will be discussed later, have a very important bearing on the maximum that can be

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expected. With proper feed-water conditions, using mechanical atomizing burners, the maximum possible capacities are as high or higher than are being obtained with forced blast stokers over peak load periods in central station work.

With a natural draft of one inch available the capacity with mechanical atomizing oil burners with the number of burners that can be installed in the ordinary setting for coal firing, would be limited to approximately 200 per cent. of rating. At higher capacities than 200 per cent., forced blast at the burners should be used with mechanical atomizing burners.

While the application of mechanical atomizing oil burners to stationary boilers is still in the development stage, the tests made by Messrs. Jacobus and Lewis above alluded to, together with results secured at several plants where the burners have been commercially installed, indicate that they will take the place of steam atomizing burners for many classes of service. Mechanical atomizing oil burners are just as easy to operate as those of the steam atomizing type, in fact, experience in marine work and in the few installations so far installed for stationary boilers has shown that they are less likely to give trouble than steam atomizing burners. There is less trouble through carbonization or clogging up and when a burner has to be changed less time is required for a mechanical atomizing burner than for a steam atomizing burner.

Where a change is made from coal to oil firing the mechanical atomizing oil burners are especially adapted as in many cases the stokers can be protected with brickwork and the oil burned directly above the stokers without removing the stokers. By keeping the stokers in place the boilers can readily be converted back to coal firing if desired. The same blast that is used for forced draft stokers may be employed for the mechanical atomizing burners, thereby saving the expense of a portion of the oil burning equipment.

The reason that higher capacities are available with mechanical atomizing burners than with steam atomizing burners is on account of the quicker combustion secured with the mechanical atomizing burners and because the steam atomizing burners do not utilize the full furnace volume. With mechanical atomizing burners the oil is expelled under a heavy pressure from the burner tips in a fine mist and intimately mingled with the air for combustion directly at the burners, which leads to the greater part of the oil

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being consumed in short cone-shaped flames. With steam atomizing burners the oil is not divided into anywhere near as fine a spray and there is no quick and thorough mingling of the oil and air, which leads to the long flames characteristic of the type. To secure a good combustion with steam atomizing burners the flames should have a natural path of travel, the air for combustion being admitted beneath the flames. It can be readily appreciated that only, say, one-half of the total furnace volume may be occupied by the flames with steam atomizing burners and that much of the furnace volume may be ineffective. With the mechanical atomizing burners there are, therefore, two effects which lead toward a greater capacity being developed with a given furnace volume, the first being the quickness of combustion due to the better atomization and thorough mingling of the oil and air at the burners, and the second the form of the flames which make available a greater proportion of the furnace volume.

In marine practice the furnace must necessarily be kept down to the minimum size that can be employed in order to save space and the best commercial results are secured by so proportioning the furnace that a good efficiency will be secured at the ordinary loads carried, or at cruising speeds, with the burners arranged so that in an emergency the boiler may be run at a considerably higher capacity. In land work where every cu. ft. of space occupied need not be considered as carefully as in marine work, it is obvious that a furnace can be installed to advantage of a larger size than in marine work. Again, in stationary work poor feed water has often to be contended with, and with poor feed water a larger volume assists materially in reducing the tube losses. Experience with oil burners has indicated that for a given overload capacity there will be more tube difficulties in case the feed water is not thoroughly clean than with a properly arranged coal fired furnace. This comes through the higher furnace temperature with oil firing, which leads to a higher absorption rate in the tubes which come next the furnace. With poor feed water oil fired boilers must be operated at a lower rating than coal fired boilers if tube difficulties are to be minimized, and in some cases it may be desirable not to run at an average load of very much over rating.

To run at the higher capacities great care must be exercised in the operation. An entirely different field is entered when capacities are carried of 300 per cent. or over, and it would be

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folly to attempt to operate at these capacities without most competent operators, clean feed water and the strictest attention to all details.

In marine practice the make-up water is distilled. The make-up water in some stationary plants is now distilled, and as this can be done without any material loss of heat to the system, it is a practice which undoubtedly will be followed to a greater extent in the future. The use of mechanical oil burners reduces the amount of make-up water required through eliminating the loss of the steam that is blown to waste with steam atomizing burners. Where the make-up water is secured from an evaporator it is obviously advantageous to use mechanical atomizing burners to reduce the amount of make-up water required. This element has an important bearing on the use of mechanical atomizing burners for marine work.

Where steam atomizing burners are used in connection with a boiler fitted with an economizer the steam carried away in the flue gases increases the tendency to condense a portion of the vapor from the flue gases on the colder parts of the economizer, thereby producing a sweating action on the exterior of the economizer. Mechanical atomizing oil burners are preferable in this respect as they do not add any steam to the gases.

The mechanical atomizing oil burners that have been developed and patented by the Babcock & Wilcox Company for stationary work are shown in Figs. 112 and 113.

Fig. 112 shows a Babcock & Wilcox mechanical atomizing oil burner of the Lodi design for use without a forced draft and Fig. 113 the same burner for use with a forced draft. The corresponding parts are similarly numbered in each of the two figures.

The fire brick moulded tile (1) are held in place by the cast iron grid (2). The cast iron bladed cone (3) conducts the air and atomized oil to the furnace. The main register casting (4) is bolted through the boiler front plate to the cast iron grid (2), thus holding the cone (3) in place. The main register casting (4) is fitted with four automatic air doors (5), by the use of which the quantity of air supplied to the burner may be regulated or shut off entirely if desired. These doors are so designed that they will close automatically in case of a flare-back in the furnace or the bursting of a boiler tube, thus protecting the fireman who may be standing in front of the boiler.

To the front of the register casting is fastened a cover plate (6),

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this cover plate being secured with studs to the main register casting and holding in place the radiation guard (19) and the spider casting (7). In case a double front is used for forced blast the radiation guard is replaced by the outer front plate of the double front, as shown in Fig. 3.

The spider casting (7) has four cams which control the operation of the automatic air doors (5). Fastened to the spider casting (7) and passing through a slot in the coverplate (6) is a handle (8) to operate the spider. By moving the handle (8) to the extreme right the air doors are all closed, and by moving it to the left the doors are gradually opened.

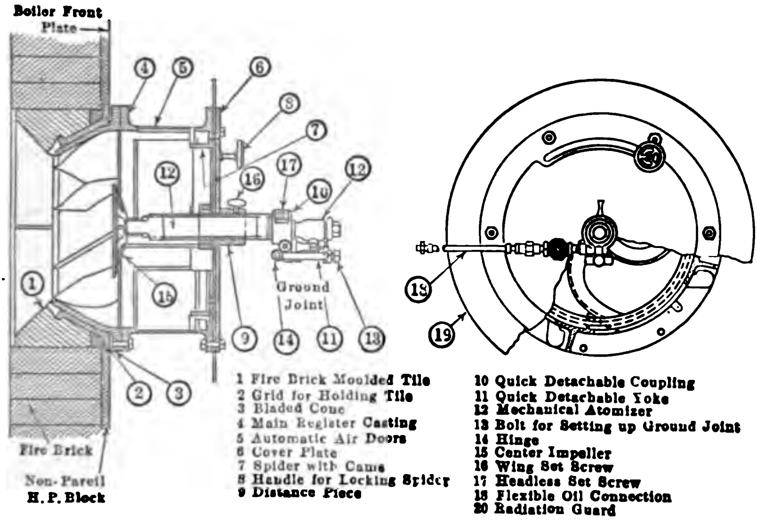

Fig. 112.—The Babcock and Wilcox mechanical oil burners showing single front construction for natural draft.

Passing through the center opening in the cover plate (6) is a distance piece (9), to the outer end of which is fastened a quick detachable coupling (10) and yoke (11).

To the other end of this distance piece is fastened an aluminized steel conical shaped impeller plate (15) for regulating the distribution of air at the nozzle of the burner. Passing through the distance piece (9) is the mechanical atomizer (12), this atomizer being held in place and connected to the fuel oil supply line through the quick detachable coupling (10) and yoke (11), thus making the atomizer (12), the distance piece (9), and the center impeller (15) a rigid unit when in operation.

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The distance piece (9) is so designed that it may be moved along its axis, thus moving the impeller plate (15) in and out with reference to the balded cone (3), and decreasing or enlarging at will the clear area for the passage of air around the outside of the impeller plate.

By means of a set screw (16) the impeller plate (15), together with distance piece (9) and atomizer (12) may be fastened in any desired position. To adjust the distance between the tip of the atomizer (12) and the center opening in the impeller plate (15), a headless set screw (17) is unscrewed, then by holding the

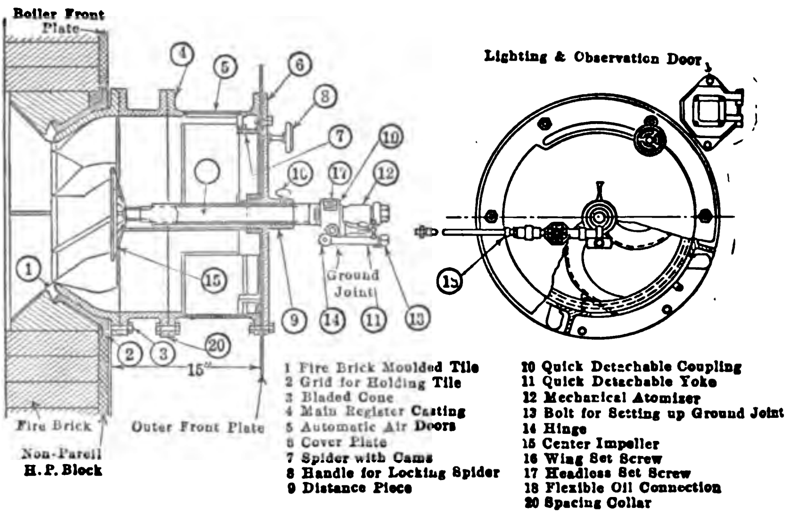

FIG. 113.—The Babcock and Wilcox mechanical oil burners showing double front construction for forced draft.

distance piece (9) with the set screw (16), the coupling (10) may be rotated on the distance piece (9) to bring it to any position desired.

The distance between the tip of the atomizer (12) and the central opening in the impeller plate (15) should be maintained at approximately ¼ inch, and when the coupling (10) is moved up on the distance piece (9) sufficiently to give this distance, the headless set screw is driven home.

In Fig. 113 a spacing collar (20) is shown which is used when the number of burners is such that the outer front plate must be moved out to give the proper flow area for the air supplied to the burners by means of the forced draft.

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The curve in Fig. 111 for the efficiency secured at different ratings with steam atomizing burners is based on tests that were made on a 14-high B. & W. boiler in 1907 at the Redondo plant of the Pacific Light & Power Company. In these tests California oil of about 14° Baume was burned which has a heat of combustion of approximately 18,000 B.t.u. per pound.

The curve for mechanical atomizing oil burners shown in Fig. 111 is based on tests made on a 14-high B. & W. boiler at the Bayonne works of the Babcock & Wilcox Company with Mexican oil 14 to 16° Baume, having a heat of combustion of approximately 18,300 B.t.u. per pound. The conditions existing in the tests of the mechanical atomizing burners were as follows:

The oil pressure at the burner varied from 100 to 200 lbs. per sq. in., depending on the capacity at which the boiler was operated and upon the size of the sprayer or orifice plates which are used inside the tip of the mechanical atomizers. The best atomization was obtained when the oil was heated to give a viscosity of from 3° to 5° Engler, which required a temperature of from 220° to 250°F. Live steam was used for heating the oil from about 110° to the temperature specified, the amount approximating one half of one per cent. of the total steam generated.

In the tests the amount of steam required for driving a rotary pump which was used for pumping the oil amounted to from 1 to 1½ per cent. of the total steam generated. The pump was considerably larger than required and therefore wasteful. In a large plant the amount of steam required for pumping the oil would be in the neighborhood of ½ of 1 per cent. of the total steam generated and the exhaust steam from the pump could be used for the preliminary heating of the oil.

The number of burners for use in a given boiler may be determined on the basis of one burner per 120 to 130 rated boiler h.p. This capacity may be exceeded in certain instances.

The variation in load is taken care of by adjusting the oil pressure at the pump between the limits of 100 to 200 lb. per sq. in. and by cutting burners in and out. Throttling the oil at any individual burner should not be done and a valve to a given burner should remain always wide open or tight shut.

It is not advisable to attempt to regulate the air to any great extent with the air doors on individual burners. The best way to adjust the air is through the use of the boiler dampers for natural draft installations. Where a forced draft is employed

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the air is regulated through the use of the boiler dampers and the adjustment of the air blast.

A boiler fitted with mechanical atomizing burners requires more draft to operate it at a given rating than one fitted with steam atomizing burners, as the air for combustion must be drawn through the burner registers at a velocity that will cause it to mingle intimately with the atomized oil. With no forced draft the amount of draft suction required at the damper of a 14-high B. & W. boiler for drawing the air through the burner registers and for drawing the gases through the boiler when operating at rating, 150 per cent. of rating, 200 per cent. of rating and 250 per cent. of rating is 0.25, 0.6, 1.0, and 1.65 in. water column, respectively. Where a forced draft is used at the burners, the draft suction required at the boiler damper to overcome the resistance of the gases flowing through the boiler is 0.15, 0.20, 0.35, and 0.50 in. water column, respectively, and when operated at 250 per cent. of rating the forced draft required at the burners is 1.15 in. water column.

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CHAPTER XXIII

RULES FOR EFFICIENT OPERATION OF OIL FIRED BOILERS

Since the advent of steam turbines the relative importance of the fireroom crew as a factor in economical operation has increased considerably compared with the engine-room crew. This is because the steam turbine, after it has once been properly set up, operates continuously at a fixed steam consumption for a given load and nothing can be done to improve its economy other than to keep the turbine, condenser and auxiliaries clean and in good operative condition. In the boiler room, on the other hand, continual watchfulness is necessary to keep the boilers operating at good efficiency, and the slightest laxity in attention to the various details results in a large waste of fuel. To assist the operators of oil-fired boilers in obtaining the best economy possible, the writers have prepared the following set of rules, which if carefully followed will bring the daily operating efficiency of the plant very close to test results:

1. Regulate Air to Suit Load.—The regulation of the air supply is one of the most important things in the operation of oil-fired boilers. If there is not enough air, a great waste of fuel may occur as part of the atomized oil will simply pass up the chimney unburned. On the other hand, it is possible to waste just as much fuel by allowing too much air to enter the furnace as all of the extra air is heated up and passes out at the temperature of the chimney gases, carrying away with it an enormous amount of heat. To determine accurately the amount of air required for the best conditions it is necessary to analyze the flue gases.

Many plants, however, are not provided with the apparatus necessary for this, and in such cases the air may be regulated with a fair degree of accuracy by an observation of the smoke discharged from the stack. For perfect combustion there should be no smoke, and if any smoke appears it means incomplete combustion and not enough air. If there is no smoke, however, it does not follow that the conditions are right, as no smoke may mean either just the right amount of air or a large excess of air.

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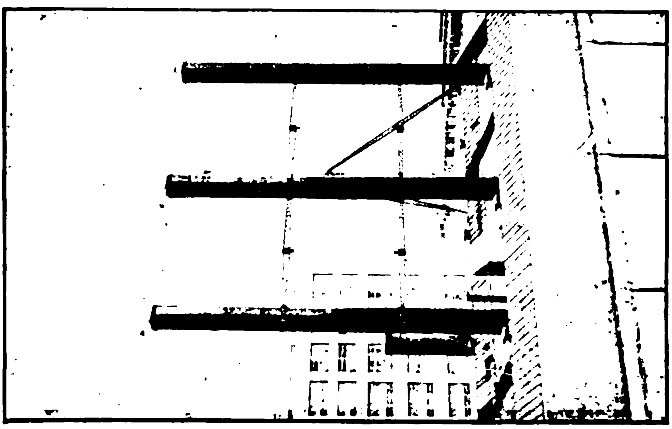

FIG. 114.

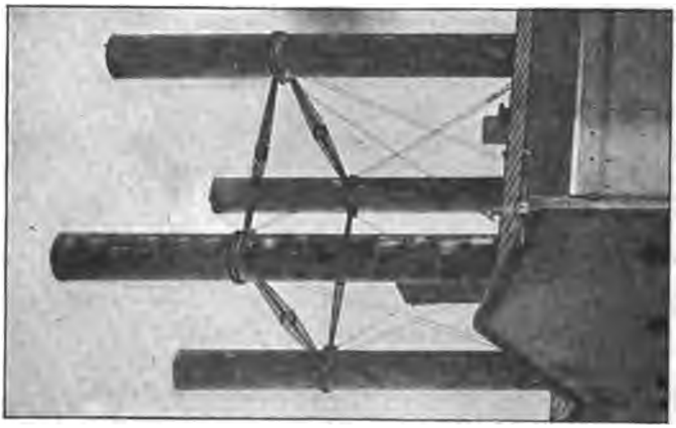

FIG. 115.

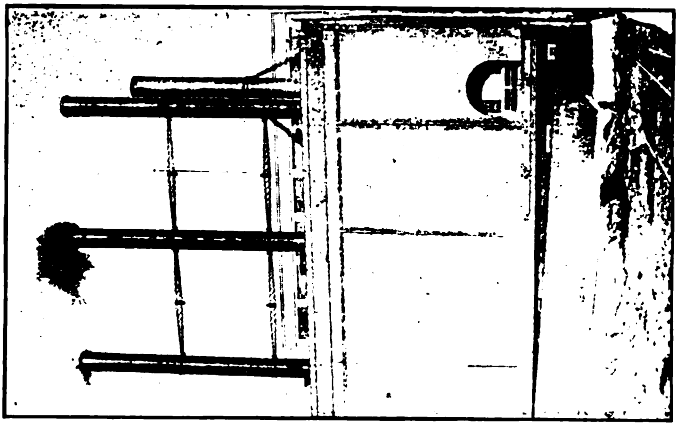

FIG. 116.

FIGS. 114–116.—Stacks of station A, Pacific Gas and Electric Company, San Francisco. This plant is one of the largest fuel oil operated power plants in the world. The first view shows improper combustion, the second proper combustion, and the third instances of proper and improper combustion in varying degrees.

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To properly regulate the air, therefore, if the boiler is operating with no smoke the damper should be gradually closed until light gray smoke just begins to appear; if then the damper is opened very slightly, this smoke will be barely perceptible and the conditions for the most economical operation will be obtained.

2. Prevent Air Leakage Through Setting.—Air leaks may be detected by holding a candle flame at the cracks in the boiler setting. If the flame is drawn in, it shows that there is an air leak which should be stopped up as far as possible.

A good coat of fire-resisting paint over a brick setting is a good investment, for the bricks themselves are quite porous as can be demonstrated by dropping one in water and noting the air bubbles driven from it.

There are two ways by which the air can be regulated, namely, by the damper at the outlet of the boiler or by the ash-pit doors. If the air is regulated by the ash-pit doors, the damper being left wide open, there will be a strong draft within the setting, tending to cause air to leak in through the cracks in the brickwork. The strong draft also tends to pull the gases through the setting by the shortest paths, so that some of the heating surface is not swept by the gases.

If, on the other hand, the ash-pit doors are left wide open and the air is regulated by partly closing the damper, the draft inside the boiler setting is very slight so that the air leakage is reduced to a minimum. There is little force tending to change the direction of the flow of the gases so that they travel of their own momentum to the furthermost corners and fill out the setting completely, thus coming in contact with all the heating surface of the boiler. It is, therefore, much better to regulate the air by means of the damper than by means of the ash-pit doors. In the case of very light loads, however, it is best to use both the damper and the ash-pit doors because if the damper alone is used there may be a positive pressure produced in the upper part of the setting, causing gas and smoke to leak out into the fire room.

3. Analyze Flue Gases Frequently.—The exact position of the damper to suit different loads can only be determined by flue-gas analysis. The CO_2 should be kept as high as possible without producing CO. If it is found impossible to secure 13½ or 14 per cent. of CO_2 without a trace of CO, then there is something wrong with the furnace or the burners and an investigation should

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be made. The proper method of sampling exit gases should be used as it is quite possible through improper sampling to obtain very poor CO_2 readings even though the furnace may be operating quite effectively. If correctly measured, the greater the CO_2, up to a certain limit, the greater the efficiency.

4. Burner Must Be Suited to Furnace.—The function of the oil burner is to atomize the oil, and the most efficient burner is the burner that will atomize the oil with the least quantity of steam. Burners are of three general types—steam jets, mechanical-pressure jets and air jets. The air jet is largely used in metallurgical work while the mechanical jet, which delivers a conical flame, is used principally in marine work and to some extent in stationary practice. The steam atomization burner is used in stationary and locomotive practice almost universally. The choice of burners for central-station furnaces therefore lies between the steam jet and the mechanical, the shape and design of the furnace influencing the type selected.

The furnace arrangement is the most important part of the boiler so far as economy of operation is concerned. If the air and oil are not properly mixed, it will be impossible to obtain proper combustion. It is important that the oil be completely burned before the cooling effect of the heating surfaces can operate to quench the fire. A large combustion chamber is of great advantage in an oil-burning furnace, as the larger the combustion chamber the more complete the combustion will be before the gases are cooled.

5. Keep Boilers Clean and Maintain Furnaces Properly.—Oil burners must be carefully watched as they are liable to become clogged with foreign matter or coated with carbon. This may cause the flame to shoot sideways, resulting in smoke and poor efficiency. Therefore a spare burner should always be kept on hand, so that as soon as any burner gives trouble it can be removed, taken apart and cleaned. All burners should be cleaned at regular intervals and adjusted so that the flame will not be projected directly against the boiler tubes or against the boiler setting, and so the flames from different burners will not interfere with one another.

The checkerwork in the furnace floor must be carefully placed and maintained in good condition. The air openings often become slagged over or stopped up by pieces of brick breaking off. Unless they are cleaned out occasionally and kept open for their

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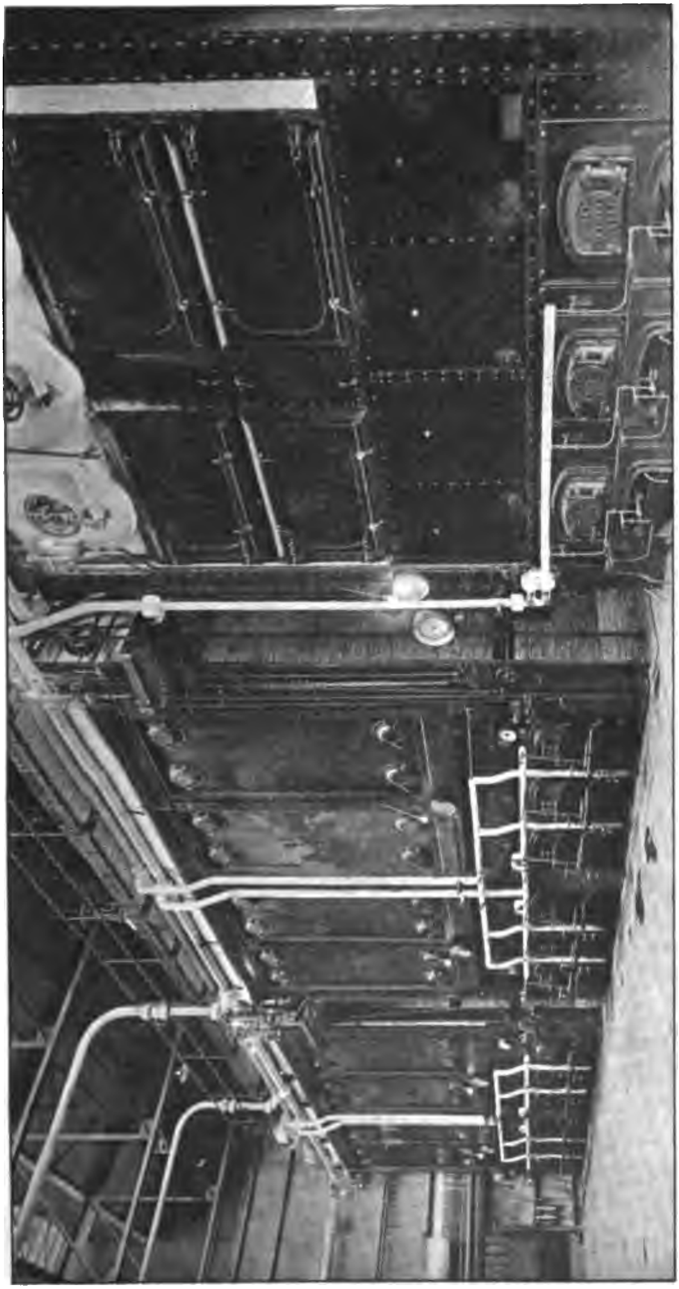

FIG. 117.—One of the four lines of boilers, fuel oil operated at station C, Pacific Gas and Electric Company, Oakland, California. Note the metering equipment and the equipment for regulating the draft and quick operation of ash pit doors as installed in this place.

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full area, both the efficiency and capacity of the boiler will be reduced.

6. Regulate Atomizing Steam to Suit Oil.—The regulation of the quantity of steam used for atomizing the oil is a matter of very great importance, for if more steam is used than is actually needed there is not only a waste of the excess quantity of steam but there is also a loss of the heat required to raise the temperature of this extra steam up to the temperature of the escaping gases. With careless operation the quantity of steam supplied to the burners sometimes amounts to as much as 5 per cent. of the total steam generated by the boiler, whereas with proper care in operating this quantity can be reduced below 1 per cent.

A simple way to adjust the quantity of steam supplied is to gradually close down on the steam valve to the burner until drops of oil fall on the furnace floor. The drops burn and scintillate and can be readily seen, and this scintillation indicates that there is not sufficient steam to atomize the oil. As soon as this point is reached the steam valve should be opened just enough to stop the scintillating action.

The quantity of steam supplied to the burner bears an important relation to the furnace arrangement and the air supply, as both the shape and character of the flame change when the quantity of steam is varied. With too much steam an intense white flame is produced, which has a tendency to cause localization of heat on the brickwork or the tubes. With the proper amount of steam and correct air regulation a soft orange-colored flame is produced which fills out the furnace and has an appearance similar to that from a soft-coal fire. This flame will sometimes appear smoky in the furnace, but the smoke disappears before the gases reach the stack. It is therefore unnecessary to have an absolutely clear flame in the furnace.

A simple method of preventing too much steam being used for atomizing where boilers are operated at a fairly steady load is to provide a disk with a small hole in it in the steam to burner line. This disk restricts the quantity of steam that can pass through the burner. The size of the hole in the disk depends on the steam pressure used and on the capacity required from the boiler and must be determined by experiment. In a plant using 200 lb. (14 kg.) steam pressure a hole $\frac{5}{16}$ in. (7.9 mm.) in diameter has been found large enough to supply all the atomizing steam required for a 600-h.p. boiler. A by-pass should be pro-

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vided on the steam line so as to pass steam around the disk in case it is found necessary to force the boiler at any time above its normal capacity. By providing the by-pass with a valve having a rising stem it can be seen at a glance whether the valve is open or shut.

Superheated steam should be used for atomizing wherever it is available, as the higher the temperature of the steam the more complete the atomization will be. The steam to the burner should always be cut down whenever the oil is cut down. Too often, however, the firemen leave the steam flowing full blast even after the oil has been reduced or shut off altogether. This is an absolutely unnecessary waste, though it is often necessary to keep a little steam flowing through a burner that is not in use, to keep it cool enough to prevent damage resulting from the heat of the furnace.

7. Heat Oil to Proper Temperature for Atomization.—The quantity of steam required for atomizing depends largely on the temperature of the oil. The hotter the oil the less steam is required. In central station work the oil should be heated up to about 180°F. on the pressure side of the pumps, the pressure carried running from 40 to 60 lb.

Oil should never be heated above its flash point, as in case of a leak in the oil pipe there will be danger of fire. However, atomization does require heating the oil to a temperature at which the viscosity is sufficiently lowered to make the oil very fluid. Over heating of an oil, due to its expansion, reduces the burner capacity to a considerable extent and also introduces the element of risk.

The proper temperature of the oil differs for different oils. California oils require heating to between 150° and 180°F. (66° and 82°C.). The higher the temperature of the oil, the less steam is required to atomize it.

8. Boilers Should not be Forced Excessively.—Boilers should never be forced unless necessary. If there are not enough boilers in the plant to maintain steam pressure without forcing them, more boilers should be installed or the steam requirements should be reduced. In case of a variable load, it is permissible to force the boilers for short periods during peak load, but during the periods of lighter loads the same number of boilers should be kept in operation without forcing. Forcing of boilers results in high flue-gas temperature and should not be resorted to unless the plant is specially designed with this end in view and

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provided with economizers to absorb the excess temperature of the gases.

9. Shutting Down Boilers for Short Periods Should be Avoided. In case of a variable load it is more economical to keep the same number of boilers in operation continuously than to fire up extra boilers for the peak load. This means forcing the boilers somewhat during the peak, but it avoids the loss due to heating up cold boiler settings, which is a considerable waste. It usually requires eighteen to twenty hours to heat boiler settings to their maximum temperature, so it is desirable to keep them in operation as steadily as practicable. In standby plants and plants that must be partly or wholly shut down during a portion of each day special precautions must be taken to prevent waste of oil during the shut-down period. As soon as the oil burners are shut off the ash-pit doors and dampers should be shut tight. If the dampers leak, they should be made tight. A boiler with a tight damper will maintain its full steam pressure for several hours after the fires are put out if no steam is drawn off. If the damper leaks, the steam pressure will begin to drop immediately, and much more oil will be required to heat up the boiler when it goes back into service.

10. Oil Should not be Sprayed into Furnace Unless There is a Fire.—In operating oil-fired boilers it is extremely important to avoid any accumulation of gas in the boiler setting because of the danger of explosion; consequently no oil should be allowed to get into the furnace unless there is a fire to ignite it and no more oil should be fed into the furnace than can be burned with the available quantity of air and atomizing steam.

To light an oil fire under a boiler, first open the damper and ash-pit doors; next place a lighted torch in front of the burner; next turn on the atomizing steam; next turn on the oil, which should immediately ignite. If the oil does not ignite immediately, turn it off at once and investigate to find out the reason before turning it on again.

11. Feed Water Uniformly.—Water should be fed to the boilers continuously and uniformly. If it is fed intermittently it may become too hot in the feed-water heater to absorb all of the heat in the exhaust steam, and at other times it will be too cold, thus causing considerable waste. Intermittent feeding also results in unsteady steam pressure, which in turn means frequent altering of the fires, variable furnace temperature and increased difficulty

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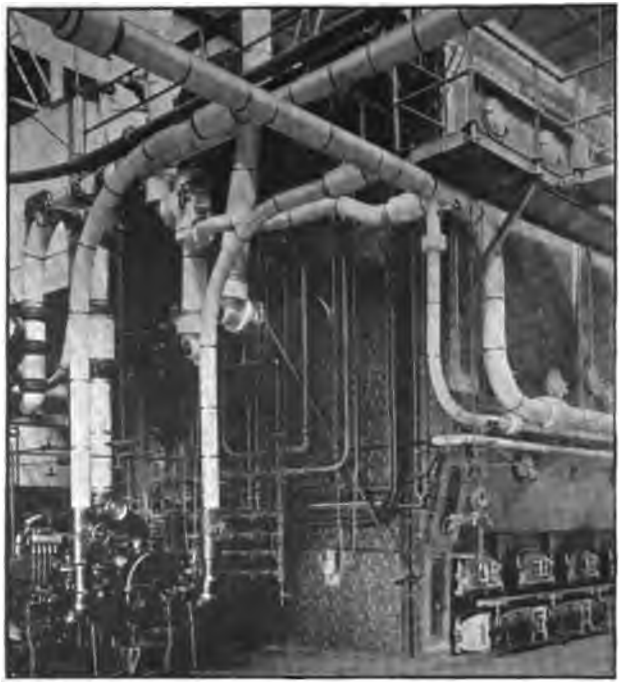

FIG. 118.

FIG. 119.

FIGS. 118–119.—Economy measuring apparatus.

To save in the boiler room the soot should be carefully blown from time to time and an accurate record kept of the various factors that lead to a knowledge of the state of combustion in the furnace. In the view to the left is shown a soot blower installation, while to the right may be seen meters for recording the draft, temperature of chimney gas leaving the boiler, feed water temperature, carbon dioxide (CO_2) in the chimney gas leaving the economiser. One set of this apparatus is provided for each pair of boilers at the Long Beach Plant of the Southern California Edison Company.

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in regulating the air supply. For best efficiency the furnace conditions should be kept as steady as possible. It is better to allow the steam pressure to vary a few pounds than to be continually altering the fires.

12. Keep Boilers Clean and in Good Repair.—The importance of keeping the boiler clean both inside and outside and in good repair is recognized by every operating engineer and need not be enlarged upon here. There are some engineers, however, who, while realizing that the boilers must be kept clean, are of the opinion that the quantity of soot resulting from an oil fire is so slight as to be negligible. Experience shows, however, that this is not the case, and it is essential to provide either mechanical soot blowers or some sort of steam lance with all oil-burning boilers. Furthermore, it is essential that these devices be used at intervals frequent enough to keep the heating surface of the boiler free from deposits of soot. In actual practice it is found that blowing the soot from the tubes of an oil-fired boiler results in a reduction in flue-gas temperature of 80° to 100°F. (45° to 55°C.), which means an increase in efficiency of 1½ to 3 per cent.

The inside of the boiler must be thoroughly washed out and all scale removed at regular intervals, the length of interval depending on the quality of feed water used. Scale should not be allowed to accumulate more than ⅛ in. (3.2 mm.) thick. The soot on the outside of the boiler should be blown off at least once a day. Scale and soot constitute extremely efficient heat insulation materials, the thermal conductivity of scale being about one-fifth that of steel so that ¼ in. (6.4 mm.) of scale is equivalent to a boiler tube 1¼ in. (3.2 cm.) thick.

The boilers should be blown down regularly at least once a day, and oftener if the water contains soluble salts which tend to cause foaming. About half a gage glass of water should be blown out each time, and this should preferably be done when there is no fire under the boiler. Samples of water drawn from the boiler should be tested for salt, which should not be allowed to concentrate above 150 grains per gallon (2.53 gm. per cu. dcm.).

13. Maintain Baffles and Flame Plates in Good Condition.—If boiler baffles are allowed to become displaced, the gases will short-circuit from one pass to another, making some portions of the heating surface ineffective. This results in high stack temperature and poor economy. The baffles should be kept gas-

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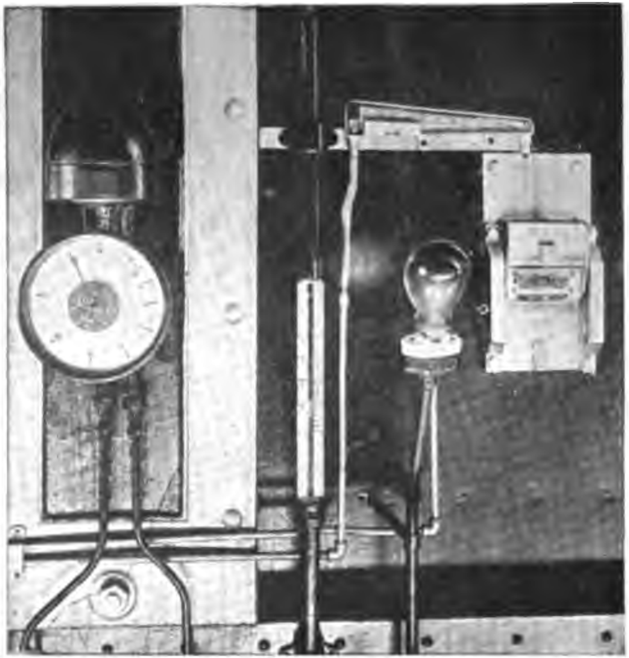

Fig. 120.

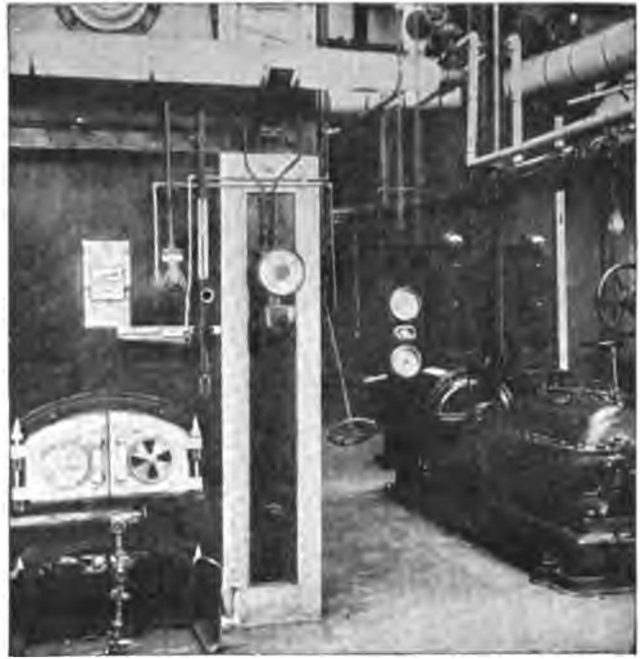

Fig. 121.

Figs. 120–121.—Economy measuring apparatus.

Meters for individual units often prove helpful, as do particular sets of instruments for steam supply to the larger power units. In the view to the left is shown the steam flow meter, carbon dioxide (CO_2) indicator and draft gage installed on the front of each boiler unit. Note that the CO_2 meter registers 14 per cent. and the draft 0.23 in. To the right is seen a steam flow meter, a draft gage and indicating carbon dioxide (CO_2) meter on the boiler front and the Venturi meter for measuring the total feed water of eight boilers on a 20,000 kw. steam turbine at the Long Beach Plant of the Southern California Edison Company.

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tight at all times, and any baffle tiles that have been disturbed by the replacing of tubes, gas explosions or from any other cause should be put back into proper condition before the boiler is fired up. The openings through the baffles for the passage of gases should be maintained the right area to suit the capacity required and the draft available.

14. Use Recording Instruments Wherever Practicable.—Reliable records of the actual performances of a plant are of great aid to its efficient operation. By this means guess work is avoided and it can be determined accurately what method of operation or which man produces the greatest efficiency. To obtain efficient results means not only the installation of efficient machinery and recording apparatus but constant attention to operation.

Recording CO_2 meters or an Orsat apparatus and draft gages are essential for efficient furnace operation, because they show excess air entering the furnace—the greatest single avoidable loss in the combustion of fuel. Recording steam gages and flow meters are also of considerable assistance, the latter particularly because they show the quantity of steam generated by each boiler. A pyrometer or flue-gas thermometer should be used to measure the temperature of the exit gases. An additional steam-flow meter should be connected to the atomizing steam pipe and a record kept of the quantity of steam used by the burners. Thermometers and recording pressure gages on the steam and oil lines close to the burners also give valuable information.

15. Determine Efficiency Daily from Records.—The actual evaporation per pound of oil may be obtained by means of water meters and oil meters. The quantity of steam used by the burners may be obtained by means of a steam-flow meter in the steam-to-burner line. The greater the net evaporation per pound of oil, the higher is the efficiency of the boiler.

By keeping accurate records of the daily performances of boilers it is possible to compare one day's results with another. By trying different methods of operation, different furnace arrangements and different intensities of draft and comparing one with another, the best and most economical method of operating may be determined with certainty.

16. Fire Boilers Scientifically.—Boilers should be fired scientifically; that is, by basing the method of operation on the flue-

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gas analysis, temperature of escaping gases and results of tests and plant records. With careful attention paid to the draft readings and adjustments of dampers, this method will usually result in considerable saving. The recognition that trained engineers are rapidly receiving is most convincing testimony that the value of scientific management is becoming increasingly appreciated.

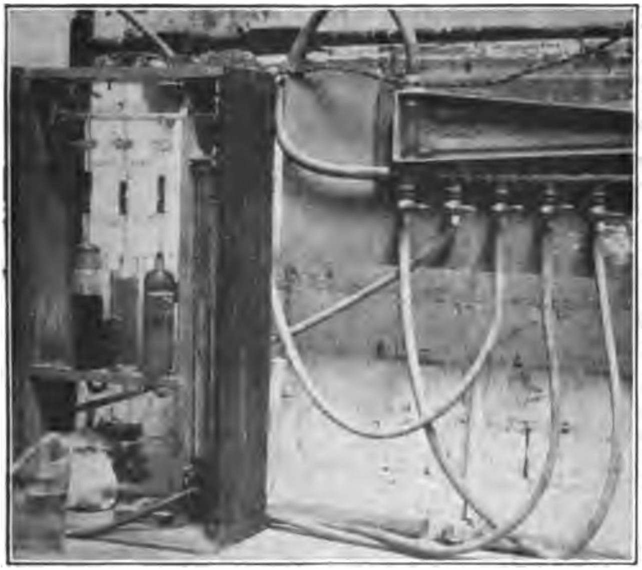

FIG. 122.—Five outlets for measuring chimney draft pressures with one draft gage.

Co-operation with the employer and with other employees is absolutely necessary for successful operation of the plant. If the plant requires new equipment or repairs to old equipment in order to improve its economy, these improvements should be suggested to the employer, with an explanation of the resulting advantages.

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CHAPTER XXIV

FUEL OIL BURNING APPLIANCES

In its course from the point of delivery at the plant to the burners the oil must pass through a number of appliances, which are necessary for the complete equipment of any oil burning plant. In this discussion we shall follow the oil in its journey through the plant and describe briefly the various appliances required for handling it.

Fig. 123.—The cars are shown on the unloading track of the Redondo Steam Plant of the Southern California Edison Company and the oil is emptied from them into the flume which runs beside the tank; thence it goes into a small underground tank from which it is pumped into the main storage tank.

Oil may be delivered at the plant either by rail in tank cars, or by water in barges or tank steamers specially constructed for the purpose. From these it is pumped into large storage tanks, which may be of either concrete or steel.

Storage Tanks.—For power plants large cylindrical steel storage tanks are used. These are usually set on the ground

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outside the plant, and are built in any desired size up to 50,000 bbl. capacity. They are built up of riveted steel plates, the thickness of plate and strength of riveted joint being proportioned in accordance with the usual safety rules based on the internal pressure due to the head of oil inside the tank. Thus if the tank is 30 ft. high the internal pressure will be that due to 30 ft. head of oil, or approximately 15 lb. per square inch. It is customary to surround the storage tank by a concrete wall about 10 ft. high far enough away from the tank so that the entire

Fig. 124.—Main oil storage tanks at Station C, Pacific Gas and Electric Company, Oakland, California. Tank in foreground is set low for fire protection purposes, while tank in rear is surrounded by concrete retaining walls.

contents of the tank will be held in by the wall in case of a leak in the tank. This is to prevent the oil from leaking out to the surrounding country.

The size of storage tank required depends on two factors:

1. Quantity of oil to be burned.
2. Availability of oil supply.

This second factor depends on the location of plant, the method of delivery, and the probability of interruptions in delivery, all of which matters must be carefully considered in determining the number of days oil supply that should be carried at the plant. Most power plants are provided with tanks of sufficient size to enable them to keep from 10 to 30 days' supply of oil on hand.

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This storage capacity should preferably be divided among two or more tanks rather than all concentrated in a single tank, as this will enable one tank to be emptied for cleaning and repairs without shutting down the entire plant.

In built up districts within the fire limits of cities it is not permissible to locate the storage tanks above ground. The National Board of Fire Underwriters have adopted certain rules for the location of oil storage tanks, which will be found in

FIG. 125.—One of the pumps which handle the oil from the main storage tank to the auxiliary tank. This pump was originally arranged for belt drive but was found unsatisfactory. The gearing is now direct.

Appendix III, page 415. Similar rules have been adopted by several cities. In general these rules provide that within the fire limits of cities the tank must be located so that its top is at least 3 ft. below the level of the fireroom floor and below the lowest pipe in the building to be supplied. The tank must be set on a firm foundation and covered with soft earth or sand, no air space being allowed immediately outside the tank.

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Every oil storage tank must be provided with the following attachments:

Filling pipe
Suction pipe
Vent pipe
Smothering pipe
Overflow pipe
Measuring rod or chain

For tanks less than 1000 gal. capacity the filling pipe and vent pipe may be on the same connection. For larger tanks separate connections are required. The vent pipe must extend from the top of the tank to a point outside the building at least 12 ft. above the top of the highest tank car from which the storage tank may be filled. All outlets on the tank should be located on top, the suction pipe running down inside the tank to near the bottom. The smothering pipe consists of a small steam pipe through which steam can be blown in case of fire, thus keeping air away and effectually smothering the flames. The overflow pipe is arranged to carry back to the storage tank all oil not used. An automatic relief valve is provided on the oil pump discharge set to open at a predetermined pressure, and discharging through the overflow pipe back to the storage tank. All pipes should be run as direct as possible and pitched toward the storage tank. The oil in the tank may be measured by means of a rod or chain let down through the top of the tank. The use of gage glasses should be avoided as they are liable to break, causing leakage of oil.

Many power plants are provided with a service tank located under the fireroom floor, in addition to the main storage tank outside the building. The service tank is filled at intervals from the storage tank, and the oil pumps take their supply from the service tank and distribute it to the oil burners.

Measurement of Oil.—Oil is ordinarily measured by passing a rod or chain down through the top of the storage tank, the rod being marked off in feet, inches and fractions of an inch. By sounding to the bottom of the tank, the depth of oil can be determined very accurately. A more convenient method, though not quite so accurate, is to use a float with a chain passing over a pulley at the top of the tank, the outer end having a

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pointer which indicates the height of oil in the tank on a suitably calibrated scale. The height of oil in the tank may also be determined by an indicating or recording pressure gage, which depends for its operation on the hydrostatic pressure produced by the oil.

After determining the height of oil in the tank it is necessary to convert the measurement into gallons or barrels of oil. To do this it is necessary to carefully calibrate the tank either by pumping into it known quantities of oil, or by taking careful measurements and calculating its contents. This latter method is very simple if the tank has vertical sides. If, however, a cylindrical tank lying horizontally is used, the calculation is somewhat complicated. The table in Fig. 127 giving the capacity of horizontal cylindrical tanks per foot of length for various depths of liquid will be of assistance in this connection.

FIG. 126.—Gage to measure height of oil in 30,000 barrel tank at North Beach Steam Plant of Sierra and San Francisco Power Company, San Francisco. The rod inside the pipe is calibrated in feet and inches, passes down into the tank and rests on a float supported by the oil. An electric lamp attached to counter weight enables it to be seen at night.

After determining the volume of oil in a tank it is necessary to make correction for its temperature, for, like everything else, oil expands and contracts with changes of temperature and the volume measured at one temperature will not be the same as the volume of the same weight of oil at another temperature. The amount of variation of the volume of the oil depends on its coefficient of expansion. The coefficient of expansion varies for different oils, but the average value for California oils is usually taken at 0.0004 for each degree Fahrenheit change in temperature; that is, the oil expands four ten thousandths of its volume for each degree F. rise in temperature. This is equivalent to 0.00072 per degree Centigrade.

In practice 60°F. has been adopted as the standard tempera-

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Table of Capacity in Gallons of Cylindrical Tanks per Foot in Length Laying Horizontally

Diam. of Tank	Depth of Liquid 6"	1'-0"	1'-6"	2'-0"	2'-6"	3'-0"	3'-6"	4'-0"	4'-6"	5'-0"	5'-6"	6'-0"	6'-6"	7'-0"	7'-6"	8'-0"	8'-6"	9'-0"	9'-6"	10'-0"
6"	146																			
1'-0"	294	588																		
1'-6"	385	9.36	1322																	
2'-0"	460	1175	1885	2158																
2'-6"	522	1371	2307	3149	3673															
3'-0"	581	1541	2640	3747	4105	5288														
3'-6"	630	1695	2947	4246	5501	6566	7196													
4'-0"	678	1937	3216	4697	6186	7562	8722	9402												
4'-6"	721	1967	3465	5101	6797	8930	9926	111.8	118.9											
5'-0"	765	2090	3705	5485	7338	9108	109.9	123.9	139.1	146.9										
5'-6"	800	2207	3925	5834	7854	99.12	119.4	138.5	155.6	169.7	177.7									
6'-0"	838	2319	4136	6164	8336	105.8	128.1	149.8	170.1	188.3	203.1	211.5								
6'-6"	875	2416	4331	6485	8796	111.9	136.3	160.2	183.5	204.9	221.6	239.4	248.8							
7'-0"	915	2521	4562	6789	92.25	117.8	143.6	169.8	195.9	219.9	242.6	262.5	279.0	287.9						
7'-6"	942	26.11	4705	7069	9649	123.4	151.2	179.1	207.2	234.1	259.5	283.5	304.6	310.9	336.6					
8'-0"	980	2708	4877	7545	100.2	128.8	158.1	187.7	217.6	274.2	275.8	302.2	326.9	348.6	366.1	376.2				
8'-6"	10.0	2799	5042	7650	104.1	133.9	164.7	195.9	228.1	259.6	290.6	320.1	348.5	374.0	396.4	414.4	424.5			
9'-0"	10.4	28.84	5206	7861	107.8	138.4	171.1	204.3	237.9	271.5	309.7	336.6	368.1	397.2	423.4	446.6	465.5	475.9		
9'-6"	10.7	2970	5367	81.12	110.9	143.5	177.3	211.9	247.4	282.9	318.2	352.9	386.6	413.8	448.9	476.5	500.5	519.6	530.2	
10'-0"	10.9	3057	5466	85.62	114.8	148.5	183.2	219.8	256.9	293.7	331.1	368.0	404.2	435.2	471.7	501.7	532.2	556.9	576.5	587.6

Example; Diam. of Tank 6'-0". Length of Tank 8'-0" Depth of Water 2'-0" From table, opposite 6'-0" and under 2'-0" the capacity for one foot in length is 61.64 gallons. For 8'-0" in length, the capacity = 61.64 × 8 = 493.12 gallons for 2'-0" in depth

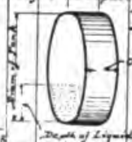

Fig. 127.—Tabulated data on cylinder volumes.

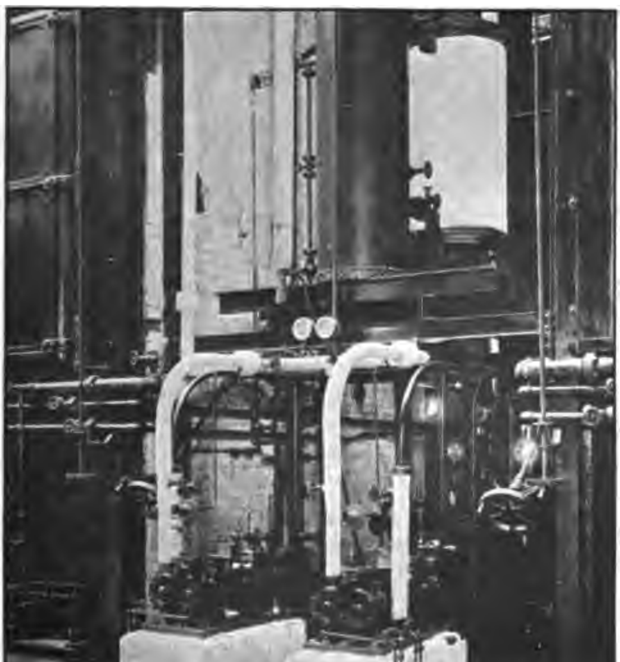

Fig. 128.—Fuel oil pumps and heaters, Pacific Gas and Electric Company, station C, Oakland. Below are shown the duplicate reciprocating fuel oil pumps. Above is a large air receiver and a coil pipe oil heater.

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ture, and to reduce the measured volume to the true volume at 60°F. the following formula is used:

$$V_{60} = \frac{V_t}{1 + [(t - 60) \times 0.0004)]}$$

where V_{60} = the volume of oil at 60°F.

V_t = the volume of oil as measured, both expressed in barrels, gallons or cubic feet as the case may be.

t = temperature of the oil when measured, in degrees F.

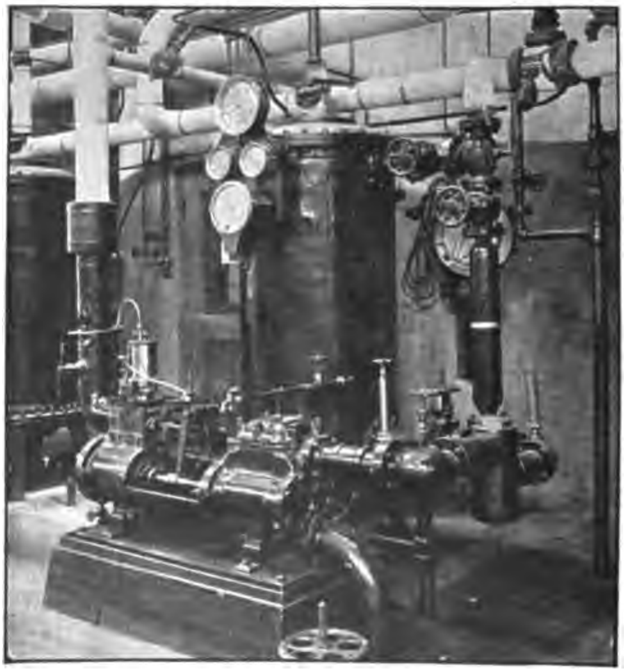

Fig. 129.—One unit of a duplicate set in the Long Beach Steam Plant of the Southern California Edison Company, consisting of a fuel oil pump in the foreground with the oil heater immediately behind it.

Thus if the sounding in a tank before and after filling show that 960 bbl. have been added, and the observed temperature of the oil was 80°F., then the true volume at 60°F.

$$V_{60} = \frac{960}{1 + (20 \times 0.0004)} = 952 \text{ bbl.}$$

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If instead of 80°F. the temperature of the oil had been 50°F., the true volume at 60°F. would be

$$V_{60} = \frac{960}{1 + [-10 \times 0.0004]} = 964 \text{ bbl.}$$

Oil Pumps.—The oil is taken from the supply tank by the oil pump. The type of pump usually used for this purpose is the ordinary duplex steam driven reciprocating pump. The pump

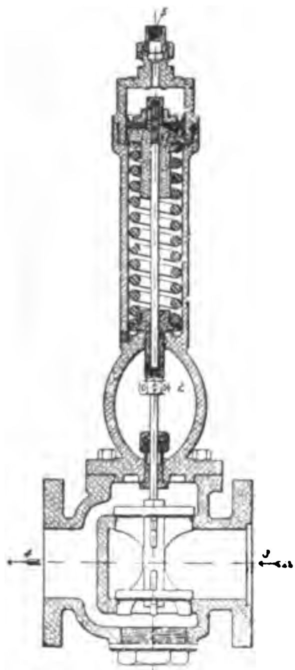

FIG. 130.—The Witt pump governor. It controls the running of a pump and will hold a steady pressure on the discharge line.

must have brass valves, and metallic or other packing that will not be affected by the oil. The pump should be provided with a large air chamber to prevent pulsations of oil pressure due to the strokes of the pump. It is customary to install the pumps in duplicate so that one may be kept shut down at all times ready to go into service immediately if the other has to be shut down for repairs. The pumps should be of sufficient size to deliver the maximum quantity of oil required when operating at compara-

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tively slow speed—not more than 15–20 strokes per minute. The pump must not be set too high above the oil tank, for oil cannot be raised by suction as high as water. The maximum suction lift permissible is about 16 ft.

The oil pumps should be provided with a pump governor for the purpose of maintaining a steady oil pressure. An example of this is the Witt pump governor shown on page 209. It consists of a double ported throttle valve placed on the steam

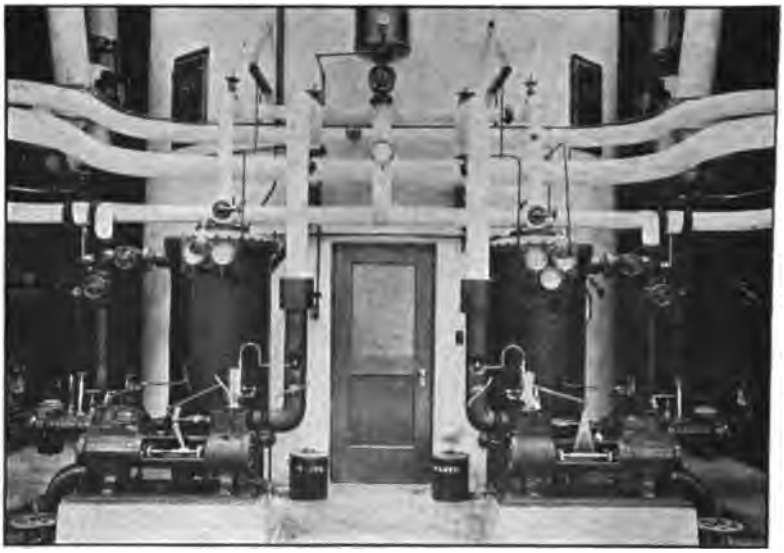

Fig. 131.—Fuel oil pumps for turbine No. 3 at the Long Beach Plant of the Southern California Edison Company. Note the neat and attractive appearance throughout in this interesting installation, quite typical where oil is used as fuel.

line supplying the pump. The valve stem is attached to a spring loaded piston which is actuated by the oil pressure. If the oil pressure increases the valve partially closes, thus slowing down the pump. If the oil pressure drops the valve opens wider and the pump is speeded up. Any predetermined pressure may thus be maintained by adjusting the spring.

Strainers.—Every oil burning plant must be provided with some form of strainer to remove the dirt and foreign matter which would be liable to cause stoppage of the burners. The strainer may be placed either in the suction line between the supply tank and the pump, or in the discharge line after leaving

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the pump, or both. The strainer usually consists of a perforated metal basket mounted in a suitable container, arranged so that the basket can be readily removed for cleaning. The Staples

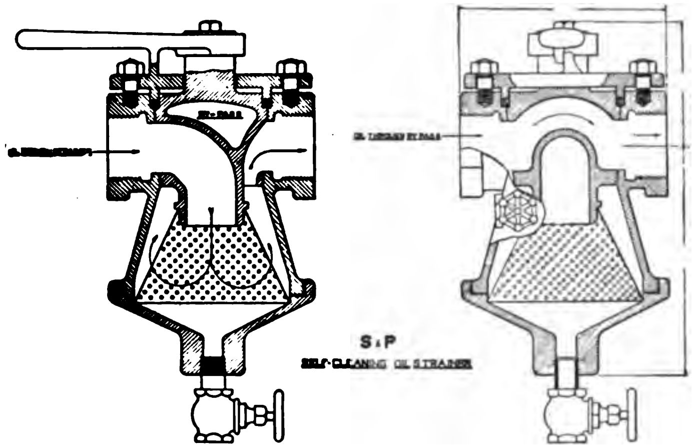

FIG. 132.—Staples and Pfeiffer self-cleaning oil strainer.

and Pfeiffer so-called self-cleaning strainer, which is shown in the illustration, is provided with a by-pass and arranged so that the dirt can be blown out by a steam jet without removing the basket.

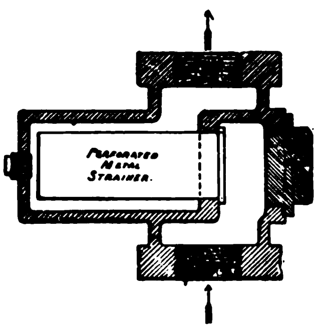

FIG. 133.—This is a simple form of a strainer in which the basket can be readily removed for cleaning.

FIG. 134.—The Elliott twin oil strainer.

In addition to the main strainer on the oil line it is advisable to provide a small fine mesh strainer at each oil burner. To clean this strainer it is only necessary to turn the handle on top so as to run the oil through the by-pass, place a bucket under the

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blowout valve at the bottom and blow steam through by opening the small valve on the side.

Oil Heaters.—Before reaching the oil burners the oil must be passed through an oil heater to bring it up to a temperature suitable for atomizing. The oil heater is usually placed between the pump and the oil burners, a convenient method being to mount the pumps over the heater, as shown in Fig. 135 the exhaust steam from the pump being utilized as the heating medium.

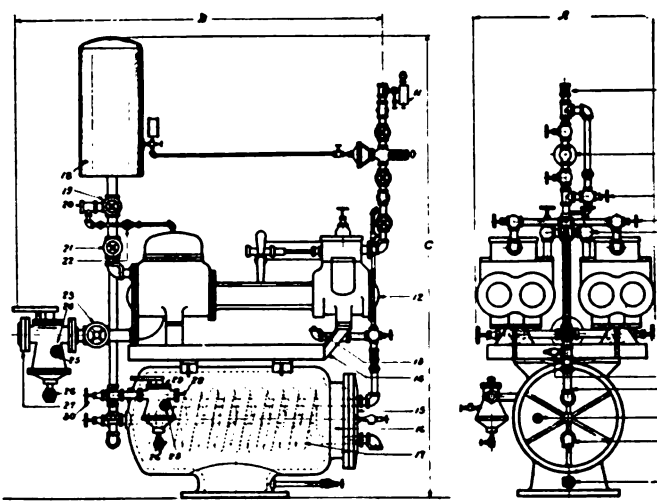

Fig. 135.—Staples and Pfeiffer oil pump and heater unit.

1. Steam to pumps
2. Pump governor and pressure regulator
3. Steam by-pass valve
4. Steam valves
5. Exhaust steam valves
6. Pop safety valve in exhaust line
7. Exhaust steam in heater
8. Pop safety valve for heater
9. Exhaust from coil to trap or atmosphere
10. Heater oil drain and blow-out
11. Side feed lubricator
12. Duplex pumps
13. Independent exhaust cone
14. Drip pan
15. Heater head
16. Oil heater
17. Copper coil
18. Air cushion tank
19. Automatic relief valve
20. Oil return to tank
21. Oil discharge valve
22. Gas relief valves
23. Oil suction valves
24. Suction oil strainer
25. Steam blow-out valves
26. Strainer blow-out valves
27. Oil suction from tank
28. Oil discharge to burner
29. Oil discharge strainer
30. Cold oil by-pass valve

Heaters invariably consist of a series of tubes or coils with oil on one side of the metal and steam on the other, the heat passing through the metal from the steam to the oil. There are several different ways in which this may be accomplished: thus the heating surface may be composed of either a coil or a number of straight tubes; the oil may flow through the tubes with the steam outside, or the oil may surround the tubes with the steam

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on the inside. All of these methods are used in different heaters now on the market.

The size of heater is determined by the formula

$$S = Hk\,(t_s - t_o)$$

where S = heating surface in square feet.
H = heat absorbed in B.t.u. per hour.
k = coefficient of heat transfer.
= B.t.u. absorbed per hour per square foot per degree difference in temperature.
t_s = mean temperature of steam, deg. F.
t_o = mean temperature of oil, deg. F.

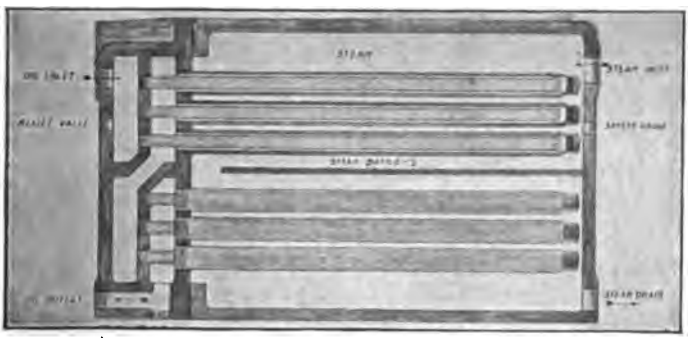

FIG. 136.—Nelson Fuel Oil Heater. This is a multipass heater using the principle of "porcupine" tubes, the tubes being free at one end for expansion and contraction.

The quantity H, heat absorbed per hour, is found readily by the formula

$$H = W\,(t_2 - t_1)c$$

where W = weight of oil heated per hour in pounds.
t_1 = initial temperature of oil, degrees F.
t_2 = final temperature of oil, degrees F.
c = specific heat of oil = 0.498.

The quantity k, coefficient of heat transfer varies with the difference in temperature between the steam and the oil, and with the velocity of oil in passing through or around the tubes. This velocity is of great importance, as the greater the velocity the better is the oil scraped from the side of the tube, thus allowing colder oil to come in contact with the hot surface. It is evident

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therefore that a high velocity of oil is desirable. There is a limit, however, to the velocity attainable, as the higher the

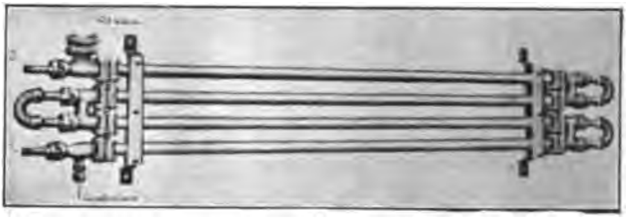

FIG. 137.—The "Coen" multiunit oil heater.

velocity the greater is the drop in pressure of the oil in passing through the heater. The drop in pressure in turn depends largely on the viscosity of the oil, so that viscosity has an important bearing on the heat transfer. The value of the coefficient, k, therefore varies between wide limits and for ordinary conditions may be said to lie between 15 and 50 B.t.u. per hour per square foot per degree difference in temperature.

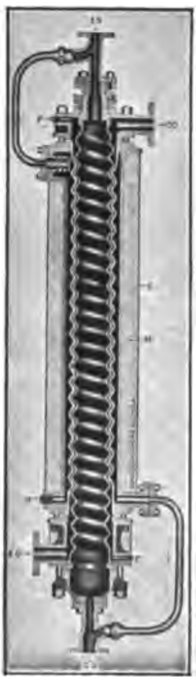

FIG. 138.—The Koerting fuel oil heater.

For a heater in which the oil flows through the tubes it is a simple matter to divide the flow into passes so as to obtain the required velocity, the oil passing through first one group of tubes and then another. If the heater is designed to contain the oil in a shell outside the tubes, there is a tendency for the oil to short circuit across from the inlet to the outlet, leaving portions of the heating surface surrounded by dead or stagnant oil. To overcome this it is necessary to place baffles in the shell causing the oil to travel back and forth, and producing what is known as turbulent flow. If the baffles are properly designed and the oil flow is sufficiently agitated it is possible to obtain as good heat transfer for a given drop in pressure by this means as by passing the oil through the tubes.

While oil heaters usually consist of a shell containing a series of tubes or coils, there are on the market a few special designs.

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One of these is the Coen multiunit oil heater, which is illustrated in Fig. 137. This heater is similar in design to the ordinary ammonia condenser used in ice machines, and consists of a series of double pipes, one inside the other, connected together by standard ammonia fittings. This heater may be constructed in any length or number of legs as desired. The oil passes through the inside pipe, and the steam is in the annular space between the two pipes.

Another heater of unusual design is the Koerting Fuel Oil Heater, illustrated in Fig. 138. This heater consists of a pair of spiral corrugated tubes, one inside the other, and both inclosed in a shell. The oil enters at *EO* and passes up the thin annular space, leaving at *DO*. The steam enters at *ES*, and is carried both inside the inner tube and outside the outer tube. For cleaning, the inner tube may be removed, or steam blown through the plugged openings *FF*.

Oil Burners.—After leaving the heater the oil is led through piping and suitable regulating valves to the oil burners or atomizers, where it comes in contact with the atomizing agent and is delivered to the furnace in the form of a fine spray. A description of a number of different types of oil burners will be found in Chapters XXI and XXII.

Oil Piping.—Ordinary wrought iron or steel pipe is used for oil, the smaller sizes being screwed and the larger flanged. Gaskets of corrugated copper or compressed asbestos fibre are used. The size of pipe in most power plants is such as to give the oil a velocity of not more than 2 ft. per second.

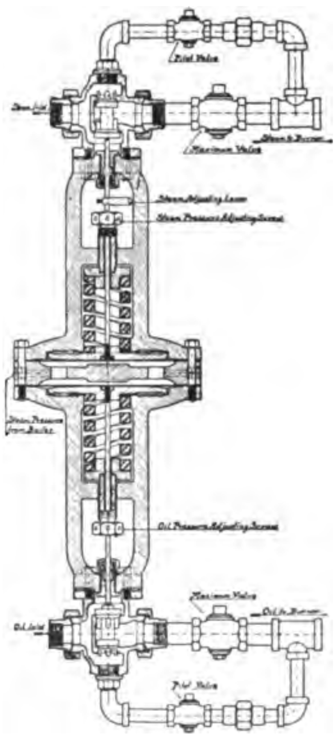

Fig. 139.—G. E. Witt Co.'s improved oil burner governor. A device for automatically controlling the flow of steam and oil to burners.

Automatic Regulators.—While in the majority of plants the oil is regulated by means of hand operated valves, automatic

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regulation has met with great success and is used quite extensively.

The Witt improved oil burner governor shown in the illustration on page 215 consists of two independent diaphragm operated valves, controlled by springs, mounted so as to have the boiler steam pressure between the two diaphragms. This governor regulates both the oil supply and the steam supply to the burners.

Fig. 140.—On the left may be observed oil-to-burner Moore regulator controlling the steam pressure and the oil pressure while on the right may be seen the damper controller that works by hydraulic cylinder actuation at the Arizona Power Company, Phoenix, Arizona.

It is provided with pilot valves which prevent the fire going out when the load is light, and with maximum valves which prevent the fires becoming larger than a predetermined point. The governor, therefore, regulates the oil and steam between these two extremes.

The Moore automatic fuel oil regulator, which is illustrated in Figs. 9, 140, 141 and 142 regulates not only the oil and steam but also

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the air required for combustion, thus controlling the three essential elements for firing the boiler. This apparatus consists of three separate regulators, one for the oil, one for the atomizing steam and one for the air. These regulators are made up on the principle of the well known Spencer damper regulator, and the set of three can be arranged by suitable piping and shafting to control the firing of a number of boilers, and in many cases of the whole plant. In the oil regulator the diaphragm is operated by

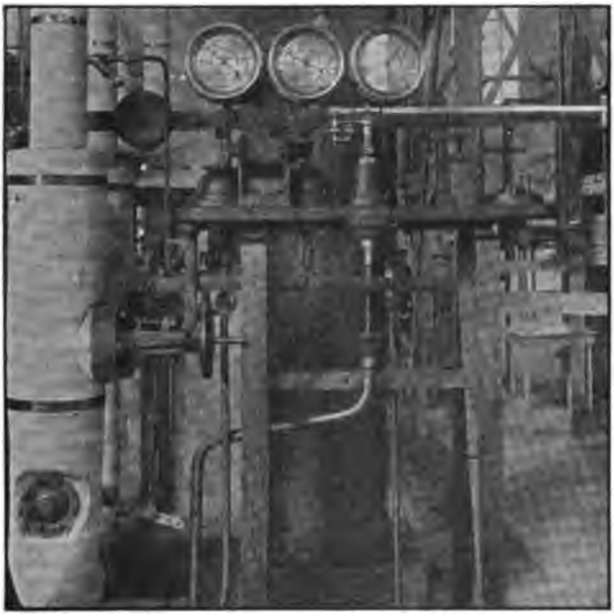

FIG. 141.—Moore steam to burner regulator controlling the atomizing steam at the Arizona Power Company's Plant, Phoenix, Arizona.

the boiler steam pressure, and the power lever is used to control a regulating valve in the main oil pipe supplying the burners. By this means a slight change in the boiler pressure is made to cause considerable change in the oil pressure, and as the quantity of oil supplied to the burners varies with the oil pressure, the fires in all boilers are increased or diminished gradually and simultaneously. This variable oil pressure is then made to act on the diaphragms of the other two regulators.

In the atomizing steam regulator there are two diaphragms, one acted on by the controlling oil pressure and the other con-

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nected to the atomizing-steam pressure near the burners. These diaphragms are connected by levers which, acting through the water motor and connecting rod operate a chronometer valve in the atomizing steam main. Thus any increase in oil pressure causes a definite fixed increase in atomizing steam pressure. The steam pressure required has been found by experiment to be a multiple of the oil pressure plus a fixed pressure. This relation-

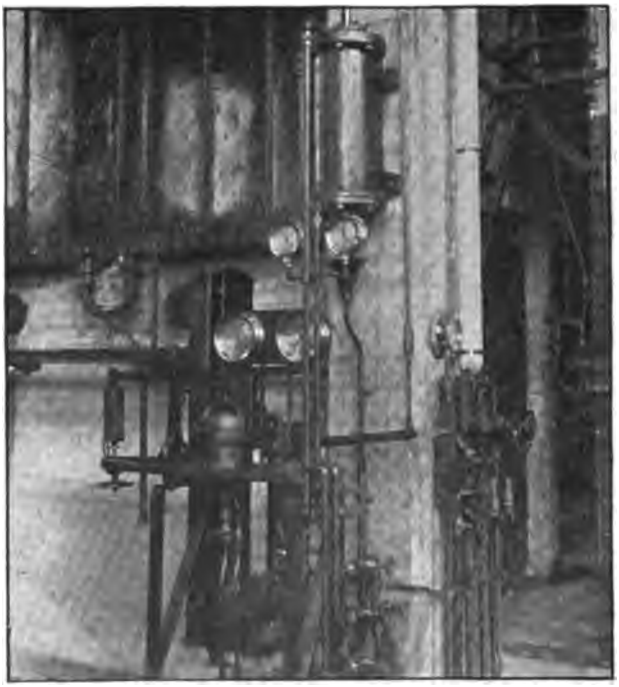

FIG. 142.—Damper regulator for Moore automatic oil firing system at the Arizona Power Company's Plant, Phoenix, Arizona.

ship is maintained by the regulator, the proportion being varied by the adjustable fulcrum and weights to suit the requirements of the type of burner employed.

The air is controlled by the third regulator which operates a rock shaft connected to the dampers of all boilers. In this regulator the motion caused by the diaphragm, which is acted on by the oil pressure, is resisted by a coil spring. The amount of movement of the lever is, therefore, proportional to the oil pressure.

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This movement is communicated by means of a controlling valve and differential lever to a hydraulic cylinder, which in turn operates the rock shaft connected to the dampers.

A more complete description of the Moore Automatic Regulator will be found in a paper by C. R. Weymouth on "Unnecessary Losses in Firing Fuel Oil" published in Vol. 30 of the Transactions of the American Society of Mechanical Engineers.

The Merit automatic oil stoking system, which is illustrated in Figs. 143 and 144, operates on the principle of controlling the fires in a series of steps, the fire jumping from a small fire to

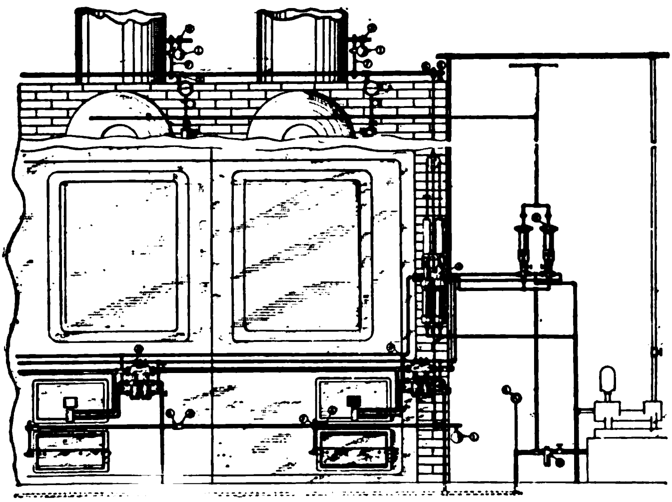

FIG. 143.—A typical automatic system of control.

Diagrammatic view, showing manner of control for the oil, the ashpit and the damper:

A. Master controller
B. Double oil strainer
C. Oil gage
D. Regulator
E. Single bearings
F. Damper arms
G. Clevices
H. Damper hubs
I. Damper weights
J. Interlocking damper
K. Special brackets

an intermediate fire and then to the maximum fire. The oil, atomizing steam and air are all three controlled by this regulator.

A diagram of this regulator is shown in Fig. 143. The boiler steam pressure acts on the diaphragms of the master controller set shown in Fig. 144, which consists of two parts, one for the maximum fire and one for the medium fire. Each of these is piped up to a damper operating device, and to a regulating device on each burner, which operates both the steam and the oil valve to the burner. For convenience, fuel oil from the main oil supply pipe is used as the operating fluid, returning to the oil

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pump suction when used. If the boilers have full steam pressure up they will be working on the small fire. If the steam pressure begins to drop the first master controller diaphragm comes into play, opening the dampers to their medium position, and opening the intermediate oil and steam valves to each burner. These valves are always open or shut, the amount of opening being fixed by adjustable auxiliary valves. If the steam pressure

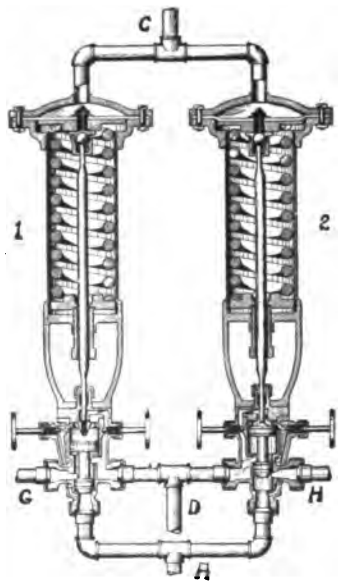

FIG. 144.—At the Bush Street Station of the Great Western Power Company in San Francisco, oil under pressure is used to operate the regulators and the interlocking damper devices as directed by the master controller shown in the illustration. This master controller controls the flow of oil for the automatic opening and closing of dampers and burners.

continues to drop, the second master controller comes into play opening the damper to its full open position, and opening the remaining oil and steam valves to each burner, thus placing the maximum fire in operation. This condition will continue until the steam pressure rises sufficiently to shut off the maximum fire, when the boilers will return to the intermediate fire, and the operation will be repeated.

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CHAPTER XXV

CHANGING FROM COAL TO OIL

When it is contemplated to change from coal firing to oil firing, the first thing to be considered is the relative cost of the two fuels. This does not mean merely the cost of a ton of coal compared to the cost of a ton of oil, because oil has a far greater heating value than an equal weight of coal. Again, while oil has a fairly uniform heating value, there are great differences in the heating values of different kinds of coal. Consequently in making the comparison it is necessary to know the kind of coal under consideration, and its heating value per pound. Even then we have not gone quite far enough, for the boiler efficiency is not the same for all grades of coal, and is higher for oil than for coal. Thus, with a good grade of semi-bituminous coal an efficiency of 75 per cent. is readily obtainable, whereas with a low grade bituminous coal or lignite it is difficult to obtain more than 60 per cent. efficiency under ordinary methods of firing. With oil on the other hand, tests have shown net efficiencies of over 80 per cent. and with careful operation it is readily possible to maintain 78 per cent. efficiency in regular plant operation.

Knowing the relative prices and heating values, and the probable boiler efficiency, it is a simple matter to calculate the saving that may be effected by changing from coal to oil. Suppose, for example, that the owner of a plant is purchasing coal at $6.00 per ton of 2000 lb. and that this coal contains 6 per cent. moisture and has a heating value of 13,000 B.t.u. per pound dry. He is considering changing over to oil which he can purchase for $1.50 per bbl. of 42 gal. The oil has a gravity of 16°Bé. and therefore weighs 336 lb. per bbl.; it contains 1 per cent. water and its heating value when free from water is 18,500 B.t.u. per pound.

Since the coal contains 6 per cent. moisture it is 94 per cent. dry, and 1 ton of coal contains

$$2000 \times 0.94 \times 13{,}000 = 24{,}440{,}000 \text{ B.t.u.}$$

Similarly 1 bbl. of oil contains

$$336 \times 0.99 \times 18500 = 6{,}153{,}840 \text{ B.t.u.}$$

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If both fuels could be burned with the same efficiency, then by dividing 24,440,000 by 6,153,840 we would find that one ton of coal is equivalent to almost 4 bbl. of oil. However, if the oil can be burned with an efficiency of 78 per cent. and the coal with an efficiency of only 69 per cent., we find that the useful heat in one ton of coal is

$$0.69 \times 24,440,000 = 16,863,600 \text{ B.t.u.}$$

and the useful heat in 1 bbl. of oil is

$$0.78 \times 6,153,840 = 4,800,000 \text{ B.t.u.}$$

Consequently one ton of coal is equivalent for steaming purposes to

$$\frac{16,863,600}{4,800,000} = 3.5 \text{ bbl. of oil}$$

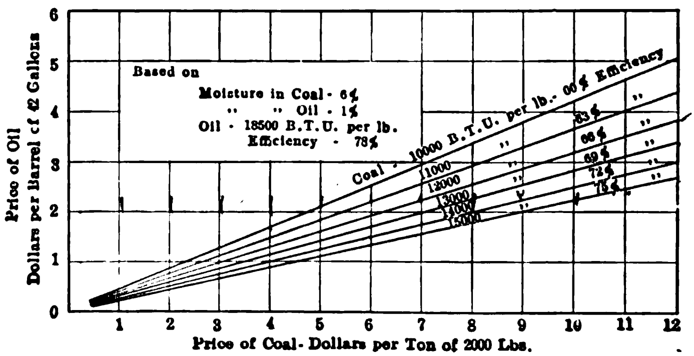

FIG. 145.—Comparison of fuel oil with coal of various heating values.

The cost of 3.5 bbl. of oil at $1.50 per barrel is $5.25, and since the coal costs $6.00 per ton, the saving would be $0.75 for each ton of coal. If the plant in question is a 100 h.p. plant, burning, say, 10,000 tons of coal per year, the saving would amount to $7500 per year for the particular conditions assumed.

In the set of curves shown in Fig. 145 a comparison is given of fuel oil with coal of heating values varying from 10,000 to 15,000 B.t.u. per lb. The heating value of the oil is taken as 18,500 B.t.u. per lb. which is a fair average value for California oil, the variation from this value being small. The efficiency of 78 per cent. for oil firing assumed in these calculations, can readily be maintained in normal service provided proper attention is paid to the furnace design and the regulation of the fires. In order to make this comparison fairly correct for the different grades of coal, an efficiency of 60 per cent. has been assumed for coal having 10,000 B.t.u. and 75 per cent. for coal having 15,000 B.t.u. per lb. with intermediate values for coals that lie between these extremes. As the heating value of coal is usually given on the basis of dry coal, and as coal when purchased invariably contains a considerable proportion of moisture, it has been assumed that the coals considered in this comparison contain 6 per cent. moisture. In the case of fuel oil the water content does not usually exceed 1 per cent., and this value has been assumed in the comparison. It will be observed from the diagram that oil at $1.50 per bbl. is equivalent in price to 14,000 B.t.u. coal at $6.00 per short ton. Oil at $1.50 per bbl. is also equivalent to 12,000 B.t.u. coal at $4.60 per short ton.

In addition to the saving in cost of fuel there will always be a saving in labor on changing from coal to oil, as the operation of firing is much simpler, there are no expensive coal elevators and conveyors to be kept up and there are no ashes to handle. On the other hand, the interest and other fixed charges on the investment required to change over, reduce the saving to some extent. Both of these items, however, are small compared to the cost of fuel, and as they tend to neutralize each other they may be safely neglected except in special cases.

The apparatus required to change a coal burning plant into an oil burning plant consists of the oil storage tank, oil pumps, oil heater, oil burners with the necessary interconnecting piping, strainers, regulating valves, etc., all of which have been described in Chapter XXIV. In addition an automatic oil firing system may be installed if desired.

The furnaces under the boilers must be altered to suit the new fuel. If the boilers are hand fired this is a simple matter, for all that is necessary is to cover the grates with firebrick, leaving suitable openings for the admission of air, and install the burners properly housed and protected from the heat. A furnace similar to that illustrated on page 158 may then be used, the grates acting as supports for the checkerwork in the furnace floor. Boilers larger than 300 h.p. should have a furnace length not less than 10 ft., so in many cases where the grates are shorter than this it will be necessary to extend them. For the additional length necessary pieces of pipe or I-beams may be used to support the furnace floor, instead of grates.

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Fig. 146.—Typical boiler room for easy changoover from oil to coal.

Many power plant operators prefer to so design the fuel oil operation that quick change over to coal operation may be accomplished should oil later involve less economic operation than coal. The view shown is that of the boiler room of the steam electric plant for the lighting department of the city of Seattle—East Lake Avenue and Highland Place.

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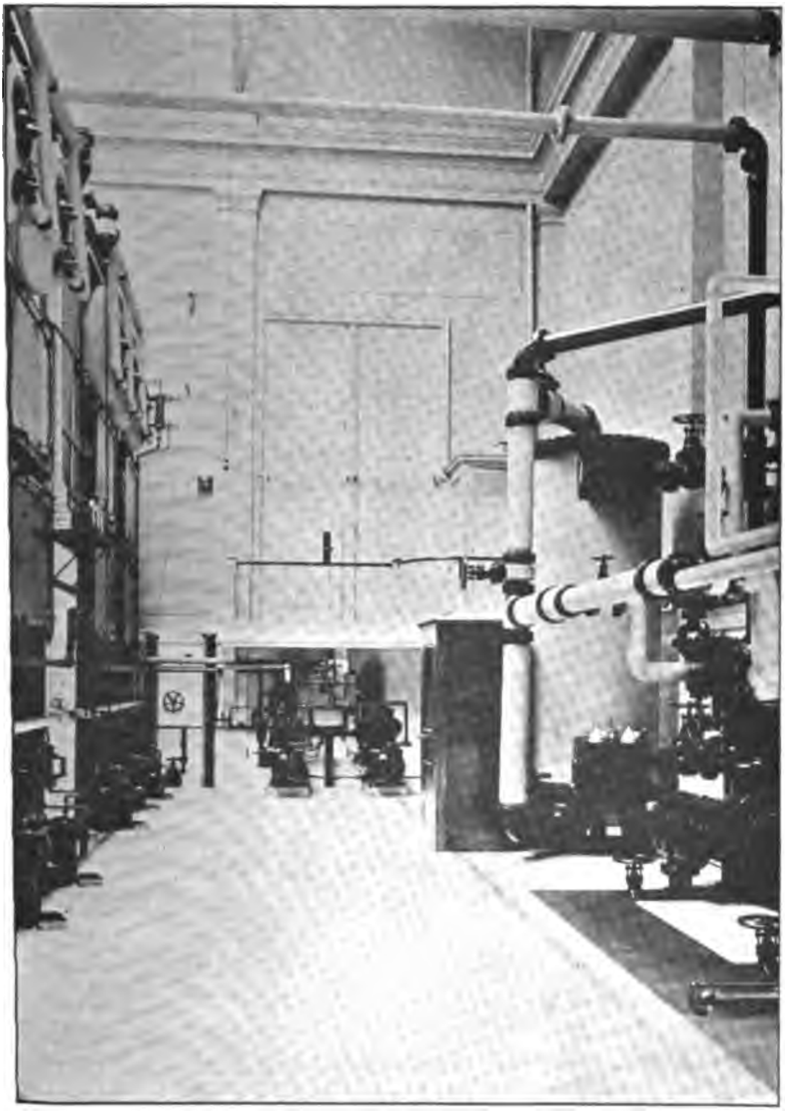

Fig. 147.—Oil fired steam heating station, Pacific Gas and Electric Company, station S, San Francisco. B. & W. boilers are to the left, fuel oil pumps and heaters in the center background, feed water pumps and heater to the right.

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For stoker fired boilers the design of furnace to be adopted will depend largely on the kind of stoker, and the arrangement of coal furnace and ashpit. If the use of coal is to be abandoned altogether the stokers should be removed, and an oil furnace installed of the general design indicated in Chapter XX. If the boilers are provided with basement ash pits advantage should be taken of this to increase the furnace volume by placing the furnace floor below the level of the fireroom floor, allowing the air to come up through the ashpit from the basement.

If oil is to be used only temporarily, and it is expected at some future time to go back to coal burning, it will be possible in many cases to leave the stokers in place, placing the oil burner at the rear of the furnace, and protecting the stoker from the heat by means of firebrick supported by structural material, leaving a dead air space between the stoker and the firebrick. The practicability of this arrangement will depend on the type of stoker, kind of boiler, and space available, and each case requires special study to secure the proper design.

The question whether steam atomizing or mechanical atomizing burners should be adopted will depend on local conditions. In general it may be said that steam atomizers should be used wherever they are applicable, as they have proved their practical value by years of application to stationary work. In special cases where steam atomizers are unsuitable the mechanical burner may be used to advantage. This would include plants so located that the waste of fresh water is a serious matter. Plants in which it is necessary to force the boilers up to 300 or 400 per cent. of their rated capacity may also find the mechanical atomizing burner more suitable, and this will be especially true if a steady load is carried and if the plant is already equipped with forced draft apparatus.

Number of Men Required for Operating Oil Fired Boilers.—The number of men required to operate boilers fired by oil is much less than the number required to operate a coal burning plant. In an oil burning central station a fireman can operate six or seven large boilers having three oil burners each, and in addition attend to the feeding of the boilers with water. In other words, a plant having 26 or 28 boilers would require only four firemen on a watch besides a man to look after the feed pumps, oil pumps and keep records of oil consumption, temperatures, etc.

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CHAPTER XXVI

THE GRAVITY OF OILS IN FUEL OIL PRACTICE

Fuel oil is classified, marketed, and designated by its gravity. Gravity is denoted in two distinct ways. The scientific method of notation is known as the "specific gravity," which is the ratio of the weight of a given volume of the oil to that of an equal volume of pure water. There has, however, grown up in practice an empirical method of representing the gravity of oil by what is known as the Baumé scale. This scale has two separate and distinct formulas for its conversion to specific gravity readings. One formula is for liquids heavier than water and the other for liquids lighter than water. In each instance the scale is graduated to 100 degrees and overlaps 10 degrees.

The use of the Baumé scale should be abandoned as it is not only unscientific, but confusing. However, as its use is universal in the oil industry, and it has obtained such a firm foothold among both producers and users of fuel oil, it is described and used freely throughout this book.

Antoine Baumé, a French chemist of the eighteenth century, distinguished for his success in the practical application of the science, was the inventor of the so-called Baumé scale now universally adopted in fuel oil practice for denoting the gravity of crude petroleum.

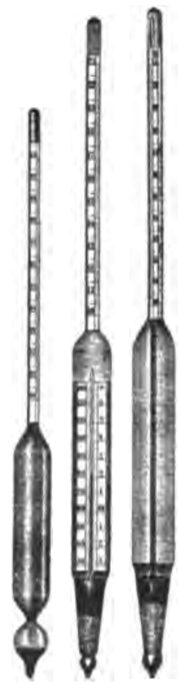

FIG. 148.—Baumé hydrometers.

The Scale for Liquids Heavier Than Water.—Baumé hit upon a unique plan for the establishment of his scale. Certain fixed points were first determined upon the stem of the instrument. The first of these was found by immersing the hydrometer in pure water, and marking the stem at the level of the surface. This formed the zero of the

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scale. Fifteen standard solutions of pure common salt in water were then prepared, containing respectively 1, 2, 3......15 per cent. (by weight) of dry salt. The hydrometer was plunged in these in order and the stem having been marked at the several surfaces, the degrees so obtained were numbered 1, 2, 3......15.

The instrument thus adapted to the determination of densities exceeding that of water was called the hydrometer for salts.

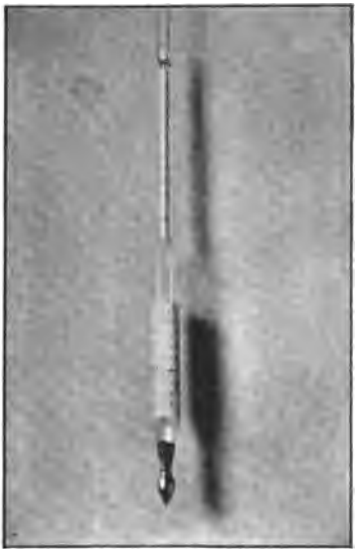

Fig. 149.—Hydrometer for obtaining gravity of fuel oil.

Expressed mathematically in its relationship with the specific gravity S, the Baumé degree reading B becomes for liquids heavier than water:

$$S = \frac{145}{145 - B} \qquad (1)$$

The Scale for Liquids Lighter Than Water.—Since practically all grades of crude petroleum are lighter than water, we are most interested in the method of expression for this latter phase of gravity denotation.

The original Baumé hydrometer intended for densities less than that of water, or the hydrometer for spirits, as it was called, was constructed on a similar principle to that for the hydrometer for salts above described. The instrument was so arranged that it floated in pure water with most of the stem above the surface. A solution containing 10 per cent. of pure salt was used to indicate the zero of the scale, and the point at which the instrument floated when immersed in distilled water at 10°R. or 54½°F. was numbered 10. Equal divisions were then marked off upwards along the stem as far as the 50th degree.

The Confusion in Expression for Specific Gravity and Baumé Readings.—Modern gravities are expressed for liquid temperatures of 60°F. instead of 54½°F. as above set forth. This fact together with other inconsistencies and errors in observation have led to the invention of some seventeen different mathematical expressions, by various investigators and scientific bodies,

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to properly set forth a relationship between specific gravity and Baumé readings for liquids lighter than water. The contest has simmered down to two equations in American practice.

The formula in general use since 1851, and which has been adopted as the "American Standard" by the U. S. Bureau of Standards is as follows:

$$S = \frac{140}{130 + B} \tag{2}$$

The other formula, which is used by C. J. Tagliabue whose hydrometers have been adopted as standard by the United States Petroleum Association, is as follows:

$$S = \frac{141.5}{131.5 + B} \tag{3}$$

A full discussion of the relative merits of these two formulas is given in Circular No. 59 of the Bureau of Standards from which the following extract is taken:

"At the time the Bureau of Standards was contemplating taking up the work of standardizing hydrometers (about 1904), diligent inquiry was made of the more important American manufacturers of hydrometers as to the Baumé scales used by them. Without exception they replied that they were using the modulus 145 for liquids heavier than water, and 140 for liquids lighter than water. These scales, the "American standard," were therefore adopted by the Bureau of Standards and have been in use ever since.

There having been no objection or protest from any manufacturer or user of Baumé hydrometers at the time the scales were adopted by the Bureau, it was assumed that they were entirely satisfactory to the American trade and were in universal use. Such, in fact, appears to be the case with the scale for liquids heavier than water, but in the case of the scale for liquids lighter than water a disturbing element has arisen which threatens to some extent the uniform practice that has heretofore existed.

The exact date of this disturbing influence can not be fixed with certainty, but it was first noticed some four or five years ago, and has been quietly at work since then to break down the uniformity of practice previously existing and to counteract as far as possible the influence of the Bureau of Standards in the interest of uniformity.

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It appears that a certain manufacturer of hydrometers, especially those used in the oil trade, discovered that his Baumé hydrometers were not graduated in accordance with the American standard Baumé scale in general use based on the modulus 140. This discovery made necessary for the manufacturer one of two things: Either he must consider his instruments in error, by the amount of the difference, or he must change the basis of the scale to conform to his instruments. The manufacturer in question, C. J. Tagliabue, chose the latter course.

The developments of the problem confronting Mr. Tagliabue are well shown by the various editions of his Manual for Coal Oil Inspectors. The first few editions of this publication contained the regular American standard Baumé table, modulus 140. Then came the discovery that his instruments did not fit the table, and an attempt was made to make a table to fit the instruments. The result was an irregular table with no definite modulus. This was published in at least two editions of the manual. Then followed the table which is now published by Mr. Tagliabue in the eighth edition of his manual, based on the modulus 141.5, which more nearly fits his standard hydrometers for petroleum oil.

A small pamphlet prepared by Mr. Tagliabue has recently been widely distributed in which the impression is given that the modulus 141.5 was adopted by the United States Petroleum Association in 1864, and has been in use in the petroleum trade ever since, and that lately the modulus 140 has been proposed and that great confusion may result from its use. That such is by no means the case has been shown by the foregoing references and historical matter.

There can be little doubt that when the United States Petroleum Association adopted as standard the hydrometers made by Jarvis Arnaboldi, who was later succeeded by C. J. Tagliabue, it was believed by all concerned that the instruments were based on the American Standard Baumé scale."

The Limitations of the Hydrometer.—The hydrometer method of ascertaining the gravity of crude petroleum is at best only approximate, as one may readily surmise. In order then to ascertain the gravity of oil with scientific accuracy, a more refined method is necessary. This is usually accomplished by determining the specific gravity of the oil with whatever moisture content it may contain by means of an actual water equivalent

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comparison, and then converting this into degrees Baumé. This roundabout method once again emphasizes the uselessness of employing the Baumé scale. If the moisture content of the oil has been ascertained, a computation is then made in order to arrive at the actual specific gravity or Baumé reading for the moisture free oil.

The Method of the Westphal Balance for Exact Measurement. Let us then examine in detail such a method. The Westphal

Fig. 150.—A commercial balance for determining specific gravity of oil.

The common hydrometer is not of sufficient accuracy to determine the specific gravity of oil used in fuel oil tests. A simple and accurate method for such determination is accomplished by the employment of a Westphal Balance as shown in the illustration. The specific gravity is first ascertained by comparison of the oil with a water standard and then by means of the mathematical relationship connecting specific gravities and Baumé readings, the latter gravity reading is ascertained.

balance is a convenient and accurate method by which the specific gravity of fuel oil may be obtained to four decimal points. As shown in Fig. 150, the apparatus necessary consists of a balance arm, supported on knife edges, from one end of which is

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hung a glass bulb, the other end being counter-weighted. Along the balance arm are nine notches, the hook supporting the glass bulb being in the position of the tenth notch. The glass bulb has a displacement of exactly five grams of pure water at 4°C., which is the point of maximum density of water, the density for which scientific gravity comparisons are made. Hence if the bulb above described were so immersed in water at 4°C. a five gram weight would establish equilibrium if hung from the hook. This would indicate a specific gravity of 1.0000.

The zero point of the balance is adjusted by turning a thumb screw, which forms one point of the three point support shown in the figure, until the pointers are opposite each other before the bulb is immersed. For specific gravities less than 1.0000 the five gram rider called the unit weight is hung in a notch such that equilibrium is nearly reached, never exceeded. This gives the first decimal place. The $1/10$, $1/100$, and $1/1000$ unit weights are then hung respectively in notches so that equilibrium is finally established. The specific gravity is then read directly to four decimal places by noting the notches in which the riders hang, commencing with the largest rider. Thus when the unit weight hangs in the ninth notch, the $1/10$ weight in the sixth notch, the $1/100$ weight in the seventh notch, and the $1/1000$ weight in the third notch, the specific gravity is evidently 0.9673.

Details of Procedure.—Before proceeding with a gravity determination, the oil sample should be allowed to stand in the laboratory several hours in order that any drops of water in the oil may settle. A small quantity is then poured from the sample can into a suitable glass jar. The Westphal balance, having been dusted with a soft brush, is then adjusted to equilibrium and the specific gravity of the sample obtained. The temperature is also ascertained by means of the thermometer inserted in the oil sample. Since specific gravities of fuel oil are by common practice referred to at a temperature of 60°F., it is now necessary to make a second determination at a temperature differing by 15° to 20°F. from the first, in order that we may have sufficient data with which to compute what the gravity would be at 60°F. temperature.

To take this second reading the temperature of the sample in the jar may be raised by immersion in a water bath. In doing this great care must be taken to allow no water to get into the oil.

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Computations Involved.—Let us next illustrate the computations involved in a gravity determination. Let us assume that by means of the Westphal balance, the oil sample is seen to have a specific gravity (S_1) of 0.9644, at a temperature (t_1) of 68.9°F., and a specific gravity (S_2) of 0.9587 at a temperature (t_2) of 86.6°F. Since the specific gravity has changed ($S_1 - S_2$) over a temperature change of ($t_1 - t_2$) the change for 1°F. would be $\frac{(S_1 - S_2)}{(t_1 - t_2)}$. This change in specific gravity for 1°F. is the coefficient of expansion (C_e), for the oil and may be expressed by the formula

$$C_e = \frac{(S_1 - S_2)}{(t_1 - t_2)} \tag{4}$$

In the particular case then we now find that

$$C_e = \frac{0.9644 - 0.9587}{68.9 - 86.6} = -0.00032$$

The coefficient is thus seen in this case to represent an intermediate value, for in practice we find that in different oils C_e varies from (-0.00027) to (-0.00042).

From the fundamental definition of the coefficient of expansion it is now seen that at 60°F., the specific gravity becomes

$$S = S_1 + C_e(60 - t_1) \tag{5}$$

Consequently by making the proper substitutions for the case cited we find that the numerical value of the specific gravity of this oil sample for 60°F. is

$$S = 0.9644 + [-0.000322 \times (-8.9)] = 0.9673$$

In order to cônvert this specific gravity to the Baumé scale we now, by substituting in formula given above for such conversion, find that

$$B = \frac{140}{0.9673} - 130 = 14.73°$$

Assuming that this particular oil sample has been found to contain 0.5 per cent. by weight of moisture and 0.484 per cent. by volume, let us now see how we should find the specific gravity of the dry oil. Let V_w represent the percentage of water by volume and S_w, S_o, S_m represent respectively the specific gravity of the water, dry oil, and moisture. Then we may write the following relationship:

$$S_m = S_o\left(\frac{100 - V_w}{100}\right) + S_w\left(\frac{V_w}{100}\right) \tag{6}$$

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From scientific tables we find that S_w at 60°F. has a value of 0.9990, and from the Westphal balance S_m has been found to be 0.9673. By transforming the formula above it is seen that

$$S_o = \frac{S_m - S_w\left(\frac{V_w}{100}\right)}{1.00 - \frac{V_w}{100}} \tag{7}$$

Consequently S_o may now be computed numerically.

$$S_o = \frac{0.9673 - 0.00484 \times 0.9990}{1.00 - 0.00484} = 0.9671$$

If it is desirable to ascertain the Baumé reading for the dry oil, we next ascertain its value from the above relationship of specific gravity and the Baumé scale from equation (2).

$$B = \frac{140}{S} - 130 = \frac{140}{.9761} - 130 = 14.8°$$

According to formula (3) this Baumé reading would of course be computed as follows:

$$B = \frac{141.5}{S} - 131.5 = \frac{141.5}{0.9671} - 131.5 = 14.8°$$

When a large quantity of oil is to be purchased and it is desirable to carry the Baumé reading to still further decimal points, the two formulas will not of course check; hence, one or the other of these formulas should be agreed upon prior to a purchase of any magnitude.

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CHAPTER XXVII

MOISTURE CONTENTS OF OILS

From our previous discussion of steam generation in the modern central station it was found that something over a thousand heat units are necessary to convert one pound of water at ordinary temperatures into saturated steam. When moisture appears in the oil used for heat generating purposes in the furnace it is evident, then, that large heat losses may thereby be involved. For, not only must this moisture be converted into saturated steam, but this steam itself must be superheated to the temperature of the outgoing chimney gases, thus dissipating energies that should go toward steam generation in the boiler.

Hence the water involved in fuel oil composition is a dead loss which should be avoided as far as possible. Settling tanks accomplish much in drawing off the water content, but when the water appears in the oil as an emulsion it is almost impossible to commercially segregate it from the oil. Since, then, all fuel oils contain a certain amount of moisture, the careful determination of its exact proportions often becomes an important problem in efficient steam engineering performance.

FIG. 151.—An electrically driven oil centrifuge.

In this centrifuge the four arms—two plain and two graduated—are caused to rotate by electric power and the water thus caused to separate from the oil. The consequent measurement of the moisture present is then easily ascertained.

Summary of Methods Employed in Determining the Moisture Content.—There are ten methods by which the moisture content of oil can be ascertained with approximate accuracy. For detailed information on this subject the reader is referred to Technical Paper No. 25 of the United States Bureau of Mines entitled, "Methods for the Determination of Water in Petroleum and its Products." These methods may be briefly summarized as follows:

The moisture content of heavy oils and greases may be approximately ascertained by the loss of weight due to heating.

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The moisture content of oil may be approximately obtained by diluting a sample with a sulphate and then causing separation by action of gravity. A diluent is to be avoided in this process, as inaccuracies are liable to be introduced.

Again by diluting with a solvent and separating the moisture content by means of a centrifuge, the moisture content is determined with a slightly greater degree of accuracy than by either of the above methods.

By treating a sample with calcium carbide, another convenient method is also arrived at, and its accuracy is approximately within 3 per cent. of the water percentage if care is observed. The sample, too, may be treated with sodium and a convenient and accurate method results.

A color comparator is sometimes used, but the method is only approximate, as is also the method of treating a sample with normal acids. The electrical treatment, on the other hand, is successful in breaking up an emulsion on a commercial scale, or reducing the water content of an oil to such a condition that it may be successfully treated in some other manner. An emulsion is a physical condition of the oil and water wherein the water is held in such intimate contact with the oil ingredients as not to be readily separated by gravity or other ordinary means.

Again, too, distilling a sample mixed with a non-miscible liquid proves accurate to 0.033 grams of water per 100 cc. of benzine and oil in the distillate.

The most reliable method, however, is that accomplished by directly distilling off the water. This method is convenient and accurate to about 0.003 grams of water in the distillate, if the water is cooled to about 35°F.

The Approximate Method of Treatment.—The method hinted at above wherein the sample is treated by a foreign agent will now be briefly set forth, since such a preliminary determination often proves sufficiently accurate for the issues involved.

The method here outlined is especially applicable for the lighter oils. A burette graduated into 200 divisions is filled to the 100 mark with gasoline, and the remaining 100 divisions with the oil, which should be slightly warmed before mixing. The two are then shaken together and any shrinkage below the 200 mark filled up with the oil. The mixture should then be allowed to stand in a warm place for 24 hr., during which the water and silt will settle to the bottom. Their percentage by volume can then

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be correctly read on the burette divisions, and the percentage by weight calculated from the specific gravities.

Details Involved in Determination of Distillation.—Since the method of determination by distillation is to be recommended above all others, we shall now proceed to the details of its accomplishment. Stated in simple words, the method consists in heating a sample slightly above the boiling point of water but not so high as to cause the vaporizing of other ingredients of the oil. As a consequence, the water passes over and leaves water-free oil in the sample.

FIG. 152.—A Goetz attachment for water determination.

By attaching the pipe shown in the lower part of the figure to a faucet, sufficient power is obtained from the city main to cause the rapid rotation of the two arms shown in the figure. This high rotative speed, due to the centrifugal force developed, causes the separation of the moisture from the oil.

The Apparatus Involved and Preliminary Proceedings.—To quantitatively determine the moisture content the sample is placed in a copper vessel known as a still, which is about 4 in. in diameter and 6 in. high. The still is then placed in an asbestos hood through which a projecting stem connects to a condenser and a burette where the condensate is measured in a graduated tube. The can from which the sample is to be drawn is first immersed in a water bath with its cover released. After the water constituting the bath has been raised to a temperature of 150° to 170°F., the cover to the can is fastened tightly, and the can agitated for several minutes in order that any water that may have settled at the bottom may be thoroughly mixed with the oil. For successful agitation the sample can should not be filled more than two-thirds with oil. 100 cc. of oil sample, measured in a graduated jar, are now poured into the still. The exact measurement of the oil is difficult without experience, as froth

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collects on the surface of the oil and tends to obscure any definite meniscus.

The jar is next washed with 50 cc. of benzol and 50 cc. of toluene. The washings are poured into the still. Since toluene has a tendency to absorb small quantities of water, accurate results may be interfered with if the toluene is not previously

Fig. 153.—Still with hood used for water determination.

Many methods are utilized in determining the water content of oil. The simplest and most accurate method for fuel oil tests is that of distillation. In this method a sample of the oil is poured into a still and raised sufficiently in temperature to evaporate the water and not the ingredients of the oil. By condensing the moisture and ascertaining its proportions the moisture content is easily ascertained.

saturated. In order to avoid such a possibility when opening a fresh bottle, 5 to 10 cc. of water should be added. The presence of water in the bottom of the bottle shows that the toluene is

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saturated, but care must be taken not to pour this water into the still when washing with the toluene.

The still must be gently shaken without splashing in order that its contents may be well mixed, and then placed in the asbestos hood and connected with the condenser. A hood and cover are provided, as shown in the illustration, to surround the still with a blanket of air at a uniform temperature. The still is then heated gradually to a temperature of about 300°F., which is usually accomplished after about fifteen minutes of heating. Since the boiling point of water has been now exceeded, the moisture in the sample begins to pass over into the condenser, and after the lapse of another fifteen minute period the distillation is complete. A thermometer for temperature control is seen at the right side of the asbestos hood in the illustration.

The Process of Distillation.—The process of distillation is interesting. At about 176.7°F. the benzol first passes over. This wets the condenser tube so that the moisture which is soon to follow will not readily cling to the tube but the more easily pass down into the measuring burette. The toluene follows at 230.5°F., and carries down with it any water which happens to remain in the condenser tube. The toluene does not, however, pass over in its entirety, since usually from 15 to 20 cc. remain in the still with the oil. In order to make up this deficiency in toluene about 15 cc. are poured down the condenser tube to free any small drops of water that may persist in remaining. This, however, does not affect the accuracy of the work, since the water content is finally separated by filtration and the water content thus obtained is alone measured.

The still while at a high temperature is drained, and, as its contents are now entirely free from water, it may be used again without additional cleaning.

Any small drops of water that cling to the side of the graduated measuring tubes must be released by a short wire. If now the resultant water is read in cc. the percentage of water by volume in the oil is easily obtained, since the water is separated from the mixture of benzol and toluene in the filter bottle.

A Numerical Determination.—Let us then follow this process by means of an illustrative example. Let us assume that 100 cc. of oil have been drawn as a sample, that 100 cc. of benzol and toluene have been poured with it into the still, and that the

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resultant distillate shows 0.484 cc. of water. It then follows directly that the percentage of water by volume is 0.484.

Error in Assuming Percentage by Weight is Same as Percentage by Volume.—The percentage of water by weight is not exactly equal to its percentage by volume, but may be taken equal to it for all practical purposes of boiler testing. This error is then nominal except with very light oils or any oil with considerable moisture content. Thus, if an oil sample of 100 cc. contains 0.50 cc. of water at 60°F., it will weigh $0.484 \times 0.999 = 0.4835$ grm.

The percentage of water by weight is therefore 0.4835 divided by 0.9673, which equals 0.50 per cent. The factor, 0.9673, appearing in the above, is the specific gravity of the oil sample at 60°F. This was ascertained by means of a Westphal balance, which is shown in detail in the preceding chapter.

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CHAPTER XXVIII

DETERMINATION OF HEATING VALUE OF OILS

To determine the efficiency of boiler operation it is necessary to know the heat producing value of the oil used in firing. Again, since oil is usually sold commercially by the barrel, the heat producing value of the product must be known in order that the

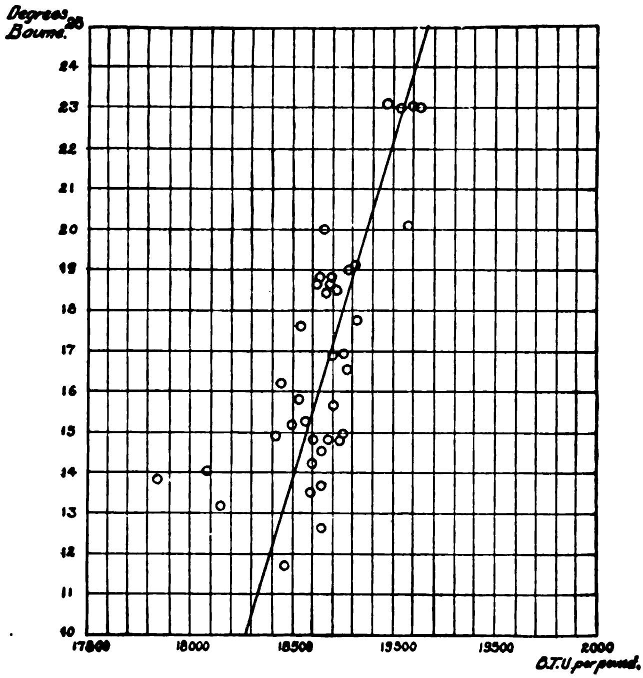

FIG. 154.—The Graphic Law for calorific value of fuels.

In this illustration is shown how a large number of experimental values often enable the engineer to ascertain an empirical law for setting forth experimental data. By plotting the heat determinations for fuel oil against their gravity expressed in Baumé readings, the experimenter deduced an equation for determining the calorific value of water free oil when its gravity Baumé is known.

engineer may ascertain the economic value the product may prove to his client in its use in the power plant for the generation of steam.

An Approximate Method Based on the Baumé Scale.—The heat producing value of oil is usually expressed in the number of heat units per unit of mass that the oil will give out when it is completely burned in a furnace. In engineering practice this is usually expressed in B.t.u. per pound of oil so burned.

There are various methods of ascertaining this value. An approximate method is that based upon the gravity of the oil. To establish this method a large number of samples with the gravities of the oil free from moisture expressed in Baumé readings were accurately determined as to their heating value. These values were plotted on a chart and it was found that the following relationship is approximately true in which H represents the heat units in B.t.u. liberated per pound of fuel burned, and B represents the gravity of the oil in degrees Baumé:

$$H = 17680 + 60B \tag{1}$$

Thus in analyzing a composite sample of forty samples of Kern River oil, the United States Bureau of Mines found that its calorific value was 18,562 B.t.u. per lb. of oil, in which the oil had 0.5 per cent. moisture, and that the Baumé reading of this oil when free from water was 14.78°. According to the formula above, which was first announced by Professor Joseph N. LeConte of the University of California, the heating value of this oil when free from moisture should be

$$H = 17680 + 60 \times 14.78 = 18{,}566 \text{ B.t.u. per lb.}$$

In this instance then it is seen that this approximate method checks with considerable accuracy, since the water-free oil showed by actual test to have a heating value of 18,658 B.t.u. per lb. In the utilization of this formula, however, it must be remembered that the oil must be taken as anhydrous, or in other words that the oil sample is moisture free.

Dulong's Formula Based on the Ultimate Analysis.—The second method of arriving at the calorific value of crude petroleum is by means of Dulong's formula. This formula is based upon the ultimate analysis of the oil in which the heat value of carbon, hydrogen, and sulphur are taken into account.

In the burning with oxygen of one pound of carbon, one pound of hydrogen, and one pound of sulphur it has been established experimentally that 14,600, 62,000, and 4000 B.t.u. of heat energy are respectively given out. Hence it is evident that if a one-

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pound sample of fuel oil has C proportions by weight of carbon, H proportions by weight of hydrogen and S proportions by weight of sulphur, the total heat given out by the one-pound sample will be

$$H = 14{,}600C + 62{,}000H + 4000S$$

In the chemical analysis of fuels a certain amount of oxygen (O) is always encountered. This of course kills, as it were, its combining weight of hydrogen. Since oxygen unites with one-eighth of its weight of hydrogen, the net hydrogen available for heat generating purpose is $\left(H - \frac{O}{8}\right)$.

Hence we have Dulong's formula

$$H = 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4000S \qquad (2)$$

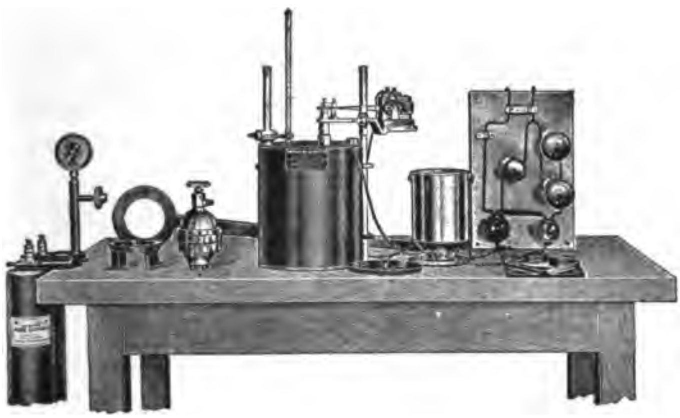

FIG. 155.—The Emerson fuel calorimeter.

In this type of calorimeter the fuel sample is placed in the bomb, the bomb inverted, as shown in the sketch, and filled with oxygen which is accomplished by means of the spindle valve at the top of the bomb. After filling the calorimeter with distilled water and firing the sample by means of an electric circuit, the rise in temperature of the water in the calorimeter is ascertained, and the calorific value of the fuel thus determined.

For California oils, Dulong's formula seems to indicate a heat value per pound of about 5 per cent. in excess of the true value. In other words, it indicates a heating value of about 19,500 B.t.u. per pound of California crude oil, while a great number of calorific tests have shown that the average value is about 18,500 B.t.u. per pound.

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The Fuel Calorimeter.—The most accurate method of determining the heating value of a sample of oil is by the employment of some form of calorimeter, wherein a sample of definite mass is burned and the heat given out ascertained. The fuel calorimeter is an entirely different instrument from the steam colorimeter used for measuring the moisture of steam, which was described in an earlier chapter. The fuel calorimeter is a true instrument for measuring heat as its name implies. Calorimeters in general may be divided into two classes, the one known as the continuous method and the other as the discontinuous method. In the former instance a sample is continually burned, and the average results ascertained over a considerable period. This method is only applicable for gases and some unusual types of oils. The discontinuous process is on the other hand the most advantageous for the determination of the heating value of crude petroleum.

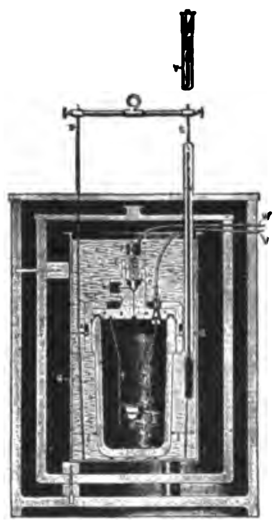

Fig. 156.—The Atwater-Mahler bomb calorimeter.

This type of calorimeter is applicable to the highest scientific work. It permits of determining the exact amount of water and carbon dioxide in the products of combustion, thus enabling the error due to the condensation of the water in the bomb to be overcome and therefore making it possible to calculate the exact amount of heat the fuel should produce under boiler conditions.

Several methods are employed in the application of the discontinuous calorimeter. Most forms of such calorimeters consist essentially of a strong combustion chamber with a crucible for holding the sample; valves for charging the chamber with oxygen in order to properly burn the sample; a method of igniting the sample; and a vessel of water in which the bomb or explosion chamber is immersed in order that the resultant heat may be absorbed by this water and thus carefully measured. This latter vessel is usually situated in a second compartment which serves as a jacket. The main principle upon which such calorimeters depend is based upon the fact that the burning of carbon, hydrogen, and sulphur with an artificial supply of oxygen presents the most accurate method of liberating the latent heat in the fuel and the ascertaining of its quantitative proportions. Types of this calorimeter familiar in the market are known as

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the Mahler, the Hempel, the Atwater, the Emerson, and the Carpenter.

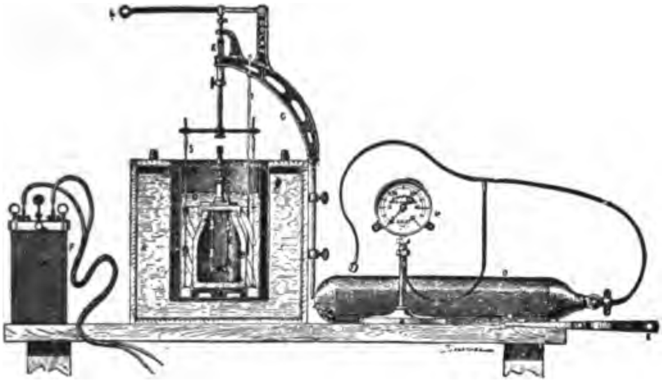

FIG. 157.—The Mahler bomb calorimeter.

This type of calorimeter represents one of the most accurate for the determination of the calorific value of fuel oil. The bomb is of enameled steel. The burning of the oil sample is accomplished by supplying an outside source of oxygen as in the Emerson Calorimeter.

The Parr Calorimeter.—In the commercial determination of the heating value of crude petroleum, however, it is often incon-

FIG. 158.—The Parr calorimeter unassembled.

In this type of calorimeter a carefully weighed oil sample is burned with a chemical agent without the use of free oxygen. The ease with which it may be manipulated commends its use for commercial application. For scientific work, however, a type of the bomb calorimeter is to be preferred.

venient to secure oxygen under the proper pressure required for the successful operation of this type of calorimeter. In recent years there has appeared upon the market a much simpler design of calorimeter which seems to have sufficient accuracy for most commercial uses and is indeed quite simple in operation. This is known as the Parr calorimeter and is the invention of Professor S. W. Parr of the University of Illinois.

The Principle of Operation.—In the Parr calorimeter a definite mass of oil is introduced into a strong cylinder of metal called the cartridge, along with some accelerator together with a measure of potassium peroxide. The potassium peroxide furnishes oxygen for combustion and the accelerator, which is usually potassium chloride, insures that all the fuel may be burned. The ignition is effected electrically by the burning out of a fine iron wire immersed in the mixture.

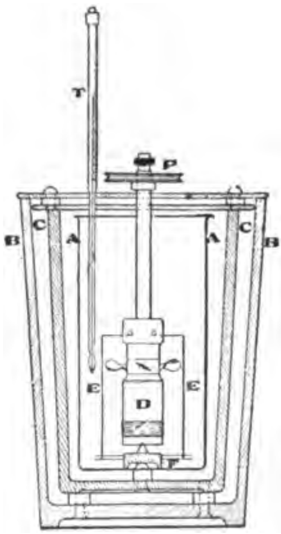

FIG. 159.—Cross-sectional view of the Parr calorimeter.

As shown in the illustration, the cartridge D, in which the sample is placed, is closed up, inserted into a can of water A, and the whole placed in a fibre vessel B, which thus brings about careful heat insulation. After causing an explosion by means of an electrical contact spark in the cartridge D, the cartridge is given a rotary motion by means of the pulley P and the heat which is given out from the cartridge due to the burning of the ingredients is rapidly absorbed by the water in the vessel A. If then we know the mass of the sample burned, and the mass and temperature of the water before and after the explosion, we can compute the heat value of the fuel.

Detailed Operation of the Parr Calorimeter.—Let us now go into the details of this calorimeter operation. A well lighted closet should be used for all calorimeter work so that air currents which might otherwise prevent uniform radiation can thus be eliminated. The outside of the calorimeter cup and of the fibre insulating case should be entirely free from moisture for the same reason. The calorimeter cup A is filled with 2000 grams of water. The cartridge or bomb in which the sample is placed has a water equivalent of 135 grams; that is, it absorbs the same

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amount of heat as 135 grams of water would under the same range of temperature. Hence, the total water equivalent W_e is 2135 grams. As the mass of oil is also determined in grams, the water equivalent W_e divided by the mass of oil fired, W_o, becomes an abstract ratio, and, if this ratio is multiplied by the rise in temperature of the water in degrees Fahrenheit, the result is heat units per pound of oil, or, if the temperature is expressed in degrees Centigrade, the result becomes calories per gram.

The water is best measured in a 2000 cc. flask. About 2003 cc. of water are used instead of an exact 2000 cc., since the specific gravity of water at ordinary room temperatures is slightly less than unity and this increased volume is necessary to measure weights volumetrically.

The thermometer which is employed in temperature measurements has a range of from 65°F. to 90°F. and is standardized by the Bureau of Standards at Washington. Graduation errors are known to within 0.01°F. The thermometer scale is divided to 0.05° and with care may be read to 0.005°. The greatest chance for error in fuel calorimeters is in reading temperatures, since it is difficult to avoid parallax. Consequently as the rise in temperature seldom exceeds 5°, an error of 0.01° is equivalent to 0.2 per cent. error in the work.

Preliminary Precautions.—Before placing the sample, the cartridge should be wiped clean and dry, as moisture will condense on it if it has been standing for some time. The top and bottom pieces, as well as the gaskets and electrical terminals, should be dry, since the moisture on them takes part in the chemical reaction and thus introduces considerable error. The cartridge should be tightly assembled, and 1.500 grams of accelerator (potassium chloride), weighed to the nearest reading of 0.005 grams, placed therein. The oil is weighed in a small flask with an eye dropper and about 0.04 to 0.05 grams (8 to 12 drops) dropped into the cartridge upon the accelerator which absorbs the oil. Upon reweighing the flask of oil and the dropper, the net weight of the oil sample is at once obtained. A measure full of sodium peroxide is added and the contents thoroughly mixed with a stiff wire. With care no oil and very little peroxide will adhere to the wire. The sodium peroxide should be supplied by the calorimeter manufacturer, as inferior grades are apt to contain variable and detrimental products of combustion.

About 3 in. of No. 4 iron wire for firing the charge are next

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looped on the firing terminals and tested out to insure a good electrical contact. The firing current is usually supplied by a few dry cells or from a storage battery.

The stem of the bomb is next fastened in place and the vanes attached. The cartridge is placed in the calorimeter cup, the cover and pulley attached, and the cartridge stirred by a small motor for about five minutes. The motor may be of the toy variety and is usually placed in the lighting circuit with a lamp resistance. The electric circuit is controlled with a two-throw switch so that the motor may be cut in and out without interfering with the illumination in the closet. The motor speed should be as nearly constant as possible, since a variable speed will cause a variable rate of radiation from the calorimeter. The rotating bomb should have from 100 to 150 revolutions per minute.

The Explosion of the Charge and the Taking of Temperatures. The thermometer is next placed into the water bath through a hole in the cover and should be supported so that it does not touch the metal cup which contains the water. After a steady initial temperature has been reached, the firing circuit is completed through the pulley, and the resulting temperatures read every minute for the succeeding ten minutes, in order to ascertain the correction to be made for uniform radiation. This series of readings is taken in order to ascertain the law of radiation and then to make a proper correction for the error involved.

Thus, for a period of about five minutes the temperature will rise until a maximum is reached, after which it will begin to fall. The radiation during the first five minutes is assumed to be at the same rate as that observed during the entire radiation period. Let us assume the following experimental data:

Water equivalent of calorimeter	135 grams
Weight of water used	2000 grams
Weight of oil used	0.3765 grams
Per cent. moisture in oil	0.5%
Weight of accelerator	1500 grams
Room temperature	70°F.

Temperature of mixture when fired, 73.665°F.

Combustion Period

1 min	77.45
2 min	78.15
3 min	78.42
4 min	78.44
5 min	78.45

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Radiation Period

6 min	78.44
7 min	78.42
8 min	78.40
9 min	78.385
10 min	78.370

The Correction for Temperature Readings.—Since from the above it is seen that the temperature falls off from its highest reading, t_h, or 78.45°F. to 78.37°F. in five minutes, it is evident that in one minute it would fall off 0.016°F. As a consequence at the end of the combustion period, in reality the thermometer should have read greater than t_h or 78.45°F. by an amount equal to the radiation t_r which is (0.080) over the first five minute period. In addition to this correction, by consulting a correction scale furnished by the Bureau of Standards, the thermometer should be corrected for 78.45°F. by an amount equal to t_{c_2} or (− 0.053°) and for the minimum temperature t_m or 73.665°F. equal to t_{c1} or (− 0.043°). From the instrument maker there has also been furnished data indicating a correction for the chemicals and wire employed, amounting to t_w or (− 0.373). Hence the true maximum temperature t_2 and the true minimum temperature t_1 are ascertained by the formulas:

$$t_2 = t_h + t_{c2} + t_r + t_w \tag{3}$$

$$t_i = t_m + t_{c1} \tag{4}$$

Substituting in the particular case cited, we have

$$t_2 = 78.450 - 0.053 + 0.080 - 0.373 = 78.104$$
$$t_1 = 73.665 - 0.043 = 73.622$$

Since a careful comparison of this calorimeter with the most accurate type of calorimeter known in the laboratory has shown that the heating value per pound of oil is 0.73 of the total heat liberated, we have

$$H = \frac{0.73W_e}{W_o}(t_2 - t_1) \tag{5}$$

We now have in this instance

$$H = \frac{0.73 \times 2135(78.104) - 73.622}{0.3765} = 18{,}562 \text{ B.t.u. per lb. of oil as fired.}$$

If it is desired to ascertain the heating value of this oil when free from moisture, it is only necessary to divide by the percentage of dry oil in the fuel. Thus if the oil sample contained 0.50

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per cent. of moisture we find that the heating value per pound of dry oil would be according to this calorimeter determination 18,562 divided by 0.995 which is 18,658 B.t.u.

Higher and Lower Heating Value.—In the operation of the calorimeter the gases produced by the combustion of the sample of oil are cooled down to the temperature of the water in the calorimeter. In the case of carbon which, on igniting with oxygen produces CO_2, this cooling of the gas has no important effect since CO_2 remains a gas at all ordinary temperatures. Hydrogen, on the other hand, on uniting with oxygen forms steam, H_2O, which is condensed to water in the calorimeter as soon as its temperature drops below 212°F., and in condensing gives up its latent heat to the calorimeter. When fuel oil is burned under a boiler the gases are always discharged at a temperature higher than 212° so that the latent heat of steam formed by the combustion of the hydrogen content is not available and cannot be absorbed by the boiler. Hydrogen combines with eight times its weight of oxygen so that for each pound of hydrogen burned 9 lb. of water are formed, and as the latent heat of steam is 970 B.t.u. per pound, there are approximately $9 \times 970 = 8730$ B.t.u., which cannot be recovered unless the gases are cooled below 212°F. Deducting this from 62,000 B.t.u., the heating value of 1 lb. of hydrogen determined by a calorimeter, gives 52,270 B.t.u., which is called the **lower heating value** of hydrogen.

Since oil contains a considerable proportion of hydrogen it has a lower heating value as well as the ordinary or higher heating value. If a sample of oil contains 12 per cent. of hydrogen, and the higher heating value by calorimeter test is 18,562 B.t.u. per pound, then the lower heating value is $18{,}652 - 0.12 \times 8730 =$ 17,515 B.t.u. per pound. In boiler testing work it is the universal custom to base calculations on the higher heating value as given by the calorimeter, but the lower heating value is ordinarily used when calculating the efficiencies of gas engines.

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CHAPTER XXIX

THEORY OF CHIMNEY DRAFT

It is a well known fact that transference of heat by convection currents accomplishes the freezing of lakes in the mountains and the boiling of the teakettle in the kitchen. In the latter instance it will be recalled that the layer of water along the portion of the teakettle exposed to the heat application becomes heated and expands, thus making its density less than the density of the water in the layer above; consequently this lower water immediately travels to the water surface to make way for the cooler, heavier water above. In the succeeding course of events this water too becomes heated, expands and travels upward to make way for other cooler water which in turn is heated and forced to the surface. This circulation or transference of heat eventually raises the temperature of the water to such heights that steam generation occurs.

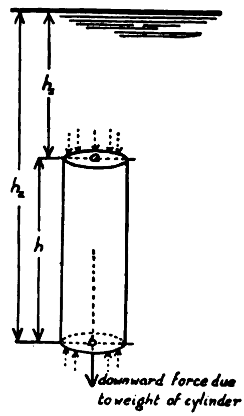

FIG. 160.—An analogy in which the chimney is considered as a vertical cylinder immersed in water.

The forces that act under such conditions are quite comparable to the forces acting upon a cylindrically shaped chimney. The downward pressure of the air upon the chimney from above plus the weight of the cylindrical column of heated gases within the stack are not equal to the pressure of the heavy dense air entering the boiler furnace, so the entire cylindrically shaped column of heated gases is forced upward and we have chimney draft.

In looking into the elementary theory of chimney draft it is found that exactly similar convection currents take place. The air or gas in the chimney is raised to a high temperature by the absorption of enormous quantities of heat given out by the fuel in combustion and consequently this air expands and becomes lighter in density than the air in the atmosphere without. Hence this heavier air pushes up through the grate or furnace entrance and expels the heated air within only to find itself a few moments later also forced up through the chimney by still other heavier air without. And so the process continues.

The Law of Pressures in Chimney Draft.—Let us consider a volume of gas or air housed up in a chimney of area A and height

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h. We may consider this cylinder of air as immersed in a sea of air with its faces respectively h_1 and h_2 ft. below the surface. This is quite approximately the actual conditions in the air surrounding the modern power plant.

Let w = weight of a cubic foot of air without.
w_1 = weight of a cubic foot of heated air within.
Let p_1 = the unit pressure of atmosphere at a.
Then evidently $p_1 = wh_1$
Also p_2 = The unit pressure of atmosphere at b.
Then $p_2 = wh_2$

Hence

Total downward pressure at $a = p_1A = wh_1A$
Total upward pressure at $b = p_2A = wh_2A$
Total down pressure due to weight of air in chimney $= w_1Ah$.

Total

Net upward press. $= wh_2A - wh_1A - w_1hA$
$= wA\ (h_2 - h_1) - w_1Ah$
But $h_2 - h_1 = h$
$\therefore$ Total net upward press. $= wAh - w_1Ah$

Let W = weight of a cylinder of outside air of dimensions of chimney $= w\ A\ h$

W_1 = weight of a cylinder of inside air of dimensions of chimney $= w_1\ A\ h$

$$\therefore \text{Total net upward pressure} = W - W_1 \qquad (2)$$

Hence the total pressure that tends to force the heated gases out of a chimney is computed by subtracting the weight of the chimney gases in the chimney from the weight of an outside volume of air equal to the volume of the chimney.

The Theoretical Draft.—In thermodynamics, we find that the weight of a so-called perfect gas, in which classification chimney gases are placed, may be computed from the formula

$$pV = WRT \qquad (3)$$

in which p is the pressure in pounds per square foot, V the volume considered in cubic feet, W the weight in pounds, R a constant (which for air is 53.3) and T the absolute temperature of the gas in degrees Fahrenheit.

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Let us assume that the pressure of the atmosphere is 14.7 lb. per square inch or 14.7 × 144 lb. per square foot, that V is unity or in other words 1 cu. ft., and that the temperature of the entering air is 62°F. or on the absolute scale is (459.6 + 62)F. We now have that 1 cu. ft. of entering air weighs

$$W = \frac{pV}{RT} = \frac{14.7 \times 144 \times 1}{53.3 \times 521.6} = 0.0761 \text{ lb.}$$

The average temperature of the outgoing chimney gases is in economic practice about 500°F. or 959.6°F. on the absolute scale. Hence a cubic foot of this air will weigh on its emergence from the stack

$$W = \frac{pV}{RT} = \frac{14.7 \times 144 \times 1}{53.3 \times 959.6} = 0.0414 \text{ lb.}$$

The difference between 0.0761 and 0.0414 is 0.0347 lb. If now we assure that the chimney stack is 100 ft. high we find that the entering air forces itself inward with an unbalanced force of 3.47 lb. over every square foot of cross-sectional area in the chimney.

In engineering practice this draft pressure is not usually recorded in pounds pressure per square foot of chimney area. Instead of this unit, the engineer measures the height to which a column of water would be forced under this pressure. In other words, since water, one square foot in cross-sectional area, at the temperature of the boiler room usually weighs 62.0 lb., for every foot in height, to bring about a pressure of 3.47 lb. per square foot, the water would have to rise to a height of

$$\frac{3.47}{62.0} \text{ ft., or } \frac{3.47}{62.0} \times 12 \text{ in.} = 0.67 \text{ inches of water.}$$

Hence it is seen that under normal conditions of operation in the modern chimney, the theoretical draft should read 0.67 inches of water, when the stack is 100 ft. high.

This theoretical intensity of draft can never be actually observed with a draft gage or any recording device. If, however, the ash pit doors of the boiler are closed and there is no perceptible leakage of air through the boiler setting or flue, with a stack 100 ft. high filled with gases at 500°F., and with external air at 62°F., a draft gage connected to the base of the stack will read approximately 0.67 inches.

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DRAFT FORMULA FOR THE MODERN POWER PLANT

Assuming that the density of the chimney gas is the same as air under the same conditions of pressure and temperature, we can at once by following the simple numerical illustration given in the last discussion develop a formula not only for a chimney 100 ft. in height, but for any height H and absolute temperature T of entering air and absolute temperature T_1 of stack gases.

By substituting in the elementary formula for chimney gases connecting pressures, volumes, and absolute temperature as set forth in the last discussion we have, where W_1 is the total weight of air within the chimney and W the weight of an equal volume of air without the chimney that

$$W_1 = \frac{pV}{RT_1} \text{ and } W = \frac{pV}{RT}$$

Hence the net force pressing inward from the outside heavier air over the entire chimney area A sq. ft. is

Net force in lb. =

$$W - W_1 = \frac{pV}{RT} - \frac{pV}{RT_1} = \frac{pV}{R}\left(\frac{1}{T} - \frac{1}{T_1}\right)$$

If the height of the chimney is H ft. and its area of cross-section A sq. ft. we have, since $V = AH$,

$$\text{Net force in lb.} = \frac{pAH}{R}\left(\frac{1}{T} - \frac{1}{T_1}\right)$$

Then the net force F over one sq. in. of chimney cross-section would be $\frac{1}{144A}$ of the above, which is

$$F = \frac{1}{144A} \times \frac{pAH}{R}\left(\frac{1}{T} - \frac{1}{T_1}\right) = \frac{pH}{144R}\left(\frac{1}{T} - \frac{1}{T_1}\right)$$

Since p is the atmospheric pressure in lb. per sq. ft., we have that $p = 144P$, wherein P is the atmospheric pressure in lb. per sq. in., and also, since $R = 53.3$ we have

$$F = \frac{144HP}{144 \times 53.3}\left(\frac{1}{T} - \frac{1}{T_1}\right) = \frac{HP}{53.3}\left(\frac{1}{T} - \frac{1}{T_1}\right)$$

Converting F from pounds pressure per sq. in. to inches of water by substituting the weight of water at the boiler room temperature which is 62.0 pounds per cubic foot, we proceed exactly as in the numerical example of the last discussion and find that

$$D = 0.52\, HP\left(\frac{1}{T} - \frac{1}{T_1}\right) \qquad (1)$$

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By means of this formula, we are enabled to compute drafts for any temperatures and pressures. Thus a chimney situated 10,000 ft. above sea-level, where the atmospheric pressure is 10 lb. per sq. in. with a stack of 100 ft., and entrance and exit temperature of 62°F. and 500°F. respectively, would have a draft of

$$D = 0.52 \times 100 \times 10\left(\frac{1}{62 + 459.6} - \frac{1}{500 + 459.6}\right)$$
$$= 0.46 \text{ in. of water.}$$

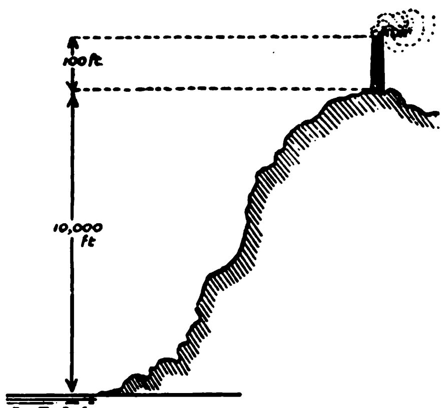

FIG. 161.—The theoretical chimney draft.

A stack 100 feet high situated 10,000 feet above sea level, with entering air at 62°F. and exit gases at 500°F. with ash pit door closed and no perceptible leakage through the boiler setting, should register 0.46 inches of draft.

It is seen that altitude has much to do with chimney drafts, for this identical chimney was previously shown to have a draft of 0.67 inches of water at sea level.

The above formula gives the theoretical draft—that is, the draft that would be obtained under perfect conditions if there were no losses. In practice, however, there is a considerable reduction in draft on account of the friction of the gases passing up through the stack. The greater the velocity of the gases, the greater is the friction. By increasing the diameter of the chimney for the same capacity the velocity of gases is reduced, and it is evident, therefore, that the diameter of the chimney has an important bearing on the net effective draft obtained at its base.

The friction of the gases passing up the chimney can best be calculated by means of formulæ based on the flow of water

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through pipes, for the chimney can be considered to be a vertical pipe with an upward stream of gases flowing through it. Gebhardt[1] gives the following formula based on Chezy's formula for hydraulic flow:

$$D_c = k\,\frac{W^2HT_1}{d^5}$$

where D_c = draft loss in the chimney, in. of water
W = weight of flue gas, lb. per sec.
H = height of chimney, feet
T_1 = absolute temperature of chimney gases, deg. F.
d = diameter of chimney, inches.

The constant k is given by different writers varying from 1.6 to 3.0, depending on the assumed coefficient of friction of the gases against the walls of the stack. For practical purposes it is sufficiently accurate to take the mean of the above values, giving the value $k = 2.3$.
The formula, therefore, becomes:

$$D_c = 2.3\,\frac{W^2HT_1}{d^5} \tag{2}$$

Deducting the draft loss in the chimney from the theoretical draft, we have the available draft, D_1, at the base of the chimney

$$\begin{aligned} D_1 &= D - D_c \\ &= 0.52\,HP\left(\frac{1}{T} - \frac{1}{T_1}\right) - 2.3\,\frac{W^2HT_1}{d^5} \end{aligned} \tag{3}$$

Expressed in words this formula means: to find the available draft at the base of the chimney, first calculate the theoretical draft by means of formula (1), and then deduct the friction loss as determined from formula (2). For example, let us determine the available draft at the base of a chimney 5 ft. diameter and 100 ft. high when operating at sea level and passing 20 lbs. of flue gas per second at a temperature of 500°F.

From Formula (1) we find, as in the previous example, that the theoretical draft is 0.67 inches of water. From formula (2) we find that the draft loss in the chimney

$$D_c = 2.3 \times \frac{20^2 \times 100 \times (500 \times 461)}{60^5} = 0.11 \text{ in. of water.}$$

The available draft at the base of the chimney is therefore 0.67 − 0.11 = 0.56 in. of water.

[1] Steam Power Plant Engineering—Geo. F. Gebhardt—5th Ed., p. 289.

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In the table in Fig. 162 the available draft is given for chimneys of various diameters all 100 ft. high, with given quantities of flue gas passing per hour based on a temperature of gases of 500°F. For any other height of chimney the draft may be obtained directly from the table by simply multiplying the draft given in the table by the actual height, and dividing by 100.

DRAFT PRODUCED BY CHIMNEY 100 FEET HIGH IN INCHES OF WATER

Average Temperature of Chimney Gases, 500°F.

Flue gas per hour, lb.	Diameter of chimney in inches																			
	36	42	48	54	60	66	72	78	84	90	96	102	108	114	120	132	144	156	168	180
20,000	**.55**	.62																		
30,000	.42	**.55**	.60																	
40 000	.21	.46	**.56**	.61																
50,000		.34	.49	**.57**	.61															
60,000		.19	.42	**.53**	.59	.61	.63													
80,000			.23	.43	**.53**	.58	.61	.63												
100,000				.29	.45	**.53**	.58	.61	.63	.64										
120,000					.35	.47	**.54**	.58	.61	.63	.64	.65								
160,000						.31	.43	**.52**	.56	.59	.62	.63	.64	.65	.65					
200,000							.30	.43	.50	**.55**	.59	.61	.62	.63	.64	.65				
250,000								.30	.41	.49	**.54**	.57	.60	.61	.63	.64	.65			
300,000									.31	.41	.48	**.52**	.56	.59	.61	.63	.64	.65		
350,000											.40	.47	**.52**	.56	.58	.62	.63	.65	.65	.66
400,000												.42	.48	**.52**	.56	.60	.62	.64	.65	.66
450,000													.43	.49	**.53**	.58	.61	.63	.64	.65
500,000														.44	.49	**.56**	.60	.62	.64	.65
550,000														.39	.45	.53	.58	.61	.63	.64
600,000															.41	.50	**.57**	.60	.62	.63

Figures underlined represent draft for chimney of least first cost.
For other heights multiply the draft given in the table by the height above the point at which the draft is to be determined, and divide by 100.

FIG. 162.

The set of curves in the diagram in Fig. 163 gives the same information, with the addition of extra scales which enable the draft to be determined for temperatures of 300°F. and 700°F., as well as 500°F. By interpolating between the results so obtained, the draft may be quickly determined for any temperature between these limits.

This table and diagram may be used for determining the height and diameter of chimney required for any boiler plant, to produce a given amount of draft, regardless of the kind of fuel used, pro-

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vided the quantity of flue gas passing per hour can be approximately estimated.

The weight of flue gas passing per second or per hour depends, for practical purposes, on just two things:

(1) The weight of fuel burned.

(2) The per cent. excess air used in burning it.

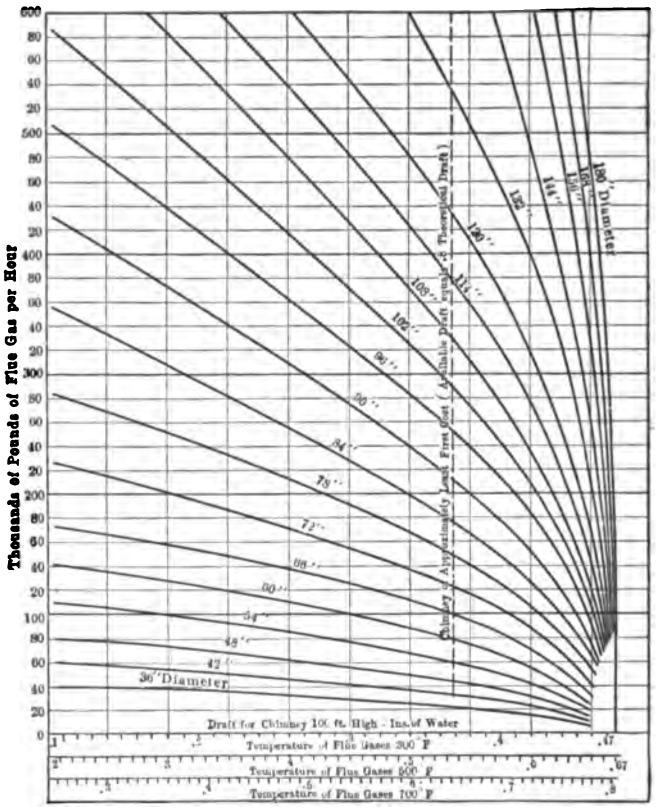

FIG. 163.—Diagram showing available draft at chimney base for various rates of gas flow and chimney diameters.

Other influences, such as moisture or oxygen in the fuel, or steam used to atomize oil, may be neglected, as their effect is small.

The weight of oil burned under a boiler, operating at 70 per cent. efficiency, amounts to 2⅔ lb. per hour per blr. h.p. As this efficiency is easily obtained with ordinary care, this is a liberal figure to use for design purposes.

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Less

Oil can be burned with no more than 15 per cent. excess air, corresponding to 14 per cent. CO_2, but as extreme care is required to attain this result the chimney should be designed on a more liberal basis. A safe figure to use is 50 per cent. excess air, corresponding to 10 per cent. CO_2. The theoretical amount of air required runs from 13 to 14 lb. per pound of oil, depending on the chemical constituents of the oil. Taking 14 lb. as the theoretical amount of air, and adding 50 per cent. excess, we have a total of 21 lb. of air for each pound of oil, and as the pound of oil is completely gasified, there are 22 lb. of flue gas per pound of oil. Multiplying this by the $2\frac{2}{3}$ lb. of oil per h.p. gives 58.7, or approximately 60 lb. of flue gas per hour per boiler h.p. Consequently the weight of flue gas per hour passing up a chimney may be estimated by multiplying the actual boiler horsepower by 60.

In practice it is customary to overload boilers up to 150 per cent., 200 per cent. and sometimes as high as 300 per cent. of their rated capacity. The actual horsepower may, therefore, be very different from the installed capacity. In designing the chimney it is essential to know what capacity is expected from the boilers, and be guided accordingly.

An Example of Chimney Design for Sea-level Installation. Let us from this data ascertain the diameter of a 100-ft. chimney capable of properly creating a draft for a 1000 h.p. boiler installation so that the stack is of sufficient size to accommodate a 50 per cent. overload. It is assumed that the stack is centrally located and that it has short flue connections with ordinary operating boiler efficiency and that a draft of 0.5 in. at the base of the chimney is sufficient. Since the 1000 h.p. boilers are to operate at 50 per cent. overload, the actual horsepower will be 1500. Multiplying this by 60 gives 90,000 lb. of flue gas per hour. From the diagram it is seen at a glance that the chimney must be a little more than 60 inches in diameter to give the required draft of 0.5 in. with gases at an average temperature of 500°F.

It will be noted from the table that it is possible to get several different combinations of heights and diameters of stacks, to produce a given draft for any particular weight of flue gas considered. For instance the diagram shows that a stack 54 in. diameter and 100 ft. high will give a draft of 0.36 in. when passing 90,000 lb. of flue gas per hour at a temperature of 500°F. If the height of this stack is increased to 140 ft., the draft obtained would

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be $0.36 \times {}^{140}/_{100} = 0.50$ in., which is the same draft for which the 60 in. $\times$ 100 ft. stack was figured in the previous example. A number of other combinations of height and diameter can be made that will give the same draft. The choice of the actual chimney to be used must therefore depend on other considerations besides the draft and in practice this point is settled by selecting the chimney that will be the cheapest to build. The cost of a

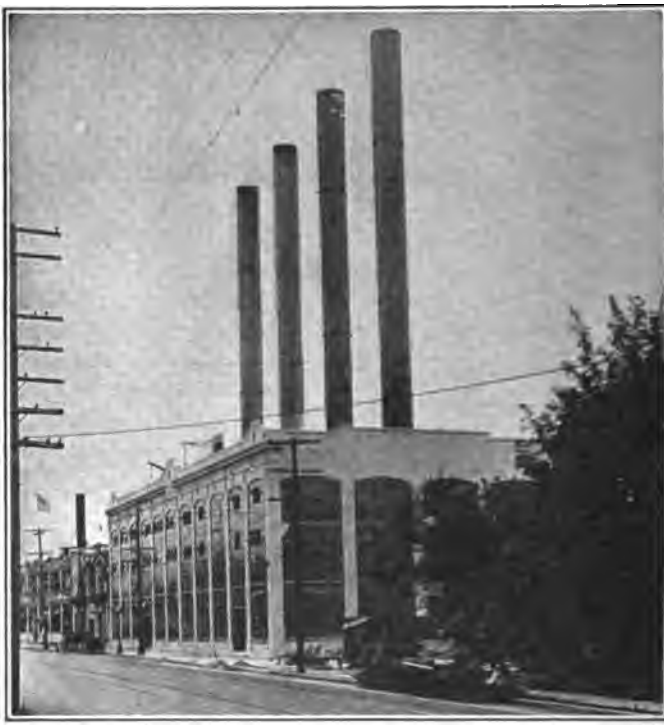

Fig. 164.—Typical sea-level installation in metallic chimney design for fuel oil practice. The view shown is that of the steam power plant auxiliary for the city of Seattle, Eastlake and Nelson Place.

chimney is roughly proportioned to its height times its diameter, and on this basis it has been found that the most economical chimney to build, for a given set of conditions, is that in which the available draft is equal to eight-tenths of the theoretical draft. In the diagram in Fig. 163 this relationship is indicated by the vertical dotted line, and the proper diameter of stack is found at a glance by noting the point at which the diameter curves cross this

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line. For example, considering once more the case of 1000 h.p. oil-fired boilers operating at 50 per cent. overload, and requiring 0.5 in. draft at the chimney base, we have, as before, 90,000 lb. of flue gas per hour at a temperature of 500°F. Following the horizontal line corresponding to 90,000 lb. per hour, until it crosses the vertical dotted line, we find that the cheapest chimney to build for this capacity will be between 60 in. and 66 in. diameter. It is customary to build stacks with diameters equal to some multiple of 6 in. We may therefore select 66 in. as the required diameter. The diagram shows that under the assumed conditions the draft for a 66-in. stack, 100 ft. high, will be 0.555 in. Since the draft required is only 0.5 in., the height of the stack may be reduced in proportion, making it 90 ft. high instead of 100 ft. It is thus found that the proper size of chimney for the conditions of the problem is 66 in. diameter and 90 ft. high.

FIG. 165.—The power plant in the Woolworth Building, New York, has the tallest chimney in the world. The height of the stack is determined by that of the building, but in some cases the stack must be built extra high in order to discharge well above surrounding buildings.

In the table in Fig. 162 the chimney of least first cost is indicated by printing in bold type the draft obtained from the proper size of chimney for each given capacity.

Sometimes stacks must be built higher than would otherwise be necessary, in order to discharge well above surrounding buildings, in which case a smaller diameter may be used than would be required for a lower stack. In tall office buildings the height of the stack is determined by the height of the building itself. It is not generally known that the tallest chimney in the world is the one provided for the power plant of the Woolworth Building in New York.

Chimneys that have small diameters in proportion to their

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height are somewhat objectionable on account of the variability of the draft. At the full load of 1000 boiler h.p., as we have seen, a 54 in. × 140 ft. stack gives the same draft as a 66 in. × 90 ft. stack. At light loads, however, the 54 in. × 140 ft. stack will give a much greater draft, for it still has its full height, but the friction loss is much less. This increase of draft at light loads requires special care on the part of the boiler fireman to adjust his dampers for proper air regulation.

Corrections in Chimney Height for Altitude.—We shall next consider the necessary corrections to be made in the dimensions of proposed chimneys in their relation to altitude above the sea. All chimney dimensions and tables have been computed on the basis of sea-level pressures. From our equation of draft readings previously derived, it is seen that the draft depends directly upon the atmospheric pressure. Hence it is evident that since the higher the altitude, the less the pressure, the stack must be lengthened in proportion to the barometric readings. Thus if H is the proper height of a chimney at sea level or barometric pressure P_o, then H_1 the proper height at the altitude P_1 above sea level is as follows:

Fig. 166.—Atmospheric barometer.

$$\frac{H_1}{H} = \frac{P_o}{P_1}$$

or if r is a factor obtained by dividing the barometric reading at sea level by the barometric reading at the proposed point of installation,

$$H_1 = rH$$

This reasoning is based on the assumption of constant draft measured in inches of water at the base of the stack for a given rate of operation of the boilers regardless of altitude.

An important point to consider in the construction of the stack is how the altitude will affect the cross-sectional area. At high altitudes the air becomes less dense, hence the area should be larger in order to pass the required weight of air needed in com-

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bustion of the fuel, for the same weight of air is needed for proper fuel combustion, no matter what the altitude may be.

In the flow of gases through pipes, it has been found that the weight passing any given section per minute is

$$W = K\left(\frac{pDd^5}{\left(1+\frac{3.6}{d}\right)L}\right)^{\frac{1}{2}}.$$

Where K is constant; p is the difference in pressure between two ends of pipe; D the density; d the diameter of pipe in inches; and L the length of pipe. Since these quantities will later disappear in self-cancelling pairs from this equation, the noting of the particular units involved in measurement is not necessary. In applying this formula to gases flowing through a stack, the quantity $\left(1+\frac{3.6}{d}\right)$ is practically unity, the quantity L becomes equal to H and H_1 in the respective cases, and p is the same in each case. Hence we have

$$W = K\left(\frac{pDd^5}{H}\right)^{\frac{1}{2}} \text{and } W_1 = K\left(\frac{pD_1d_1^5}{H_1}\right)^{\frac{1}{2}}$$

But W must equal W_1 for the same economy of fuel burning. Hence

$$K\left(\frac{pDd^5}{H}\right)^{\frac{1}{2}} = K\left(\frac{pD_1d_1^5}{H_1}\right)^{\frac{1}{2}}$$

Also

$$\frac{D}{D_1} = r \text{ and } \frac{H_1}{H} = r$$

Therefore, substituting and cancelling

$$\frac{Dd^5}{H} = \frac{D_1d_1^5}{H_1}$$

$$\frac{rD_1d^5}{H} = \frac{D_1d_1^5}{rH}$$

$$r^2d^5 = d_1^5 \therefore d_1 = dr^{\frac{2}{5}}$$

Rule for Altitude Correction.—Hence to properly proportion a chimney for a given altitude above sea level, first pick the height and diameter for the boiler capacity on the assumption that the installation is to be made at sea level. Next determine the height for the altitude desired by making the ratio of the new height to the sea-level determination inversely proportional to the barometric readings. The stack diameter is then increased so that

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the stack at the higher altitude should have the same frictional resistance as that used at sea-level. This new diameter is determined by multiplying the diameter obtained on the basis of sea-level assumption by the ratio r of barometric heights raised to the $\frac{2}{5}$th power as above deduced.

An Example of Chimney Design at Altitude.—Since it is now seen that the factor or ratio of sea-level pressure to the pressure at altitude enters as a first and a two-fifths power, a chart is herewith given by means of which this factor may be quickly raised to the power desired for altitudes up to 10,000 ft., without any reference to barometric pressures.

As an example, let us find the proper proportions of a chimney to amply provide for a 1000 boiler horsepower installation situated 8000 ft. above sea-level.

We have hitherto found that the proper dimensions at sea-level for such an installation are 66 in. in diameter for a height of 90 ft. Applying our rule set forth above, we find from the chart that r for 8000 ft. is 1.357. Hence the proper height is 122 ft. at this altitude, and since r raised to the $\frac{2}{5}$th power is found from the chart to be 1.130 the proper diameter is 74.5 in.

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CHAPTER XXX

ACTUAL DRAFT REQUIRED FOR FUEL OIL

For every kind of fuel and rate of combustion there is a certain draft with which the best general results are obtained. A comparatively light draft is best for burning bituminous coals and the amount to use increases as the percentage of volatile matter diminishes and the fixed carbon increases, being highest for the small sizes of anthracites. Numerous other factors such as the thickness of fires, the percentage of ash and the air spaces in the grates bear directly on this question of the draft best suited to a given combustion rate.

For fuel oil, the question of draft required is greatly simplified by the fact that the air does not have to be drawn in through a thick bed of fuel and there are no ashes or clinkers to further complicate the matter. The resistance offered to the entrance of air to the furnace is caused by the checkerwork furnace floor, and as the openings in the checkerwork can be altered at will, it is evident that the amount of draft required in the furnace will depend largely on the arrangement of checkerwork adopted.

For a furnace arrangement such as shown on page 158, in which the total net area of free air space amounts to 3 to 3½ sq. in. per rated horsepower of the boiler, the draft required in the furnace amounts to the following, approximately:

Per cent. of rating	Draft in furnace, inches of water
100	0.05
150	0.10
200	0.25

The draft in the furnace is only a small proportion of the total draft that must be supplied by the chimney, for it is necessary to add to the furnace draft the draft loss caused by the friction of the gases in passing through the boilers, breechings and flues leading to the chimney.

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DRAFT LOSSES IN STEAM POWER GENERATION

The loss of draft is greatest in boilers having the longest path of gases, the greatest velocity, and the greatest number of changes in direction of flow of gases. A boiler having a single pass with the hot gases entering at the bottom and leaving at the top has a minimum draft loss. In most designs of boilers, however, this arrangement cannot be adopted as the area of gas passage would be too large. This would result in the gases short circuiting, that is passing in a narrow stream from one corner to the other without coming in contact with all of the heating surface. To make the heating surface effective in absorbing heat from the gases it is therefore necessary to provide baffles in the boiler, which deflect the gases and cause them to travel back and forth until their temperature has been reduced as much as possible.

The arrangement of baffles is a feature of boiler design and need not be entered into here. It is well, however, to refer briefly to the general principle involved, namely, that the higher the velocity of gases traveling over the heating surface the greater will be the coefficient of heat transfer. Consequently it would seem that in order to insure maximum efficiency of the boiler there should be a large number of passages of small area, so as to insure high velocity to the gases. This is true up to certain limits, but unfortunately it is soon found that the additional loss of draft caused by increased friction and extra changes in direction of the gases makes the production of the required draft both difficult and expensive.

In the majority of water tube boilers the baffles are arranged for three passes, that is the gases are forced to travel the length or height of the boiler setting three times before reaching the stack. With this arrangement the areas of passes are such as to give the gases a velocity of 10 or 15 ft. per second when the boiler is operating at its rated capacity. By increasing the number of passes to four or five the velocity may be increased to 20 or 30 ft. per second. This results in a higher rate of heat transmission so that more heat is absorbed from the gases, reducing their temperature and resulting in less waste to the chimney.

To enable the number of passes in a boiler to be increased the chimney must be designed to suit the increased loss of draft that will occur. Thus in every case the actual draft loss should be determined as closely as possible, and the actual figures for the

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particular case in hand used in designing the chimney. It is desirable in all cases to design the stack for a greater draft than is expected, for it is a simple matter to reduce the draft by closing in on the damper, whereas if the draft is insufficient nothing can be done to increase it. Again, it may be desired at some future time to increase the number of passes in the boiler, or otherwise modify the baffles in such a way as to require more draft. This would be impracticable unless the stack is large enough to produce a surplus of draft.

In order to give the reader some general ideas of computations involved in ascertaining draft losses assumed in design we shall now pass to a brief consideration of this problem.

Loss of Draft in Boilers.—The loss of draft through a boiler proper will depend upon its type and baffling, and will increase with the per cent. of rating at which it is run. For design purposes, it may be assumed that the loss through an oil fired boiler between the furnace and the damper will be 0.15 in. when it is run at its rating, 0.35 in. at 150 per cent. of its rating and 0.60 in. at 200 per cent. of its rating.

Loss in Flues and Turns.—With circular steel flues of approximately the same size as the stack or when reduced proportionally to the volume of gases they are to handle, a convenient rule is to allow 0.1 in. draft loss per 100 ft. of flue length and 0.05 in for each right angle turn. These figures are also good for square or rectangular steel flues with areas sufficiently large to provide against excessive frictional loss. For losses in brick or concrete flues these figures should be doubled.

Thus the loss in draft flues and turns for an installation having a flue 100 ft. long and containing two right angle turns is

Loss for flues, per 100 ft.	0.1 in.
Turns 2×0.05	0.1 in.
	0.2 in. loss

Total Available Draft Required.—We are now enabled to compute the total available draft required for a boiler installation by summing up the separate components required for the furnace, for the boiler, for the flues and for the turns.

Thus, for an oil fired boiler to operate at 200 per cent. of its rated capacity, connected to a chimney through a flue 100 ft.

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long and containing two right angle turns, we have the following:

Draft in furnace	0.25 in.
Draft loss in boiler	0.60 in.
Draft loss in flue	0.20 in.
Draft required at base of chimney	1.05 in.

The size of chimney required to produce this draft may be determined by the method described in the last chapter. Thus, let us suppose we are considering an installation of 2500 h.p. At 200 per cent. of rating there will be 5000 h.p. actually developed, and the quantity of flue gas produced by the oil fires will be $5000 \times 60 = 300{,}000$ lb. per hour. From the diagram on page 258 we find that a chimney for this capacity should be 102 in. diameter. We may assume that at the capacity considered the average temperature of chimney gases will be 600°F. The diagram on page 258 gives the draft for a chimney 102 in. diameter and 100 ft. high with 300,000 lb. per hour of flue gas, 0.525 in. for 500°F., and 0.63 in. for 700°F. Interpolating we have a draft of 0.58 in. for a temperature of 600°F. Since the draft required is 1.05 in., the height of the chimney must be $100 \times \frac{1.05}{0.58} = 181$ ft. This is the height of the chimney above the point at which the flue enters. If the flue enters the chimney 14 ft. above the ground, then the total height of the chimney must be 195 ft.

Artificial Draft.—As we have seen draft in a stack is caused by difference in pressure between the gases inside and outside, resulting in a flow of air from the higher external pressure to the lower internal pressure. A similar difference in pressure, and consequent flow of air, may be produced by a fan or blower instead of by a chimney. When this is done we have what is known as Artificial Draft.

There are two forms of artificial draft known as Forced Draft and Induced Draft, the distinguishing feature between the two being the location of the fan in respect to the boiler.

In the case of Forced Draft the fan sucks air direct from the atmosphere and delivers it to the boiler, under pressure somewhat greater than that of the atmosphere. In the case of Induced Draft the fan is located between the boiler and the stack, sucks the gases of combustion out of the boiler and discharges them to the stack.

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Since Forced Draft produces a pressure greater than atmospheric, its use is confined principally to forcing air through a thick bed of fuel on the grates. It is used largely in connection with certain kinds of stokers, and frequently with hand fired boilers using low grade coals. Forced draft is not suitable for steam atomized oil fired boilers, because there being no fuel bed on the grates to offer resistance, the positive pressure from the fan discharge would be carried up into the boiler setting. This would cause the gases to leak out into the boiler room, and in some cases would result in excessive furnace temperature and

FIG. 167.—Exterior view of San Francisco's new high pressure salt water pumping plant, showing chimney design for fuel oil practice.

Here is a view of the housing for the Townsend Street high pressure salt water pumping station at San Francisco. The chimney stack as shown illustrates the best and most permanent type of design for fuel oil practice. No artificial means of producing a draft is employed in this installation. It is a plant kept in eternal readiness, should disaster ever again visit San Francisco as happened during April, 1906, in the days of the great fire.

burning out of the brickwork. For satisfactory operation of stationary boilers it is necessary to keep the pressure of the gases within the boiler setting slightly lower than atmospheric pressure. When forced draft is used the positive pressure should not extend beyond the ash pit. Forced draft is used extensively with Mechanical Atomizing oil burners, where owing to the small area for air admission it is impossible to get into

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the furnace enough air for high overloads by means of natural draft.

Induced draft can be used instead of natural draft wherever desired. It is cheaper to install than a high stack, but the power required to drive the fan makes it more expensive to operate. It is of especial advantage where the gases escaping from the boiler are passed through an economizer to absorb some of their heat, before they are allowed to reach the chimney. The economizer introduces added frictional resistance to the gases so that extra draft is required. Besides this, the economizer reduces the temperature of the gases to such an extent that to obtain sufficient draft without a fan would require a stack of excessive height. By installing an induced draft fan between

FIG. 168.—Exterior view of San Bernardino Plant of Southern Sierras Power Company where artificial draft is employed.

the economizer and the stack, ample draft can be obtained regardless of the height of the stack or the temperature of the gases.

Induced draft is also of value, even when economizers are not employed, in cases where it is impracticable or undesirable to build a stack of normal height. An example of this is found in the power plant of the University of California, where a high unsightly stack would seriously interfere with the architectural features of the university buildings. By building a stack only 50 ft. high, and supplementing it with an induced draft fan, this difficulty was overcome.

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CHAPTER XXXI

CHIMNEY GAS ANALYSIS

We have found in preceding discussions that for practical purposes the gases passing out through a chimney from the central station boiler are usually considered to be composed of carbon dioxide, oxygen, carbon monoxide and nitrogen. Since these constituents are usually determined volumetrically we shall represent them by the symbols V_1, V_2, V_3, and V_4, respectively. We shall now proceed to a discussion of the usual methods employed in determining the flue gas analysis during the boiler test.

FIG. 169.—A carbon dioxide recorder.

The Taking of the Flue Gas Samples and Analysis.—Certain solutions have been found in the chemist's laboratory that will absorb carbon dioxide and will not absorb oxygen, carbon monoxide or nitrogen. Again another solution has been found that will absorb oxygen but will not absorb carbon monoxide or nitrogen. And still a third solution has been found that will absorb carbon monoxide but will not absorb nitrogen. If then a contrivance can be set up so that a flue gas sample may be successively washed in these solutions, a means is provided for determining an analysis by volume.

Orsat Apparatus.—Let us then see how the flue gas analysis is taken. The apparatus commonly called the Orsat Apparatus (see Fig. 171) consists of a wooden case with removable sliding doors which contain a measuring tube or burette B, three absorbing bottles or

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pipettes, P', P'', and P'''. In addition a leveling bottle A and connecting tube T are also provided.

The tube E is connected to the point in the flue at which the sample is to be taken. The instrument is first set in

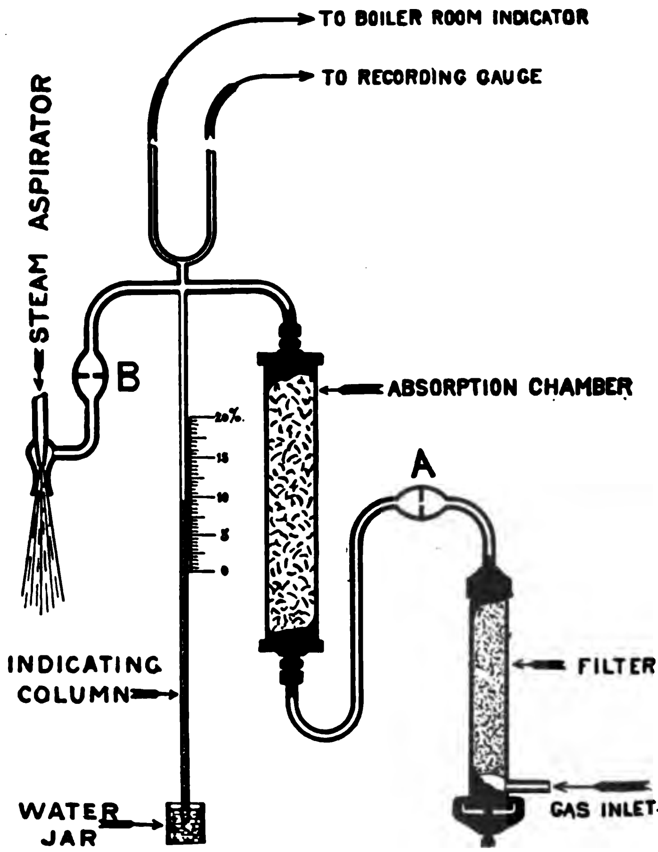

FIG. 170.—A recorder for combustion operation.

From the discussion in the text it may be inferred that a knowledge of the carbon dioxide component of the flue gas enables us to judge concerning the combustion taking place in the furnace. The principle involved in the type of carbon dioxide recorder as shown is that a change of volume in a gas produces a change of pressure. A continuous sample of the flue gas enters at A and in passing through the absorption chamber the carbon dioxide is absorbed and consequently a reduction in pressure takes place. By the calibration of suitable manometer tubes the instrument may be made to read the carbon dioxide component direct.

operation by closing the stop-cocks f, g, and e, d being open. By lowering the leveling bottle A, a sample of the gas is drawn into the burette B. This preliminary sample is then

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expelled to the atmosphere by raising the bottle *A* and allowing the gas thus put under pressure to pass out through a by-pass at *d*. This process is continued until it is considered that an average sample has been drawn into the burette *B*. The leveling bottle *A* is next lowered so as to cause the water in burette *B* to come to its zero mark. By raising the bottle *A* the water is again forced into burette *B* and the gas sample expelled through stop-cock *e* into the pipette *P'*, in which there is a chemical solution

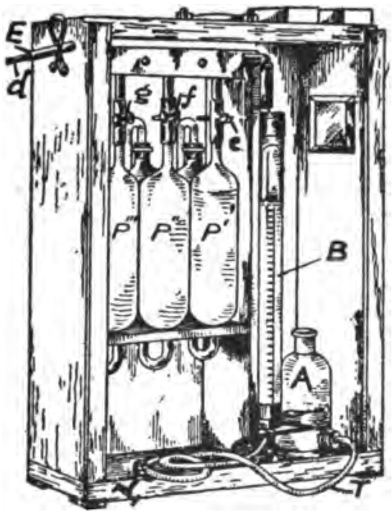

FIG. 171.—The Orsat apparatus.

The Orsat apparatus is a portable instrument contained in a wooden case with removable sliding door front and back, as shown in its simplest form in this illustration, taken from the report of the Power Test Committee of the American Society of Mechanical Engineers. It consists essentially of a measuring tube or burette, three absorbing bottles or pipettes, and a leveling bottle, together with the connecting tubes and apparatus. The bottle and measuring tube contain pure water; the first pipette, sodium or potassium hydrate dissolved in three times its weight of water; the second, pyrogallic acid dissolved in a like sodium hydrate solution in the proportion of 5 grams of the acid to 100 cc. of the hydrate; and the third, cuprous chloride. These chemicals are sold by most of the large dealers. Details of how this apparatus is used to determine the chimney gas analysis were set forth in a previous discussion.

that absorbs carbon dioxide, but will not absorb oxygen, carbon monoxide or nitrogen.

To Ascertain the Carbon Dioxide Content of a Flue Gas.—Exactly 100 cc. of gas were originally drawn into the burette *B*. If now the leveling bottle *A* is again lowered to draw the gas back through stop-cock *e*, the volume in the burette will be found to have lessened in quantity so that instead of reading zero it now reads N_1 which indicates directly the volume of car-

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bon dioxide that was present in the gas, for evidently this volume has been absorbed in the pipette P'. Hence, we have

$$V_1 = N_1 \tag{1}$$

To Ascertain the Oxygen Content of a Flue Gas.—In a similar manner the gas sample in the burette B is now forced through pipette P'', in which is a solution that will absorb free oxygen in the sample but will not absorb carbon monoxide or nitrogen. By means of the leveling bottle A, the sample is next drawn back into the burette B and a reading N_2 noted. It is now evident that the oxygen content of the flue gas may be computed from the formula

$$V_2 = N_2 - V_1 \tag{2}$$

To Ascertain the Carbon Monoxide Content of a Flue Gas. The pipette P''' similarly contains a solution which readily absorbs the carbon monoxide present in the gas, but will not absorb nitrogen. Hence we proceed as in the two former instances and return the gas sample to the burette which now reads N_3. Consequently the carbon monoxide which was present in the flue gas is obtained from the formula

$$V_3 = N_3 - (V_1 + V_2) \tag{3}$$

To ascertain the Nitrogen Content of a Flue Gas.—We assume that all of the gas which remains in the sample is nitrogen. Consequently the nitrogen content is obtained from the formula

$$V_4 = 100 - (V_1 + V_2 + V_3) \tag{4}$$

An Approximate Check on the Orsat Analysis.—Air is found by weight to have 76.85 per cent. nitrogen and 23.15 per cent. oxygen. By volume this analysis will be found to be 79.09 per cent. nitrogen and 20.91 per cent. oxygen. Since 1 unit by volume of oxygen forms 1 unit by volume of carbon dioxide in the burning of pure carbon the actual percentage of nitrogen in the chimney gases is not altered but should remain 79.09 per cent. if perfect combustion is maintained.

On the other hand, when imperfect combustion is under way, or in other words, when some carbon monoxide is being formed 1 unit by volume of oxygen forms 2 units by volume of carbon monoxide. Hence when pure carbon is the fuel, the sum of the percentages of carbon dioxide, oxygen, and ½ the carbon monoxide must be in the same ratio to the nitrogen present as the oxygen

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in the air is to the nitrogen component, namely as 20.91 : 79.09. This is a convenient check upon a flue gas analysis in the progress of the experiment. Thus if an analysis of chimney gas is found to contain by volume 9.5 per cent. carbon dioxide, 10.2 per cent. carbon monoxide, 5.2 per cent. oxygen, and 75.1 per cent. nitrogen, according to this proportion, we should have

$$9.5 + 5.2 + \frac{10.2}{2} : 75.1 = 20.91 : 79.09$$

Upon investigation this will be found to be approximately true and well within the limit of experimental accuracy.

Fig. 172.—Hays gas analyzer, convenient for carrying from place to place.

As California crude oil contains usually about 11 per cent. of hydrogen, the ready checking above indicated proves of no avail since the hydrogen content is not taken account of in the Orsat or flue gas analysis. As the relationship serves, however, to clinch our ideas of volumetric proportions of entering air and outgoing flue gases, it is well to bear it in mind.

In boilers fired by coal containing little hydrogen the CO does not usually exceed 1 or 2 per cent. and the sum of the Orsat readings $CO_2 + O + CO$ is usually beween 20 and 21 per cent. When burning oil, on the other hand, the sum of these readings may be as low as 16 or 17 per cent. due to the large proportion of hydrogen in the fuel, which means an apparent nitrogen content of 83 or 84 per cent. The reason for this is that the water vapor formed by the burning of hydrogen condenses in the Orsat apparatus and occupies practically no volume, but the oxygen which unites with the hydrogen brings with it the same proportion of nitrogen as does the oxygen that unites with the carbon. Consequently the Orsat indicates a larger proportion of nitrogen than would occur if the fuel were pure carbon.

Chemical Formulas for Preparing the Absorption Solutions. The bottle *A* and the measuring tube or burette *B* contain pure

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water only, while the first pipette P' in which carbon dioxide is absorbed contains sodium hydrate dissolved in three times its weight of water. The second pipette P'' in which oxygen is absorbed contains Pyrogallic acid dissolved in sodium hydrate in the proportion of five grams of the acid to 100 cc. of the hydrate, and in the third pipette wherein carbon monoxide is absorbed cuprous chloride is contained. These chemicals are sold by most of the large dealers.

Another series of formulas which work equally well and in many cases are more easily prepared, are the following:

To absorb the carbon dioxide, potassium hydroxide is used, and is made by diluting 500 grams of commercial potassium hydroxide in one quart of water. To absorb the oxygen, potassium-Pyrogallite is used wherein five grams of solid acid in 100 cc. of potassium hydroxide above mentioned is prepared. When over 28 per cent. of oxygen is present, it is necessary to use 12 grams of commercial potassium hydroxide to 100 cc. of water. To absorb the carbon monoxide, cuprous chloride is used which is prepared by covering the bottom of a quart measure with cuprous chloride (CuO) to a depth of $\frac{3}{8}$ths of an inch. The measure is then filled with hydrochloric acid, shaken and allowed to stand until it becomes colorless. The copper wire is then placed in the solution and left to stand for a number of hours.

The Hemphel Apparatus for Determining the Hydrogen Content.—It is seen from the above description that no means are provided to ascertain whether or not the hydrogen content of the fuel is being properly consumed. This determination can only be made by the refined laboratory apparatus of the chemist. The authors consider that such a test is beyond the scope of this work, hence the description of the Hemphel apparatus and its operation will not be undertaken in these pages. Standard works on this subject are, however, available in all chemical engineering libraries for those who desire to go into this subject. Except for refined tests covering certain particular problems in combustion the Orsat analysis of flue gases is considered sufficiently accurate for power plant practice. Indeed, in most instances, as we shall see, the determination of the carbon dioxide component alone gives us sufficient information for ordinary operating conditions.

Gas Analysis in the Power Plant.—The simple Orsat apparatus is used very extensively in many power plants. It is reliable and

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accurate and its only objections are that it is a somewhat delicate instrument and requires careful manipulation.

There are on the market other instruments that are more rugged in construction and therefore more suitable for power plant work, by which it is possible to determine the CO_2 only. While for any scientific investigation or accurate test it is necessary to determine the oxygen and CO as well as the CO_2, there are many cases in practical operation where a determination of the CO_2 alone is very valuable. Therefore, the simple instrument by which this can be done has a useful place in power plant work.

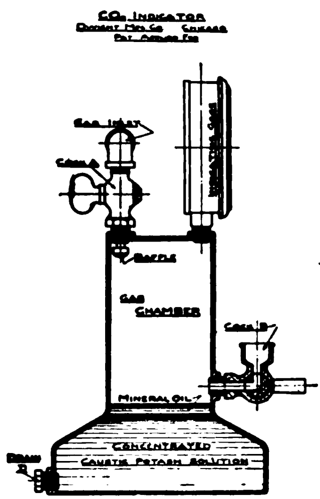

FIG. 173.—CO_2 indicator. Dwight Mfg. Co., Chicago, Pat. applied for.

One of these instruments which is illustrated in Fig. 173 is known as the Dwight CO_2 Indicator. This instrument consists of a metallic vessel, into which is pumped a charge of the flue gas to be analyzed. The vessel contains a small quantity of Caustic Potash solution for the purpose of absorbing the CO_2 in the sample. As soon as the gas is taken into the receiver the cocks are closed, and the instrument is shaken to thoroughly mix the gas with the absorbent solution, thus removing the CO_2 content. A small amount of mineral oil floats on top of the Caustic Solution to keep the gas from coming in contact with the caustic until after the cocks are closed and the instrument shaken. Removing the CO_2 in the gas reduces either its volume or its pressure. In this case, the volume remains constant, and consequently the removal of the CO_2 reduces the pressure, in accordance with Boyle's Law (page 44). The reduction in pressure is measured by a small vacuum gauge, which is calibrated to read direct in percentage of CO_2.

Another instrument known as the Pocket CO_2 Indicator is illustrated in Fig. 174. This is a compact portable instrument that operates on the same principle as the Orsat, and makes accurate CO_2 determinations.

Conclusion on the Orsat Analysis.—By care and a little patience, the experimenter will find that the Orsat analysis as

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above set forth can be taken easily and quite accurately, and thus a splendid lot of data obtained wherewith steam boiler economy and operation can be checked. If wrong conditions of combustion are found to prevail the proper adjustments can then be made in the furnace and its accessories.

We shall next proceed to formulate some equations whereby the data gained from the flue gas analysis may be thrown into more useful analytical form.

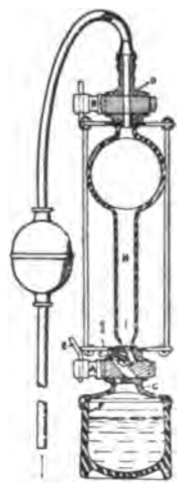

FIG. 174.—Pocket CO_2 indicator (patented) made by Bacharach Industrial Instrument Co., Pittsburg, Pa.

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CHAPTER XXXII

ANALYSIS BY WEIGHT, AND AIR THEORETICALLY REQUIRED IN FUEL OIL FURNACE

In the last discussion it was found that Orsat analyses of chimney gases are always made volumetrically. In computing combustion data from these analyses, however, it is often necessary to have the proportions or percentages by weight instead

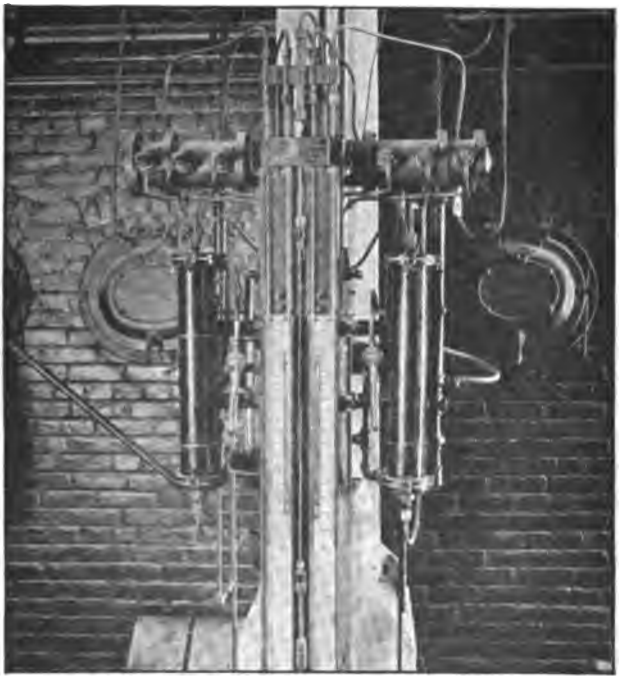

Fig. 175.—Carbon dioxide recording machine located in the Long Beach Plant of the Southern California Edison Company.

of by volume. The volumes of carbon dioxide, oxygen, carbon monoxide, and nitrogen which constitute the chimney gas analysis of a sample volume by means of the Orsat apparatus will be represented by V_1, V_2, V_3, V_4, respectively in this discussion.

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Let us now see how we may transfer this relationship so that proportions by weight of M_1, M_2, M_3 and M_4 pounds may respectively set forth the constituents of a flue gas sample of weight M pounds. Since we are only in search of proportions by weight—that is a ratio of M_1 to M, M_2 to M etc., it is evidently not necessary to actually know the quantitative values of the weights involved.

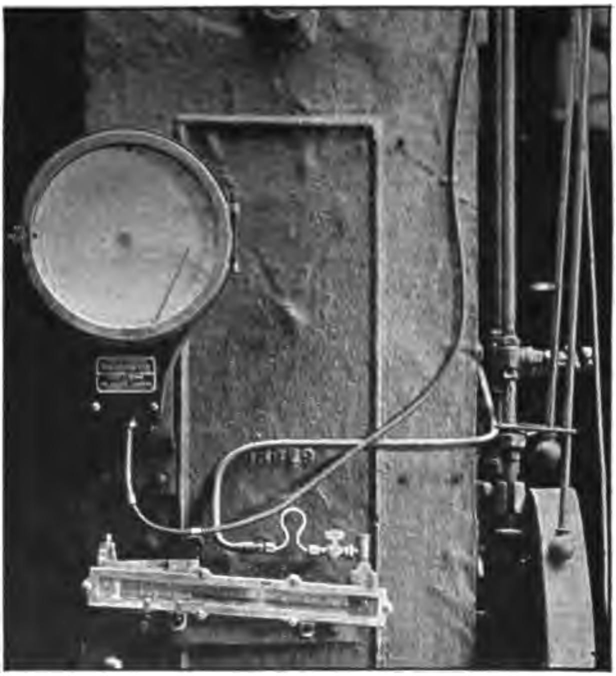

FIG. 176.—Recording thermometer and draft gage on Stirling boilers. Pacific Gas and Electric Company, station C, Oakland, Cal.

Fundamental Laws Involved.—In a previous discussion we found (see page 48) that all perfect gases follow the composite law—namely, that at any particular state the product of its pressure p and volume V is equal to the product of its weight M and absolute temperature T multiplied by a constant R, or mathematically expressed

$$pV = MRT$$

Hence, we may at once write the respective mathematical rela-

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tionships for the carbon dioxide, oxygen, carbon monoxide, and nitrogen of the flue gas.

It is to be remembered that in the case under consideration the pressure p and the temperature T have the same value for each component in the flue gas; consequently, we shall not put any individual subscript for the pressure p and temperature T, so that we may write these individual expressions as follows:

$$pV_1 = M_1R_1T \text{ or } M_1 = \frac{pV_1}{R_1T}$$

$$pV_2 = M_2R_2T \text{ or } M_2 = \frac{pV_2}{R_2T}$$

$$pV_3 = M_3R_3T \text{ or } M_3 = \frac{pV_3}{R_3T}$$

$$pV_4 = M_4R_4T \text{ or } M_4 = \frac{pV_4}{RT_4}$$

and for the gas as a whole, we have

$$pV = MRT \text{ or } M = \frac{pV}{RT}$$

In our previous discussion on the elementary laws of gases, it was also found mathematically that the constant R for any perfect gas is obtained by dividing 1544 by the molecular weight of the gas in question (see page 47).

From any book on elementary chemistry we find the molecular weight m of carbon dioxide (CO_2) is 44, that of oxygen (O_2) is 32, that of carbon monoxide (CO) is 28, and that of nitrogen (N_2) is 28.

Relationship of a Component Weight to the Whole.—Bearing this in mind, it is seen from the above mathematical relationships that, since $R = \frac{1544}{m}$, we have

$$M_1 = \frac{pV_1}{R_1T} = \frac{m_1pV_1}{1544T} = Km_1V_1 \text{ if } K = \frac{p}{1544T}$$

$$M_2 = \frac{pV_2}{R_2T} = \frac{m_2pV_2}{1544T} = Km_2V_2$$

$$M_3 = \frac{pV_3}{R_3T} = \frac{m_3pV_3}{1544T} = Km_3V_3$$

$$M_4 = \frac{pV_4}{R_4T} = \frac{m_4pV_4}{1544T} = Km_4V_4$$

$$M = \frac{pV}{RT} = \frac{mpV}{1544T} = KmV$$

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But $M_1 + M_2 + M_3 + M_4 = M$

$$\therefore \quad M = Km_1V_1 + Km_2V_2 + Km_3V_3 + Km_4V_4 = K(m_1V_1 + m_2V_2 + m_3V_3 + m_4V_4)$$

Let $C_s = m_1V_1 + m_2V_2 + m_3V_3 + m_4V_4$

$\therefore M = KC_s$

Also $M_1 = Km_1V_1$

Hence
$$\frac{M_1}{M} = \frac{Km_1V_1}{KC_s} = \frac{m_1V_1}{C_s} \qquad (1)$$

FIG. 177.—View of the float arrangements, showing the valves which control the inlet and outlet of oil from storage tanks of Long Beach Plant, Southern California Edison Company.

A Concrete Rule for Conversions.—This last equation now gives us a simple and ready rule for determining proportions by weight if the proportions by volume are given. In other words, this rule may be stated as follows:

In any analysis by volume, the analysis by weight is found by first summing the products formed by multiplying each component volume by its particular molecular weight. If now this summation C_s is divided into the product of a component volume

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and its particular molecular weight, the proportion by weight of that component is at once ascertained.

An Illustrative Example.—Thus, a flue gas analysis shows the following proportions by volume: carbon dioxide (CO_2) 0.086; oxygen (O_2) 0.110; carbon monoxide 0.011; and nitrogen (N_2) 0.793 per cent. Let us determine the proportions by weight present in this particular flue gas.

Since the molecular weights of carbon dioxide, oxygen, carbon monoxide and nitrogen are respectively 44, 32, 28, and 28, we find that m_1V_1 is 3.782, m_2V_2 is 3.520, m_3V_3 is 0.308, and m_4V_4 is 22.200. The sum of these products C_s is found to be 29.810. Hence since m_1V_1 is 3.782, we now find that the carbon dioxide component obtained by dividing 3.782 by 29.810 is 0.1270. Similarly for the oxygen component the proportion by weight is 0.1182; for the carbon monoxide component it is 0.0103; and for the nitrogen component we have 0.7453. As a check on our work we find that the sum of these separate components is unity as it should be. Or expressed in percentages, we would have for a volumetric analysis consisting of 8.6 per cent. carbon dioxide, 11.0 per cent. oxygen, 1.1 per cent. carbon monoxide, and 79.3 per cent. nitrogen, that the percentages by weight become 12.70 per cent. carbon dioxide, 11.82 per cent. oxygen, 1.03 per cent. carbon monoxide, and 74.53 per cent. nitrogen, which foot up 100 per cent. in either case and thus check our work.

A Suggested Form of Tabulation.—To expedite computation the work set forth in the above discussion may be tabulated. Below we have a form of tabulation which will prove useful for such transformations:

Constituents	Volume	Mol. Wt.	mV	$\frac{mV}{C_s}$
CO_2	0.086	44	3.782	0.1270
O_2	0.110	32	3.520	0.1182
CO	0.011	28	0.308	0.0103
N_2	0.793	28	22.200	0.7453
	1.000		C_s = 29.810	1.0000

Weight of Air Theoretically Required for Perfect Fuel Oil Combustion.—For economic combustion in the furnace a certain percentage of air over and above that theoretically required for

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perfect combustion is necessary. This is due to the fact that it is practically impossible to bring all of the entering air into intimate contact with the heated carbon, hydrogen, and other combustible ingredients of the fuel; consequently, unless an excess of air is admitted some of these ingredients will pass out of the chimney unconsumed. Good practice dictates from 15 to 20 per cent. excess of air as the proper ratio for economic fuel oil consumption in the furnace.

In order then to know when this ratio is properly established we must have some means of ascertaining the air theoretically required for perfect combustion as well as that actually used in the furnace per pound of fuel.

Correction for Oxygen Appearing in Fuel Analysis.—In the composition of fuels varying quantities of oxygen (O) are found by analysis to be present. While in a sense this is in a free state, still the hydrogen content is reduced in heating value by an amount equal to the combining weight of this oxygen (O) with the hydrogen (H). Experimentally we find that 8 lb. of oxygen combine with 1 lb. of hydrogen. Hence, so far as heating value is concerned and indeed so far as outside oxygen may be required for combustion of the hydrogen, the actual hydrogen content is reduced in value to $\left(H - \frac{O}{8}\right)$, where H represents the proportion by weight of hydrogen and O the proportion by weight of oxygen present in the fuel.

Oxygen Theoretically Required for Fuel Combustion.—The oxygen theoretically required is computed from a consideration of the fundamental chemical reactions that take place in the furnace.

Thus, from chemistry we learn that to completely burn 1 lb. of pure carbon $^{32}\!/_{12}$ths of a pound of oxygen are required. Again to burn 1 lb. of pure hydrogen 8 lb. of oxygen are required. And in the third place to burn 1 lb. of pure sulphur 1 lb. of oxygen is required.

If now 1 lb. of fuel oil is found by analysis to contain C parts by weight of carbon, H parts by weight of hydrogen, O parts by weight of oxygen, and S parts by weight of sulphur, it is evident that the weight of oxygen required per pound of fuel oil for perfect combustion is from the above discussion

$$\frac{32}{12}C + 8\left(H - \frac{O}{8}\right) + S$$

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Air Required per Pound of Fuel Burned.—Since air is composed of 0.2315 parts by weight of oxygen, the theoretical weight of air M_{ta} necessary to supply the oxygen above required for perfect combustion is

$$M_{ta} = \frac{32}{12} C \times \frac{1}{0.2315} + 8 \left(H - \frac{O}{8} \right) \frac{1}{0.2315} + S \times \frac{1}{0.2315}$$

$$\therefore M_{ta} = 11.52C + 34.56\left(H - \frac{O}{8}\right) + 4.32S \qquad (2)$$

An Illustrative Example.—Fuel analyses are always given in proportions or percentages by weight. In a certain boiler test a sample pound of the fuel oil analyzed as follows: carbon 81.52 per cent.; hydrogen 11.01 per cent.; sulphur 0.55 per cent.; and oxygen 6.92 per cent. Let us then compute the weight of air M_{ta} theoretically required to burn a pound of this oil.

In applying the formula above deduced, it must be remembered that the symbols there given for hydrogen, oxygen, and sulphur contents are in proportions and not percentages. Bearing this in mind we have by substitution—

$$M_{ta} = 11.52 \times 0.8152 + 34.56\left(0.1101 - \frac{0.0692}{8}\right) + 4.32 \times 0.0055 = 12.92 \text{ lb.}$$

Having now learned how to convert the Orsat analysis by volume into proportions by weight and also to ascertain the air theoretically required per pound of fuel, we shall in the next discussion determine actual combustion data by means of these stepping stones in computation.

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CHAPTER XXXIII

COMPUTATION OF COMBUSTION DATA FROM THE ORSAT ANALYSIS

Fig. 178. — Boiler front—oil fired; showing gage for measuring chimney draft.

IN the last chapter the reader was shown in detail how to convert the Orsat analysis by volume to an analysis by weight. We now assume that the volumetric content of a sample of flue gas has been taken and that V_1, V_2, V_3, and V_4 represent quantitatively the carbon dioxide, oxygen, carbon monoxide, and nitrogen contents respectively.

If accurately ascertained these components of the flue gas enable the engineer, as has been previously hinted, to compute important economic conclusions on the combustion phenomena that are taking place in the boiler furnace. It is important to know, for instance, not only the air that is theoretically required for perfect combustion, but the actual weight of air that is being admitted to the furnace per pound of fuel consumed. The weight of the flue gases per pound of fuel burned is, too, of importance, as well as many other details that may now be ascertained. Several different methods of utilizing the flue gas analysis have been proposed to arrive at combustion data. Let us now proceed to their consideration and discussion.

Air Actually Supplied to Furnace per Pound of Fuel Burned. There are three formulas that enable us to compute the actual quantity of air entering the furnace if we know the analysis of the chimney gases.

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From volumetric experiments in chemistry we learn that V_1 units by volume of oxygen form V_1 units by volume of carbon dioxide, thereby burning V_1 units by volume of carbon in the fuel. The unit volume of carbon is, of course, to be considered as a gas and not in its solid state as ordinarily encountered. Since carbon does not exist in a gaseous state at ordinary pressures and temperature, the reasoning involved is, of course, incorrect in that particular. However, for purposes of deriving the formula there is no inaccuracy introduced by the assumption that carbon in a gaseous state acts according to all well known laws of chemistry.

On the other hand, volumetric experiments in chemistry tell us that $\frac{V_3}{2}$ units by volume of oxygen form V_3 units by volume of carbon monoxide, thereby burning V_3 units by volume of carbon in the fuel.

Again, V_2 units of oxygen appearing in the flue gas have evidently necessitated the entrance of V_2 units of oxygen from the air without, but have required no carbon from the fuel.

Summing up, we find that $(V_1 + \frac{V_3}{2} + V_2)$ units by volume of oxygen have required $(V_1 + V_3)$ units by volume of carbon in the fuel for the formation of the particular analysis shown in the flue gas.

Therefore, one unit by volume of carbon in the fuel would require

$$\frac{V_1 + \frac{V_3}{2} + V_2}{V_1 + V_3}$$

units by volume of entering oxygen.

A unit volume of carbon, however, weighs 12 pounds, while a similar unit volume of oxygen weighs 32 pounds. Hence, for every unit weight of carbon consumed in the furnace

$$\frac{32}{12} \times \frac{V_1 + \frac{V_3}{2} + V_2}{V_1 + V_3}$$

units by weight of oxygen are required. Air has by weight 0.2315 proportions of oxygen. Hence if C units by weight of carbon are found in each pound of fuel, the actual weight of

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air M_c admitted to the furnace to burn carbon per pound of fuel burned is

$$M_c = \frac{32}{12} \times \frac{1}{0.2315} \times \frac{V_1 + \frac{V_3}{2} + V_2}{V_1 + V_3} \times C$$

$$\therefore M_c = 11.52 \times \frac{V_1 + \frac{V_3}{2} + V_2}{V_1 + V_3} \times C \qquad (1)$$

It is to be remembered that the derivation of this formula thus far has not taken into account the hydrogen content of the fuel. Since the Orsat analysis condenses the water vapor formed by the burning of the hydrogen with the entering oxygen of the air as the sample enters the first tube of the apparatus, the actual Orsat analysis indicates volumetric proportions for the dry flue gas only. We can, however, if we know the hydrogen content present in the fuel, make a correction or rather addition to the above formula so that the relationship will correctly represent the total admission of air into the furnace under test.

In the previous chapter it was shown that if a fuel analysis indicates H units of hydrogen by weight and O units of free oxygen by weight that the actual hydrogen available for combustion is $\left(H - \frac{O}{8}\right)$ units by weight.

It has been seen too that 1 lb. of hydrogen requires for its burning 8 lb. of oxygen, and that air contains 0.2315 proportions by weight of oxygen. Hence the weight of air necessary to burn

$\left(H - \frac{O}{8}\right)$ pounds of hydrogen is $\frac{8}{0.2315}\left(H - \frac{O}{8}\right)$ or

$$34.56\left(H - \frac{O}{8}\right) \text{ pounds.}$$

Therefore, the total air M_a admitted to the furnace per pound of fuel oil burned is

$$M_a = 11.52 \times \frac{V_1 + \frac{V_3}{2} + V_2}{V_1 + V_3} \times C + 34.56\left(H - \frac{O}{8}\right) \qquad (2)$$

An Illustrative Example.—A certain California oil by chemical analysis is found to contain 81.52 per cent. carbon; 11.01 per cent. hydrogen; 0.55 per cent. sulphur; and 6.92 per cent. oxygen.

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The flue gas of a boiler under test using this oil was found by Orsat analysis to contain 8.6 per cent. carbon dioxide; 9.0 per cent. oxygen; 1.1 per cent. carbon monoxide; and 81.3 per cent. nitrogen. Let us by means of the above formula compute the air actually admitted to the furnace per pound of oil burned in the test. Before substitution we must remember that the above formula is expressed in proportions and not in percentages. Substituting then, with this in mind, we have

$$M_a = 11.52 \times \frac{0.086 + 0.0055 + 0.09}{0.086 + 0.011} \times 0.8152 + 34.56 \times \left(0.1101 - \frac{0.0692}{8}\right) = 21.1 \text{ lb.}$$

A Second Formula for Ascertaining Air Actually Admitted to the Furnace.—Let us next deduce a formula recommended by the American Society of Mechanical Engineers for the determination of the air supplied to the furnace per pound of fuel consumed.

In the deduction of the last formula it was seen that the entering oxygen combines with $(V_1 + V_3)$ units by volume of carbon in the fuel. With this entering oxygen, however, is associated V_4 units by volume of nitrogen. Hence for each unit by volume of carbon consumed in the furnace $\left(\frac{V_4}{V_1 + V_3}\right)$ units by volume of nitrogen enter with the oxygen into the furnace. Since one unit of carbon by volume weighs $^{12}/_{28}$ths of the weight of one unit by volume of nitrogen, we have that for every pound of carbon burned in the furnace

$$\frac{28}{12} \times \frac{V_4}{V_1 + V_3}$$

pounds of nitrogen enter from without. But air is 0.7685 parts by weight nitrogen. Hence if fuel oil contains C parts by weight of carbon, for every pound of fuel oil burned in the furnace, the weight of air M_a drawn in from without is

$$M_a = C \frac{28 V_4}{12(V_1 + V_3)} \times \frac{1}{0.7685}$$

$$\therefore M_a = 3.032 \left(\frac{V_4}{V_1 + V_3}\right) C \qquad (3)$$

An Illustrative Example.—Taking as an example the data set forth in illustrating the first formula deduced for ascertaining

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the air actually admitted to the furnace, we have by substituting in this second formula that

$$M_a = 3.032\left(\frac{0.813}{0.086 + 0.011}\right)0.8152 = 20.7 \text{ lb.}$$

Weight of Dry Flue Gas per Pound of Fuel.—Since the entering air above computed combines with one pound of fuel, we may ascertain the weight of flue gas per pound of fuel oil consumed by simply adding unity to the weight of air actually admitted to the furnace.

Let us, however, deduce a formula directly for this computation and check by numerical comparison the results obtained by the former methods.

To convert an analysis by volume into an analysis by weight it has been shown that if these components, carbon dioxide (CO_2), oxygen (O_2), carbon monoxide (CO), and nitrogen (N_2) are respectively V_1, V_2, V_3 and V_4 by volume, then by weight according to the formula derived in the last discussion, they will prove to be

$$\frac{m_1V_1}{C_s}, \frac{m_2V_2}{C_s}, \frac{m_3V_3}{C_s}, \text{ and } \frac{m_4V_4}{C_s},$$

wherein C_s is obtained by summing up all products formed by multiplying each component volume by its molecular weight. For every pound of carbon dioxide (CO_2) formed $^{12}\!/_{44}$ lb. of carbon are consumed in the fuel oil. Hence to form $\frac{m_1V_1}{C_s}$ lb. of carbon dioxide it is evident that $\left(\frac{12}{44} \text{ of } \frac{m_1V_1}{C_s}\right)$ lb. of carbon are consumed in the fuel oil. But m_1 for carbon dioxide (CO_2) is 44. Hence this quantity becomes $\frac{12V_1}{C_s}$.

Similarly, since each pound of carbon monoxide (CO) in its formation requires $^{12}\!/_{28}$ lb. of carbon from the fuel oil, for the formation of $\frac{m_3V_3}{C_s}$ lb. of carbon monoxide (CO), we must burn $\left(\frac{12}{28} \text{ of } \frac{m_3V_3}{C_s}\right)$ lb. of carbon. But m_3 for carbon monoxide (CO) is 28.

Hence this quantity becomes $\frac{12V_3}{C_s}$.

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The free oxygen and the free nitrogen in the flue gas have of course required no carbon of the fuel. Therefore, for M lb. of chimney gas there will be required

$$\frac{12V_1}{C_s} + \frac{12V_3}{C_s} = \frac{12}{C_s}(V_1 + V_3)$$

units of carbon; or reciprocally, one pound of carbon will, of course, form $\frac{C_s}{12(V_1 + V_3)}$ units by weight of flue gas and C parts of carbon by weight in the fuel will form M_g lb. of flue gas which may be computed from the formula

$$M_g = C\frac{C_s}{12(V_1 + V_3)}$$

But the molecular weight for carbon dioxide is 44, for oxygen it is 32, for carbon monoxide it is 28, and for nitrogen it is 28, therefore for C_s we have

$$C_s = 44V_1 + 32V_2 + 28V_3 + 28V_4$$

$$\text{or } M_g = C\frac{44V_1 + 32V_2 + 28V_3 + 28V_4}{12(V_1 + V_3)}$$

$$\therefore M_g = C\frac{11V_1 + 8V_2 + 7V_3 + 7V_4}{3(V_1 + V_3)} \qquad (4)$$

An Illustrative Example.—Let us now use the same experimental data as in the previous examples and compute the pounds of dry flue gas that were formed, as indicated by the data from the Orsat analysis.

It is to be noted in passing that all of these computations based simply upon the Orsat analysis give the weight of dry flue gas only. If the moisture present in the flue gas is to be taken into consideration, a correction should be made by noting the hydrogen present in the fuel, the moisture in the entering air, and the steam used in atomization. In ordinary practice these factors are not used, for, as we shall see in the discussion of the heat balance in a later chapter, the moisture content is properly cared for under separate headings. Returning to our example, then, we find that the weight of dry flue gas M_g per pound of fuel burned is

$$M_g = 0.8152\frac{11 \times 0.086 + 8 \times 0.09 + 7 \times 0.011 + 7 \times 0.813}{3(0.086 + 0.011)} = 20.8$$

Ratio of Air Drawn Into Furnace to that Theoretically Required. We have already derived sufficient relationships to compute the ratio of air drawn into the furnace to that theoretically required.

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Let us, however, proceed to the derivation of another formula that is recommended for use by the American Society of Mechanical Engineers in the testing of boilers.

Assuming that perfect combustion is taking place and neglecting the hydrogen content of the fuel which we know disappears in the Orsat apparatus before our analysis really begins, we have that for $\frac{m_4V_4}{C_s}$ lb. of nitrogen drawn into the furnace $\frac{m_4V_4}{C_s}\frac{1}{0.7685}$ lbs. of air must have passed in. Similarly, if no carbon monoxide is formed in the flue gas, for $\frac{m_2V_2}{C_s}$ lb. of free oxygen appearing in the flue gas, then evidently $\frac{m_2V_2}{C_s} \times \frac{1}{0.2315}$ lb. of excess air must also have been drawn into the furnace. Hence we have that the total air M_a drawn in is expressed by the formula

$$M_a = \frac{m_4V_4}{C_s}\frac{1}{0.7685}$$

While the air M_{ta} theoretically required is

$$M_{ta} = \frac{m_4V_4}{C_s}\frac{1}{0.7685} - \frac{m_2V_2}{C_s}\frac{1}{0.2315}$$

Therefore, the ratio r_a of the air actually supplied to that theoretically required may be derived as follows:

$$r_a = \frac{M_a}{M_{ta}} = \frac{\frac{m_4V_4}{C_s}\frac{1}{0.7685}}{\frac{m_4V_4}{C_s}\frac{1}{0.7685} - \frac{m_2V_2}{C_s}\frac{1}{0.2315}}$$

$$= \frac{m_4V_4\frac{1}{0.7685}}{m_4V_4\frac{1}{0.7685} - m_2V_2\frac{1}{0.2315}}$$

Since $m_4 = 28$ and $m_2 = 32$, we have

$$r_a = \frac{V_4}{V_4 - \frac{32V_2}{28} \times \frac{0.7685}{0.2315}} = \frac{V_4}{V_4 - 3.782V_2} \qquad (5)$$

An Illustrative Example.—Using the same data as employed in previous examples, we have

$$r_a = \frac{0.813}{0.813 - 3.782 \times 0.09} = 1.72$$

The air theoretically required in this example was computed on page 217 and found to be 12.92 lb. Hence for the three formulas derived we arrive at the following values for r_a:

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From formula (2) M_a was found to be 21.1.

$$\therefore r_a = \frac{21.1}{12.92} = 1.63$$

From formula (3) M_a was found to be 20.7.

$$\therefore r_a = \frac{20.7}{12.92} = 1.60$$

And from formula (4) M_g was found to be 20.8. Hence M_a is 19.8 and

$$\therefore r_a = \frac{19.8}{12.92} = 1.53$$

It is difficult to pick the most correct formula to use in any given instance. If the analyses are obtained with precision

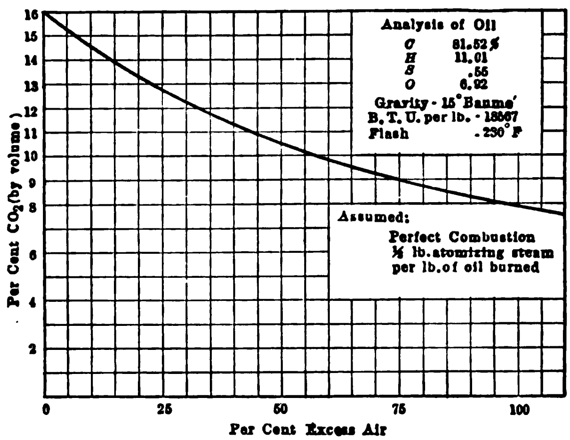

FIG. 179.—Chart showing excess air admitted for varying amounts of CO_2 in fuel oil practice.

undoubtedly results will be obtained that will check quite closely, indeed well within the degree of precision of the other factors that enter.

With the combustion data thus obtained, we are now in position to proceed to the determination of the heat balance which, as we shall see later, tells us in detail just what disposition is being made of the vast quantities of heat that are generated in the furnace due to the burning of the fuel oil.

Combustion of one Pound of Oil.—It is sometimes convenient to be able to determine in advance the maximum amount of

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CO_2 that can be expected in the flue gas when burning oil with a given quantity of excess air. In the following table three different cases of the combustion of one pound of oil are worked out, the excess air being taken at zero in the first case, 50 per cent. in the second and 100 per cent. in the third. In all three cases it is assumed that perfect combustion is obtained, thus eliminating CO from the calculations and that ½ lb. steam per pound of oil is used for atomization. It is also assumed that the fuel oil contains 82 per cent. Carbon and 11 per cent. Hydrogen, the other ingredients being neglected. The calculations do not require any explanation, as they can be easily followed, the ingredients of the air and fuel being segregated and combined in proper proportions according to chemical formulæ, and then converted from weight to volume, and to per cent. by volume as ordinarily obtained in flue gas analysis.

Combustion of One Pound of Oil

			No excess air	50% excess air	100% excess air
	Air supplied per lb. oil	Lbs.	13.21	19.81	26.42
	Oxygen supplied per lb. oil	Lbs.	3.06	4.59	6.12
	Nitrogen supplied per lb. oil	Lbs.	10.15	15.22	20.30
	Oxygen used per lb. oil	Lbs.	3.06	3.06	3.06
Gases of Combustion—Weight	Oxygen free	Lbs.	0.	1.53	3.06
	CO_2 produced	Lbs.	3.00	3.00	3.00
	H_2O from combustion	Lbs.	0.99	0.99	0.99
	H_2O from atomizing steam	Lbs.	0.5	0.5	0.5
	Nitrogen supplied	Lbs.	10.15	15.22	20.30
	Total weight of gases	Lbs.	14.64	21.24	27.85
Gases of Combustion—Volume	Oxygen—Vol. at 32°	Cu. ft.	0.	17.1	34.3
	CO_2—Vol. at 32°	Cu. ft.	24.3	24.3	24.3
	H_2O—Vol. at 32°	Cu. ft.		Negligible	
	Nitrogen—Vol. at 32°	Cu. ft.	129.2	194.1	258.5
	Total Vol. at 32°	Cu. ft.	153.5	235.5	317.1
Gases of Combustion—Per Cent.	Oxygen—Per cent. by volume		0.	7.26	10.81
	CO_2—Per cent. by volume		15.8	10.32	7.66
	Nitrogen—Per cent. by volume		84.2	82.42	81.53

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CHAPTER XXXIV

WEIGHING THE WATER AND OIL IN BOILER TESTS

There are various types of commercial water meters and water weighers on the market. Some of these are quite accurate

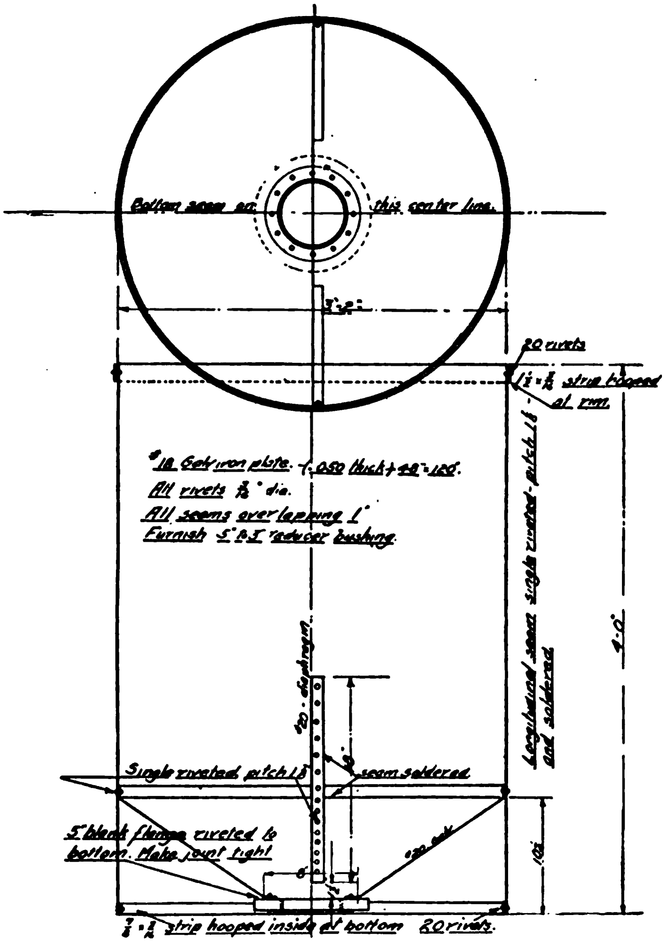

FIG. 180.—An excellent design for a measuring tank.

for certain investigations. For boiler performance, however, they are not to be recommended.

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Volumetric Method of Water Measurement.—Water may also be measured quantitatively by taking its volumetric proportions. Its weight is then computed after ascertaining its specific density. The reverse of this principle is used, for instance, in measuring the volumetric clearance of a steam engine, wherein water is poured into the cylinder ports when the piston head is at its dead end and the water afterwards drained out and weighed. From the weight of the water so used the volume of the clearance is computed. In rough measurements of engine and boiler performance the water is sometimes measured by filling a tank or barrel of known volumetric proportions, and by keeping account of the number of barrels so filled and dumped into the sump, sufficient data is obtained to compute the weight.

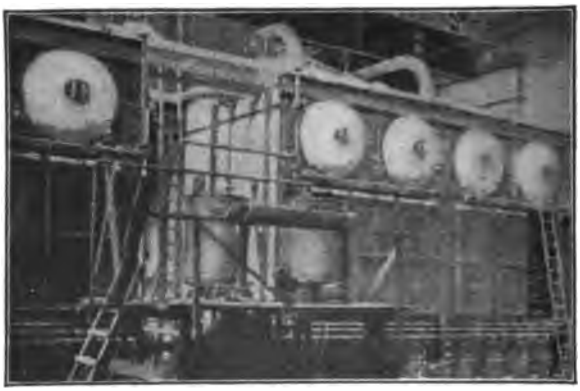

FIG. 181.—Platform scales and tanks for water measurement.

The boiler immediately to the right of the platform scales is under test. The tank below the platform scales into which the water is emptied after being weighed, is utilized to furnish all water for the boiler during the test. At the beginning of the test a hooked gage registers the height of the water in this tank, and at each hourly period thereafter sufficient water is weighed and emptied into it from the tanks above to maintain this exact level. By means of these data, properly taken, the factor of evaporation and the boiler horsepower are easily computed.

The Method of Standardized Platform Scales.—It is now universally recognized, however, that carefully weighing the water on carefully standardized scales is the only safe and reliable method of ascertaining the water fed to a boiler under test.

Let us then see how the details are arranged for the weighing of the water used in steam generation.

A large square metallic tank about 5 by 5 by 4 ft. in dimensions is usually constructed. From the bottom of this tank all feed water for steam generation in the boiler under test is drawn. At

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the beginning of the test the water level in this tank is accurately measured by means of a hook gage situated within the tank. At the end of each hourly period of the test and at the conclusion of the test this exact level is also maintained.

The control for the water supply is accomplished by two or three vertical cylindrical tanks that have a conically shaped outlet at the bottom. These tanks are located on standardized scales immediately above the main supply tank that has just been described. The complete installation is shown in the illustration. At the beginning of the test the height of the water in the boiler is noted on the gage glass in front of the boiler and as near as

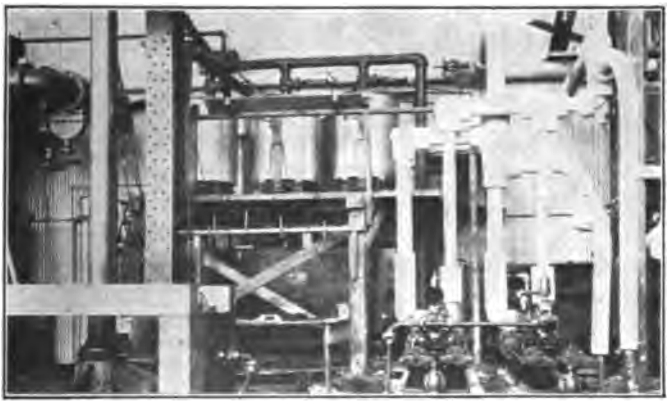

FIG. 182.—A design for a weighing tank in a boiler test.

In order to assure the rapid passage of water or oil from the tank upon the platform scales into the container below, the employment of steel tanks with conical shaped bottoms is most effective. The outlet for the oil or water should be controlled by quick-opening valves.

is possible the feed pump is regulated in its operation so as to maintain this level. At the instant of conclusion the water level is most carefully adjusted to meet the condition of boiler water level prevailing at the beginning of the test.

As the water is drawn from the feed tanks beneath the platform scales the operators fill the tanks on the scales above and note the weight before and after emptying their contents into the tank below. Thus with ease the water surface in the tank below may be kept at the constant hook gage reading desired, and the net weight of water fed to the boiler ascertained at any time during the test.

The improvised desk boards shown in the illustration assist

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materially in aiding the water weighing operators to perform their task with ease and without confusion.

In order to prevent wastes and leakages of water, it is well to disconnect the outlets from the blowoff pipes of the boiler during the period of the test. All outlets from the water columns and gage glasses should also be carefully watched.

The Weighing of the Oil.—For the careful weighing of the oil fed to the furnace a similar device is constructed as in the case

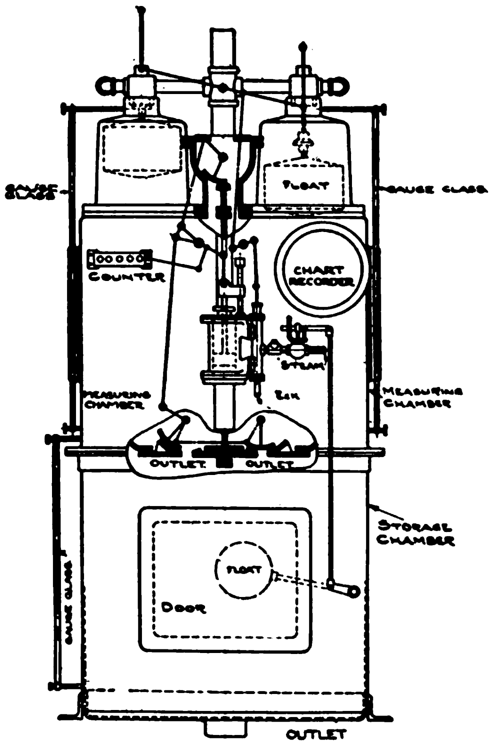

Fig. 183.—An excellent water measurer.

While the automatic water measurer is not as accurate as the standardised scale method, still it finds many applications in the testing laboratory.

of the water determination. A metallic tank is constructed from which the oil supply is pumped to the furnace through the oil heater. The oil pump is best fitted with a governor and an automatic relief valve. By this means a constant pressure may be maintained on the oil line to the burners. The discharge from the relief valve is led back to the tank from which the supply to

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the pump is taken. Within the tank is situated a hook gage, the reading of which is carefully ascertained at the instant of the beginning of the test. This exact reading is maintained throughout the hourly progress of the test, and indeed at any other period if so desired.

This is accomplished by means of a tank situated above the main supply tank. This tank is installed on standardized scales. Previous to the discharge of the oil into the tank below, the scales are read and when the oil is brought to the proper hook gage reading in the tank below, the scales are again read. By subtracting these two readings, the net oil supply is ascertained.

Sampling the Oil Supply.—As the fuel is poured into the tank upon the standardized scales, a dipperful of the oil is set aside in a convenient receptacle. After a sample has thus been obtained from each tank, as it is weighed, the entire quantity is then thoroughly mixed. Three parts of this mixture are then put into separate cans and sealed. One part is analyzed by the party or company for whom the test is being performed, the second is analyzed by a disinterested party, and the third is retained in case of disagreement.

General Sampling of Fuel Oil for Purchase.—The question of determining a proper sample for commercial valuation of oil is one of patient care. The United States Bureau of Mines has evolved careful instructions to accomplish this in their technical paper No. 3, from which the following is largely an excerpt:

The accuracy of the sampling and, in turn, the value of the analysis must necessarily depend on the integrity, alertness and ability of the person who does the sampling. No matter how honest the sampler may be, if he lacks alertness and sampling ability, he may easily make errors that will vitiate all subsequent work and render the results of tests and analyses utterly misleading. A sampler must be always on the alert for sand, water and foreign matter. He should note any circumstances that appear suspicious, and should submit a critical report on them, together with samples of the questioned oil.

Sampling With a Dipper.—Immediately after the oil begins to flow from the wagon to the receiving tank, a small dipper holding any definite quantity, say 0.5 liter (about 1 pint), is filled from the stream of oil. Similar samples are taken at equal intervals of time from the beginning to the end of the flow—a dozen or more dipperfuls in all. These samples are poured into a clean drum

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and well shaken. If the oil is heavy, the dipperfuls of oil may be poured into a clean pail, and thoroughly stirred. For a complete analysis the final sample should contain at least 4 liters (about 1 gallon). This sample should be poured into a clean can, soldered tight and forwarded to the laboratory.

It is important that the dipper be filled with oil at uniform intervals of time, and that the dipper be always filled to the same level. The total quantity of oil taken should represent a definite quantity of oil delivered and the relation of the sample to the delivery should be always stated, for instance: "1-gallon sample representing 1 wagon-load of 20 barrels."

Fig. 184.—The viscosimeter.

The design of this viscosimeter is based upon a thorough knowledge of lubricating oils and of the requirements of manufacture and trade. It is made to meet all demands as a measure of viscosity, and is without the many objections that may be made to all other devices for this purpose. The viscosity of any oil is shown by the number of seconds required for a certain number of cubic centimeters to run through the open faucet. This corresponds to the most generally approved standard now in use by the largest refiners. (See page 128.)

Continuous Sampling.—Instead of taking samples with a dipper, it may be more convenient to take a continuous sample. This may be taken by allowing the oil to flow at a constant and uninterrupted rate from a ½-inch cock on the underside of the delivery pipe during the entire time of discharge. The continuous sample should be thoroughly mixed in a clean drum or pail, and at least 4 liters (about 1 gallon) of it forwarded for analysis. A careful examination should be made for water, and if the first dipperful shows water this dipperful should be

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thrown into the receiving tank and not mixed with the sample for analysis.

Mixed Samples.—The all-important point is that the gross sample, whatever the manner of sampling, shall be made up of equivalent portions of oil taken at regular intervals of time, so that the sample finally forwarded for analysis will truly represent the entire shipment.

Water or earthy matter settles on standing. Hence, before a large stationary tank or a reservoir is sampled, the character of the contents at the bottom should be ascertained by dredging with a long-handled dipper, and the contents of the dipper should be examined critically. If a considerable quantity of sediment is brought up, it should be cause for rejecting the oil.

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CHAPTER XXXV

MEASUREMENT OF STEAM USED IN ATOMIZATION

As has been previously set forth, there are three methods used in pulverizing or atomizing the fuel oil in the industries for heat generating purposes, namely: by compressed air, by steam, and by some mechanical operation.

In any one of these instances the actual expenditure of energy necessary to accomplish this result when converted into heat units should be charged as a loss in furnace operation, when the efficiency of the boiler as a whole is being determined. And if this energy is taken from the steam that is being generated in the boiler, then the net steam energy should be computed by subtracting from the gross production such steam as may be used in atomization.

It then becomes the task of the steam engineer to construct some accurate and convenient apparatus whereby this may be easily and accurately accomplished.

There are steam meters on the market that may be utilized for this purpose, and if a careful design is picked, the measurement may be relied upon. Many engineers, however, prefer the use of a standardized orifice or the construction of an apparatus of their own whereby this important data may be ascertained with accuracy.

Mathematical Expression for Flow of Steam.—In the mathematical considerations involved in establishing a formula for steam flow through orifices, a rather unique incident is encountered. When the pressure of the lower medium into which the steam empties itself is less than 58 per cent. of the higher pressure, a certain formula applies. And the rather remarkable thing is that below this point the flow is neither increased nor decreased by a reduction of the external pressure, even to the extent of a perfect vacuum. This was the basis upon which Napier's formula was derived in the chapter on Steam Calorimetry, wherein a formula was given to compute the steam utilized for operating the calorimeter. In this formula it was

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seen that, if W is the weight of the steam in pounds per second flowing into the atmosphere, p the absolute pressure in pounds per square inch in the steam main, and a the area of orifice in square inches, we have

$$W = \frac{pa}{70} \tag{1}$$

FIG. 185.—Steam flow meter and draft gage on the left with Venturi meter on the right. This apparatus is at the Redondo Steam Plant of the Southern California Edison Company.

For steam flowing through an orifice from a higher to a lower pressure where the lower pressure is greater than 58 per cent. of the higher, we have the formula

$$W = 1.9\ AK\sqrt{(P-d)d} \tag{2}$$

wherein W is the weight of steam as discharged in pounds per minute, A the area of orifice in square inches, P the absolute initial pressure in pounds per square inch, d the difference in pressure between the two sides in pounds per square inch, and K

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is a constant which has a value of 0.93 for a short pipe and 0.63 for a hole in a thin plate or a safety valve.

This latter formula is applicable in the measurement of steam to burner utilized in the atomization of fuel oil. In the following lines a method will be outlined setting forth the necessary apparatus involved in determining the variables in the formula. Instead of actually substituting and solving numerically, however, it is far simpler to construct a chart and pick from this the

FIG. 186.—Steam flow meter with integrating device for registering total quantity of steam passed.

steam consumption for any given steam pressure and pressure difference in an orifice placed in the main.

Here then is presented a ready and accurate means of steam measurement for atomization purposes. A diaphragm with an orifice opening of 0.5 of a sq. in. in area is inserted in the steam line. On both sides of this diaphragm are drilled holes which are tapped for a ¼-inch pipe. The pipes are then connected to both legs of a manometer filled with mercury. A manometer is

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nothing more nor less than a U-tube filled with mercury. When these two ends are connected with pipes of varying pressures, the mercury in the U-tube will of course be raised to a higher point in one leg of the U-tube than in the other. The difference in this height represents in inches of mercury the difference in pressure between the two sides of the diaphragm. If now a steam gauge be inserted in the steam main on the boiler side of the diaphragm, we are enabled by means of the atmospheric barometer reading to express these pressures in absolute pressure units as set forth in the chapter on pressures. On the burner

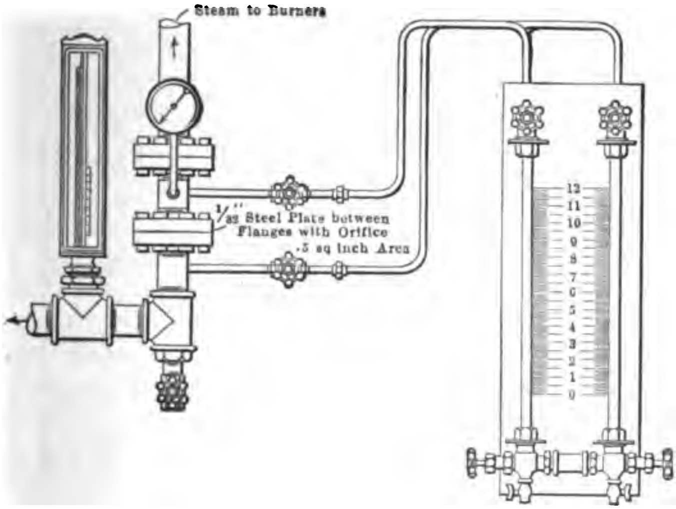

Fig. 187.—Apparatus employed in measuring steam in atomization.

The flow of steam through an orifice wherein a slightly lower pressure is maintained on the further side of the orifice, is found experimentally to be proportional to the difference in mercury heights indicated on the manometer shown on the right in the illustration. By calibrating these readings prior to a test the steam used in atomization may be conveniently and readily determined during a test.

side of the steam main a thermometer is inserted as shown in order to measure the temperature of the steam fed to the furnace, as this steam in many instances is superheated and hence the pressure reading does not indicate the temperature existing.

A manometer is accurately calibrated prior to the test by allowing the steam to be discharged into a barrel for a period of time under varying manometer readings. A curve is then plotted similar to the one shown in the illustration, which sets forth the pounds of steam passing per minute for any particular manometer reading in inches of mercury. If, then, readings are taken

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every fifteen minutes during the test, the testing engineer notes at such intervals the steam that has passed during the preceding fifteen-minute period. In such a manner the total quantity of steam used in atomization is ascertained.

Thus in a test at the Fruitvale Station of the Southern Pacific Company, the pressure of the steam at the burner was found to be 168 lb. per sq. in. The temperature of the steam at the burner was 440°F., which indicated a superheated condition of 65°F. The total steam used by the burners for a ten-hour test

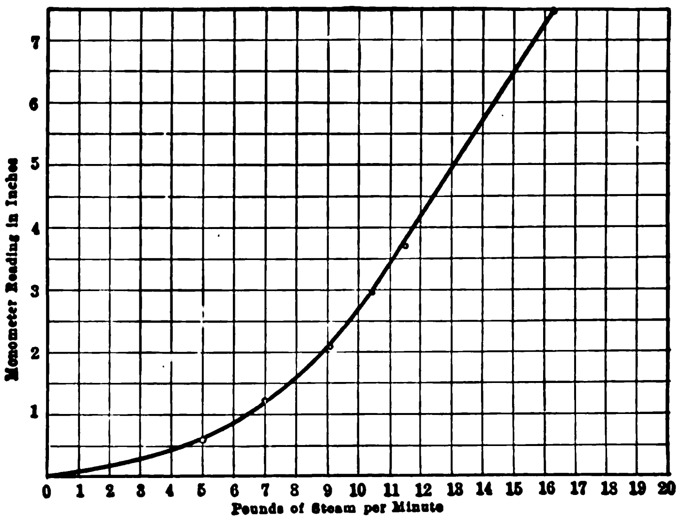

FIG. 188.—Calibration of orifice for measurement of steam used in atomization.

Previous to a boiler test the manometer which registers the pressure difference at the faces of the orifice is carefully calibrated by condensing the steam flow and weighing the hourly condensate. These data when plotted on a curve as shown above enable the engineer to quickly ascertain the steam used in atomization at any time during a test.

was found by the above means to be 7441 lb., while the total weight of water fed to the boilers proved to be 180,240 lb. Hence the percentage of total water evaporated by the boilers used in atomization is determined by dividing 7441 by 180,240, which is 4.16 per cent.

The total weight of oil fired was 14,093 lb. during the test of 10 hr. Hence, the pounds of steam utilized for atomization per pound of oil fired is obtained by dividing 7441 by 14,093, which proves to be 0.528 lb.

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CHAPTER XXXVI

THE TAKING OF BOILER TEST DATA

In previous chapters we have touched upon all the important points involved in tests on boiler economy. These, however, have been considered under separate headings and of necessity in a somewhat disconnected manner. In this and the succeeding chapters, we shall endeavor to link these items into a connected unit. This chapter will be concerned with the gathering of the data and the next with its computation.

The Object.—"What can you do?" applies equally well to the rating of inanimate objects as well as to the accomplish-

FIG. 189.—A portable pyrometer outfit.

For the ready measurement of temperatures in and about the power plant, a portable type of pyrometer is often convenient. In the illustration shown temperatures may be read from 200°F. to 2200°F. Such an instrument as the one indicated is convenient in ascertaining the flue gas temperatures when the Orsat analysis is being taken.

ment of human endeavor. And so the object of boiler testing is to try out the latent steaming qualities of the boiler and test its strength both for sudden calls and for endurance. The manner in which the mechanical design of the boiler can withstand such tests and especially the efficiency with which it can perform its function of transforming the heat energy of the fuel into energy latent in the steam sent forth from the boiler are as a rule the factors that either add lustre to the name of the manufacturer or else relegate the type of steam generator under test to the scrap heap.

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The Instruction for Boiler Tests.—The minute details that should be satisfied in order to secure accurate data wherewith to rate the boiler and scientifically set forth its commercial worth are elaborately set forth in instructions issued by the American

FIG. 190.—Saybolt water indicator, a device whereby the water contents of oil may be conveniently indicated.

FIG. 191.—An oil sampler, a device which may be easily lowered into storage tanks of oil and a sample drawn from any particular strata in the tank.

Society of Mechanical Engineers, compiled by their Committee on Power Tests. In any case of actual test, the steam engineer should be provided with a copy of these instructions, which he

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can secure from the secretary of the society by the payment of a small fee.

Since these instructions require many pages wherein to set forth the details of a test, it cannot, of course, be expected that anything beyond a general outline of procedure in boiller testing be set forth in this article. Still it has been the experience of the authors that if the steam engineer gets a thorough picture of the test details as a whole he is well equipped, with the assistance of a nearby copy of the detailed instructions, to properly understand the procedure.

The Test for Efficiency Under Normal Rating.—It has been seen in the chapter on Rating of Boilers that the manufacturer or builder rates the output of the boiler on the basis of the boiler heating surface presented to the furnace gases. For each 10 sq. ft. of boiler surface so exposed to the furnace gases, the boiler is said to have one boiler horsepower. A test for boiler efficiency under this normal condition of operation is one of the most important to be ascertained in boiler performance. In order to accomplish this result, the steam engineer usually computes the total weight of water the boiler would approximately have to evaporate into steam per hour under the conditions of entering feed-water temperature, boiler pressure, and quality of steam generated to satisfy the builder's rating. Having made a careful estimate of this quantity he then proceeds to operate the boiler as nearly as possible to meet this condition.

Time of Duration of Test.—The generation of steam is maintained as uniformly as possible over a period of from 8 to 10 hrs.

The Beginning and Stopping of a Test.—At the beginning of the test the level of water in the boiler is noted on the water glass and at the completion the water is brought to the same height.

In the testing of boilers fired by fuel oil, the boiler is brought up to and continued at normal operating conditions until the furnace wall and boiler room temperatures are at their normal reading. Then the test is started by feeding weighed water and fuel oil. At the end of the test, all conditions of pressure, temperature and rate of steam generation should be as nearly as possible the same as at the beginning.

The Weighing of the Water.—Several tanks are placed upon carefully calibrated scales and all water entering the boiler from the instant the test starts to its closing point is carefully weighed.

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The details of the methods involved in the weighing of water have appeared in a previous chapter.

The Heat Represented in the Steam Generated.—The temperature of the entering water and the pressure of the steam generated are noted at frequent intervals. The quality of the steam as to whether it is wet, dry saturated, or superheated, is also carefully determined quantitatively by methods outlined in previous chapters.

With these data at hand the steam engineer is enabled by deductions to be set forth in the chapter on Heat Balance to compute the actual heat energy absorbed by the entering water in the production of steam.

The Oil, Its Measurement and Analysis.—At the same time that the steam generating functions of the boilers are being ascertained, it is of course necessary to weigh the fuel oil admitted to the furnace for firing purposes and to draw frequent samples for the composite sample to be used in ascertaining the heat producing value of one pound of fuel. The method of weighing the oil and drawing the oil sample has been set forth in a previous chapter.

Having determined the calorific value of one pound of fuel by methods previously described the total heat put into the furnace by the fuel during the test is computed.

In former chapters are to be found discussions which fully set forth the methods utilized in determining from the oil sample its calorific value, its moisture content, and its gravity under standard conditions which are necessary to compute the total heat producing value of the oil used in firing the boiler under test.

The Steam Used in Atomization.—In most central station practice wherein fuel oil is consumed for heat generation, the atomization of the fuel oil is accomplished by blowing into the furnace through the oil burner a certain quantity of steam that is being generated in the boiler. To obtain the useful and economic quantity of steam generated by the boiler we should then subtract this steam used in atomization from the total steam generated in the test. A practical method of obtaining experimentally the steam used in atomization has been described in Chapter XXXV.

The Boiler Efficiency.—Having thus obtained the net heat absorbed by the boiler under test and also the heat given out by

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the fuel oil sprayed into the furnace, the ratio of the former to the latter gives us the efficiency of the boiler as set forth in the chapter on Heat Balance.

In central station practice on the Pacific Coast the gross boiler efficiency in the best installations ranges from 81 to 83 per cent. under test conditions. The atomization of the steam lowers this efficiency by about 2 per cent., thus making the best net boiler efficiencies range between 79 and 81 per cent.

The Overload Test.—The sudden demand for power during certain hours of the day make an elasticity in boiler steaming qualities absolutely imperative. Otherwise, a great additional expense would be involved in the cost and installation of additional steaming units. Hence the overload steaming qualities of a boiler are of utmost importance, especially in central station or steam auxiliary practice.

As an instance of performance of a boiler under overload conditions on the Pacific Coast, an authentic case is on record where a boiler of 773 rated horsepower developed an overload of 75.7 per cent. for 5 hours and still maintained a gross efficiency of 80.62 per cent.

The Quick Steaming Test.—In other instances the ability of a boiler to hastily get into action is of prime importance. This is especially true in cases where boilers are held in readiness for pumping station operation for fire protection. In San Francisco, California, for instance, is located a high-pressure water system whereby pumps stand eternally ready to deliver 12,000 gal. of water per minute to a height of 700 ft. should disaster by fire ever again visit that municipality. The boilers that operate the pumping station have by test demonstrated that full boiler pressure and steaming conditions can be accomplished in less than thirty minutes time.

Again, other features of test are under special cases desirable to attain. But the two most important tests are those of ascertaining the conversion ratio of heat represented in the steam to the heat supplied by the furnace under normal conditions of operation and under certain definite overload guarantees—in a word, the ascertaining of boiler efficiency for normal rating and for conditions of overload.

Observations Necessary.—A complete tabulated list for final test computation is set forth in the book of instructions previously mentioned as approved or advised by the American Society

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of Mechanical Engineers. Let us now look into some of the details necessary to obtain this recorded data.

In the first place, one should note on a log sheet the general observations such as date of test, duration of test, type of oil burner, make of oil burner, number of burners used, and with this information should be compiled sufficient physical dimensions of the boiler to enable one to compute the builder's rating both for the boiler and for the superheater. An illustration of this computation was set forth under the chapter on Rating of Boilers.

During the test period, observations are usually taken every fifteen minutes, simultaneously if possible.

Pressure Readings.—The pressure of the atmosphere is read in inches of mercury and the steam gauge readings of the boiler and superheater having been duly calibrated or corrected for mechanical inaccuracies, are then reduced to absolute pressure readings as set forth in the chapter on pressures.

The pressure of the oil under which it is forced into the furnace is also usually noted, although it has no bearing on data computation.

The pressure of the draft at various parts of the ash pit, furnace, breeching, and chimney are also noted by means of a multiple cock arrangement shown in Fig. 122. This arrangement makes possible the quick ascertaining of various draft readings by means of one instrument.

The pressure of the saturated steam and also that of the superheated steam is ascertained by inserting carefully calibrated steam gages, the one in the saturated steam compartment and the other in the superheater compartment. These pressures are then converted into absolute pressure readings by correcting for atmospheric pressure as set forth in Chapter III.

Temperature Readings.—A thermometer is usually located in the atmosphere without to ascertain general external temperature conditions of the day. One is also placed in the boiler room to ascertain the temperature of the entering air passing into the furnace.

To ascertain the temperature of entering feed waster and fuel oil, thermometer wells with thermometers are also installed at nearby points of entrance.

It is often desirable to ascertain the temperature of the furnace gases at various points in their journey. To accomplish this thermo-couples are installed at the points desired previous to

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the firing of the boilers and during the test an electrical pyrometer is advised, especially if other high temperatures are to be taken in various points of flue gas passage.

The Flue Gas Analysis.—Simultaneously with the taking of the temperatures, pressure and other readings of the test, the flue gas analysis is ascertained at frequent intervals. The detailed method of taking these data has been fully set forth in previous chapters and methods of computation of combustion data explained. The Heat Balance will be set forth in full in a later chapter.

The Test as a Whole.—The reader has now before him the taking of the test as a whole. At this point he should carefully review all the previous chapters alluded to in this discussion so as to weld into a solid chain the links that go to make up the boiler test in fuel oil practice.

Having thus in mind a complete idea of the various details involved in the taking of the boiler test data, we are now in position to link together the computed data involved in formulating the engineer's report of the economic results of the test.

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CHAPTER XXXII

PRELIMINARY TABULATION AND CALCULATION OF TEST DATA

Fig. 192.

THE systematic construction of a log sheet that will show in the minutest detail every incident in the progress of the boiler test is of prime importance. It is far better to overdo then to underdo in the gathering of detail data of this kind. The notation of remarks from time to time upon the log sheet concerning relevant observations during the progress of the test is of much service to the engineer when he finally comes to decide fine points in economic boiler performance.

No straight and narrow schedule or log sheet can be set forth to meet all types of boiler test. Each particular test as a rule involves its own particular tabulation. Let us, however, consider a series of tabulation sheets for boiler tests in which oil is used as a fuel. The suggestions that will be set forth illustrate a carefully evolutionized scheme of tabulation for such data that may be well followed in guiding one in the construction of his own individual log form should occasion arise.

The Log Sheet for Weighing the Water.—In the previous chapter we have seen that the water is brought to a definite height in the supply tank the instant of starting the test. Above this supply tank are located standardized scales upon which the water is weighed before emptying into the supply tank below. As a rule, at the closing of each hourly period, water readings are computed in order that the engineer may get a preliminary idea of the progress of the test. Blank sheets are given each water

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weigher, one to be used for each hourly period. Each sheet sets forth general information indicating the kind of boiler under test, the date of test, the name of the observer, and the particular tank at which each is stationed. A column is devoted to the number of the scale reading, a second to the gross weight of the water and tank before emptying into the tank below, the tare to be subtracted from the gross weight, which is the weight of the upper tank after the water is emptied into the tank below, and a fourth column setting forth the net weight or difference of the two preceding columns. This sheet will have somewhat the following appearance.

Log Sheet for Water Fed to Boiler

Kind of boiler...

Method of Starting Test..

Date...

Observers at Scales for Water..................................

Reading	Time	Gross	Tare	Net	Temp. of water	Remarks
1.						
2.						
3.						
4.						
5.						
6.						
7.						
8.						

Total——— ——— ———Signature;

Fig. 193.

By using the type of log sheet above indicated, it is evident that the engineer has a check on his water computation, for in the line marked "total" the footing for the gross weight should exactly equal that for the sum of the tare weight and the net weight.

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A place is also given for a signature to be appended by the one responsible for the weight notation.

Log Sheet for the Fuel Oil Fed to Furnace.—Simultaneously with the weighing of the water, a similar log sheet is kept by another set of observers setting forth the weight of fuel oil fed to the furnace. As the weighing proceeds, a periodic sample is taken to make up a composite sample for the detemination of the calorific value of the oil as set forth in the preceding chapter. The log sheet for the oil is quite similar to that used for the water and should be footed up at the end of each hourly period so that the engineer may have some definite idea of preliminary economic results. A suggestion for this log sheet is as follows:

LOG SHEET FOR OIL FED TO FURNACE

Type and Location of Boiler. . . . .
Type of Burner. . . . .
Type of Furnace. . . . .
Date. . . . .
Observers at Scales for Oil. . . . .

Reading	Time	Gross	Tare	Net	Temp. of oil	Remarks
1.						
2.						
3.						
4.						
5.						
6.						
7.						
8.						

FIG. 194.

Other Data to be Taken.—The tabulation of data to determine the steam used for atomization and the analysis of the flue gases require special treatment, depending upon the particular method decided upon by the engineer to ascertain these factors. Pre-

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THE GENERAL BOILER TEST LOG SHEET

General		Pressures						Temperatures									Water			Oil			Remarks
Reading	Time	Atmospheric barometer	Steam pressure saturated (gage)	Steam pressure superheated (gage)	Oil pressure at burner	Force of draft in in flue	Force of draft in in ashpit	External air	Fire room	Air entering ashpit	Furnace 6 ft. from burner	Temp. of stack	Temp. of boiler at any other pt. desired	Oil at burner	Feed water entering boiler	Superheated steam	Gross	Tare	Net	Gross	Tare	Net	
1																							
2																							
3																							
11																							
12																							
Totals																							
A vera	ge																						

Signature of Observer.

FIG. 195.

vious chapters have already set forth in detail suggestions for the ascertaining of these quantities, and the reader is now advised to re-read them in order to correlate in his mind, as it were, all the data, that must be taken in order to ascertain the economic performance of the modern boiler utilizing crude petroleum as a fuel.

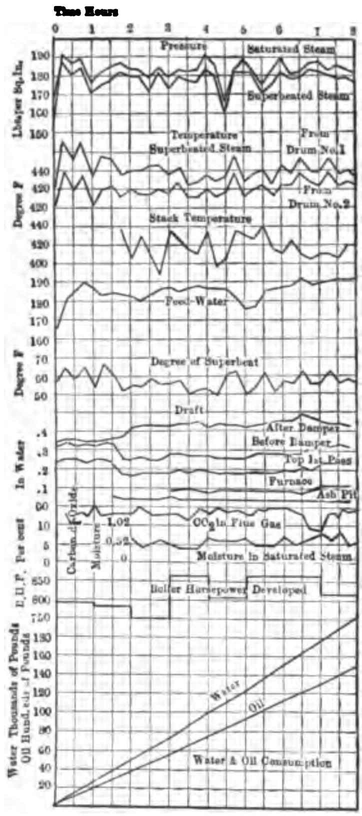

FIG. 196.—The graphic log sheet for fuel oil tests.

During the progress of a test a graphic plot most conveniently sets forth the behavior of the variables under observation. The above shows a typical graphic log sheet and its method of construction for fuel oil tests.

The General Log Sheet.—In addition to the two log sheets just described, a general log sheet is necessary upon which to note the temperatures, pressures, flue gas analysis and other information desired.

Here is an illustration of a suggestion for such a log sheet. At the completion of the test an average is easily obtained for

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the various readings by footing up the total and dividing by the number of readings noted. The columns for the water fed to the boiler and the oil fed to the furnace are footed and as in the hourly sheets previously described, the totals from these sheets which are noted on this general log sheet should now check—that is, the total gross should equal the sum of the total tare and total net columns. The reader is to bear in mind that the actual notations to be made in any particular test are not all set down in this general log sheet suggestion, for the information desired and the purpose of the test must in each given case determine these factors. The sheet will, however, serve as a general guide for such matters.

The Plotting of Test Data.—As the test proceeds hour by hour, it is very instructive and helpful to keep a diagrammatic log sheet also. By this means a glance will often reveal certain irregularities that may be righted at their incipiency. Such a log sheet is shown in Fig. 196 and by reference to it the reader will observe how the history of a test may be simply and clearly set forth.

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CHAPTER XXXVIII

THE HEAT BALANCE AND BOILER EFFICIENCY

The steaming qualities of a boiler are best set forth by measuring its so-called efficiency. The efficiency of a boiler is the relationship between the heat absorbed per pound of fuel fired and the calorific value of 1 lb. of fuel. Thus although each pound of fuel consumed in steam production is found to have a calorific value of 19,450 B.t.u. in the numerical illustration for this chapter, that portion alone of this heat which is actually represented in the steam itself is of economic value.

In the illustrative test which is made use of throughout this chapter, it will be found that of this 19,450 B.t.u. represented in each pound of oil only 14,076.56 go toward power generation. It is then useful and instructive to analyze the losses in a boiler and see through what channels this heat has been dissipated. The major portion of these losses may be easily computed by means of data taken in the test. Those which cannot be mathematically computed are thrown under the column entitled "Stray Losses," and are made to represent such an amount that the total losses together with the useful heat generated in the boiler represent the heat from 1 lb. of fuel.

Let us then examine the various channels of heat transfer going on in the boiler and see how the details of the heat balance are set forth. In this discussion H_o will represent the calorific value of 1 lb. of fuel oil under test.

(*a*) a. **The Total Heat Absorbed by the Boiler.**—As has been previously shown, the equivalent evaporation of a boiler per pound of oil represents the number of pounds of water which would be evaporated into steam per pound of oil if the water was at 212°F. and under atmospheric pressure, and this water then converted into dry saturated steam at the same temperature and pressure. It is self-evident then that the total heat absorbed by the boiler for each pound of oil burned in the furnace is equal to the equivalent evaporation multiplied by the heat necessary to convert 1 lb. of water into steam under conditions just mentioned.

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This quantity of heat has been found by Marks and Davis to be 970.4 B.t.u., as set forth in previous discussions. Representing this in a formula the total heat H_t absorbed by the boiler per pound of fuel is

$$H_t = M_e \times L_e \qquad (1)$$

in which H_t is the total heat absorbed by the boiler per pound of dry fuel, M_e the equivalent evaporation per pound of oil, and L_e the latent heat of evaporation at 212°F., which is 970.4 B.t.u. Hence, if the equivalent evaporation of a boiler is found by test to be 28,225 lb. of water per hour, and if the measurement of oil shows that 1872 lb. of oil have been consumed

$$M_e = \frac{28{,}225}{1{,}872}$$

$$\therefore H_t = \frac{28{,}225}{1{,}872} \times 970.4 = 14{,}639$$

b. **Heat Absorbed by Boiler for Atomization.**—In ordinary practice of fuel oil combustion, there are three methods of atomization employed. In the larger power plants the use of steam for atomization purposes, or in other words, the diverting of steam from the boiler into the furnace in order to atomize the oils, seems to have by far the preference. It is proposed to alter the rules of the American Society of Mechanical Engineers so that the heat represented by the steam used in atomization must be subtracted from the total heat absorbed by the boiler in order to compute the net evaporative power of the boiler. Hence to make this computation we must know the number of pounds of steam used in atomization per pound of oil burned. Methods of arriving at this result have been described in Chapter XXXV.

Calling M_s the pounds of steam used in atomization per pound of fuel burned, H_s the total heat per pound of steam so used, and h_1 the heat in the entering feed water, and H_a the heat absorbed by the boiler per pound of fuel in atomizing the oil, it is evident that

$$H_a = M_s(H_s - h_1) \qquad (2)$$

Thus it has been found in the test under description that 0.530 lb. of steam were utilized in atomization per pound of oil. Saturated steam at a temperature of 381.9° was used. From the steam tables such steam is found to have a total heat of 1198.08

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B.t.u. The entering feed water was at a temperature of 169.1°F. and has a heat of liquid amounting to 136.87 B.t.u. We find by substitution that the heat absorbed in atomizing the oil is computed as follows:

$$H_a = 0.530\,(1198.98 - 136.87) = 562.44 \text{ B.t.u.}$$

c. **Net Heat Absorbed by Boiler for Power Generation.**—Since then the heat utilized in atomization must be subtracted from the total heat absorbed by the boiler, to ascertain the net heat H_n absorbed by the boiler for power generation, we have the following formula:

$$H_n = H_t - H_a \qquad (3)$$
$$\therefore H_n = 14{,}639 - 562.44 = 14{,}076.56 \text{ B.t.u.}$$

(*b*) **Loss Due to Water in the Fuel.**—All fuels contain a certain amount of moisture. It is evident that since it requires considerable heat to convert this moisture into steam and then to send it forth from the chimney in a superheated condition, a definite loss is thereby sustained in boiler operation. This moisture must first be raised to 212°F., then converted into steam, and then heated to the temperature of the outgoing chimney gases. If we let M_w be the proportion by weight of moisture in the 1 lb. of fuel, t_o the temperature of the oil entering the burner, t_g the temperature of the escaping gases, and H_m the loss due to moisture in the fuel per pound of fuel burned, we may write at once an equation representing this loss.

Thus

$$H_m = M_w\,[212 - t_o + 970.4 + 0.47\,(t_g - 212)] \qquad (4)$$

The reasons for this formula are seen by inspection. To raise each pound of moisture from t_o to 212° F. would require as many B.t.u. as the raise in temperature, in other words $(212 - t_o)$ B.t.u. Again, to evaporate each pound would require 970.4 B.t.u., and as 0.47 of a B.t.u. are required to superheat 1 lb. of steam 1° in temperature at atmospheric pressure, each pound of steam superheated to the temperature of the outgoing chimney gases would require $0.47\,(t_g - 212)$ B.t.u. Therefore, the total heat required for M_w pounds would be as indicated in the formula above by summing up these separate components.

Thus in the test under consideration, let us assume that the fuel contains 1 per cent. of moisture; that its entering temperature

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is 96°F., and that the temperature of the escaping gases is 400°F. Hence

$$H_m = 0.01\,[212 - 96 + 970.4 + 0.47\,(400 - 212)] = 11.67 \text{ B.t.u.}$$

(*c*) **Loss Due to Water Formed by Burning Hydrogen.**—In the chapter on chimney gas analysis, it was seen that the Orsat Apparatus is so constructed that the vapor or superheated steam formed by the burning of the hydrogen content in the fuel is condensed into water upon entering the burette; hence the Orsat analysis indicates only dry flue gases and takes no account of the percentage of steam actually present in these gases. It is seen then that the moisture formed by the burning of hydrogen must also create a loss as it journeys upward through the boiler. Assuming H_h to be the heat lost due to the moisture formed by the burning of hydrogen by following identically similar processes of reasoning just employed in the considerations of the loss due to the moisture in the fuel, we find that each pound of moisture formed by the burning of hydrogen requires

$$[212 - t_o + 970.4 + 0.47\,(t_g - 212)] \text{ B.t.u.}$$

From the principles of chemistry each pound of hydrogen combines with 8 lb. of oxygen, thereby forming 9 lb. of water or steam. This relationship gives us a ready means of computing the weight of water vapor formed by the burning of hydrogen, although the Orsat analysis failed to do so. Assuming M_h to be the proportion by weight of hydrogen per pound of fuel oil burned, we have

$$H_h = 9M_h\,[212 - t_o + 970.4 + 0.47\,(t_g - 212)] \qquad (5)$$

By referring to the test data, we find that the fuel analysis shows 0.11 lb. of hydrogen per pound of fuel, that the temperature of entering air is 84° and the temperature of the escaping gases 400°, therefore

$$H_h = 9 \times 0.11\,[212 - 84 + 970.4 + 0.47\,(400 - 212)] = 1166.97 \text{ B.t.u.}$$

(*d*) **Loss Due to Heat Carried away by Dry Gases.**—From the Orsat analysis, as was seen in Chapter XXXIII on the Computation of Combustion Data, the pounds of dry gas passing up the chimney per pound of fuel burned may be easily computed by means of several different formulas. It is found by experiment

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that it requires 0.24 B.t.u. to raise one pound of chimney gas 1° in temperature. Hence if M_g be the pounds of dry chimney gas per pound of fuel, the total heat wasted H_g in raising the temperature of these dry gases is seen to be

$$H_g = 0.24\,(t_g - t_a)M_g \tag{6}$$

In this particular instance, let us assume that by the application of our formula we find that 19.83 lb. of dry chimney gas are formed per pound of fuel burned; that the temperature of the entering air is 84°, and that of the outgoing chimney gases 400°. Hence

$$H_g = 0.24\,(400 - 84)\,19.83 = 1503.91 \text{ B.t.u.}$$

(*e*) **Loss Due to Carbon Monoxide.**—In the burning of every pound of carbon to carbon dioxide, 14,600 B.t.u. are liberated. When the carbon is not completely burned but passes up the chimney in the form of carbon monoxide only 4450 B.t.u. per lb. of carbon so burned are liberated. Hence whenever carbon monoxide appears in the gas analysis it is evident that a definite loss is being sustained due to this incomplete combustion of the carbon.

For every pound of carbon which passes up the chimney as carbon monoxide, a net loss of 10,150 B.t.u. are thus uselessly thrown away. Let us assume that 1 lb. of carbon volumetrically produces V_1 units by volume of carbon dioxide and V_3 units by volume of carbon monoxide. If this is true it is evident that in every pound of carbon so burned $\frac{V_3}{V_1 + V_3}$ pounds are converted into carbon monoxide, which represents a loss of 10,150 B.t.u. per pound. Hence if there are C units of carbon by weight in each pound of the fuel, the formula to be applied to ascertain the loss due to incomplete combustion H_c is

$$H_c = C\frac{V_3}{V_1 + V_3} \times 10{,}150 \tag{7}$$

In the particular case cited above the fuel has 0.86 proportions by weight of carbon and 0.01 proportions by volume go out of the chimney in the form of carbon monoxide and 0.0979 proportions by volume in the form of carbon dioxide. Then the total loss is evidently

$$H_c = 0.86\frac{10{,}150 \times 0.01}{0.0979 + 0.01} = 801.82 \text{ B.t.u.}$$

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(*f*) a. **Loss Due to Generating Steam for Atomization.**—By referring back to (*a*) *b* in this discussion, we find that the loss due to generating steam used in atomization is represented by the formula

$$H_a = M_s (H_s - h_1) \tag{8}$$

and in the particular instance in question it is 562.44 B.t.u. per pound of fuel burned. Where the steam used in atomization is brought from an outside source, it would, of course, be necessary to neglect the correction made under (*a*) b, although the quantity under this heading must still be taken into account.

b. **Loss Due to Superheating Steam used for Atomization.**—If the steam has been injected into the furnace in atomization, it is clearly evident that for every pound so injected, 0.47 of a B.t.u. are required in superheating it to the temperature of the outgoing chimney gases. Hence the loss so sustained is seen at once to be computed from the formula:

$$H_{sa} = 0.47 M_s (t_g - t_s) \tag{9}$$

in which H_{sa} is the loss due to superheating steam due to atomization per pound of fuel burned; M_s is the proportion by weight of steam used in atomization per pound of oil; t_g the temperature of escaping flue gas; and t_s the temperature of steam used in atomization.

Since we have found that 0.53 lb. of steam were used per pound of oil in atomization and the temperature of the outgoing chimney gases was 400°, and that of the inlet temperature of the steam 381.9°, we see at once that

$$H_{sa} = 0.47 \times 0.53(400 - 381.9) = 4.51 \text{ B.t.u. per lb. of oil burned.}$$

c. **Total Loss in Atomization.**—If now the steam supply in atomization is taken from the boiler under test, or even brought from a separate supply, it is clear that the total loss so sustained is the sum of H_a and H_{sa}. Hence the total loss H_{ta} in atomization is

$$H_{ta} = H_a + H_{sa} \tag{10}$$

In the case at issue then,

$$H_{sa} = 562.44 + 4.51 = 566.95 \text{ B.t.u.}$$

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(*g*) **Loss Due to Moisture in Entering Air.**—All air drawn into a furnace holds in suspension a certain amount of moisture. In previous instances of moisture entering the flue gas it is seen that a loss is sustained in superheating this moisture content to the temperature of the outgoing chimney gases. Let M_a be the pounds of air that enter the furnace per pound of fuel burned, and let K be the proportion by weight of moisture in this entering air then the loss in heat units H_{ma} due to this moisture may be expressed at once by the formula

$$H_{ma} = 0.47\ M_a K\ (t_g - t_a) \tag{11}$$

In the illustration cited in this case it was found that there were 22.82 lb. of chimney gas formed, which means that 21.82 lb. of air were drawn into the furnace to burn 1 lb. of fuel oil; that the entering moisture represented 0.75 per cent. of the entering air which found its way into the furnace at a temperature of 84° and escaped from the chimney at a temperature of 400°.

Therefore

$$H_{ma} = 0.47\ (21.82) \times 0.0075\ (400 - 84) = 23.18 \text{ B t.u.}$$

(*h*) **Stray Losses.**—In order to make a perfect balance between all of the various factors entering a heat balance, the residual heat of each pound of oil not otherwise accounted for is thrown into a column headed "Stray Losses." It is clearly evident that this loss is equal to the calorific value of the fuel per pound less the sum of all the heat accounted for in the various columns cited above. Hence if H_s represents the stray losses per pound of fuel, and H_o the calorific value of 1 lb. of fuel oil under test, we may write the formula as follows:

$$H_s = H_o - (H_n + H_m + H_h + H_g + H_c + H_{ta} + H_{ma}) \tag{12}$$

and in the case at issue by summarizing the columns we find this to be 18151.06 B.t.u.

$$\therefore H_s = 19{,}450 - 18{,}151.06 = 1{,}298.94 \text{ B.t.u.}$$

(*i*) **Total Calorific Value or Summary.**—We are now in a position to summarize the complete heat balance. The various items just discussed will be seen to be represented both in B.t.u. per pound and in percentages, as follows:

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SUMMARY FOR HEAT BALANCE

		Losses		Heat available
		B.t.u.	per cent.	
Total B.t.u. in 1 pound water free oil				19,450
(*a*) *a.* In total heat absorbed by boiler	14,639.00			
b. Heat absorbed for atomization	562.44			
c. Net heat absorbed for power		14,076.56	72.37	
(*b*) Loss due to moisture in fuel		11.67	0.06	
(*c*) Loss due to moisture of burning H		1,166.97	6.00	
(*d*) Loss due to heat carried away by gases		1,503.91	7.73	
(*e*) Loss due to incomplete combustion of C		801.82	4.12	
(*f*) *a.* Loss due to generation of steam for atomization	562.44			
b. Loss due to superheating of steam for atomization	4.51			
c. Total loss due to atomization		566.95	2.92	
(*g*) Loss due to moisture of entering air		23.18	0.12	
(*h*) Stray losses		1,298.94	6.68	
		19,450.00	100.00	19,450

The Net Boiler Efficiency.—In fuel oil central station practice, due to the fact that a portion of the steam generated in the boiler is used for atomization, we need further definition for true boiler efficiency than the notation set forth in the Rules for Boiler Tests advised by the Power Test Committee of the American Society of Mechanical Engineers. Further comment on this point will be made in the next chapter. Suffice it to say here, however, that the net boiler efficiency B_{ne} for the boiler will be considered as that resulting from taking the ratio of the heat H_n represented in the useful steam evaporated by the boiler per pound of oil fired to the total heat H_o given out by each pound of oil burned.

Thus

$$B_{ne} = \frac{H_n}{H_o} \qquad (13)$$

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In the data set forth in the heat balance just computed we find then that

$$B_{ne} = \frac{14{,}076.56}{19450} = 72.37 \text{ per cent.}$$

The Boiler Efficiency as a Steaming Mechanism.—In case, however, it is desired to ascertain the boiler efficiency B_e as a steaming mechanism, it would then of course be proper to compute this boiler efficiency B_e by taking the ratio of the total heat H_t absorbed by the steam for each pound of oil fired to the total heat H_o actually given out by each pound of fuel oil fired. Thus

$$B_e = \frac{H_t}{H_o} \tag{14}$$

Under such a definition the boiler data set forth in the heat balance would indicate a boiler efficiency, thus

$$B_e = \frac{14{,}639}{19{,}450} = 75.27 \text{ per cent.}$$

The data from which the heat balance and boiler efficiency illustration was computed in this chapter is summarized as follows:

Summary of Data Used

Calorific value of dry fuel oil per pound	19,450 B.t.u.
Equivalent evaporation of water per hour	28,225 lb.
Consumption of dry fuel oil per hour	1,872 lb.
Steam used in atomization per lb. of dry fuel oil	0.520 lb
Temp. of saturated steam used in atomization	381.9°F.
Temp. of feed water	169.1°F.
Per cent. of moisture in fuel oil	1.0%
Temp. of entering fuel oil	96°F.
Temp. of flue gases	400°F.
Hydrogen content of fuel	11.0%
Carbon content of fuel	86%
Temp. of entering air	84°F.
Weight of dry chimney gases per lb. of dry fuel	19.83 lb.
Weight of entering air per lb. of dry fuel oil	21.82 lb.
Carbon dioxide in flue gas	9.79%
Carbon monoxide in flue gas	1.00%
Moisture of entering air from boiler room	0.75%

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CHAPTER XXXIX

SUMMARY OF SUGGESTIONS FOR FUEL OIL TESTS AND THEIR TABULATION

The rules for conducting boiler performances, as advised by the Power Test Committee of the American Society of Mechanical Engineers, covers in wonderful detail the setting forth of apparatus and tabulation of data for such performances, when coal is employed as a fuel. Only brief mention is, however, made for alterations necessary when crude petroleum is used as a fuel. Since a greater number of engineers would probably be inconvenienced than those actually benefited by attempting to make a set of rules broad enough to cover both performances by coal and by oil as fuels, an appendix should be drawn up to satisfy standardized conditions of test for oil fired boilers. This lack of standardized performance has caused considerable confusion in those communities where oil is used as a fuel.

The most glaring source of confusion is that relating to boiler efficiency. Some engineers maintain that boiler efficiency is the ratio of heat actually transferred from the fuel through the metallic parts of the boiler to the total quantity of heat given out by the fuel. When coal is used as a fuel this definition is perfectly proper, but when oil is the fuel employed confusion is at once introduced, due to the fact that as a rule a certain amount of the steam generated must be utilized to atomize the oil in the furnace. In the last chapter it was shown that the efficiency of an oil fired boiler computed on one assumption in a specific instance is 75.27 per cent., and on another assumption it becomes but 72.37 per cent.

Let us then discuss some of the points wherein additional instructions are desirable to properly conduct boiler tests where oil is used as the fuel for heat production.

Efficiency for Oil Fired Boilers Defined.—Perhaps the most important point is to come to some definite decision relative to an exact manner of arriving at the efficiency of the boiler as above alluded to. In this work we shall consider that the true

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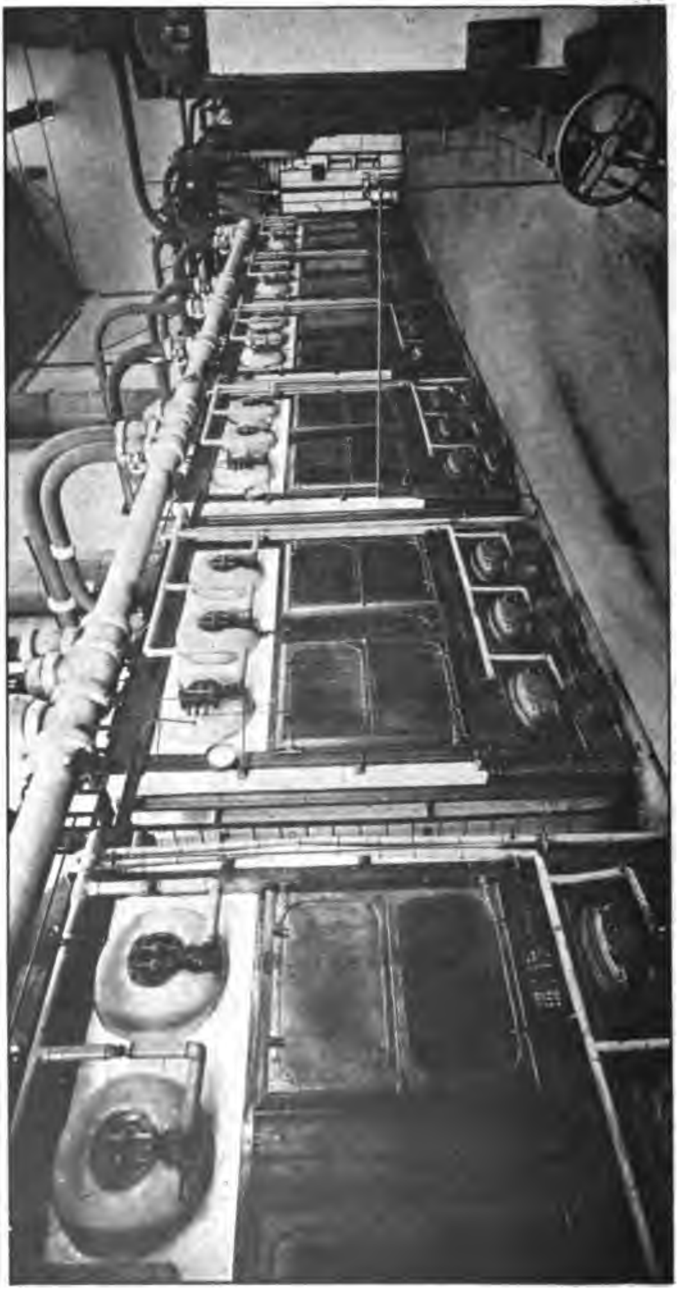

FIG. 197.—Oil fired Babcock and Wilcox boilers at station A, Pacific Gas and Electric Company, San Francisco. These boilers as here shown are operated by hand control in the feeding of the fuel oil to the burners, but quite recently have been equipped with the Merit automatic control.

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Fig. 198.—Boilers equipped with Moore automatic oil firing system.

Here may be seen an improved system for regulating the fuel that is fed to the boiler furnaces of the new installation of the Inspiration Copper Company, Miami, Arizona. The oil supply is controlled from one central point, and with it, the steam used in atomizing the oil and the quantity of air admitted to the furnace are also under delicate control. Any fluctuation in steam pressure operates a governor whose power arm controls a bleeder valve on the oil pump discharge line, thus cutting off the oil supply, if the steam pressure is too high and reducing it if too low. Any change in pressure in the oil main, in turn, controls the amount of steam for atomizing and of air for burning the oil.

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	B.t.u.	Per cent.
(a) a. Total heat absorbed by boiler.........		
b. Heat absorbed for atomization.........		
c. Net heat absorbed for power...........		
(b) Loss due to water in fuel oil..............		
(c) Loss due to water from burning H..........		
(d) Loss due to heat carried away by dry gases..................................	For numerical	
(e) Loss due to carbon monoxide.............	example com-	
(f) a. Loss due to generation of steam for atomization...........................	pletely solved, see page 327.	
b. Loss due to superheat of steam used for atomization......................		
c. Total loss due to atomization...........		
(g) Loss due to moisture in entering air........		
(h) Stray losses..........................		
(i) Total calorific value of 1 lb. of fuel oil free from water (item 42a)		100

TABLE 2.—**Principal Data and Results of Boiler Test**

(1) Oil Burners.—No............Type..........Make...............
(2) Total heating surface......................................sq. ft.
(3) Date..
(4) Duration...hr.
(5) Kind and gravity of fuel oil...................................
(6) Steam pressure by gage............................lb. per sq. in.
(a) Oil pressure at burner.......................lb. per sq. in.
(7) Temperature of feed water entering boiler.....................deg.
(a) Temperature of oil at burner..........................deg.
(8) Percentage of moisture in steam or number of degrees of superheating.......................................per cent. or deg.
(9) Percentage of water in oil.............................per cent.
(10) Oil free from water per hour....................................lb.
(11) Oil free from water per hour per burner........................lb.
(12) Equivalent evaporation per hour from and at 212 deg............lb.
(13) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of heating surface...lb.
(14) Rated capacity per hour, from and at 212 deg...................lb.
(15) Percentage of rated capacity developed................per cent.
(16) Equivalent evaporation from and at 212 deg. per lb. oil free from water...lb.
(a) Per cent. of total steam used by burner............per cent.
(17) Net equivalent evaportion from and at 212 deg. per lb. of oil free from water (deducting steam used by burner)...............lb.
(18) Calorific value of 1 lb. of oil as received, by calorimeter.........B.t.u
(19) Calorific value of 1 lb. of oil free from water................B.t.u.
(20) Efficiency of boiler and furnace........................per cent.
(21) Net efficiency (deducting steam used by burners).........per cent.

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CHAPTER XL

THE USE OF EVAPORATIVE TESTS IN INCREASING EFFICIENCY OF OIL FIRED BOILERS

To the operating engineer it may seem that the somewhat elaborate rules for conducting evaporative tests of steam boilers are of little interest. It is his province to run the boilers as economically as he can, to keep them clean and in proper repair, and above all to keep the plant in continuous operation. There is one very important function of boiler tests, however, which makes them invaluable to the broad-gage operating engineer who is desirous of securing the best possible results from his plant. This is the use of the evaporative test as a guide in determining what is the best furnace arrangement, the best style of oil burner, and the best draft conditions for the particular boilers he is operating. Thus by making a careful test under certain conditions and then making another test, or sometimes a series of tests, under different conditions, it is possible to determine from the relative efficiencies obtained just how the boiler should be operated. It will not be out of place, therefore, to discuss briefly the various changes that may be made in the boiler operation, which when intelligently carried out will lead to higher efficiencies.

Furnace Arrangement.—Perhaps the most important part of an oil fired boiler is its furnace arrangement. In a previous chapter a number of different furnaces were described, but it was not stated which was the most efficient. This must be determined by testing the boiler under actual operating conditions, first with one furnace arrangement, then with another, being guided in making changes by the results obtained in the different tests. It is impossible to design a furnace that will be right for all conditions, as with different grades of fuel oil or different makes of boiler or different draft conditions, different furnace arrangements are required. Fortunately it is possible to make minor changes in the furnace very easily, as these involve usually only an alteration of the location of fire brick on the furnace

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floor. It is thus possible to increase or decrease the size of air openings, or to change them in such a way as to allow more air to enter at one part of the furnace, such as directly under the flame, and less at another part where it is not needed. It is also possible, without much difficulty, to alter a furnace that has been designed for a front shot burner and make it suitable for a back shot burner, and thus it may be found by actual tests which of these two types of furnace is best suited to the particular boiler.

In testing the different arrangements it is very important to test the boiler for capacity as well as economy, as it may some-

FIG. 199.—The furnace interior.

Here is shown the furnace interior of an oil fired boiler similar in design to the specifications given in this chapter. Note the V-shaped arrangement in the brickwork in order to admit air for the economic burning of the fuel oil.

times happen that the furnace that is most efficient at ordinary loads is not capable of forcing the boiler enough to carry the heavy loads sometimes required. In such a case it may be necessary to adopt a less efficient furnace, as it is usually of supreme importance for the boiler to be capable of carrying an overload when required.

Oil Burners.—Boiler tests are of great value in determining what make and style of oil burner is the best to use under the given conditions. In testing oil burners it is of extreme importance to measure the steam used by the burner and determine the net efficiency of the boiler; for one kind of burner may produce

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better furnace efficiency than another, and yet use so much steam for atomizing as to make it an uneconomical burner to use. After deciding on the type of burner to use, tests should be made with varying quantities of atomizing steam with the same burner, the object being not to find out the least quantity of steam that may be used for atomizing but to determine the quantity of steam that secures the best net efficiency of the boiler.

The temperature and pressure of the oil are intimately connected with the quantity of atomizing steam required. In the case of mechanical atomization, such as is used in marine work, high pressure and high temperatures are used and no steam is required. In general it may be said that the hotter the oil and the higher its pressure, the less atomizing steam is needed. Different oils require different temperatures, and the temperature should always be kept well below the flash point of the oil. By testing the boiler with the oil first at one temperature and then at another, and varying the quantity of steam to suit, much information can be obtained as to the most economical method of operation.

Apart from the quantity of steam used, other changes that may be made in the burner consist in varying the size of the steam and oil slots, altering the height of the burner in reference to the furnace floor, and changing the angle of the flame in reference to the grates.

Draft.—The quantity of air entering the furnace depends on the intensity of the draft, and the area of openings for the admission of air to the furnace. The quantity of air may be reduced by partially closing the boiler damper or the ash pit doors, or it may be increased by enlarging the openings in the furnace floor. Thus it is possible to operate with large openings and light draft, or with small openings and strong draft. A careful test of the boiler will determine at once which of these conditions gives the best results. If the load on the plant is variable it is necessary to have the air openings large enough to admit sufficient air for the maximum load at full draft. Then for lighter loads the damper or ash pit doors must be operated. When making tests the readings of the draft gage at various points in the setting should be carefully observed, and loss of draft due to the gases passing through the setting noted. Thus, if the draft in the furnace is 0.2 in and the draft in front of the damper is 0.3 in., there is loss of 0.1 in. between the damper and the furnace. This loss of

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draft varies with the volume of gases just as the drop in pressure due to steam flowing through an orifice varies with the quantity of steam flowing. If the quantity of excess air increases, therefore, the loss of draft also increases. By connecting a draft gage so as to measure the difference in the draft at the two points, it will serve as an approximate indicator of the amount of excess air.

Flue Gas Analysis for Maximum Efficiency.—The analysis of the flue gases serves as an accurate means of determining how to set the dampers, and is the most valuable guide in securing the best efficiency, both during an evaporative test and in regular operation. In general, it may be said that the best efficiency is obtained when the greatest percentage of carbon dioxide (CO_2) occurs, without the presence of carbon monoxide (CO). If CO begins to appear in the gas analysis it is useless to increase the CO_2 further, as any gain due to reducing the excess air is more than offset by the loss due to incomplete combustion. The presence of CO is always more harmful than is indicated by the calculated loss for unconsumed carbon, for if carbon is only partially consumed it is certain that some of the hydrogen is also passing off unconsumed in the form of hydrocarbons, thus causing a far greater loss. This loss due to unconsumed hydrogen does not appear in the ordinary gas analysis, and it is in connection with this item that the heat balance is of special value. Item (*h*) of the heat balance, which is found by subtracting the heat accounted for from the heat supplied, includes the loss due to unconsumed hydrogen, and if accurate tests are made it will be found that this item is always greater the more CO is found in the gases.

If the furnace is properly designed it should be possible to secure 13½ per cent. to 14 per cent. CO_2, with not over 3 per cent. oxygen, and without a trace of CO, using not over 15 per cent. or 20 per cent. excess air. These results must be secured to give the best economical results, and if they cannot be secured by changing the draft or the burners, it will then follow that there is something wrong with the furnace arrangement.

It will be found that there is a very intimate relation between the furnace, the burner, and the draft. Thus the intensity of draft and amount of atomizing steam that give best results with one furnace, may give poor results with another; yet by readjusting the dampers and burner valves to suit the new conditions, better results than ever may be obtained. With too much steam

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FIG. 200.—A single boiler unit automatically controlled.

Here is an illustration, that of the Shredded Wheat Company's plant in Oakland, California, of how automatic control for damper, ash pit, atomizing steam, and fuel oil admission is accurately accomplished under one master controller to the right as shown in the picture.

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the flame may be carried too far beyond the air openings, causing a poor mixture of air and gases. This would result in a poor gas analysis, although the total quantity of air may be correct.

There are so many variations that can be made, that it is usually impractical to make a complete evaporative test for

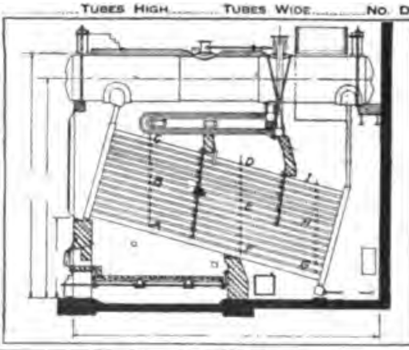
BOILER OPERATION REPORT

PACIFIC GAS AND ELECTRIC COMPANY OPERATION AND MAINTENANCE DEPT

OBSERVATIONS ON BOILER NO. .... IN STATION ....

DATE .... OBSERVATIONS BY ....

RATED H.P. .... TUBES HIGH .... TUBES WIDE .... NO. DRUMS ....



	Item No.						Item No.				
Pressures	1	Superheated Steam Lbs.				Drafts	29	Under Grate In. Water			
	2	Saturated Steam "					30	In Furnace " "			
	3	Steam at Burner "					31	At A " "			
	4	Oil in Line "					32	" B " "			
	5	Oil at Burner "					33	" C " "			
	6						34	" D " "			
	7						35	" E " "			
	8						36	" F " "			
Temperatures	9	Boiler Room F°					37	" G " "			
	10	Feed Water "					38	" H " "			
	11	Superheated Steam "					39	" I " "			
	12	Saturated Steam "					40	In Breeching " "			
	13	Degree of Superheat "					41	At Economizer Inlet " "			
	14	Fuel Oil to Burners "					42	" " Outlet " "			
	15	Gases at A "					43	In Stack			
	16	" " B "					44				
	17	" " C "					45				
	18	" " D "				Flue Gas Analysis	46	CO_2 at "C" Vol. in %			
	19	" " E "					47	O " " " " "			
	20	" " F "					48	CO " " " " "			
	21	" " G "					49	CO_2 at "I" " " "			
	22	" " H "					50	O " " " " "			
	23	" " I "					51	CO " " " " "			
	24	" Leaving Boiler "					52				
	25	" Entering Economizer "					53				
	26	" Entering Stack "					54				
	27						55				
	28						56				

FIG. 201.—Typical form for boiler operation report.

Here is how the Pacific Gas and Electric Company, a corporation operating the largest system of oil-fired steam power plants in the world, keeps its records on evaporative tests for bettering power plant economy.

each set of conditions. It is possible, however, to obtain comparative data in a single test, by varying the conditions at the end of each hour, or each two hours. By carefully observing the quantity of oil and water used each hour, a fairly accurate

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comparison of efficiencies under different conditions may be obtained. This, combined with the flue gas analysis, makes a valuable guide for efficient operation.

Regulation.—When an oil fired boiler is in operation there are three variables under control of the fireman, viz.:

The quantity of oil burned.

The quantity of atomizing steam used, and

The quantity of air supplied.

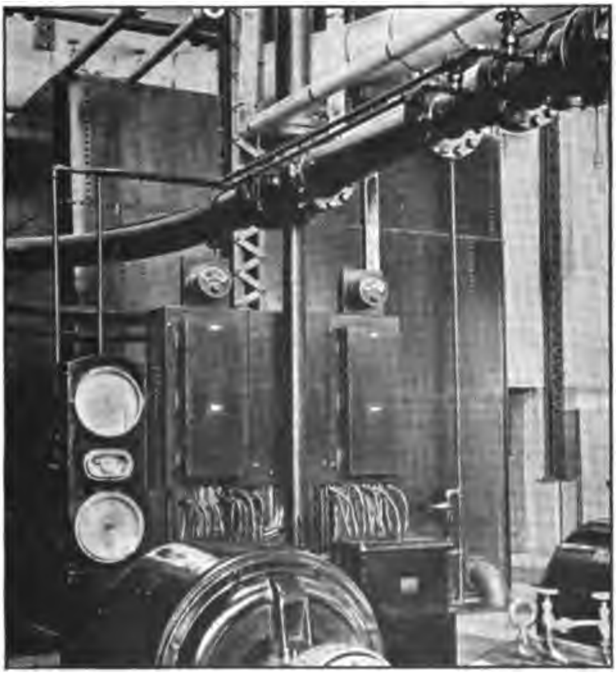

Fig. 202.—Venturi meter for measuring water supply at the Long Beach Plant of the Southern California Edison Company shown installed on piping in upper right center.

The quantity of oil burned is determined by the amount of steam required in the plant, and must be varied accordingly. When there are several boilers in battery the amount burned under each boiler may be varied by operating the oil valves at the burners, or the total amount in the plant may be changed by altering the oil pressure at the oil pump. Whenever the quantity of oil burned is varied, there should be a corresponding variation in the quantity of atomizing steam and the quantity of air.

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There are now on the market devices which regulate all three variables automatically according to the load on the plant. Illustrations of automatic firing systems are shown on pages 331 and 341. The essential requisites for a device of this kind are that it shall be reliable in operation, and that when it has once been set to give proper CO_2 readings at certain loads, it will always come back to the same position for the same load. While it is possible under test conditions to secure just as high efficiency with hand regulation as with the automatic, it will usually be found that the automatic regulator produces better every-day economy under operating conditions.

Records.—Complete evaporative tests cannot be made every day in an ordinary plant, but it is possible to take sufficient observations to secure a daily record of the important items entering into the operation of a boiler. A form that is convenient for such a record is illustrated in Fig. 201. By carefully studying these records, together with the results of evaporative tests, it is possible to maintain the operation of a boiler plant at a very efficient point.

By operation at the most efficient point we save and it is well to remember in these days of national crisis, that "to save is to serve."

PRACTICAL ILLUSTRATIONS OF ECONOMY STUDY

As an illustration of what can be accomplished in actual practice by the use of boiler tests, combined with intelligent changes of furnace arrangement and operating details, the following description of work performed in one of the large oil burning plants in San Francisco will be of interest:

In order to increase the efficiency and capacity of the boilers, certain changes were made on the boiler shown in Fig. 203. To determine what improvements had been effected a series of tests was run before and after the changes were made. The amount of oil burned in the furnace was measured by a meter and the amounts of steam generated and steam used by the burners were measured by General Electric flow meters.

General Furnace Arrangement.—Before making the changes, the furnace in the boiler was arranged as shown in Fig. 86 and the baffles between the gas passages were located as shown in Fig. 203. Most of the baffle bricks in front of the flame plates between the 1st and 2nd passes were missing, thus allowing a large percentage

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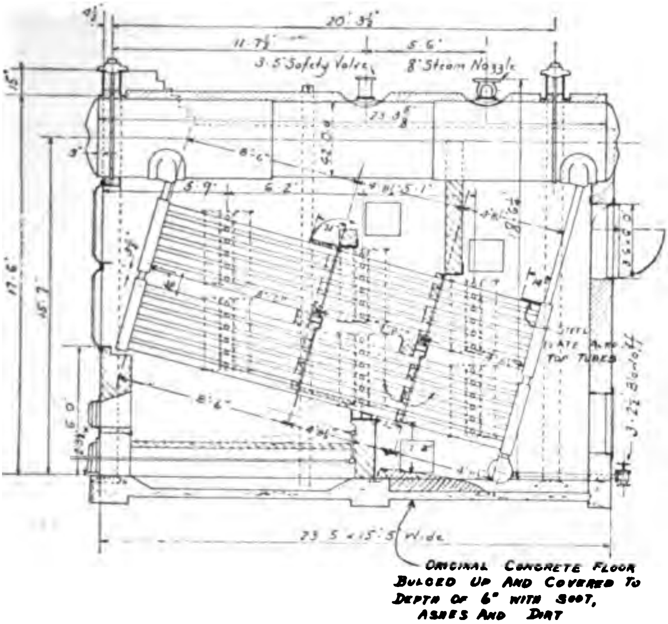

FIG. 203.—Arrangement of baffles between gas passages before making the changes for the test.

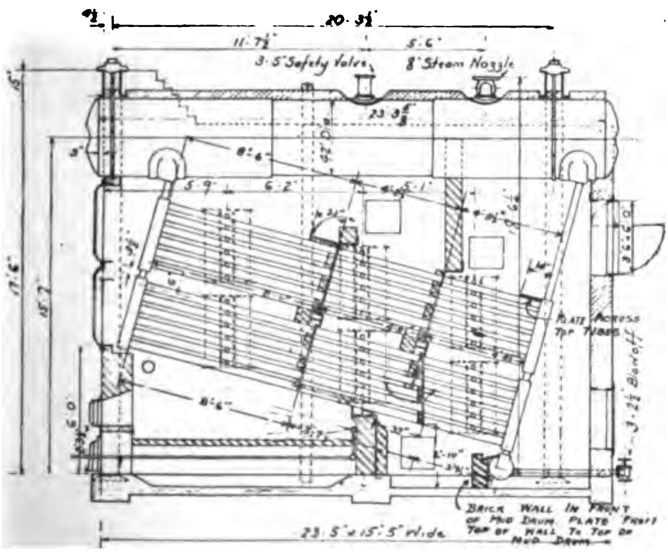

FIG. 204.—Arrangement of baffles between gas passages after making the changes for the test.

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of the gases from the furnace to pass directly into the 2nd pass, the flame plates themselves having been burned away in a number of places. The space between the bottom of the rear baffle

FIG. 205.—Checkerwork and housings installed around burners.

and the bridge wall was also very small as shown on the sketch. This was remedied by moving back the bottom of the baffle to a position as shown in Fig. 204. All of the flame plates which were

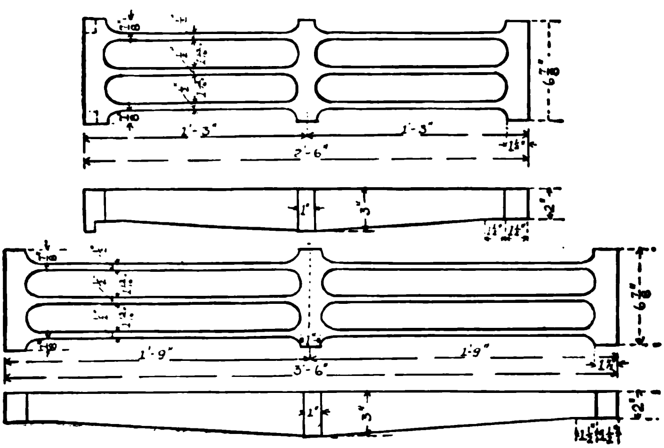

FIG. 206.—Sketch of grate bars used during test.

in bad shape were renewed and new bricks put in front of them, thus making the new baffles as tight as possible.

The furnace arrangement as shown in Fig. 86 was changed to conform to the arrangement as shown in Fig. 87. New grate bars

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	Item No.	DATE	1915	April 19 3-4 P.M.	April 19 7-7.05 P.M.	April 20 10.30-11.00 A.M.	April 30 2.30-4.30 P.M.	April 30 3-4 P.M.	May 10 2.40-4.40 P.M.	May 11 2.45-4.45 P.M.	May 13 2.15 to 4.15 P.M.	May 14 7.45-8.45 P.M.
	1	Duration of test	Hours	2	5/4	1½	1	1	2	2	2	1
	2	Air space in grate	Sq. ft.	←	See	Sketch	No. 1 →		←	See Sketch No. 2		→
	3	Oil burners	No. & make	←	3	Peabody	→		←	8 Peabody		→
Pressures	4	Barometer	Ins. mercury									
	5	Steam gauge (saturated)	Lbs.	196	197	197	196	196	198	202	1976	199
	6	Steam gauge (superheated)	Lbs.									
	7	Steam pressure at burner	Lbs.									
	8	Oil pressure at burner	Lbs.									
	9	Oil pressure on line	Lbs.	52	57	52	52	52	52	52	52	55
Drafts	10	Draft under grate	Ins. water	.10	.07	.107	.069	.078	.055	.046	.075	.108
	11	Draft in furnace	Ins. water	.096	.065	.103	.074	.083	.026	.017	.055	.096
	12	Draft in Top of 1st pass	Ins. water	.03	.00	.035	.00	.016	+.025	+.034	.00	.042
	13	Draft before damper Top of 3rd Pass	Ins. water	.19	.176	.195	.18	.21	.095	.055	.163	.271
	14	Draft after damper	Ins. water	.205	.20	.204	.194	.225	.177	.19	.181	.315
Temperatures	15	Boiler room	Degrees F.									
	16	Feed water	Degrees F.	205	200	196	200	187	181	198	184	208
	17	Superheated steam	Degrees F.									
	18	Saturated steam	Degrees F.	386	387	387	386	386				
	19	Degree of superheat	Degrees F.	0	0	0	0	0				
	20	Furnace	Degrees F.									
	21	Escaping gases	Degrees F.	565	603	542	574	610	504	483	566	642
	22	Fuel oil to burners	Degrees F.	181	183	182	170	177	176	177	179	180
Fuel oil	23	Total fuel oil used	Lbs.				1720	1970	3260	2836	4100	2300
	24	Gravity of fuel oil	Degrees B.									
	25	Per cent of water in fuel oil	%									
	26	Heating value of fuel oil ~~free from water~~	B. T. U. per lb.	Assumed			18500	18500	18500	18500	18500	18500
	27	Total fuel oil used free from water	Lbs.									
	28	Fuel oil per hour as fired	Lbs.				1720	1970	1630	1418	2050	2300
	29	Fuel oil per hour free from water	Lbs.									
Water	30	Total water used	Lbs.									
	31	Factor of evaporation		1.053	1.06	1.062	1.06	1.07	1.08	1.065	1.08	1.06
	32	Total water evaporated from & at 212° F.	Lbs.									
	33	Actual water evaporated per hour	Lbs.	18960	20500	20600	22710	24340	20280	17300	26010	31600
	34	Water evaporated per hour from & at 212° F.	Lbs.	19960	21730	21880	24070	26050	21900	18420	28100	33490
Efficiency	35	Water evaporated per lb. of oil as fired	Lbs.				13.2	12.33	12.44	12.2	12.7	13.72
	36	Water evaporated from & at 212° F. per lb. of oil as fired	lbs.				14.0	13.20	13.44	13.0	13.7	14.55
	37	Water Evaporated from & at 212° F. per Lb. of Oil Free from Water	lbs.				—	—				
	38	Per cent of total steam used by burners	%	4.4	5.17	4.04	3.75	4.28	3.15	3.06	3.12	3.37
	39	Gross efficiency	%				73.7	69.5	70.5	68.1	71.9	76.4
	40	Net efficiency (deducting steam used by burners)	%				71.0	66.5	68.3	66.02	69.3	73.82
Power	41	Standard commercial horsepower developed	H. P.	580	630	635	700	755	635	528	815	972
	42	Builders' rating	H. P.	524	524	524	524	524	524	524	524	524
	43	Per cent horsepower developed of rating	%	111	120	121	134	144	121	101	156	186
Gases	44	Carbon dioxide Top of 1st Pass	CO_2 volume in %	15.0	15.6	14.8	15.4	15.0	15.1%	14.5	15.6	14.9
	45	Oxygen " " " "	O volume in %	0.2	0.5	0.1	0.8	0.2	1.3	2.0	0.5	0.0
	46	Carbon monoxide " " " "	CO volume in %	0.8	0.6	0.8	0.0	0.8	0.1	0.0	0.1	1.3

FIG. 207.—Tabulated data on boiler overhauling test.

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having wide air spaces were also installed, a 2 ft. 6 in. bar being used next to the bridge wall and a 3 ft. 6 in. long bar next to this. These bars which are shown in Fig. 206 have a net free area of about 65 per cent. The checkerwork and housings installed around the burners are shown in Fig. 205.

New piping was installed for the steam and oil to the burners. The general view of the piping and other details of the boiler front is shown in Fig. 60. Two flanges, inserted for bringing about an arrangement for limiting the steam to burners, had a steel disc inserted between them, this disc having a $\frac{5}{16}$ in. hole drilled through it. When the boiler is running at rating all the steam for atomizing passes through this hole, the valve on the by-pass around the disc being closed. When it is desired to carry a heavy load on the boiler the lower or by-pass valve is opened and enough steam can be obtained for any overload desired. The valves having rising stems, one can tell at a glance whether the by-pass valve is opened or closed.

The damper control was brought to the front of the boiler and arranged so that the damper could be opened or closed in very small increments. It was found that the damper did not fit tight all around as proved by the draft readings taken in the boiler with the damper and ash pit doors closed as tight as possible. This defect was not completely remedied as there was still some leakage at this point during the tests.

The location of the peep hole in the south wall was changed so that a view of the front walls and tubes for some distance back from it could be obtained. This enabled the operator to see the end of the flame at all times and to determine when the fires were smoking. Peep holes were also put in the side walls close to the burners so that the flame and furnace could be observed at these points.

By building a small wall in front of the mud drum and laying a plate from the top of this wall to the mud drum, the possibility of gases getting under the drum and by-passing the 3d pass was eliminated. A plate 14 in. wide was laid on top of the upper tubes and against the back headers, the effect of this being to force the gases to pass over the entire tube surface in the rear pass.

Table Showing Economy Data.—The following table shows the operating conditions and loads which could be carried on this boiler before and after the changes were made and the boiler

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cleaned. This table shows the average of the 15 minute readings during the periods for which the tests were run. In all cases the conditions remained practically constant during the run.

The first five columns of the table show the tests made on April 19, 20 and 21 before any changes had been made on the boiler, except that before the trials of April 20 the soot was blown from the tubes. This accounts for the much lower flue gas temperature for the first test on April 20 as compared with the evening test of April 19, at approximately the same load. The test of April 21 shows the maximum load which could be obtained on the boiler prior to the changes.

The last four columns in the table show the results obtained after the changes had been made. The following summary shows a comparison of the two maximum loads run:

COMPARATIVE ECONOMIC RESULTS

	April 21, before changes had been made	May 14, after changes had been made
Boiler pressure	196 lbs.	199 lbs.
Oil pressure	52 lbs.	55 lbs.
Oil temperature	177°	180°
Draft in breeching	0.225″	0.315″
Draft in top 3d pass	0.21 ″	0.271″
Draft in bot. 3d pass	0.21 ″	0.243″
Draft in bot. 2d pass	0.155″	0.171″
Draft in top 2d pass	0.039″	0.066″
Draft in top 1st pass	0.016″	0.042″
Draft in furnace	0.083″	0.096″
Draft in ash pit	0.078″	0.103″
Temp. of flue gases	610°	642°
Load on boiler	755 h.p.	972 h.p.
Per cent. of rating developed	144%	186%
Gross efficiency	69.5%	76.4%
Steam to burners	4.28%	3.37%
Net efficiency	66.5%	73.82%

Conclusions from Test Data.—Thus the capacity of the boiler was increased from 144 per cent. of rating to 186 per cent. of rating, a net gain of 29 per cent. in the amount of steam generated. At the same time the net efficiency was increased from 66.5 per cent. to 73.8 per cent. This increase in net efficiency, it will be

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noted was helped out by the saving in the amount of steam for atomizing, the latter item having been reduced from 4.28 per cent. of the total steam generated to 3.37 per cent. These efficiencies are only comparative as a heat balance shows that the efficiency is probably higher in each case than that given. The discrepancy is due probably to an error in the oil meter.

A comparison of the tests at other ratings shows a marked improvement in the operation of the boiler, particularly in the amount of steam for atomizing the oil. Subsequent to the tests enumerated it has been found that rating could be obtained on the boiler if the steam for atomizing was supplied through only a ¼ in. hole in the disc previously mentioned. Under this condition the steam for atomizing was reduced to slightly above 2 per cent.

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CHAPTER XLI

ECONOMIES OBTAINED IN OIL BURNING PRACTICE

When a boiler is fired with oil there are no ashes to carry away unburned fuel, there are no banked fires to cause poor efficiency at light loads, and as it is possible to secure perfect combustion with very little excess air, the efficiency of an oil fired boiler is naturally higher than that of a coal fired boiler.

In Table 1, on page 357 are given the results of actual tests that have been made from time to time on oil fired boilers with steam atomizing burners at different locations and under various conditions. In Table 2, on page 358 are given the results of tests, made by the Babcock and Wilcox Co. with mechanically atomized oil burners, taken from a paper by Darrah Corbet published in the August 1920 Journal of the American Institute of Electrical Engineers. These tests give the best results that can be expected from oil fired boilers. Actual results obtained in regular operation are not usually as good as the results obtained from tests, but they can be made as good if the same care is exercised in keeping the boilers clean, regulating the air supply and all the minor details that help in securing high efficiency.

There is very little data available to show just how closely modern power plants approximate to test results in every day operation, as there are few plants that keep records complete enough and accurate enough to determine the daily boiler efficiency separate from the complete plant efficiency. The usual method of recording the daily efficiency of an oil burning central station is by the ratio of kilowatt-hours generated to barrels of oil burned. While this is an excellent method of determining the overall efficiency of the plant as a whole, it is a poor method of comparing an oil burning plant with a coal burning plant or of comparing one oil burning plant with another. There are so many factors that enter into the overall efficiency, such as pressure of steam, degree of superheat, amount of vacuum, economy of prime movers, character of auxiliaries, and station load factor, that the mere statement of kilowatt-hours generated

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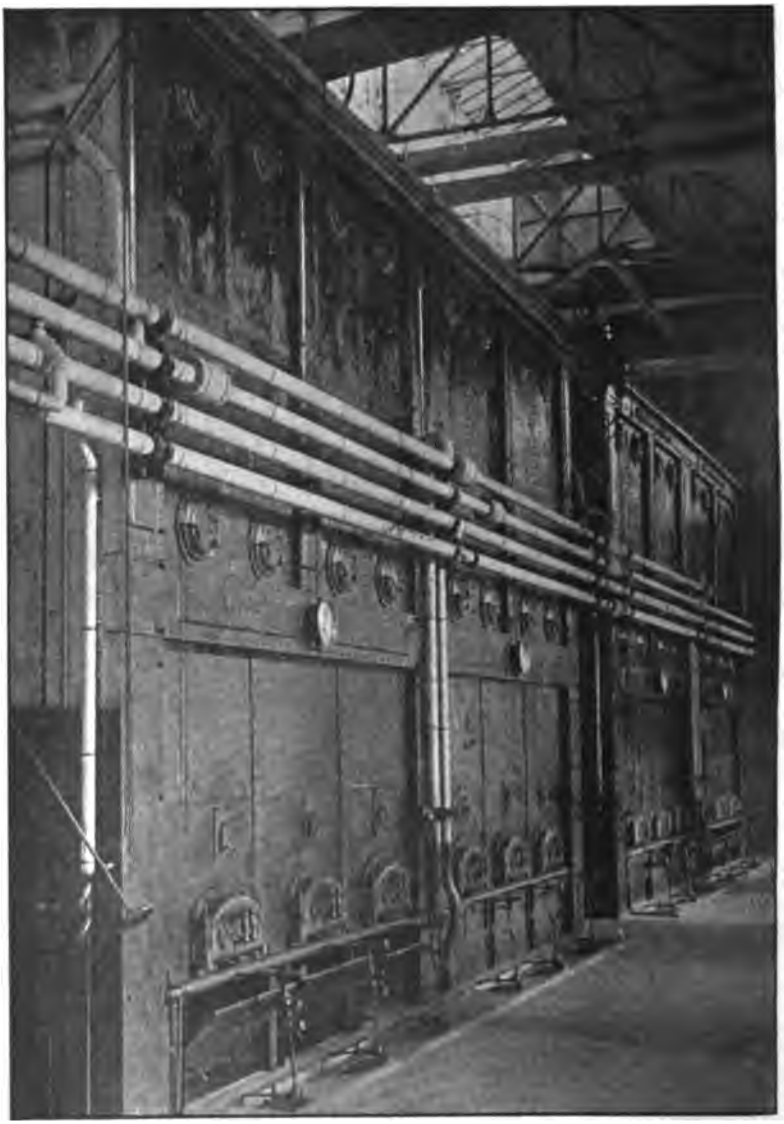

FIG. 208.—Boiler front at Arizona Power Plant, Phoenix, Arizona. A world's record was here established, wherein 333.3 kilowatt hours of electrical energy were generated per barrel of fuel oil burned.

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per barrel of oil does not give much idea of what efficiency is being obtained from the oil burning boilers unless a complete detailed description of the plant is included.

It is probable that the general average of present day oil burning central stations generate from 200 to 240 kilowatt hours per barrel of oil, when operated at a fair load factor. This is equivalent to from 25,000 to 30,000 B.t.u. per kilowatt hour.

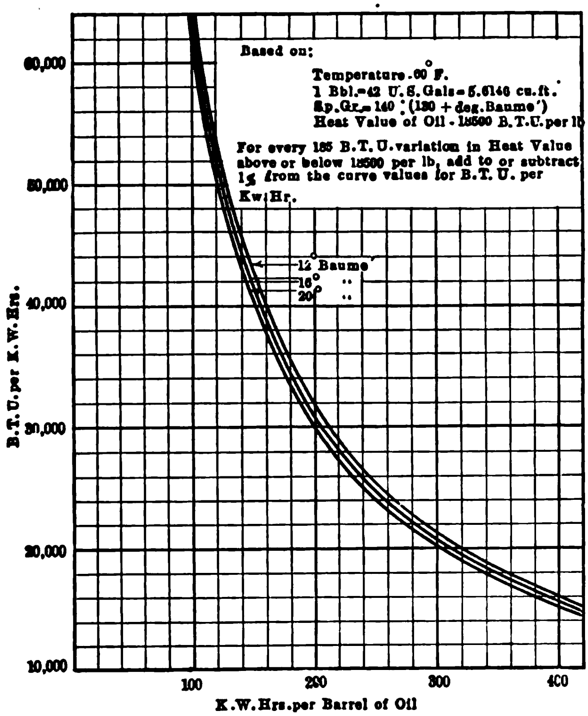

FIG. 209.—Curve showing relation between K.W.H. per barrel of oil and B.T.U. per K.W.H. for different grades of oil.

At very light loads, such as occur in standby service, the efficiency may drop to less than 100 kilowatt hours per barrel.

While records giving the actual efficiency of oil fired boilers under regular operating conditions are rare, a very close approximation of the performance of the boilers can be obtained from the flue gas analysis and the temperature of the escaping gases. It will therefore be of interest to note some of the actual Orsat readings taken from boilers in regular service.

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The following readings were obtained from a 600 h.p. Stirling Boiler with Peabody Hammel furnace. No special adjustments were made and the object of taking these readings was to obtain some idea of the performance of the boiler as usually fired:

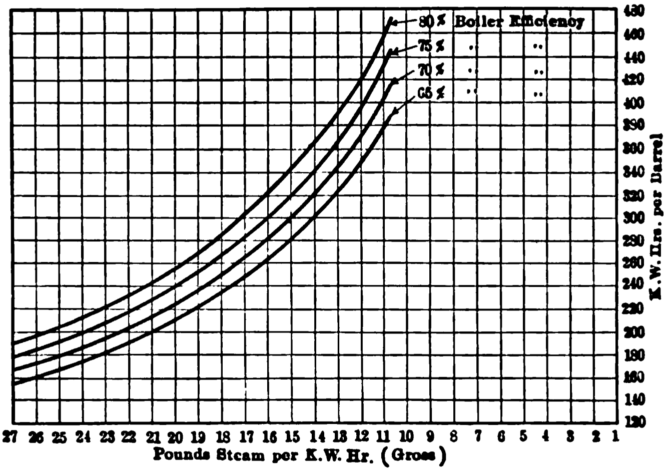

FIG. 210.—Curve showing relation between steam consumption per kw. hour and the output in kw. hours per barrel of oil at different boiler efficiencies.

	CO_2	O	CO	Per cent. excess air calculated
	13.8	3.0	0.0	16.0
	14.2	2.1	0.1	12.5
	14.0	2.2	0.0	14.0
	12.0	5.2	0.0	32.0
Average..........	13.5	3.1	0.02	18.6

The following readings were obtained from a 750 h.p. Boiler of the water leg type, with vertical baffles and burners set in the front wall:

	CO_2	O	CO	Per cent. excess air
	11.4	5.4	0	39
	12.2	4.5	0	30
	12.0	4.8	0	32
	12.3	4.5	0	29
Average..........	11.9	4.8	0	32

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At this point an effort was made to reduce the excess air by partially closing the damper, after which the following readings were obtained:

CO_2	O	CO	Per cent. excess air
13.6	2.8	0	17

The following readings were obtained from a 773 h.p. Parker boiler with horizontal baffles with the burners set in the front wall, the furnace extending the full length of the boiler:

CO_2	O	CO	Per cent. excess air
14.9	1.2	0.2	7
15.4	0.5	0.9	4

Since these readings show the presence of carbon monoxide (CO), an effort was made to obtain perfect combustion by opening the damper and ashpit doors wide, with the following result:

CO_2	O	CO	Per cent. excess air
15.2	0.6	0.5	5

Since CO was still present the fires were cut down, slightly reducing the capacity of the boiler, with the following results:

CO_2	O	CO	Per cent. excess air
14.3	2.4	0	12

It is thus seen that with boilers of several different types, and different furnace arrangements, very little effort on the part of the operators is required to obtain air regulation, corresponding to the test results shown in the table of tests as reproduced in this article.

The temperature of the escaping gases, which is the other important item affecting the efficiency of the boilers in regular operation, depends on five things:

1. Regulation of air supply
2. Cleanliness of the boiler
3. Capacity of the boiler
4. Design of the boiler
5. Condition of the boiler brickwork and baffles

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Flue gas temperatures in actual practice run all the way from 400°F. for clean boilers operating at their rated capacity with 15 or 20 per cent. of excess air, up to 800°F. for dirty boilers with leaky baffles operating at 200 per cent. of rating with 50 to 100 per cent. excess air. A general average for boilers in regular operation is about 550°F.

The following results were obtained on a 524 h.p. Babcock and Wilcox boiler set with the front headers 9 feet above the floor, equipped with a Peabody furnace. The setting was new, and the boiler had just been cleaned inside and out. The boiler was equipped with soot blowers which were operated about eight hours before the observations were taken.

Duration of test, hrs	1.0	3.0
Capacity—per cent. of rating (measured by steam flow meter)	110.0	172.0
Steam pressure, gage, lbs. per sq. in	192.0	198.0
Degree of superheat, deg. F	75.0	73.0
Draft at damper, in. water	0.01	0.23
Carbon dioxide CO_2%	14.4	14.3
Temperature escaping gases, deg. F	431.5	457.5

The importance of keeping a boiler clean is well illustrated by the effect on the flue gas temperature of blowing the soot off the tubes. There is a common belief that when burning oil there is little or no accumulation of soot. This, however, is not the case, for while the quantity of soot is much less than in a coal fired boiler it is still an ever present evil, and it is necessary to dust the tubes at least once a day if the best efficiency is to be obtained. The following readings taken in regular service, on an 823 h.p. Stirling boiler with a Peabody Hammel oil furnace, show that blowing the soot off the tubes resulted in reducing the exit temperature more than 75°, and that in six days the temperature gradually built up to more than it had been before the tubes were dusted.

In stand by plants, where it is necessary to operate the plant at no load and be ready to pick up the load at any instant, the boilers must be kept hot even though no steam is generated. With coal fired boilers this is accomplished by keeping a banked fire on the grates. With oil fired boilers the fire is put out completely, and when the steam pressure in the boiler has dropped as low as permissible, the fire is relighted and the steam brought back to full pressure. During the time the fire is out the damper

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TABLE 1.—RESULTS OF TESTS ON OIL-BURNING BOILERS WITH STEAM ATOMIZING BURNERS

	Redondo, Cal.		Fruitvale, Cal.	San Francisco, Cal.		Oakland, Cal.		Handley, Tex.		Inspiration, Ariz.	
Type of boiler	B. & W.	B. & W.	Parker	B. & W.	B. & W.	Parker	Parker	B. & W.	B. & W.	Stirling	Stirling
Type of oil burner	Hammel	Hammel	Owens	Hammel	Hammel	S. & P.	Owens	Hammel	Hammel.	...	...
Heating surface, sq. ft.	6,042	6,042	6,080	3,500	3,500	7,734	7,734	6,000	6,000	7,129	7,129
Date	8/11/10	9/5/10	3/18/11	2/9/12	2/10/12	2/24/12	2/26/12	6/25/14	6/27/14	5/29/15	6/4/15
Duration, hours	7	7	10	10	10	8	8	9	9	10	6
Gravity of oil at 60 deg. F., deg. Bé	13.3	12.9	15.37	...	...	14.32	14.32	25.9	26.3		
Specific gravity of oil at 60 deg. F.	...	...	0.9627								
Steam pressure (gage), lb. per sq. in.	184.7	184.9	179.7	179.6	179.8	185.3	194.2	148.5	161.3	187.4	182.3
Oil pressure at burner, lb. per sq. in.	25.4	61.6	92	...	...	50.4	126	...	...	22	9
Temperature of feed water, deg. F.	93.4	101.2	123.4	188.1	195.6	169.1	172	187.6	209.5	210.15	191
Temperature of oil at burner, deg. F.	142.3	141.7	113	...	...	96	90	...	...	187.8	186
Temperature of gases leaving boiler, deg. F.	406.2	537.5	394	433	...	370	440	426.8	507.6	470.5	389.8
Draft between damper and boiler, in. water	0.044	0.230	0.15	...	...	0.34	0.48	0.035	0.46	0.076	0.020
Draft in furnace, in. water	0.014	0.188	+0.016	...	...	0.06	0.057	0.02	0.20		
Draft in ash pit, in. water	0.046	0.471	0.02	...	...	0.06	0.08				
Degree of superheat, deg. F.	83.7	144.3	181.7	...	...	148	161.5	81.6	125.7	122.1	111.0
Per cent. of water in oil	0.4	0.6	0.6	0.11	0.12	0.7	0.5	...	...	0.49	0.49
Oil free from water per hour, lb.	1,439	2,869	1,401	812.3	1,546.3	1,872	3,043	1,312*	2,355*	1,666	905
Evaporation per hour from and at 212 deg. lb.	22,639	40,375	22,422	12,600	22,273	30,227	46,380	...	...	26,159	14,564
Evaporation per hour from and at 212 deg. per square foot of heating surface, lb.	3.75	6.68	3.69	3.6	6.37	3.9	6.0	3.56	6.18	3.67	2.04
Rated capacity of boiler, b.hp.	604	604	645	350	350	773	773	600	600	712.9	712.9
Percentage of rated capacity developed, per cent	108.6	193.8	100.8	102.8	182.6	113.3	173.8	103	179	106.7	59.2
Evaporation from and at 212 deg. per lb. oil free from water, lb.	15.73	14.07	16.01	15.53	14.43	16.14	15.24	16.27*	15.80*	15.70	16.1
Per cent. of total steam used by burner, per cent	2.4	2.13	4.16	1.41	0.97	...	...	2.03	1.97	...	
Net evaporation from and at 212 deg. per lb., oil free from water, lb.	14.35	13.77	15.35	15.31	14.28	...	...	15.94*	15.49*		
Calorific value of 1 lb. oil as received, B.t.u.	18,253	17,985	18,569	18,496	18,496	18,621	18,658	19,276	19,280	18,540	18,540
Calorific value of 1 lb. of oil free from water, B.t.u.	18,326	18,093	18,681	18,516	18,519	18,752	18,752	...	...	18,631	18,631
Efficiency of boiler and furnace, per cent.	83.3	75.4	83.17	81.4	75.5	83.53	78.86	81.9	79.5	81.76	83.74
Net efficiency (deducting steam used by burner), per cent	81.3	74.0	79.8	80.2	74.8	...	...	80.3	77.9		
Carbon dioxide (CO_2), per cent.	14.3	12.1	13.5	...	...	14.3	14.6	14.0	13.5	14.5	14.5
Oxygen (O), per cent.	1.8	6.8	...	...	...	...	...	...	...	2.3	2.1
Carbon monoxide (CO), per cent.	...	...	...	...	...	...	...	...	...	Trace	

* Oil as fired.

Table 2.*—Results of Tests on Oil-burning Boilers with Mechanical Atomizing Burners

OIL BURNER TEST—BABCOCK & WILCOX BOILER
BAYONNE, NEW JERSEY

450 horsepower—15.45 per cent. superheating surface
445 cu. ft. furnace volume.

Number of Test	41	42	43	44	45
Date of Test, 1919	Aug. 14	Aug. 15	Aug. 15	Aug. 21	Aug. 21
Type of Burner			Lodi		
No. and size of Sprayer Plate	3-47-2	3-40-2	3-40-2	3-104	3-25-1.1
Duration of Test—Hours	6	4	4	2.5	3.58
Steam Pressure—lb. per sq. In.	178.2	178.4	177.9	186.2	192.1
Steam Temperature—Deg. F.	453.6	466.9	482.3	504.9	530.4
Superheat—Deg. F.	74.7	88.0	103.6	122.6	146.1
Feed Temperature—Deg. F.	71.9	70.35	72.3	70.8	71.6
Factor of Evaporation	1.2394	1.2484	1.2549	1.2675	1.2798
Oil Pressure at Burners lb. per sq. in.	169.7	120.3	194.7	193.2	188.7
Oil Temperature at Burners—Deg. F.	250.7	252.6	248.2	258.6	232.7
Total Oil Burned—lb.	7016	6245	8171	6558	12663
Oil burned per hour—lb.	1169.8	1561	2042.8	2623.2	3537.1
Oil Burned per hour per burner—lb.	389.8	520.4	680.9	874.4	1179.0
Temperature of Flue Gases—Deg. F.	413	443	485	523	6.16
Temperature of Room Deg. F.	81	86	89	93	97
Draft Inside Damper—Inches of Water	.24	.40	.71	.77	.72
Draft in Furnace—Inches of Water	.14	.18	.24	.21	.17
Air Pressure in Duct—Inches of Water	.22	.44	.75	1.82	3.64
CO_2 } 3rd Pass	13.77	13.71	13.53	13.41	13.70
O } 3rd Pass	2.26	2.30	2.61	2.94	2.65
CO } 3rd Pass	.05	.06	.04	0	.05
Total Water Fed Boiler—lb	87515	77733	98928	77098	139722
Total Water From and at 212 deg.—lb.	108466	97041	124145	97722	178816
Water Per Hour From and at 212—lb.	18078	24260	31036	39089	49947
Actual Evaporation per lb. of oil—lb.	12.47	12.45	12.11	11.76	11.03
Equiv. Evaporation fr. and at per lb. of oil—lb.	15.46	15.54	15.19	14.60	14.12
Water fr. and at per sq. ft. of H. S.	4.02	5.39	6.89	8.69	11.10
Horse Power Developed	524.0	703.2	899.6	1133.0	1447.7
Per Cent. Rating	116.4	156.3	199.9	251.8	321.7
Efficiency—Per cent	81.80	81.41	79.05	77.91	75.29
Gravity of Oil-Baumé	15.3	15.3	19.5	15.3	17.0
Per cent. Moisture in Oil	.				
B.t.u. Per Lb. of Oil	18340	18530	18650	18560	18200

* From paper by Darrah Corbet published in the August, 1920, Journal of the American Institute of Electrical Engineers.

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TUBES DUSTED AFTER THE FIRST SET OF READINGS AND BOILER OPERATED FOR SIX DAYS WITHOUT FURTHER USE OF THE SOOT BLOWERS. OIL FUEL

Date, 1916	Draft		Capacity				
	Furnace, in.	Third pass, in.	Per cent. of rating, per cent.	Breeching temp., deg. F.	CO_2, per cent.	O, per cent.	CO, per cent.
Mar. 24, A.M.	0.08	0.35	145	650	15.5	0.5	0.0
Mar. 24	Tubes dusted						
Mar. 24, P.M.	0.07	0.33	143	574	14.5	2.2	0.0
Mar. 25	0.08	0.34	141	594	14.7	1.6	0.0
Mar. 27	0.07	0.37	144	607	15.0	1.05	0.0
Mar 28	0.08	0.43	146	635	14.1	2.0	0.0
Mar. 29	0.08	0.37	142	637	14.9	1.05	0.05
Mar. 30	0.08	0.40	142	668	15.2	1.0	0.05

may be shut tight, thus materially reducing the loss of heat to the chimney during this period. Tests have shown that the oil required to keep a boiler hot in this manner amounts to from 1.5 to 3 per cent. of the quantity of oil required to operate the boiler at its rated capacity. The tighter the damper the less oil will be required. In a boiler with good tight dampers the drop in steam pressure will be not more than 10 or 15 pounds per hour.

The best recorded results of oil burning plants are those obtained at the three Arizona plants described in C. R. Weymouth's paper entitled "Economy of Certain Arizona Steam Electric Power Plants Using Oil Fuel," presented at the June, 1919, meeting of the American Society of Mechanical Engineers. This paper shows that a plant having 6000 K. W. turbines operating with a steam pressure of 175 lb. and 100 deg. superheat and 2 to 3 in. absolute back presure generates from 257 to 294 kilowatt hours per barrel of oil. Another plant, having 250 lb. steam pressure and 150 deg. superheat and 1.66 in. to 2.14 in. absolute back pressure, generates from 287 to 326 kilowatt hours per barrel of oil. Monthly boiler room records of the former plant show combined net efficiency of boilers and economizers of approximately 83 per cent., which corresponds to 79½ per cent. net efficiency for the boilers alone.

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The following additional data on the economies and method of operation at one of these plants, namely the New Cornelia Copper Company, was compiled by E. A. Rogers who was for two years Chief Engineer in charge of the plant:

ECONOMIES IN THE NEW CORNELIA COPPER COMPANY PLANT

Type of Equipment.—The generating equipment consists of two General Electric turbo-generators rated at 7500 kilowatts at 80 per cent. power factor. The generators are 3-phase, 60-cycle, 2300-volt at 1800 r.p.m. The turbines are designed to use steam at 240 pounds pressure and 140 degrees superheat.

The condensing equipment consists of Wheeler surface condensers, Wheeler dry vacuum and centrifugal hot well pumps. Duplicate hot well pumps, one motor driven and one turbine driven, are provided for each condenser. Each condenser has its own motor-driven centrifugal circulating water pump for supplying the cooling water from the pond.

The boiler equipment consists of five Stirling boilers rated at 825 h.p. each. They are operated at a pressure of 255 pounds and as normally operated deliver steam at 110 degrees superheat. Being equipped with asbestos insulation and steel casings, radiation and air infiltration losses are reduced to a minimum. The gases leaving the boilers pass through Green fuel economizers which are arranged so that with four boilers in operation, each economizer takes care of two boilers. Natural draft is used and as the air temperatures are high in summer, a stack 220 feet high is necessary. This stack is of reinforced concrete, double for part of its height, and the tapered construction gives not only good mechanical proportions but a pleasing appearance. Feed water heaters are provided which utilize the steam from the rotative dry vacuum pumps and feed water pumps to heat the feed water before it goes to the economizers. The pumps for handling the feed water are turbine driven centrifugals, the exhaust steam from the turbine going to the feed heater.

BOILER EFFICIENCY

The firing of the boilers is controlled by a Moore automatic regulating system which is one of the main contributing factors in the maintaining of the high boiler efficiencies obtained at this plant. The furnaces are of the Hammel-Peabody type and Ham-

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mel oil burners are used. That the automatic control of the dampers produces excellent results is shown by the boiler efficiencies maintained. With the dampers properly set the regulator maintains a draft condition which gives almost perfect combustion as shown by the gas analyzer, the per cent. of CO_2 in the gases leaving the boilers being maintained between limits of 13.0 per cent. and 14.5 per cent. The following figures show the boiler room results for the month of December, 1919:

Load on boilers in per cent. of rating	91.0%
Gross boiler plus economizer efficiency	84.5%
Gross boiler efficiency	80.0%
Amount of steam used for atomizing oil	1.0%
Net boiler efficiency	79.0%

These percentages do not take into consideration the additional amount of steam used for atomizing the oil with hand firing, but refer to automatic control only. When the boilers were fired by hand the atomizing steam was supplied through a line on which there is no flow meter and consequently accurate data on this item is not available. It has been the writer's experience that with hand firing the atomizing steam will vary from 2.0 per cent. to 4.0 per cent. of the steam generated, depending on the skill and experience of the operator. Observations on the boilers at this plant would indicate a steam consumption of at least 2.0 per cent. for atomizing, which is a loss of 1.0 per cent. over the amount shown for automatic control.

With regard to the efficiency of the automatic control of the firing of the boilers as compared with hand firing, it has been found that the boiler room efficiency is 4 per cent. to 5 per cent. higher when the regulators are in operation than when firing and damper setting are done by hand. The comparison has been made at times when the regulators were taken out of service for inspection and general overhauling, and the firing and regulating of dampers done entirely by hand.

OPERATION OF AUTOMATIC REGULATORS

The automatic regulators do not relieve the fireman of all responsibility in connection with the firing, however, for it is necessary to watch the burners carefully to see that they are working properly. It occasionally happens that a burner becomes choked or partially so with carbon, which reduces the size

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of the flame. This results in a considerable amount of excess air in this particular boiler and, unless the burner is changed, may result in a serious loss in efficiency. The fireman must be trained, furthermore, to understand the value of the gas analysis in its relation to efficient operation. This is necessary because it is essential for them to make adjustments on the main damper regulator to compensate for changes in draft conditions due to atmospheric changes. Thus, for instance, if the dampers are set during the day to give proper combustion and at night the air cools down considerably, the stack draft increases and more air will be drawn into the boilers than is necessary. When the fireman finds that this condition exists, by turning a hand wheel and tightening a spring on the damper regulator the dampers are set for the new draft conditions and automatically take care of changes in load as before. It has been found in this plant that this adjustment for changes in draft due to atmospheric changes is a very important part of maintaining high economies.

MAINTAINING ECONOMY

In order to obtain a maximum of effort on the part of the operating force, in maintaining the proper economies, the economy is worked out for each watch and posted daily. The rivalry thus promoted has aided materially in maintaining good operating results in the plant. The results thus posted every day are:

Kilowatt-hours generated.
Barrels of fuel oil used.
Pounds of water evaporated.
Kilowatt-hours per bbl. of oil.
Pounds of water per pound of oil.

The equipment used to obtain this and other data is as follows:

For accurately measuring the fuel oil two carefully calibrated tanks are used, it being possible to obtain the oil consumption within less than one-half of 1 per cent.

A Cochrane V notch meter in the feed water heater and a Venturi meter on the feed water line to the boilers give the amount of water evaporated. A General Electric steam flow meter gives the amount of steam used for atomizing the fuel oil.

The boilers are provided with draft gauges, superheated steam thermometers, flue gas thermometers and Uehling CO_2 recorders.

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Thermometers are also provided on the economizers for gases entering and leaving, and water entering and leaving.

Readings are taken hourly and logged, of all of these instruments, and thus it is an easy matter to locate any losses of efficiency which may occur.

As the town of Ajo is situated in the desert, extreme temperatures prevail in the summer months and consequently much lower economies are obtained during this period. The circulating water for the condensers gets very warm, this resulting in a material loss in vacuum. The vacuum is also influenced by scale

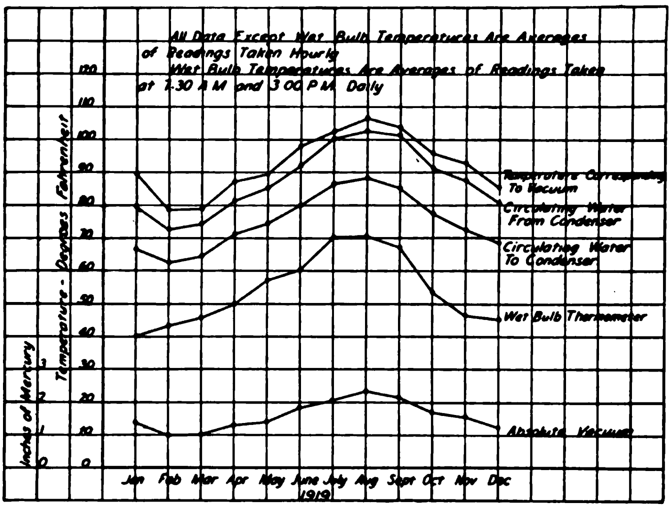

FIG. 211.—In the summer the circulating water for the condensers gets very warm, and the vacuum is influenced by a scale forming in the condenser tubes. Much lower economies prevail during the period of these extreme temperatures at New Cornelia Copper Co.'s plant, Ajo, Arizona.

forming in the condenser tubes, but this loss can be easily remedied by cleaning the tubes out with a proper cleaner. To determine when it is necessary to remove the scale, tests are run, from which a heat balance for the condenser can be worked out. From this heat balance the heat transfer through the tubes can be determined and when this figure falls below a certain point the tubes are cleaned.

In spite of the very low vacuum during the summer months, fairly good efficiencies can be obtained, as shown by the load efficiency curve. During February, March and April the load was very low, which accounts for the low efficiencies obtained,

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when the vacuum was relatively high. During the first twelve days of May the turbine was run with practically no load except the plant auxiliaries, but the plant was in readiness to pick up at any time the full load. This accounts for the low efficiency shown for this month, the average for the last nineteen days being 303 kw.-hr. per bbl. In June the load was increased practically to normal conditions, and these conditions held until in December the load was again reduced with a corresponding drop in efficiency. All efficiencies shown or given in terms of kw-hr. per bbl. of oil are net, all power used in the plant being deducted from the total generation before figuring the efficiency.

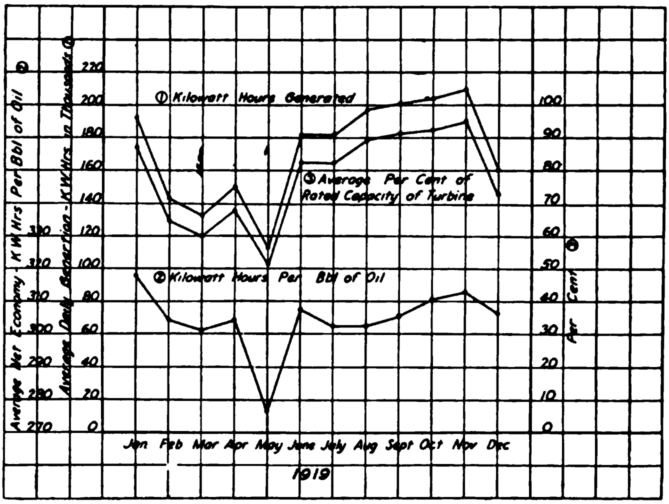

Fig. 212.—Economy curve showing the relation between the load and B.t.u. supplied per kw. hour during the year 1919, at New Cornelia Copper Co., Ajo, Arizona.

The best economy obtained by the plant during 1919 was for the last eleven days of January. During this period the load averaged 8085 kw. The temperature of the circulating water was 68.8 degrees, and the vacuum 1.28 in. absolute. The average net economy was 323.1 kw.-hr., per barrel. Individual days showed as high as 327 kw.-hr. per barrel, and individual shifts better than 330 kw.-hr. per barrel, but these results are not as accurate as those taken over a longer period. The economy curve showing the relation between the load and B.t.u. supplied per kilowatt hour generated is calculated from the results actually obtained in the plant during the year 1919. In some cases on the low loads correction was made for circulating water temperature, as the loads occurred when the water was about 75°. At the higher ratings, however, the curve corresponds to the actual existing temperatures and conditions.

Notwithstanding the fact that with a cooling pond and spray nozzles it is necessary to pump the circulating water against a considerably greater head than should ordinarily be necessary in a sea-water plant, the power used by the auxiliaries compares quite favorably with plants of the latter type. The average amount of power used by the auxiliaries during 1919 was 3.2

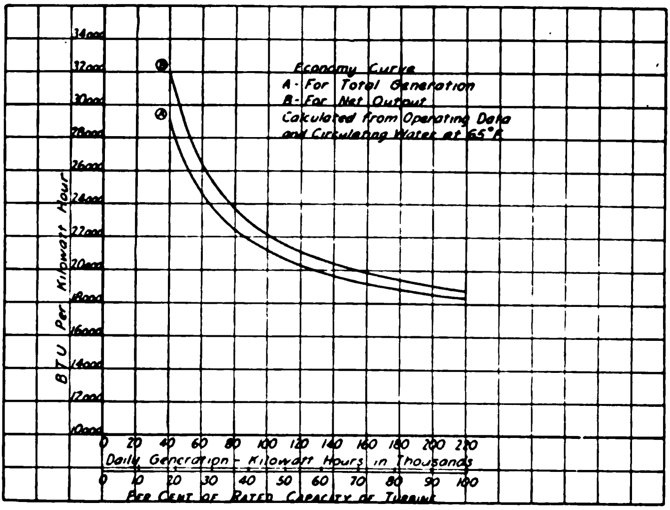

FIG. 213.—Load efficiency curve at plant of New Cornelia Copper Co. All efficiencies shown or given in terms of kw. hour per barrel of oil are net, all power used in the plant being deducted from the total generation before figuring the efficiency.

per cent. of the total generation and under normal full load this figure is 2.75 per cent. The auxiliaries include the circulating water and hot-well pumps, air washers for generators, economizer scrapers, motor-generator for switch control power, and station lights.

MOTOR-GENERATOR SETS

In addition to the generating equipment the plant contains the motor-generator sets which furnish the direct current for the electrolytic deposition of copper. There are four of these sets, three being in continuous operation and the fourth a spare. Each set consists of a 2400-h.p. synchronous motor direct-con-

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nected to an 850-kw. direct current generator on each end of the shaft. These generators are rated at 5000 amperes at 170

FIG. 214.—Exterior view of power plant of New Cornelia Copper Co., Ajo, Arizona, showing tapered construction of reinforced concrete stack 220 feet high.

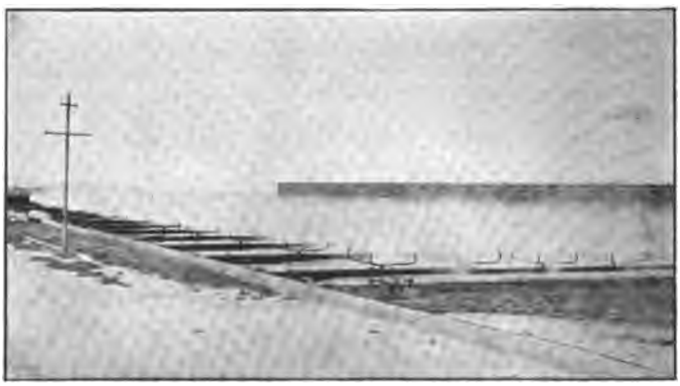

FIG. 215.—This cooling pond measures 150 by 400 feet and is one of the largest in the country. Each condenser in the plant has its own motor-driven centrifugal circulating water pump for supplying the cooling water from the pond.

volts, and as normally operated carry the full load current at 160 to 170 volts. Thus with three units in service there is a total

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load of 30,000 amperes at 160 to 170 volts on the d.c. side. As this load is continuous for 24 hrs. a day it gives the plant a very high load factor, at times 94 per cent. and averaging about 90 per cent. This aids materially in maintaining high economies. The motors being synchronous make it possible to keep the power factor at 98 to 99 per cent., which has made it possible to carry peak loads of 10,000 kw. on one of the main generators.

Fig. 216.—Turbine room, New Cornelia Copper Co., Ajo, Ariz., showing an auxiliary exciter unit in the foreground, one of the turbo-generators, and four motor generator sets in the background. Three of these sets are in continuous operation, and carry a total load of 30,000 amperes at 160 to 170 volts on the d.c. side, 24 hours a day.

Accompanying illustrations show the power plant, cooling pond and interior of the plant. The cooling pond is one of the largest in the country, being 150 ft. wide by 400 ft. long. The turbine room is shown, an auxiliary exciter unit made by the Allis-Chalmers Company appearing in the foreground, then one of the turbo-generators, and in the background the four Westinghouse motor-generator sets.

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CHAPTER XLII

MISCELLANEOUS OIL BURNING TESTS

TESTS AT LONG BEACH STEAM PLANT

General.—The following report covers a series of tests conducted on the boilers at the Long Beach Steam Plant of the Southern California Edison Company, and was compiled by H. L. Doolittle, steam power plant specialist for the company. These tests extended over a period of approximately 10 mos.

Description of Plant.—The Long Beach Steam Plant is located at the Long Beach Inner Harbor, Long Beach, California, and consists of the following equipment:

No. 1 Unit—placed in operation August 20, 1911, consisting of one 12,000-kw. General Electric Co. vertical Curtis Turbo-generator running at 750 r.p.m.; eight 777.5-h.p. Stirling boilers; duplex feed pumps; turbine driven exciter; engine driven centrifugal circulating pump; all auxiliaries steam driven excepting spare hot well and oil drain pumps which are motor driven.

No. 2 Unit—placed in operation February 2, 1913, consisting of one 15,000-kw. General Electric Co. vertical Curtis turbo-generator running at 750 r.p.m.; eight 777.5-h.p. Stirling boilers; steam turbine driven centrifugal feed pumps; turbine driven exciter; steam turbine driven centrifugal circulating pump; all auxiliaries steam driven excepting spare hot well and oil drain pumps which are motor driven.

No. 3 Unit—placed in operation March 30, 1914, consisting of one 20,000-kw. General Electric Co. vertical Curtis turbo-generator running at 750 r.p.m.; eight 850-h.p. Stirling boilers fitted with Sturtevant economizers; motor driven exciter; centrifugal boiler feed pumps; centrifugal circulating pump; all auxiliaries being motor driven with the exception of the fuel oil pumps which are duplex steam pumps.

All boilers are equipped with four Hammel oil burners and B. & W. U tube superheaters.

All units operate at the normal pressure of 225 lb. gauge and 125°F. superheat.

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Method of Testing.—In general all tests extended over a period of from 7 to 8 hrs., all observations being taken every 15 min.

All temperatures, except the stack temperatures, were taken with mercurial thermometers calibrated by means of a standard thermometer.

All pressure gauges were calibrated with the dead weight gauge tester. Small differences in pressure and small pressures were measured with mercury U tubes.

Stack temperatures for boiler tests were measured by means of a resistance thermometer consisting of copper wire installed in the path of the flue gases. The resistance of this wire was obtained by taking a cold reading after the wire was installed. Temperatures were calculated from the variation in the resistance of the wire. It was hoped that this method of measuring the flue gas temperature would be very accurate inasmuch as the wire was strung back and forth across the flue so as to measure the average gas temperature. It was found, however, that this method of measurement gave more or less erratic results which must be due to the varying velocities of the gas flowing past the wire. It appears that in general this method would give results that are too low, on account of the fact that the gas in the upper part of the flue is the hottest and travels with the greatest velocity. It is, however, believed that this method gives as accurate results as can be obtained by means of a thermometer or a pyrometer as either of these methods would be subject to the same error due to varying gas velocities. The ideal method of measuring gas temperature appears to be some system by which the product of the velocity of the gas by its temperature could be averaged for various sections of the flue.

Flue gas analyses were made with a portable Orsat flue gas testing apparatus.

Fuel oil was weighed on a 5000-lb. Fairbanks-Morse portable platform scale calibrated in place against standard weights.

The steam required for atomizing the oil in the burners was measured by installing an orifice between two flanges in the steam pipe. The drop in pressure through this orifice was measured by means of a mercury U tube. The orifice was calibrated by condensing the steam flow through the orifice during a given period of time in a tank of water and measuring the increase in weight.

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Four samples of oil were taken. These samples were composee of small amounts of oil taken approximately every hour from ths oil being fed to the boilers. The analyses of the oil sampled were made by Wrana King & Company, Chemists, Los Angeles.

The water evaporated during the boiler tests was obtained by correcting the total water weighed to the boilers for feed pump leakage, amounting to 75 lb. per hour.

In all tests the water evaporated was corrected for the height of the water in the boiler gauge glasses, the deduction amounting to 670 lb. per inch.

BOILER TESTS

A series of nineteen boiler tests was conducted on the boilers of the No. 3 unit in order to determine the most economical load at which the boilers should be operated. In the test on a single boiler, one boiler of a battery was used with the other boiler of the battery shut down. This necessarily increased the radiation losses of the boiler being tested but it was much easier to conduct the test in this manner on account of the fact that one economizer is installed to take care of two boilers. The tests made on two boilers were made with both boilers in one battery operating.

In all of the boiler tests complete readings of the temperatures, pressures and water fed to the boilers were taken so as to determine the efficiency of the boiler alone and also the combined efficiency of the boiler and economizers. All of the boilers tests, excepting that made on March 9, 1915, were made with Hammel oil burners.

The first ten tests were conducted on the boiler with the furnace as originally installed by the manufacturer. The remaining tests were made on a boiler with the furnace rebuilt to accommodate three instead of four burners. In addition to rebuilding the furnace, the boiler for the last series of tests had a slight modification in the baffling of the rear pass. This modification consisted in removing two rows of 12-in. tile directly in front of the damper opening.

In this series of boiler tests we endeavored to determine the efficiency of both the boiler and economizer at different loads, the maximum and minimum loads that the boiler was capable of carrying, the effect of hot water entering the economizer, the efficiency of the boiler during a swinging load, and also the oil

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required to bring the boiler up to header pressure after being shut down for several hours.

Boiler Efficiency.—Fig. 217 shows the gross boiler efficiency obtained for the various tests plotted against combined economizer and boiler horse power. These curves show that the efficiency is practically constant between 55 per cent. and 130 per cent. of boiler rating. The curves drawn through the points represent a fair average of all the tests. Any great departure from these curves is generally for an obvious reason; for instance, it is seen that the efficiency for the swinging load, also for the test of the boiler starting up cold, fall very far below

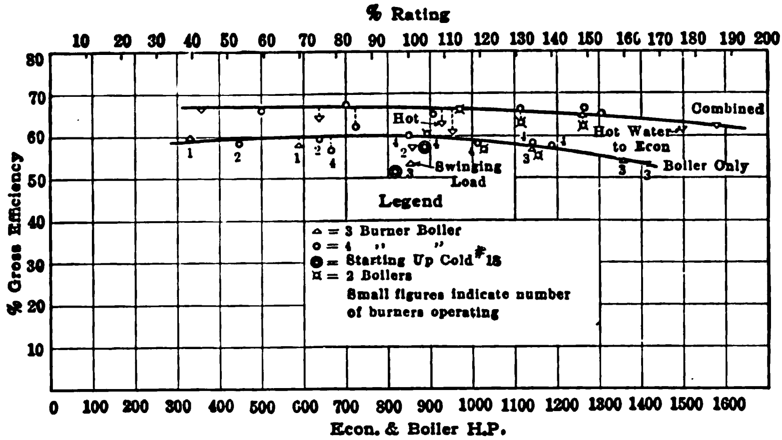

FIG. 217.—Boiler tests at the Long Beach Plant of the Southern California Edison Company, showing the relationship of gross efficiency and total rating at which boiler is operated.

the average efficiency. The points representing the tests during which hot water was fed to the economizer also result in a low efficiency. Two of the three tests on two boilers fall about 3 per cent. below the average efficiency. It is expected that the combined efficiency for two boilers operating on one economizer would be less than for a single boiler operated on the same economizer on account of there being just half the economizer surface available per boiler. It is also noticeable that the tests on the low loads which were made with fewer burners operating, gave better efficiency than those with a larger number of burners. This would indicate that it would be more economical to operate the boilers on light loads with either one or two burners instead of with all four burners.

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The tests of the boiler equipped with three burners gave practically the same efficiency as those conducted on the boilers equipped with four burners. It was also possible to obtain the same maximum capacity with three as with four burners. This would indicate that there would be some advantage in having future boilers equipped with three burners as it would make one less burner per boiler to be kept in repair. There is a possible disadvantage in reducing the number of burners, however in that the four burner furnace would permit one burner becoming inoperative without reducing the capacity of the boiler or necessitating the immediate installation of a good burner.

The curve showing the combined efficiency of boiler and economizer shows the tendency of the economizer to flatten out the efficiency curve at high load. This largely compensates for the rapid decrease in boiler efficiency as the load is increased. In general the economizer adds from 9 to 12 per cent. to the boiler efficiency and at the same time increases the boiler capacity approximately 12 per cent.

After conducting the first ten tests on the boiler as originally installed, it was found that the loss in draft in passing over the rear baffle in front of the damper opening was .2 in. at 140 per cent. rating. This had the effect of reducing the available draft on the furnace and thus limiting the boiler output. It was, therefore, decided to remove the two top rows of rear baffle tile leaving only one row above the damper opening. After making this change it was found that the available draft was greatly increased thereby making possible much higher loads on the boiler.

Figure 218 shows the draft required at the damper for the various loads on the boiler after removing the two top rows of the rear baffle. From this curve it is seen that a definite draft is required for a given load on the boiler, or for a given amount of oil burned per hour. It is rather surprising to note that the boiler can be operated up to 70 per cent. of rating with a positive pressure on the damper.

Stack Temperatures.—Figure 219 shows the stack temperatures obtained for the gases out of the boiler and also out of the economizer. These curves show only the result obtained during the tests on the reconstructed boiler. The temperatures taken during the first ten tests were found to be unreliable on account of the location of the wires used in measuring the temperature.

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The curves show that the temperature of the gases leaving the boiler vary from 400° to 700°, while the temperatures leaving the economizer are between 180° and 265°. It is noticeable that the economizer has a tendency to maintain the stack temperature at a fairly constant value. As stated before, it is questionable if the readings taken for stack temperature represent the actual values. It is, however, believed that they give a good indication of the variation in temperature as well as the actual amount.

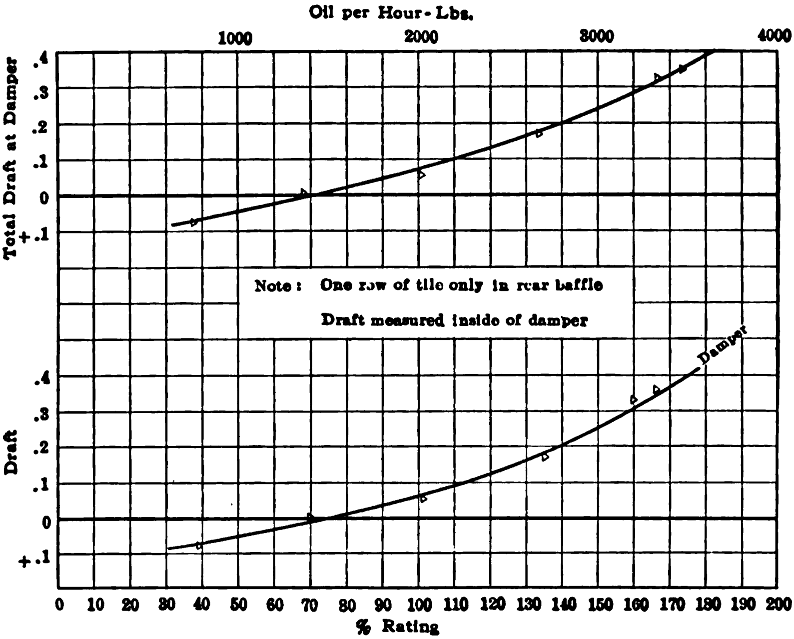

FIG. 218.—A relationship of total draft at the damper with the oil consumed per hour and also compared with the percentage of rating at which the boiler is operated. Note that the draft is measured inside of the damper.

Oil Consumed.—Figure 220 shows the oil burned for different loads on boiler and economizer. It is seen that these curves are practically a straight line passing through zero between 30 per cent. and 150 per cent. of rating on the boiler. This, therefore, indicates that the boiler operates at a practically constant efficiency between these loads. This agrees with the result shown on Fig. 217.

This matter of constant efficiency over a wide range of load has a practical bearing on the operation of the plant in that it

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would enable the plant to be operated at light loads on several boilers. The boilers would then be ready to pick up additional load on short notice.

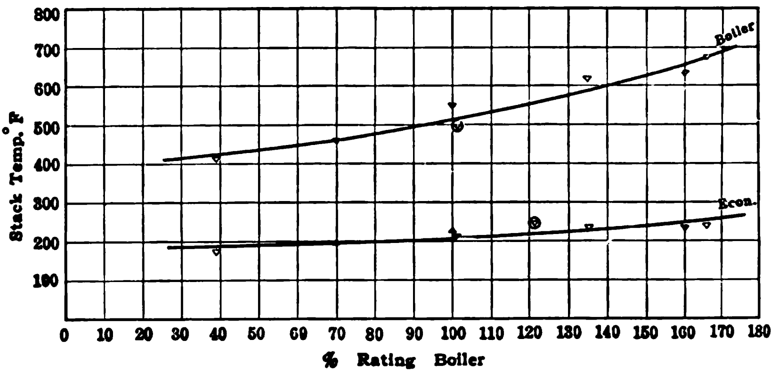

FIG. 219.—Stack temperatures in their relationship with the rating at which the boiler is operated, both with and without economiser.

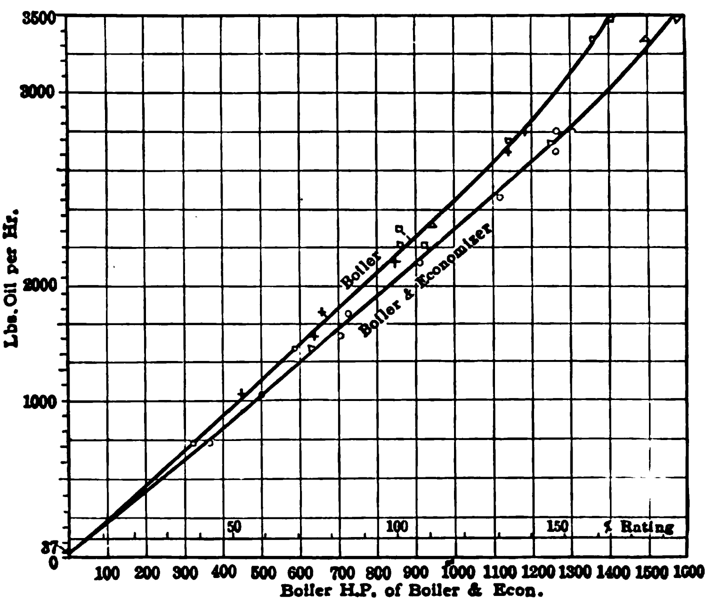

FIG. 220.—The pounds of oil per hour shown in relationship with boiler h.p. of boiler and economiser combined.

Steam to Burners.—The steam used by the burners was accurately measured by means of calibrated orifices, the difference in pressure across the orifices being measured by mercury U

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tubes. After making several attempts to find some relation between the steam required for atomization and oil burned or the load on the boiler, it finally appeared that the steam required was practically proportional to the number of burners operating.

Figure 221 was therefore prepared to show the amount of steam per pound of oil fired in relation to the number of pounds of oil burned per hour per burner. The curve shown is an equilateral hyperbola which would represent a constant amount

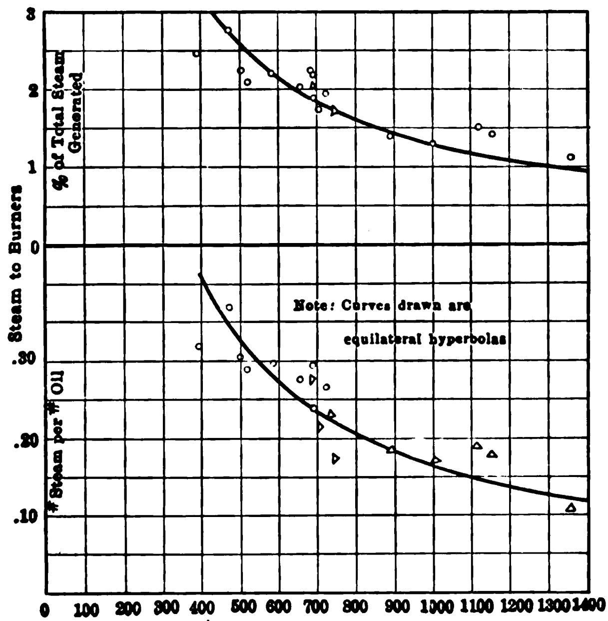

FIG. 221.—Steam to burners utilized in furnace operation shown in relationship with the oil per hour per burner. Note the interesting result obtained is that the curves are thus shown to be equilateral hyperbolas.

of steam used per burner regardless of the amount of oil burned. It is seen that the points approximately follow this curve. The amount of steam varies from 0.11 lb. to 1 lb. of oil, when the burner is handling 1400 lb. of oil per hour, to 0.4 lb. of steam to 1 lb. of oil with the burner handling 400 lb. of oil per hour. From this it is seen that a saving in steam for atomization could be affected by operating the boiler with a smaller number of burners.

Pressure Loss in Superheater.—In order to determine the pressure loss through the superheater a mercury U tube was

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connected across the two superheater headers and observations were taken at frequent intervals for one hour periods. Figure 222 shows the results of these tests and from this curve it is seen

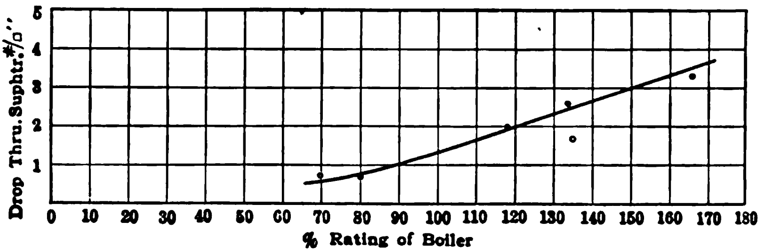

FIG. 222.—Pressure loss through the superheater in its relationship with the rating of the boiler.

that the drop across the superheater varies uniformly with the load on the boiler. The drop at 170 per cent. of rating was found to be only 3.6 lb. per sq. in.

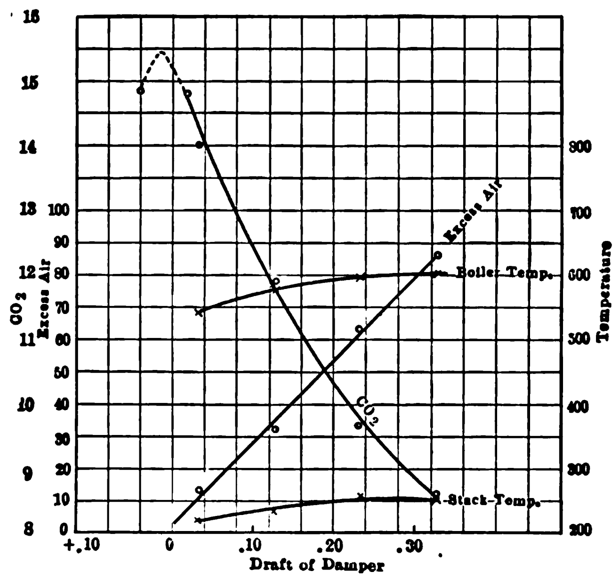

FIG. 223.—A relationship showing the varying draft of a boiler when operated at 105 per cent. rating in its relationship with draft at the damper and the temperatures of the stack.

Varying Draft.—Four 2 hr. tests were made on one boiler operated at a constant load of about 105 per cent. of rating. In these tests the total draft at the damper was varied from zero to 0.3 in. and the effect of this varying draft on CO_2 and stack temperatures was noted.

Figure 223 shows the results of these tests. From these curves it is seen that CO_2, which is an indication of the completeness of combustion, varies from 8.5 per cent. for the 0.32 in. draft to 14.7 per cent. for the 0.02 in. draft. From Fig. 218 we find that the draft required for 105 per cent. rating is .07 in., this draft corresponds to 13.2 per cent. CO_2 from the curves on Fig. 223. The average of the CO_2 obtained on all of the tests for the

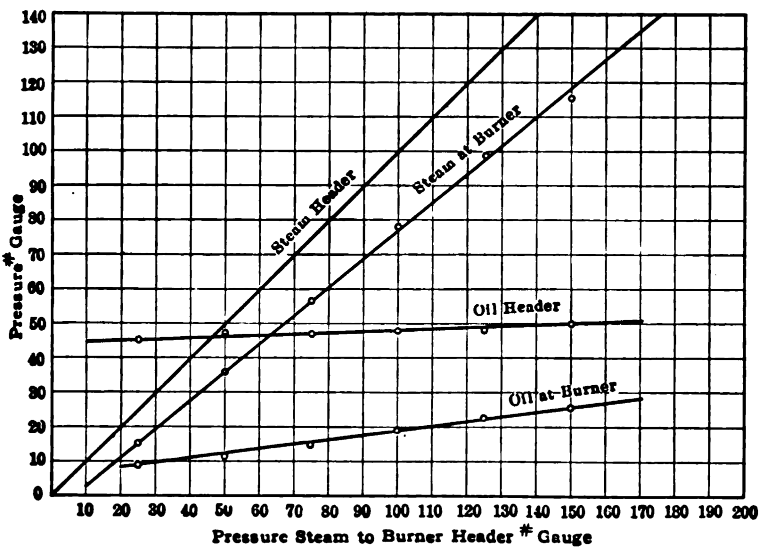

FIG. 224.—A relationship showing the effect of varying the steam header pressure with the steam and oil pressures at the burner.

boiler equipped with four burners was 14.0 per cent. and for the tests of the boiler equipped with three burners, 13.4 per cent. It is therefore seen that the curve obtained for the tests on varying draft agree very closely with those obtained during the other tests.

It is also apparent from these tests that the excess air varies directly with the draft. The excess air under normal operation at this load should amount to 20 per cent. corresponding to 0.07 in. draft. The excess air will, however, be increased to 80 per cent. by increasing the draft to 0.30 in. This gives a good indication of the unnecessary loss that would result from operating

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the boiler at drafts greater than those required to give proper combustion.

Ratio of Oil and Steam Pressures.—A short run was made to determine the effect of varying steam pressure on the steam and oil pressures at the burner. Pressure gauges were installed on the burner side of the regulating valves so as to give the actual pressure of the oil and steam at the burner tip.

Figure 224 shows the results of simultaneous readings of the pressures on the steam and oil headers and also the pressures of the steam and oil at the burner. During this test the steam header pressure was varied while all other regulating valves were left unchanged. These curves show very clearly that with an inside mixing burner, such as the Hammel, the oil pressure at the burner is affected to a great extent by any change in the steam pressure.

The boiler test made on Jan 4, 1915 consisted of three runs made with different ratios of steam and oil header pressures. These pressures on steam and oil headers were respectively, 148 lb., 50 lb.; 122 lb., 41 lb.; 102 lb., 41 lb. The combustion during these tests appeared to be practically constant as the CO_2 varied only 0.32 per cent. There was also very little difference noted in the boiler efficiency showing that variation in oil and steam pressure ratios within certain limits, have practically no effect on efficiency.

Radiation Test.—On March 24, 1915 a 6 hr. run was made to determine the amount of oil required to keep up full steam pressure with the boiler cut off of the header. In this test a small burner having an oil slot 3/8 in. wide was used. This burner was operated at intervals as required to keep the boiler pressure within the limits of 199 lb. to 212 lb. It was required to operate the burner a total of 2 hr. and 57 min. out of the 6 hr. and there was used 223 lb. of oil or an average of 37 lb. of oil per hour. This, therefore, represents the radiation losses on boilers kept up to header pressure but not delivering any steam. It is evident that the radiation loss would be somewhat greater than this with the boiler operating under normal conditions on account of the higher furnace temperature.

Swinging Load.—On March 18, 1915 a seven hour test was made on one boiler to determine the efficiency on a fluctuating load.

During this test the load varied from approximately 90 per cent. to 140 per cent. of rating and for 42 min. during the noon

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hour the fires were shut down completely. The results of this swinging load test show that the boiler efficiency was approximately 8 per cent. lower than the combined efficiency of boiler and economizer, and approximately 6 per cent. lower than the corresponding efficiency obtained under normal operation.

Starting up Cold.—In order to determine the amount of oil required to bring a cold boiler up to header pressure, a run was made on March 13, 1915 on a boiler that had been shut down for 48 hours.

The water in this boiler just before the test was at a temperature of 148°. The boiler was brought up to header pressure in 62 min. after the time of starting and during the period 1497 lb. of oil, or 190 gal. were burned. It was also noted that the water in the gauge glass rose 7½ in. from the time of starting until the time of obtaining header pressure. The pressure in the boiler started to rise 25 min. after the beginning of the test.

TESTS ON FLOW OF OIL THROUGH BURNERS

The authors are indebted to Mr. C. R. Weymouth, Chief Engineer of Chas. C. Moore & Company, for the following description and results of tests made some years ago on the flow of crude oil through orifices and oil burners, and the relative steam and oil pressures at the burners.

Influence of Load on Pressures of Oil and Atomizing Steam in Oil Burners

Very little information has been published concerning the oil pressure to operate oil burners at different rate of firing, or the steam pressure necessary to give proper atomization with a minimum quantity of steam. In the average plant, hand controlled, the oil pressure is maintained at a constant pressure by means of a pump governor, and the supply of oil to the burners is controlled at each burner by the burner oil-throttle valve, and similarly the supply of steam to burners by the burner steam-throttle valves. In times past engineers have debated the advisability of carrying higher or lower of pressures at the pumps, as influencing the economy of firing the boiler, without stopping to think that any surplus in pressure over and above that necessary to force the oil through the burner orifice, must be overcome by the friction of the oil-throttle valve, and that unless the load on the burner

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or the rate of oil firing changes, any increase in pressure at the pump above the necessary minimum, has no effect whatsoever on the performance of the burners, or the pressure between the burner-throttle valve and the tip of the burner.

Also, it is not generally known that a comparatively low steam pressure furnishes all of the steam necessary for atomizing oil at the lighter loads, and that the maximum steam pressure at the burner, generally speaking, can be considerably less than the boiler pressure. From this fact it is apparent that unless the steam-burner throttle valves are closely regulated a large waste in steam is permissible, corresponding to the difference between the steam pressure necessary to atomize the oil at a given load, and the maximum or boiler steam pressure.

TESTS WITH OIL BURNERS

With the present-day high price of fuel, and the special effort that is now being given towards the conservation of fuel and other

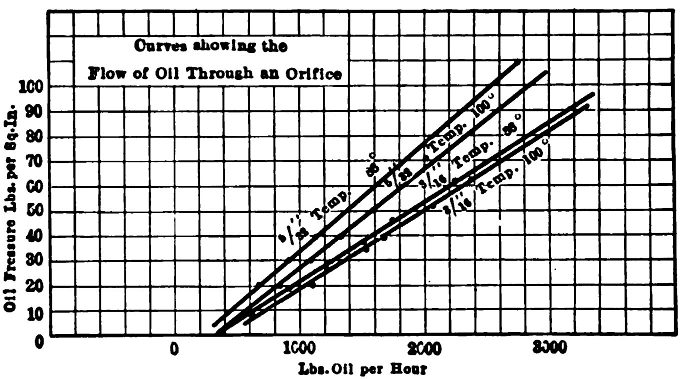

FIG. 225.—Chart showing the flow of oil through orifices of stated diameter at different temperatures. It is to be noted that for a given temperature and orifice, the rate of flow is almost directly proportionate to the pressure.

resources, it is thought that a brief review of certain test data will be of interest to fuel oil users. The test data given is taken from a report to Chas. C. Moore & Company, Engineers, dated July 1, 1907, by G. Chester Noble, then assistant professor of electrical engineering, University of California. The data from these tests was the basis of the initial design of the Moore Automatic Fuel Oil Regulating System, by Chas. C. Moore & Company, Engineers. It happens that all of the burners tested were of the

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external atomizing type, the particular burners selected being those which were available for the work, without any preference as to make of burner. The oil burned at that date was practically crude oil, which was somewhat heavier in gravity and considerably more viscous than the topped oil now generally used for fuel purposes.

Figure 225 gives data as to the flow of oil through orifices of stated diameter at different temperatures. The specific gravity and viscosity of the oil were not observed at the time. It will be seen that the plotted points fall practically on a straight line, indicating a flow of oil for a given temperature and a given orifice

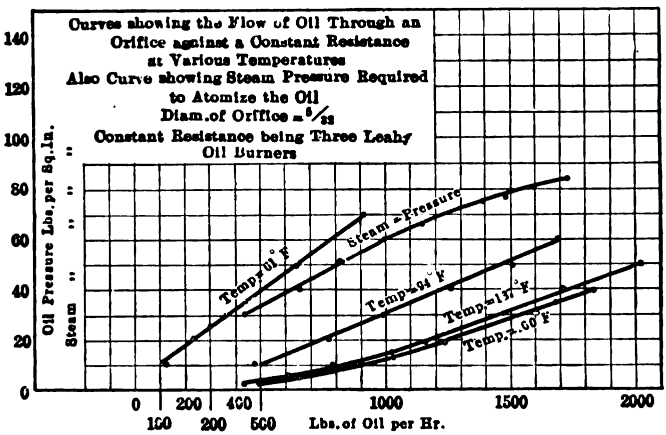

FIG. 226.—Curves showing the flow of oil through an orifice and length of piping at various temperatures. To this is added a curve showing the steam pressure required to atomize the oil. Diameter of orifice—5/32 in. A constant resistance was maintained against three oil burners connected in parallel.

nearly proportionate to the pressure; and it is interesting to note that with the oil then used a pressure gauge in the burner line so placed as to record the pressure on the oil burner orifice, was a rough index as to the rate of flow of oil, or the relative load on the boiler.

Figure 226 gives additional data, being the flow of oil through an orifice and length of piping, including also the resistance of three oil burners connected in parallel. The curve also shows the steam pressure necessary for atomizing the oil used by the burners. It will be observed that the curves of oil pressure and of steam pressure are nearly straight lines. The tests for both of the above curves were made at the University of California.

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Figure 227 gives tests showing oil pressure and steam pressure at burner at the old Third Street Plant of the Pacific Light and Power Company, Los Angeles.

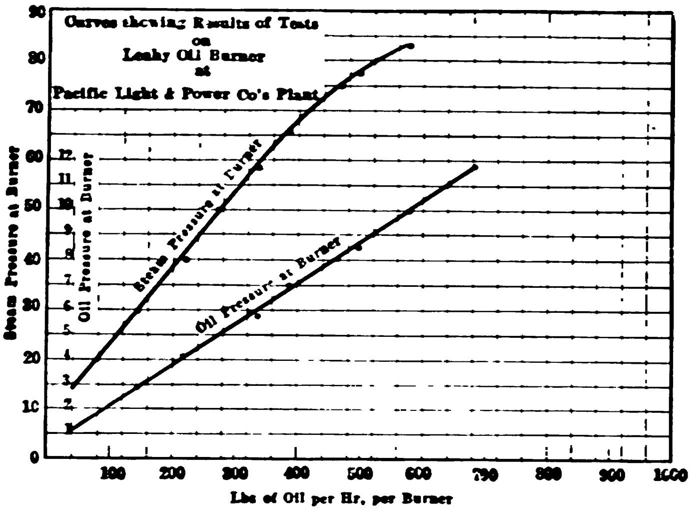

FIG. 227.—Curves showing relationships of steam and oil pressure at burner during a test on a Leahy Oil Burner at the old Third St. plant of the Pacific Light and Power Company, Los Angeles.

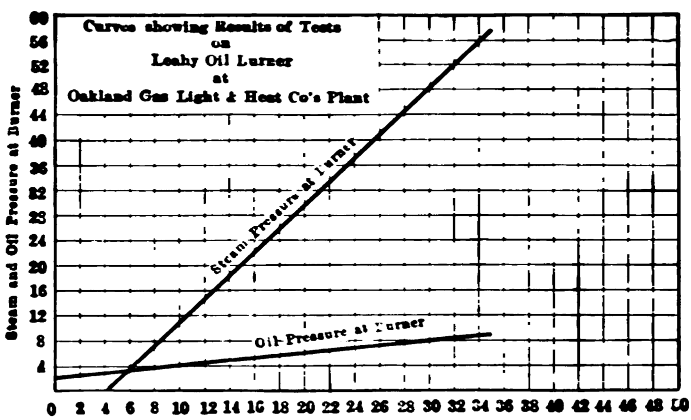

FIG. 228.—Test data showing oil and steam pressure at burner at the Oakland Gas Light and Heat Company's Plant, now station C of the Pacific Gas and Electric Company.

Figure 228 gives similar data at the plant of the Oakland Gas Light & Heat Company, Oakland, now Station "C," Pacific Gas & Electric Company.

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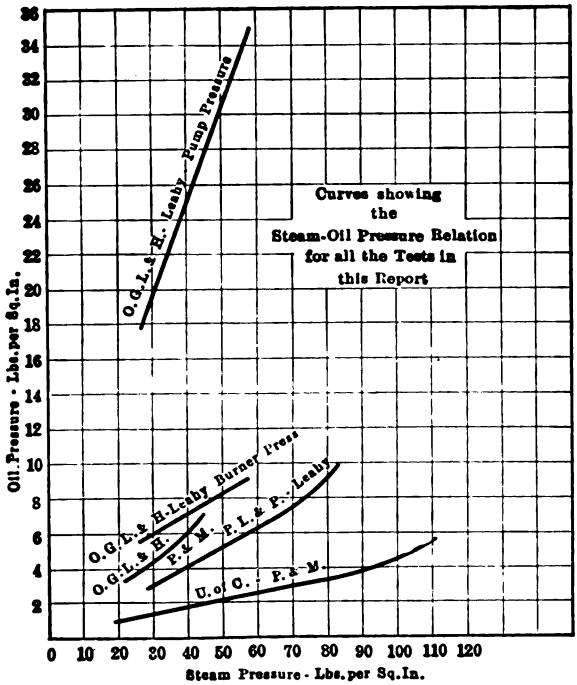

FIG. 229.—Curves showing the steam-oil pressure relation for all tests reported. Note that all these curves represent practically straight lines.

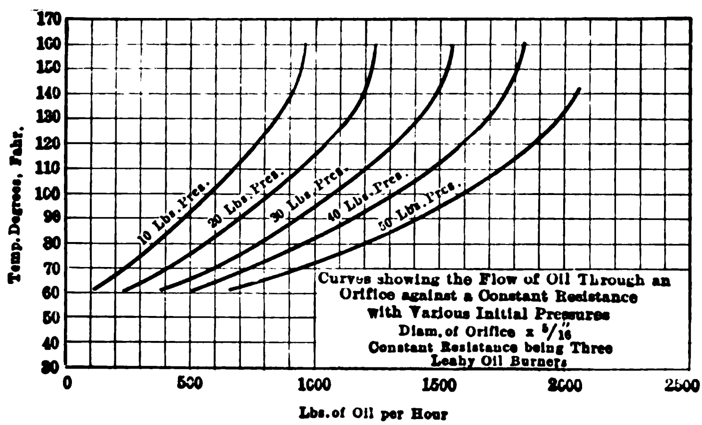

FIG. 230.—The influence of temperature on the flow of oil through an orifice against a constant resistance with various initial pressures.

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Figure 229 gives results of various tests, in which the steam-pressure and the oil-pressure on the burner are the two variables. It is apparent that the curves represent practically straight lines.

Figure 230 shows the influence of temperature on the flow of oil through an orifice, the pressure difference remaining constant.

PRACTICAL APPLICATION OF TEST DATA

To illustrate the application of the above data in the design of the Moore Automatic Fuel Oil Regulating System, now in use in a number of prominent plants in the West, it should be stated, for those not familiar with the details of this system, that it operates on the principle of central control instead of individual control of the burners and dampers. The oil burners and valves are left wide open, or nearly so, and a variable oil pressure is maintained in the oil-burner header to given an equal pressure at all burners, the pressure varying with the load, controlled automatically by throttling the supply of oil to the header, to maintain a nearly constant steam-boiler pressure. As in most plants, burners must be designed to handle very heavy oil and also to permit heavy overloads on boilers, the average pressure at the burner at normal loads is very low and but a few pounds. To build up the pressure in the oil-to-burner header, and to prevent the friction in the header causing an unequal supply of oil to all burners, a resistance, due to a diamond-ported regulating cock, is inserted in each burner-branch pipe between the main throttle valve and the tip of the burner and set to give such resistance that the pressure in the header at normal load on the boilers will be 20 or 30 lb. or thereabouts, depending upon operating characteristics, etc. Then a slight pressure drop in the header would have little effect on the unequal supply of oil to the various burners.

The oil pressure gauges connected to this header are located in the front of each battery of boilers, so that the firemen can tell approximately, from the reading of the oil-pressure gauge, the relative rate of firing of boilers.

A low pressure steam header is similarly connected to all burners, but generally without the diamond ported valve as a resistance, the pressure being high enough without this resistance. The supply of steam from the main boilers to the low pressure burner header, and its pressure, are controlled by means of a special throttle valve, generally known as a chronometer valve,

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and this chronometer valve is in turn controlled by a steam-to-burner regulator actuated by the variable oil pressure in the oil-to-burner main.

If the curve of steam and oil pressure, as mentioned above, is a straight line, then the steam pressure is equal to the oil pressure multiplied by a coefficient plus a constant. At one plant this relationship was found to be such that the steam pressure at the burner was equal to the oil pressure times three, plus thirty; thus at rating the oil pressure was 20 lb. and the steam pressure was 90 lb.; at 50 per cent. overload the oil pressure was 30 lb. and the steam pressure was 120 lb.; at half the load the oil pressure was 10 lb. and the steam pressure was 60 lb.

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CHAPTER XLIII

PRESENT STATUS OF OIL BURNING POWER PLANT DESIGN

The present shortage of hydro-electric power on the Pacific Coast has created an unusual situation in that the demand for power is so great that steam plants intended originally to act merely as standby installations and to assist the hydro-electric

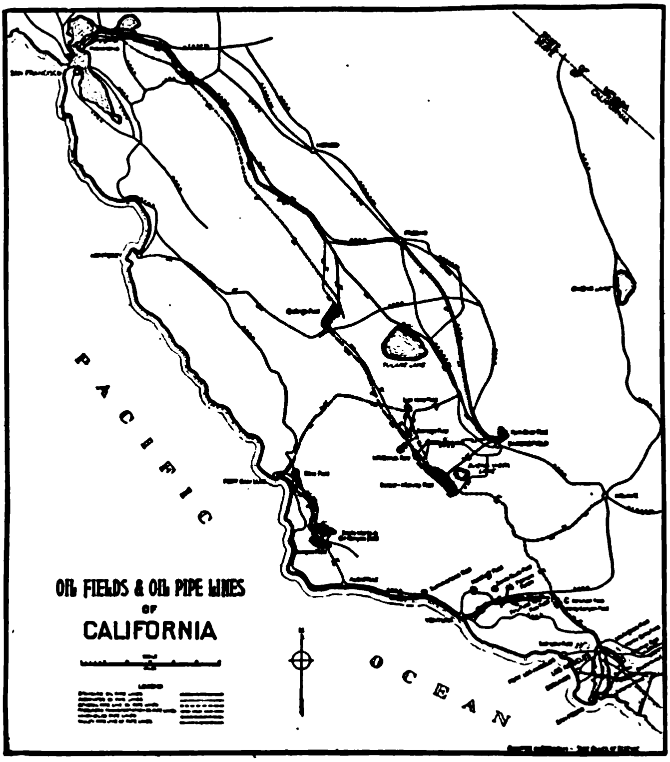

FIG. 231.—Oil fields and oil pipe lines of California.

systems in case of trouble are now being operated at full capacity and carrying a large proportion of the total load. Other steam plants are being planned, and those already under way are being

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rushed to completion so as to tide over the emergency fast becoming acute.

Fuel.—California oil is the fuel that is used almost exclusively, along the Western Coast. However, unless its production is greatly increased, in the next few years Mexican oil will be introduced into California as it has been in Arízona, Texas and the Atlantic Coast. Mexican oil is generally much heavier, dirtier and more viscous than California oil. Oil in the Panuco

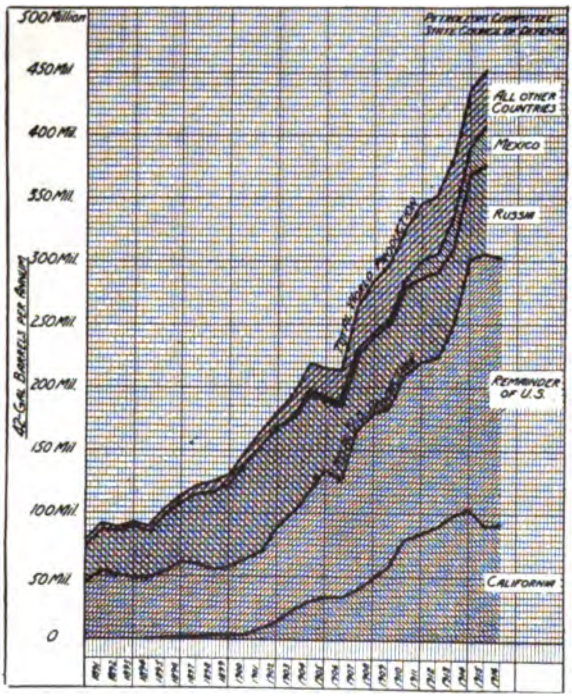

FIG. 232.—Graphical display of petroleum production.

field runs in gravity about 12° Baumé and is so viscous that it cannot be unloaded from a car without being heated and must be kept up to a temperature of 120°F. (50°C.) in the pipes in order to keep it flowing. It is therefore necessary to use large pipes, covered on the outside and containing internal heating pipes through which steam or hot water is passed. Mexican oil usually contains a large proportion of silt, and it is necessary to provide strainers at both the suction and discharge of pumps as

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well as at the burners. To burn properly it must be heated up to 190°F. (90°C.) and in some cases even as high as 250°F. (110°C.)

Natural gas is now being used in Bakersfield and in Los Angeles. This is the only available fuel that is superior to fuel oil. It has

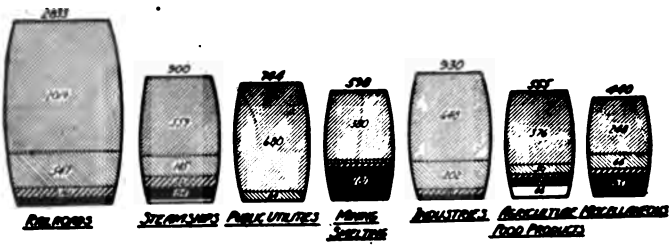

FIG. 233.—Comparative uses of crude petroleum.

all the advantages of oil, and in addition does not require atomizers or bulky storage tanks if it is piped direct from the wells.

Coal is used quite extensively in the Pacific Northwest, both in pulverized form and stoker-fired. In California, however,

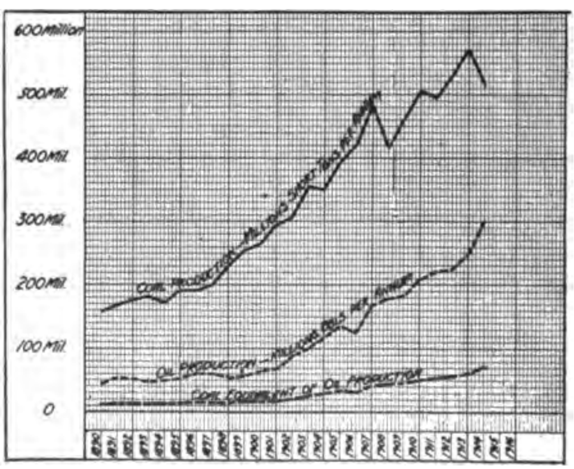

FIG. 234.—Comparison of coal and fuel oil production.

there is no coal marketed except for domestic purposes. If oil continues to advance in price, the question of the available supply of coal will become of paramount importance. The most promising source appears to be the Alaskan coal fields, which are

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known to be very extensive and to contain coal of excellent quality. Vast developments must be made, however, in the way of transport and docking facilities before Alaskan coal becomes available in quantities sufficient to represent a factor of importance in connection with power developments.

The design of a coal-burning plant differs in many respects from that of an oil-burning plant. Boilers must be set higher for coal than for oil so as to provide room for mechanical stokers or to provide ample combustion space for powdered fuel. A basement under the boilers is required for handling ashes. The building must be high enough to allow for coal bunkers and conveyors above the boilers, larger smokestacks are necessary, and forced-draft apparatus is usually required. In most cases, therefore, it would be impracticable to change over existing oil-burning plants to coal-burning plants, although, on the other hand, it is an easy matter to change a coal-burning plant to an oil-burning one.

In some cases oil-burning plants have been specially designed with a view to converting them to coal-burning at a future date. However, since it is impossible to anticipate the rapid changes that occur in engineering practice, the extra expense involved in thus attempting to design for the future is hardly warranted.

Another fuel used quite extensively in the Northwest is the refuse from sawmills, known as hog fuel. This is an extremely cheap fuel if used close to the mill. Owing to its bulk, however, it is difficult to transport, and its field of usefulness is therefore limited.

LOCATION OF STEAM-ELECTRIC POWER PLANTS

Economy of design in the vast hydro-electric transmission lines of the West, in which steam-electric generation serves as an auxiliary, necessitates the location of the steam-electric plant as near to the large industrial centers as possible. This reduces to a minimum the distance through which the steam-generated power must be transmitted, thus avoiding the necessity of burning extra fuel to make up for transmission losses. The exact location within the industrial center depends mainly on the four following factors: (*a*) An adequate supply of water for condensing purposes; (*b*) access to deep water, to enable oil to be delivered by barge; (*c*) railway facilities for the delivery of machinery and possible delivery of fuel; (*d*) proximity of the transmission

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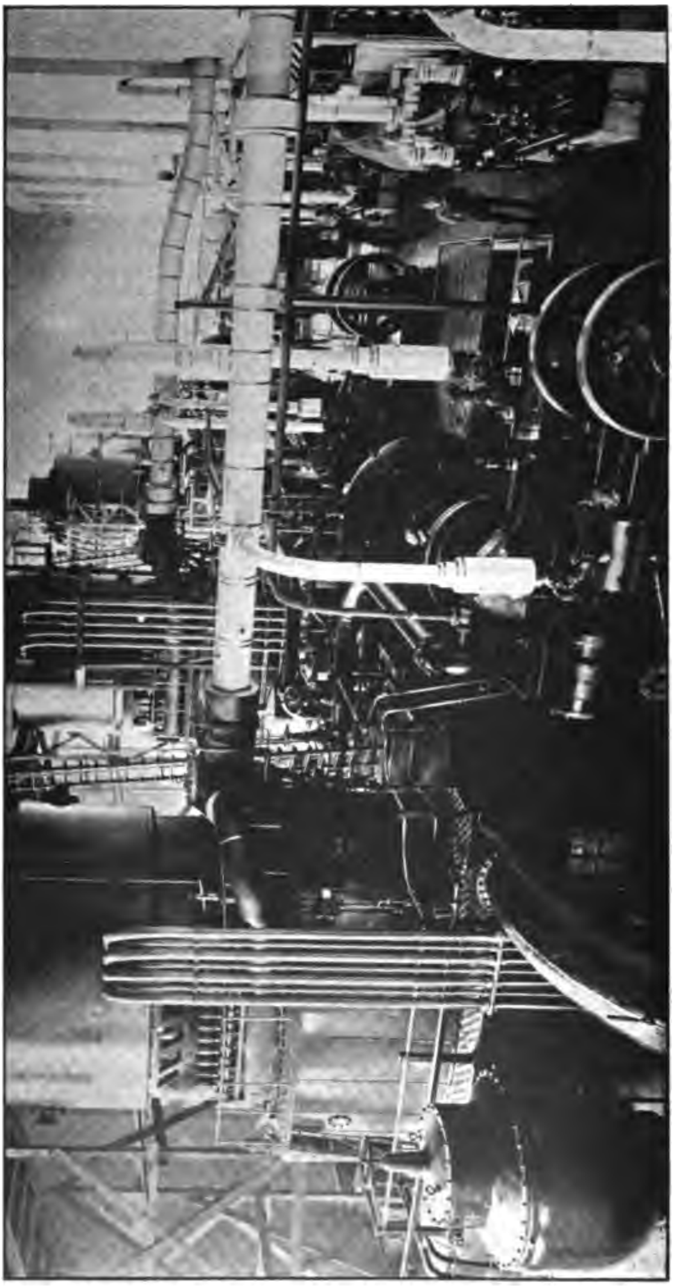

FIG. 235.—Interior view of generating room, Long Beach Plant of Southern California Edison Company. This is one of the largest fuel oil steam generated power plants in the world.

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lines through which power is delivered from the hydro-electric system.

A large interconnected hydro-electric system may have one large auxiliary steam plant or several small ones. As most of the hydro-electric systems in the West have grown to their present huge proportions through combinations of several smaller systems, in the majority of cases there are several steam plants connected to each system. While, if properly designed, a large single plant is in general more economical to operate than a number of small plants, the distribution losses may be much less

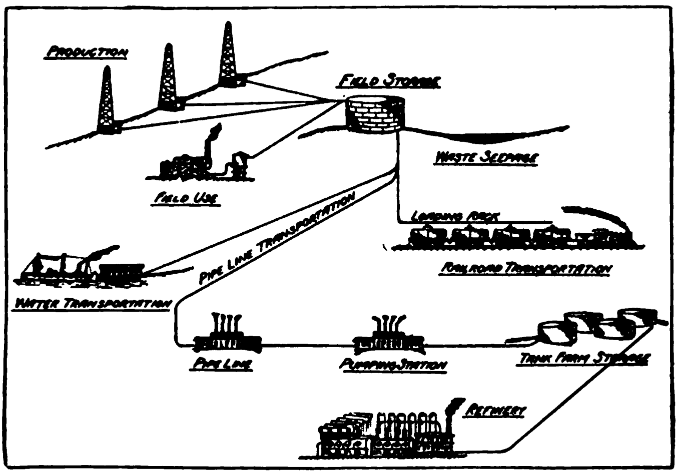

FIG. 236.—Diagrammatic representation of refining the product.

in the latter case. The question of economical distribution is, therefore, a very important factor in determining both the size and the location of a plant.

SIZE OF STEAM PLANT UNITS

The size of turbine selected for a plant of given capacity depends largely on the question of spare units required. In the ordinary isolated steam plant it is an axiom that in order to insure continuity of service it is necessary to have enough spare units to provide for repairs, adjustments, cleaning of condensers, and any other of the many causes that may necessitate the shutting down of the main unit. If the units are too large this spare requirement results in too expensive a plant. For instance, if

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the plant capacity is to be 30,000 kw. and it is considered essential to have at least one spare unit, the plant may consist of two 30,000 kw. machines, three 15,000 or four 10,000 kw. machines. It is seen at once that for the same plant capacity with one machine shut down, the first case has an installed capacity of 60,000 kw., the second case 45,000 kw. and the third case 40,000 kw. Since the cost of the plant is approximately proportional to the installed capacity it is obvious that the selection of too large a unit results in excessive first cost.

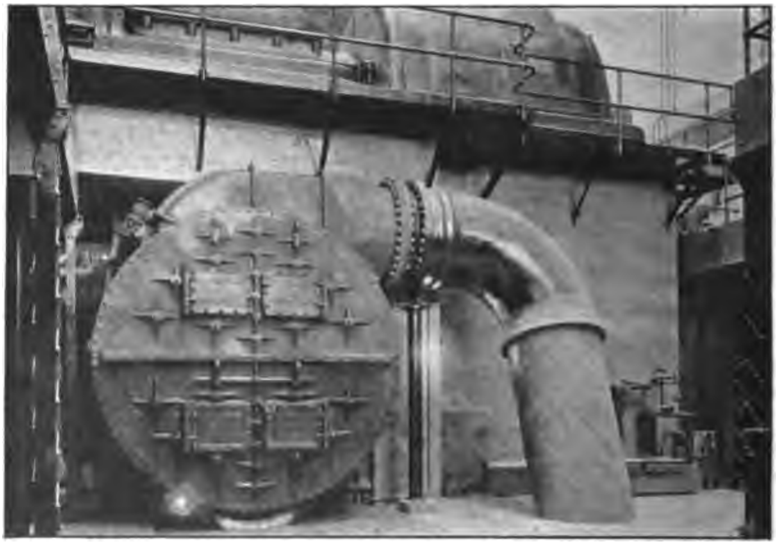

FIG. 237.—The condenser and a portion of the 42 inch discharge salt water pipe for the new 15,000 kw. turbo generator, station A, San Francisco, Pacific Gas and Electric Company.

In the case of steam plants that are interconnected with a large hydroelectric system, the necessity for spare units is not so great, as it is always possible to take the load off one plant temporarily and carry it on another. In such plants the size of unit may be as large as the system will stand—that is, it must not be so large as to cripple the system when it is shut down. The largest single unit on the Pacific Coast at the present time is 20,000 kw., although machines as large as 45,000 kw. in single units have been built and are operating in the east.

The size of the boiler unit is also affected by the necessity

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of having enough spare boilers to permit frequent cleaning and repairs. It is desirable in any central station to have at least 4 or 6 boilers. In the East many plants have boilers up to 1,500 h.p. or 2,000 h.p. each, but on the Pacific Coast 600 h.p. to 800 h.p. is the maximum. For oil burning the best results are obtained with boilers having a relatively large combustion chamber, and the present tendency is to raise boilers higher than formerly so as to increase the furnace volume and enable the boilers to be forced to high capacities. Oil-burning boilers have not as yet been forced to such high overloads as has been done in

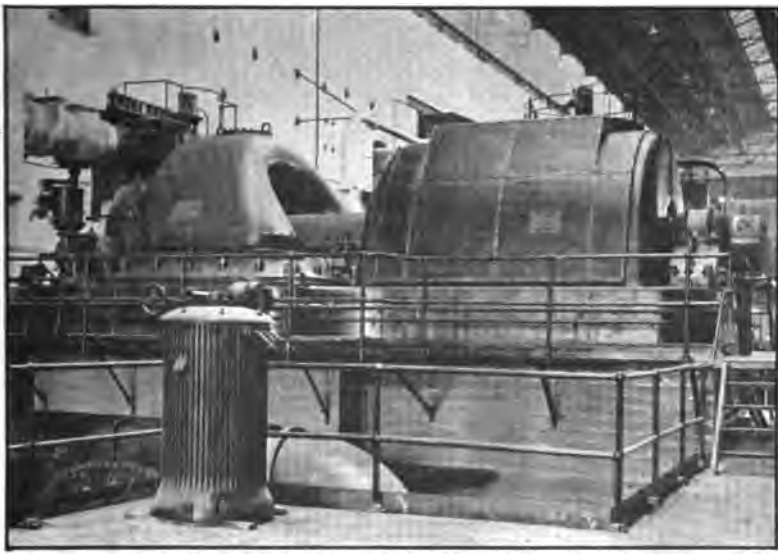

FIG. 238.—The new turbo-unit of 15,000 kw., station A, Pacific Gas and Electric Company, San Francisco.

Eastern cities with coal-burning boilers. Capacities of 200 per cent. of rating have been obtained with steam atomizing burners, and a recent plant on the Atlantic Coast firing Mexican oil with mechanical atomizing burners has operated up to 300 per cent. of rating. Mechanical atomizing burners produce a softer flame that is less damaging to brickwork than the steam atomizing burner, and give a higher boiler efficiency at high overloads, owing to the more perfect combustion maintained when firing large quantities of oil in furnaces of limited volume. They require strong forced draft, however, and the power required for

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this as well as the extra steam used for heating and pumping tends to offset these advantages.

Economizers have not made as much headway in oil burning plants as in coal burning plants. This is largely due to the fact that owing to the small amount of excess air required with oil burning, there is less heat in the gases leaving the boiler, and hence, less necessity for economizers. At present prices of fuel oil it would undoubtedly pay to install economizers in any plant

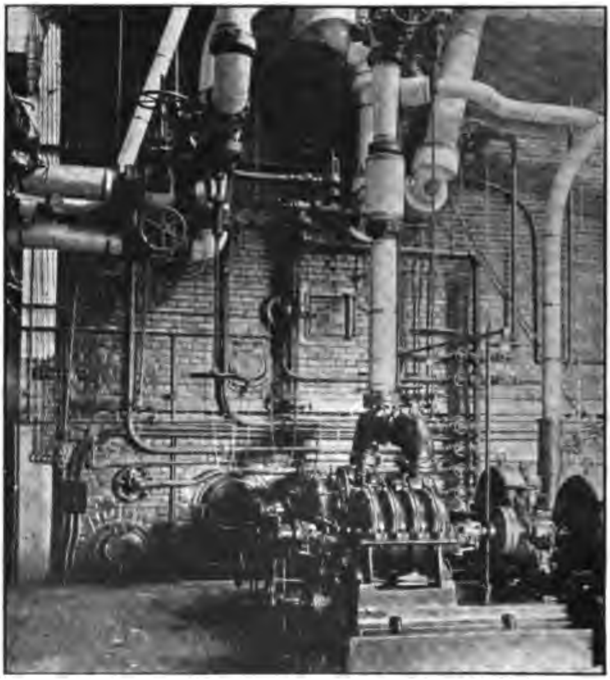

FIG. 239.—Typical view of auxiliary apparatus installed in Pacific Coast power plants. Figure shows particularly a turbo feed-water pump at the Long Beach Plant of the Southern California Edison Company.

that is to be operated at a fairly high load factor. For a poor load factor, however, and especially for a plant that is expected to act during most of its life as a standby plant, economizers are not considered to be a good investment. Where economizers are installed it is necessary to either increase the height of smoke-stack, or provide induced draft apparatus. This increases the cost of the economizer installation, and, must be considered

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when balancing interest and other fixed charges against the increased efficiency resulting from the economizers.

The modern tendency in the operation of central steam stations is toward higher steam pressure and higher superheat. Theoretically in any heat engine the maximum efficiency is obtained, as shown in the Carnot cycle, by having the tempera-

FIG. 240.—Auxiliary apparatus, including centrifugal feed pump and feed water heater at Station C., Pacific Gas and Electric Company, Oakland.

ture at which the heat is supplied to the working substance uniform at the highest attainable value and the temperature at which heat is with-drawn uniform similarly at the lowest attainable value. With steam the upper temperature range may be raised either by increasing the pressure or by increasing the superheat, or both. In neither case, however, is the upper temperature range uniform. Increase of pressure does, however, carry the elevated temperature through a longer part of the range than increase of superheat and thus more nearly approaches

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the condition for maximum efficiency with any given extreme upper limit of temperature. We should thus expect that a given extreme upper limit of temperature reached through high pressure and relatively moderate superheat would give better conditions for economy than lower pressure and higher superheat.

Fig. 241.

Fig. 242.

Figs. 241–242.—Economy measuring apparatus.

The temperature and pressure of the steam and water from the boiler down through the condenser need careful attention in the economic operation of the modern power plant. On the left is exhibited the vacuum gage, barometer and thermometer installed between the first and second pass of the steam turbine. Note the vacuum of 29.15 in. with the atmospheric barometer reading of 30.1 in. To the right may be seen recording meters for inlet and outlet temperatures of the circulating water, steam temperature, vacuum and steam pressure, and the temperature of the condensate. The Klaxon horn at the right of the meter board sounds an alarm when the oil pressure accumulator drops. This installation is at the Long Beach Plant of the Southern California Edison Company.

This may be illustrated by the following example, in which steam at 200 lb. pressure and 200°F. superheat is compared with steam at 300 lb. pressure and 166°F. superheat, both of these combinations giving actual temperature of superheated steam at 588°F.:

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Pressure, gage, lb. per sq. in	200.0	300.0
Temperature saturated steam, deg. Fahr	388.0	422.0
Degree of superheat, deg. Fahr	200.0	166.0
Temperature superheated steam, deg. Fahr	588.0	588.0
Heat per lb. steam (above 32 deg.) B.t.u	1310.0	1300.0
Heat (above 32 deg.) per lb. steam after expanding adiabatically to 1-in. absolute, B.t.u	892.0	866.0
Heat available per lb. steam, B.t.u	418.0	434.0
Heat utilized at 75 per cent. efficiency, B.t.u	313.0	325.0
Moisture in exhaust steam, per cent	9.2	11.4

It will be observed from the above comparison that while the quantity of heat present in the initial steam in the two cases is practically the same, the heat utilized in the case of steam entering the engine at 300 lb. pressure is about 4 per cent. more than in the case of the 200 lb. pressure steam. In actual practice these figures would be slightly modified by difference in efficiency of the prime mover under the two different conditions. However with turbines properly designed for the conditions under which they are to operate this difference would be small, and may be neglected in the present discussion. It will also be observed that there is more moisture in the exhaust steam in the case of 300 lb. pressure initial steam than in the case of 200 lb. initial steam. This means that there will be less work to be done by the condenser as more of the steam is already condensed. This results in a still further advantage to the higher pressure steam. On the Pacific Coast steam pressures up to 200 lb. have been used for many years, station "A" of the Pacific Gas & Electric Company, San Francisco, having been built for that pressure in 1901.

Within the last two or three years higher pressures than this have been adopted, several plants having been built for boiler pressures of 250 lb., while in the Eastern states plants are already in operation at 300 lb. pressure and pressures even as high as 500 lb. are being talked of as possibilities. The maximum limit to pressures and superheat is determined at the present time by the temperature that the materials of construction will stand. With present steels 700°F. is about the limit. This limit would be reached at 500 lb. pressure and 230°F. of superheat. The pressure is also limited, commercially, by the extra cost involved as it is possible that in some circumstances the fixed charges on the extra investment required for the high pressure apparatus may neutralize the saving effected.

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In the modern plant both steam driven and electric driven auxiliaries are used. Steam drive is always the cheapest whenever it is possible to make use of the exhaust steam for heating purposes. If however, the exhaust would be wasted it is cheaper to use electric driven auxiliaries. In an ordinary plant in which all the auxiliaries are driven by steam it is possible to utilize all of the exhaust steam for heating the feed water when a fairly heavy load is carried on the plant, but when the load is light there is less feed water to be heated and, as there is nearly as

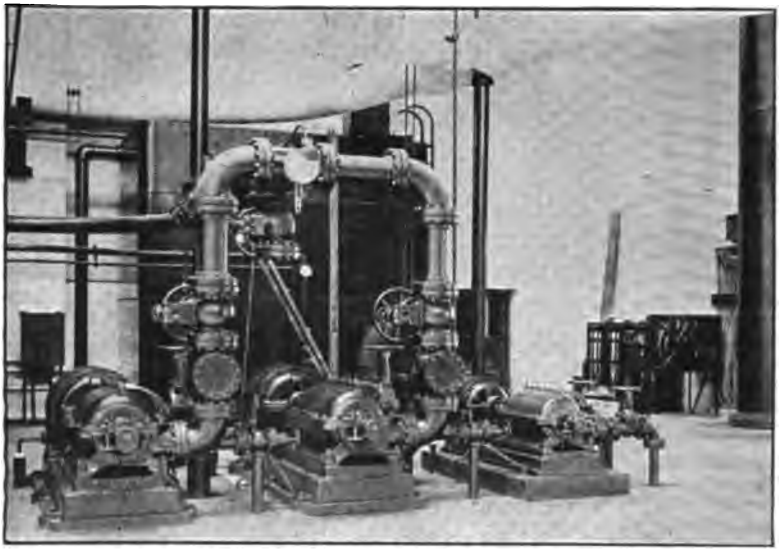

Fig. 243.—Auxiliary apparatus, turbo driven at the Long Beach Plant of the Southern California Edison Company.

much steam used by the auxiliaries as at heavy loads, there is bound to be a waste of steam. It is, therefore, more economical to use steam driven auxiliaries at the heavy load and electric driven auxiliaries at lighter loads. It is thus desirable to install duplicate auxiliaries in which one set is driven by steam and the other by electric power. The electric drive is not quite as reliable as steam and it is therefore frequently desirable to install a separate auxiliary turbine and generator to generate the current required by the auxiliaries. If this is done, all the auxiliaries may be motor driven. At times of heavy load, they would get their current from the auxiliary generator and at times of light

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load they would take their current from the main bus. The exhaust from the auxiliary generator would thus be available for heating the feed water, and as this machine could be run merely as a standby at light loads there would be no exhaust steam wasted. This system has been used to advantage at the Connors Creek plant of the Detroit Edison Company and other stations in the east, but has not yet been introduced on the Pacific Coast.

The modern tendency is to eliminate as far as possible all reciprocating pumps and at the present time centrifugal pumps are used in all cases except for fuel oil and lubricating pumps where the reciprocating type is still used. Steam turbines are used in preference to reciprocating engines even in cases where slow speed is desired such as for operating circulating pumps for condensers. In such cases it is customary to provide reduction gears so as to be able to operate the turbine at high speed and the pump at low speed thus enabling each machine to operate at the speed best suited to highest efficiency.

AUTOMATIC CONTROL

As economical operation of a plant is obtained by the careful watching of all details, it is a growing conviction that personal control under trained supervisors is the one way to produce high economy. This is especially true of the boiler room, and it is common practice in the best plants to employ a combustion engineer to make flue-gas analyses and to keep continual check on the boiler efficiency. There is a tendency toward introducing automatic control into the fire room. Automatic oil-fire regulators have been placed in service in a number of plants on the Pacific Coast and increases in efficiency as high as 3 or 4 per cent. are reported. Another tendency at the present time is to equip the plant with recording meters which register automatically all important elements of operation throughout the full twenty-four hours of the day. It was formerly the custom to operate boilers with no instruments except a steam gage, but it is now customary to install steam-flow meters which register the quantity of steam produced by the boiler and the quantity of steam uesd by the burners, and air meters which indicate the amount of air passing through the boiler settings. These instruments are of great value in assisting the combustion engineer to secure maximum efficiency from the boiler plant.

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APPENDIX I

ILLUSTRATIVE PROBLEMS

Problem No. 1.—The mean effective pressure of a single-acting oil engine cylinder under test is found from an indicator card to be 43.9 lb. per sq. in.; the cylinder has 47.5 working strokes per minute; the diameter of the cylinder is 30 in.; and the length of stroke is 30 in.

What is its horsepower?

Solution.

By reference to formula for horsepower computation, we find for

$$P = 43.9,\ L = \frac{30}{12},\ A = 0.7854(30)^2, \text{ and } N = 47.5 \text{ that } H.P. = \frac{PLAN}{33{,}000} = \frac{43.9 \times 2.5 \times 706.9 \times 47.5}{33{,}000} = 111.7$$

Problem No. 2.—In a turbine test the atmospheric barometer reduced to the 32°F. standard of measurement, read 29.93 in.

If the condenser vacuum reduced to the same standard read 28.23 in. of vacuum, what was the absolute pressure in the condenser?

Solution.

Barometer for day	29.93 in.
Vacuum maintained	28.23 in.
Pressure in condenser in inches of mercury	1.70 in.

14.696 lb. per sq. in. = 29.92 in. of mercury.

$$\therefore \frac{14.696}{X} = \frac{29.92}{1.70}$$

$$X = \frac{14.696}{29.92} \times 1.70 = 0.835 \text{ lb. per sq. in. absolute pressure in condenser.}$$

Problem No. 3.—A 10,000 kw. turbine under test operated with a gage reading of 171.5 lb. per sq. in. The gage, however, read one pound too low. The computed absolute pressure was found to be 187.2 lb. per sq. in.

What was the barometer reading for the day?

Solution.

Absolute pressure	= 187.2 lb. per sq. in.
Corrected gage pressure (171.5 + 1)	= 172.5 lb. per sq. in.
Atmospheric pressure	= 14.7 lb. per sq. in.

$$\frac{14.696}{14.7} = \frac{29.92}{X}$$

$$\therefore X = \frac{29.96}{14.696} \times 14.7 = 29.93 \text{ in. of mercury barometer reading for the day.}$$

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Problem No. 4.—A corrected atmospheric barometer reading is found to be 29.942 in. of mercury on the 32°F. standard.

How many lb. per sq. in. does this represent?

Solution.

To convert to lb. per sq. in. by formula in the chapter on pressures:

$$\frac{I_m}{P} = \frac{29.921}{14.696}$$

$$\text{or } 29.942 = 2.046\,P$$

$$\therefore P = 14.670 \text{ lb. per sq. in.}$$

Problem No. 5.—A corrected barometer reading is 29.937 in. of mercury on the 30-inch vacuum standard.

What is the pressure in lb. per sq. in.?

Solution.

To convert to lb. per sq. in. from formula in the chapter on pressures:

$$\frac{I_m}{P} = \frac{30}{14.7}$$

$$\text{or } 29.937 = 2.041\,P$$

$$\therefore P = 14.668 \text{ lb. per sq. in.}$$

Problem No. 6.—(*a*) At what temperature do the Fahrenheit and Centigrade scales read the same? Fahrenheit and Reamur? Centigrade and Reamur?

(*b*) Assuming the absolute zero of the Fahrenheit scale to be 459.6°F. compute the absolute zero on the Centigrade and Reamur scales.

Solution.

(*a*) *Fahrenheit and Centigrade.*

Relation is given by formula:

$F - 32 = 9/5C$

When the scales have identical numerical readings, then $F = C = X$

Substituting in formula

$$X - 32 = 9/5X$$

$$4X = -160 \qquad \text{or } X = -40°$$

$$\therefore -40°\text{F.} = -40°\text{C.}$$

(*Fahrenheit and Reamur.*

Relation is given by formula:

$$F - 32 = 9/4R$$

$$\text{Let } F = R = X, \quad \text{then } X - 32 = 9/4X$$

$$5X = -128, \quad \text{or} \quad X = -25.6$$

$$\therefore -25.6°\text{F.} = -25.6°\text{R.}$$

Centigrade and Reamur.

Relation is given by formula:

$$C = 5/4R$$

$$\text{Let } C = R = X, \quad \text{then } X = 5/4X$$

$$4X = 5X \quad \text{or} \quad X = 0$$

$$\therefore 0°\text{C} = 0°\text{R}$$

(*b*) Absolute zero = − 459.6°F.

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Let us substitute this value of F in the general relationship,

$$F - 32 = 9/5C$$

and we have

$$-459.6 - 32 = 9/5C$$

$$9C = -2458 \text{ or } C = -273.1^\circ \text{ absolute zero on Cent. scale.}$$

Similarly for the relationship

$$F - 32 = 9/4R$$

we have

$$-459.6 - 32 = 9/4R$$

$$9R = -1966.4$$

$$R = -218.049 = \text{absolute zero on Reamur scale.}$$

Problem No. 7.—The temperature of the steam entering a turbine during a test was found to be 521.2°F.; the correction for stem exposure of the thermometer was 5.6°F.; the corrected steam gage reading 172.5 lb. gage; and the atmospheric barometer read 14.7 lb. per sq. in.

What was the superheat of the steam?

Solution.

Thermometer reading on entering steam	=	521.2°
Correction for stem exposure	= +	5.6°
True temp. of steam entering turbine	=	526.8°F.
Absolute pressure = 172.5 + 14.7		187.2 lb.
From steam tables the temperature corresponding to this pressure of saturated dry steam	=	376.4°F.
∴ Degrees of superheat = 526.8° − 376.4°	=	150.4°

Problem No. 8.—The temperature of the superheated steam entering a turbine during a test was found to be 544.8°F. The pressure of the steam in the main was 182 lb. abs.

What was the superheat of the steam?

Solution.

By reference to Table 2 of the steam tables the temperature of saturated steam corresponding to 182 lb. pressure is found to be 374.0°F. Substract this value from the temperature of the steam entering the turbine and the result will be the degrees of superheat, or

$$544.8 - 374.0 = 170.8^\circ\text{F. superheat}$$

Problem No. 9.—Regnault's classic formula for total heat of saturated steam is:

$$H = 1091.7 + 0.305\ (t - 32)$$

Compute the total heat of saturated steam at the boiler pressure corresponding to 382.3°F.

Solution.

Substituting, we have

$$\begin{aligned} H &= 1091.7 + 0.305\ (382.3 - 32) \\ &= 1091.7 + 106.84 \\ &= 1198.54 \text{ B.t.u. per lb.} \end{aligned}$$

From tables $H = 1198.2$

$$\therefore \text{Error} = \frac{1198.54 - 1198.2}{1198.2} = \frac{0.34}{1198.2} = 0.0284\,\%$$

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Problem No. 10.—Compute the total heat of saturated steam at 382.3°F. by the formula:

$$H = 1150.3 + 0.3745\ (t - 212) - 0.000550\ (t - 212)^2$$

Solution.—Substituting the value of temperature, we have

$$\begin{aligned} H &= 1150.3 + 0.3745(382.3 - 212) - 0.000550(382.3 - 212)^2 \\ &= 1150.3 + 63.78 - 15.95 \\ &= 1198.13 \text{ B.t.u. per lb.} \end{aligned}$$

From tables $H = 1198.2$

$$\therefore \text{Error} = \frac{1198.2 - 1198.13}{1198.2} = \frac{0.07}{1198.2}$$
$$= 0.00585 \text{ per cent.}$$

Problem No. 11.—The specific volume of saturated steam is represented on page 104 of Marks and Davis Steam Tables by the formula:

$$S = 28.424 - 0.01650(t - 320) - 0.0000132(t - 320)^2$$

Find the specific volume of steam for $t = 382.3$.

Solution.—Substituting, we have

$$\begin{aligned} S &= 28.424 - 0.01650(382.3 - 320) - 0.0000132(382.3 - 320)^2 \\ &= 28.424 - 0.862 - 0.036 \\ &= 29.525 \end{aligned}$$

N. B.—This formula evidently does not check up at all for this temperature, since the specific volume for a temperature of 382.3°F. is 2.279 from the steam tables.

Problem No. 12.—The mean specific heat of steam is represented mathematically on page 92 of Marks & Davis Steam Tables by the formula:

$$C_m = 0.9983 - 0.0000288(t - 32) - 0.0002133(t - 32)^2$$

What is the mean specific heat of steam for $t = 382.3$°F.?

Solution.—Substituting, we have

$$\begin{aligned} C_m\ & 0.9983 - 0.0000288\ (382.3 - 32) + 0.0002133(382.3 - 32)^2 \\ &= 0.9983 - 0.0000288(350.3) + 0.0002133(350.3)^2 \\ &= 0.9983 - 0.0101 + 26.17 \\ &= 27.1582 \text{ Mean specific heat.} \end{aligned}$$

Evidently a mistake is made in translating the last term of this formula from its original source, for it should be $.0000002133\ (t - 32)^2$. On this basis, we have that

$$C_m = 0.9983 - 0.0101 + .02617 = 1.0346$$

In the steam tables the heat of liquid for 382° is 355.0 B.t.u. and for 383° is 356.1 B.t.u. Hence the mean specific heat C_m is approximately 1.1, which indicates that had the decimal points been carried further the specific heat approaches that set forth in the above correction.

Problem No. 13.—At a certain central station there are four 773 boiler horsepower Parker boilers. These boilers were used to give a 10,000 kw. load at the terminals of a turbine which has an over-all efficiency of 21 per cent. What was the percentage of overload on the boilers?

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Solution.

$\frac{10,000}{0.21}$ = 47,600 kw. actually taken from boilers (neglecting losses in steam mains).

Since 1 hp. = .746 kw.

$\frac{47,600}{0.746}$ = 63,800 mechanical horsepower actually taken off boilers.

From discussion in text, we have

$\frac{34.5 \times 970.4 \times 777.5}{60 \times 33,000} = 13.14$ = ratio of boiler horsepower to mechanical horsepower.

$\therefore \frac{63,800}{13.14} = 4850$ Bl. h. p. actually taken from boiler.

$4 \times 773 = 3092$ Bl. h.p. rated capacity.

$\therefore \frac{4850 - 3092}{3092} \times 100 = 56.8$ per cent. overload.

Problem No. 14.—A Parker boiler under test operated with the following conditions: Steam pressure 179.7 lbs. gage; temperature of feed water entering boiler was 123.4°F.; barometer for the day read 30.1 inches of mercury.

Find the factor of evaporation for: (*a*) steam superheated 182°F.; (*b*) dry saturated steam; and (*c*) 5 per cent. wet steam.

Solution.

$\frac{30.10}{29.92} = \frac{X}{14.696}$ or $X = 14.78$ lb. per sq. in. atmospheric reading of day.

Gage reading	179.7 pounds
Atmospheric pressure	14.78 pounds
∴ Abs. pressure of boiler	= 194.48 pounds

From Steam Tables:

h_1	= heat of liquid at absolute boiler pressure	352.45
L_1	= latent heat of evaporation at absolute boiler pressure	845.2
H_1	= total heat of steam at absolute pressure	1197.65
h_2	= heat of liquid at temperature of entering feed water	91.3
H_s	= total heat of superheated steam (194.48 lb. pressure and 182° superheat)	1297.99
X	= quality of steam	0.95

(*a*) $$F_e = \frac{H_s - h_2}{970.4} = \frac{1297.99 - 91.3}{970.4} = \frac{1206.69}{970.4} = 1.243$$

(*b*) $$F_e = \frac{H_1 - h_2}{970.4} = \frac{1197.65 - 91.3}{970.4} = \frac{1106.35}{970.4} = 1.141$$

(*c*) $$F_e = \frac{h_1 + XL_1 - h_2}{970.4} = \frac{352.45 + 0.95 \times 845.2 - 91.3}{970.4} = \frac{1065.25}{970.4} = 1.097$$

Problem No. 15.—In a boiler test, the temperature of the feed water entering the boiler was 170.7°F., the steam pressure was 144 pounds gage, and the barometer read 29.28 inches of mercury.

Find the factor of evaporation for: (*a*) dry saturated steam; (*b*) 10 per cent. wet steam; (*c*) steam superheated 125°F.

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Solution.

$$\frac{29.28}{29.92} = \frac{X}{14.696} \text{ or } X = \frac{29.28}{29.92} \times 14.696 = 14.38 \text{ lb. per sq. in. atmospheric pressure.}$$

Boiler gage reading = 144.00 lb.
Atmospheric pressure = 14.38 lb.

∴ Abs. boiler pressure = 158.38 lb. per sq. in.

From Steam Tables:

h_1 = heat of liquid at absolute boiler pressure .... = 334.7
L_1 = latent heat of evaporation at absolute boiler pressure...... = 859.6
H_1 = total heat of steam at absolute boiler pressure.......... = 1194.34
h_2 = heat of liquid at temperature of entering feed water..... = 138.57
H_s = total heat of superheated steam (158.38 lb. pressure and 125° superheat)............................ = 1263.88

(a) $$F_e = \frac{H_1 - h_2}{970.4} = \frac{1194.34 - 138.57}{970.4} = \frac{1055.77}{970.4} = 1.088$$

(b) $$F_e = \frac{h_1 + XL_1 - h_2}{970.4} = \frac{334.7 + 0.90 \times 859.6 - 138.57}{970.4} = \frac{969.13}{970.4} = 0.999$$ (where X = quality of steam)

(c) $$F_e = \frac{H_s - h_2}{970.4} = \frac{1263.88 - 138.57}{970.4} = \frac{1125.31}{970.4} = 1.159$$

Problem No. 16.—What is the equivalent evaporation in lb. of water per hr. from and at 212°F. if the water fed to a boiler has a total weight of 64,494 lb. and the factor of evaporation is 1.193 lb.?

Solution.—By applying the fundamental formula developed in the text, we have at once equivalent evaporation from and at 212°F. = 64,494 × 1.193 = 76,950 lb.

Problem No. 17.—Compute the factor of evaporation for a boiler generating dry saturated steam under a pressure of 98.1 lb. per sq. in. abs. and receiving its feed water at 58.8°F.

Solution.—Total heat of saturated steam at 98.1 lb. abs. = 1186.
Heat of liquid at temperature 58.8°F. = 26.88

$$\therefore F_e = \frac{1186 - 26.88}{970.4} = 1.193$$

Problem No. 18.—What is the weight of equivalent water evaporated to dry steam from and at 212°F., if the total weight of water actually evaporated is 53,688 lbs. and the factor of evaporation is 1.193?

Solution.—Weight of equivalent water evaporated to dry steam from and at 212°F. = 53,688 × 1.193 = 64,150

Problem No. 19.—The equivalent evaporation of a boiler under test is 5940 lbs. of water per hour, and the total heating surface of the boiler is found to be 2031 sq. ft.

What is the average equivalent evaporation per sq. ft. of water heating surface per hour?

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Solution.—The average equivalent evaporation per sq. ft. of water heating surface per hour is evidently

$$\frac{5940}{2031} = 2.93$$

Problem No. 20.—The equivalent evaporation of a boiler under test is found to be 5940 lb. of water per hour.

What is the boiler horsepower of the boiler?

Solution.—By definition

$$\text{Bl. H. P.} = \frac{5940}{34.5} = 172.2$$

Problem No. 21.—The rated horsepower of a boiler is given by the builders as 210 Bl. h. p. Under test 172.2 Bl. h.p. were actually developed.

What was the percentage of boiler capacity developed?

Solution.—Capacity of boiler as developed in percentage is

$$\frac{172.2}{210} \times 100 = 82 \text{ per cent.}$$

Problem No. 22.—What is the equivalent evaporation per lb. of coal as fired in a boiler under test when the weight of equivalent water evaporated to dry steam from and at 212°F. is 64,150, and the total weight of fuel consumed as fired is 8012?

Solution.—Equivalent evaporation per lb. of coal as fired =

$$\frac{64{,}150}{8{,}012} = 8.00$$

Problem No. 23.—From a Parker boiler test covering a period of 8 hrs., the following data were taken:

Steam pressure (gage)	185.3 lb. per sq. in.
Atmospheric barometer	30.2 in.
Temp. of water entering the boiler	169.1°F.
Temp. of steam leaving the superheater drum	527.°F.
Specific gravity of the oil at 60°F	0.9705
Percentage of water in the oil	0.7 of 1%
Calorific value of oil per lb	18,752 B.t.u.
Weight of oil as fired	15,084 lb.
Total weight of water fed to boiler	205,277 lb.

What is the degree of superheat of the steam leaving the superheater?

Solution.—Barometer reading = 30.2 in. or $\frac{30.2}{2.036}$ = 14.83 lb. per sq. in.

Steam pressure abs. = 185.3 + 14.83 = 200.13 lb. per sq. in.
From tables $t_1 = 381.9°$
Temp. of steam leaving superheater drum = 527°F.

$$\therefore 527 - 381.9 = 145.1° \text{ superheat}$$

Problem No. 24.—What is the gravity of the oil in degrees Baumé in Problem 23?

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Solution.—For light liquids:

$$\text{Sp. gr.} = \frac{140}{130 + \text{Deg. Baumé}}$$

$$0.9705 = \frac{140}{130 + \text{Deg. Baumé}}$$

$$0.9705\ (\text{Deg. Baumé}) = 140 - 130 \times 0.9705$$

$$\therefore \text{Deg. Baumé} = \frac{140 - 130 \times 0.9705}{0.9705}$$

$$= \frac{13.84}{0.9705} = 14.26$$

Problem No. 25.—What is the weight of the oil corrected for moisture in Problem 23?

Solution.—Percentage of water in the oil = .7 of 1%

Wt. of oil as fired...................... = 15,084 lb.
$\therefore$ 15,084 × 0.007 = 105.6 lb. = Wt. of water in oil
Weight of oil corrected for moisture =

$$15{,}084 - 105.6 = 14{,}978.4 \text{ lb.}$$

Problem No. 26.—What is the factor of evaporation in Problem 23?

Solution.—Factor of evaporation $= \dfrac{H_s - h_2}{970.4}$ (for superheated steam)

From problem 23, P_1 = 200.13 lb. per sq. in. and 145.1° superheat.
$\therefore$ From tables H_s = 1280.1 B.t.u.
Also from problem 23 P_2 = 14.83 lb. per sq. in.
$\therefore$ From tables h_2 = 136.96 B.t.u.

$$\text{Factor of evaporation} = \frac{1280.1 - 136\ 96}{970.4}$$

$$= 1.178$$

Problem No. 27.—Determine the equivalent evaporation from and at 212°F. from Problem 23.

Solution.—Total wt. of water fed to boiler = 205,277 lb.

Duration of test............................. = 8 hrs.

Equivalent evaporation = Water evaporation per hr. × factor of evaporation =

$$\frac{205{,}277}{8} \times 1.178 = 30{,}227 \text{ lb.}$$

Problem No. 28.—What is the boiler horsepower developed in A.S.M.E. rating in Problem 23?

Solution.

$$H.P = \frac{\text{Equivalent evaporation from and at } 212°\text{F.}}{34.5}$$

$$= \frac{30{,}227}{34.5} = 876$$

Problem No. 29.—What is the equivalent evaporation from and at 212°F. per lb. of oil as fired in Problem 23?

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Solution.—Wt. of oil as fired = 15,084 lb.

$$\text{Wt. of oil as fired per hr.} = \frac{15,084}{8} = 1,885.5 \text{ lb.}$$

Equivalent evaporation = 30,227

Equivalent evap. from and at 212° per lb. of oil as fired

$$= \frac{30,227}{1,885.5} = 16.03 \text{ lb.}$$

Problem No. 30.—What is the equivalent evaporation from and at 212°F. per lb. of oil corrected for moisture in Problem 23?

Solution.—Wt. of oil corrected for moisture = 14,978.4 lb.

Wt. of oil corrected for moisture per hr. =

$$\frac{14,978.4}{8} = 1,872.3 \text{ lb.}$$

Equiv. evap. from and at 212° per lb. of oil corrected for moisture =

$$\frac{30,227}{1,872.3} = 16.144 \text{ lb.}$$

Problem No. 31.—What is the efficiency of the boiler in Problem 23?

Solution.

$$\text{Boiler eff.} = \frac{\text{Ht. abs. by boiler per unit of time}}{\text{Ht. in fuel fed to furnace per unit of time}}$$

$$= \frac{30,227 \times 970.4}{1872.3 \times 18,752} = 83.29 \text{ per cent.}$$

Problem No. 32.—Assuming the steam was just saturated and not superheated in the above, what would be the factor of evaporation in Problem 23?

Solution.

Factor of evaporation $= \dfrac{H_1 - h_2}{970.4}$ (for saturated steam)

From Problem 23 $P_1 = 200.13$ lb. per sq. in.

$P_2 = 14.83$ lb. per sq. in.

From Tables $H_1 = 1198.1$ B.t.u.

$h_2 = 136.96$ B.t.u.

$\therefore$ Factor of evaporation $= \dfrac{1198.1 - 136.96}{970.4}$

$$= \frac{1061.14}{970.4} = 1.093$$

Problem No. 33.—Assuming the steam to be 97 % dry, what would be the factor of evaporation in Problem 23?

Solution.

Factor of evap. $= \dfrac{h_1 + XL_1 - h_2}{970.4}$ (wet steam)

From Prob. 23, $P_1 = 200.13$ lb. per sq. in.

$P_2 = 14.83$ lb. per sq. in.

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From tables h_1 = heat of liquid at 200.13 lb. per sq. in.
= 354.9 B.t.u.
h_2 = heat of liquid of entering feed water
= 136.96 B.t.u.
L_1 = latent heat of evap. at 200.13 lb. per sq. in.
= 843.2 B.t.u.
X = % dry steam = .97

$$\text{Factor of evap.} = \frac{354.9 + 0.97 \times 843.2 - 136.96}{970.4}$$
$$= \frac{354.9 + 817.9 - 136.96}{970.4}$$
$$= \frac{1035.84}{970.4} = 1.068$$

Miscellaneous Questions and Answers on Fuel Oil and Steam Engineering

1. How do you compute proper size of boiler installation for a power plant?

Answer.—In order to compute the proper size of boiler installation for a given power output, we must know three factors:

(1) The Maximum output to be anticipated.

(2) The Overall efficiency of steam boilers and power units.

(3) The Maximum overload to be allowed on boilers in steam generation.

Thus let us assume that 12,000 kw. are desired at the switchboard of a steam turbine installation and that the overall efficiency of steam boilers steam turbine and electric generation is 15%. What boiler capacity should be installed if the boilers are capable of carrying a 50% overload?

Since 1 hp. = 0.746 kw.

We proceed as follows:

$$\text{Mech. h.p. required at switchboard} = \frac{12000}{0.746}$$

$$\text{Mech. h.p. required to be delivered by boilers} = \frac{12000}{0.746 \times 0.15}$$

Since 1 boiler h.p. = 13.14 mech. hp.,

$$\text{Total boiler h.p. required of boilers} = \frac{12000}{0.746 \times 0.15} \times \frac{1}{13.14}$$

Since boilers are to run at 150% of rating, the rating of boilers must be

$$\frac{12000}{0.746 \times 0.15} \times \frac{1}{13.14} \times \frac{1}{1.50} = 5430 \text{ Blhp.}$$

Questions Answered on Fuel Oil Economy

2. What is the approximate increase, if any, in boiler repairs when coal fired boilers are changed to oil?

Answer.—The amount of boiler repairs is practically the same when burning oil as when burning coal, provided the boilers are operated at the same capacity and the oil burners are properly adjusted so as not to blow oil direct against the boiler tubes or direct against the brick walls.

3. Is the back shot method the best for Stirling boilers?

Answer.—The back shot oil burner is generally considered the best for Stirling boilers, owing to the advantage gained by the large combustion chamber.

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4. Can you give the comparative cost per 1000 pounds of steam, of coal vs. oil, including all boiler room costs? In this question I have in mind modern firing facilities. Give the unit prices for both coal and oil.

Answer.—In regard to the comparative costs of oil vs. coal the reader is referred to Fig. 145, page 222 on which the relative value of oil is plotted against coal of various qualities. From this diagram the reader can see at a glance, for instance, that oil costing $1.50 per bbl. is equivalent to coal of 10,000 B.t.u.'s costing $3.50 per short ton or coal of 14,000 B.t.u.'s costing $6.00 per short ton.

5. What quantity of oil have you found by actual tests is necessary to evaporate the same quantity of water as 1 ton (2240 lb.) of coal, giving B.t.u.'s in coal and oil?

Answer.—The quantity of oil necessary to evaporate the same quantity of water as one ton of coal depends entirely on the heating value of the two fuels and the boiler efficiency obtained. If the heating value of coal is 14,000 B.t.u.'s and the boiler efficiency is 72%, each pound of coal would evaporate 10.35 lb. of water from and at 212°. One pound of oil containing 18,000 B.t.u.'s with a boiler efficiency of 78% would evaporate 14.4 pounds of water from and at 212°. From this you can readily figure out that one ton of coal containing 2240 lb. would evaporate the same quantity of water as 4.8 bbl. of oil each containing 336 pounds.

6. Have you found mechanical atomizing to be the most economical?

Answer.—Oil is atomized by steam in nearly all stationary boiler plants due to the fact that steam is more convenient than any other method, and the atomization is very perfect. As a rule mechanical atomization is not used unless the loss of fresh water used in steam atomization is an important consideration.

7. What are the relative merits of saturated and superheated steam for atomization? Mr. Hawkins in his book, "The Economy Factor in Steam Power Plants," discounts the value of superheated steam. I quote verbatim:

"On the other hand, the action of the superheated steam appears to produce an unsteady flame—a rapid succession of small puffs rather than the steady uniform condition which is desired."

We are wondering if these puffs are occasioned by the oil temperature being raised to the flash point. We are of the opinion that superheated steam should prove more efficient than saturated, and to this end are arranging for test. Pending receipt of flow meter and calibration of our orifice, however, we will appreciate your comments.

Answer.—We use superheated steam in preference to saturated steam wherever we have super heaters. Superheated steam atomizes the oil more perfectly than saturated steam and consequently it is the general belief that less steam is required.

The quantity of steam required for atomizing can always be reduced by heating the oil to a high temperature and it is, therefore, desirable to get the oil as hot as possible without heating it above its flash point. We have never found the use of superheated steam to cause an unsteady flame such as is referred to in Mr. Hawkins' book. We have used steam having as much as 160 degrees of superheat, though normally we use about 100 degrees.

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APPENDIX II

HELPFUL FACTORS IN FUEL OIL STUDY AND CONSERVATION

The authors of the work would feel unmindful of their duty in setting forth the elements of fuel oil and steam engineering did they not at this time point out to the reader some of the helpful factors that are aiding in fuel oil study and conservation in these days of national emergency.

First and foremost must be mentioned the educative and helpful influence of the universities and technical colleges of the West—such as the University of California and Leland Stanford Junior University. These institutions are not only prepared to train technical fuel oil specialists, but the eminent scientists and engineers upon their teaching staffs are contributing noteworthy research data for the upbuilding of efficient mining and utilization of this important national resource. The Extension Division of the University of California is now serving over three hundred thousand people in the state of California. Practically every conceivable educative aid is available through this branch of university instruction. All operators or engineers interested in fuel oil, its uses and conservation, may for a small fee enjoy this excellent service. The only other requirement on the part of the applicant is that he be thoroughly in earnest in undertaking such study. To get in touch with this excellent service, a letter should be addressed to Director of Extension Division, University of California, Berkeley.

The California Railroad Commission is doing excellent service in the efficient handling of the petroleum situation. Authorized under the law to regulate the public utilities as to rates and other matters, this commission on its own initiative has worked out a scheme of interconnection of power companies in the state of California that will do much in conserving the fuel oil in that industry.

Especial attention is called to the research investigations of the United States Bureau of Mines and the United States Bureau of Standards. Much of the scientific data on fuel oil specifications and steam generation contained in this work have been gleaned from the various publications of the Bureau of Mines, while the standardization of thermometers as treated in addition to many other aids in scientific precision described, must be accredited to the helpful work of the Bureau of Standades. In discussions looking toward the production of petroleum the publications of the United States Geological Survey are timely, as are also the publications of the California State Mining Bureau.

The Book on Steam of the Babcock & Wilcox Company is perhaps the most helpful of its kind in existence in setting forth the elementary laws of steam engineering in a practical manner, and the authors are greatly indebted to it for many helpful items in the present work.

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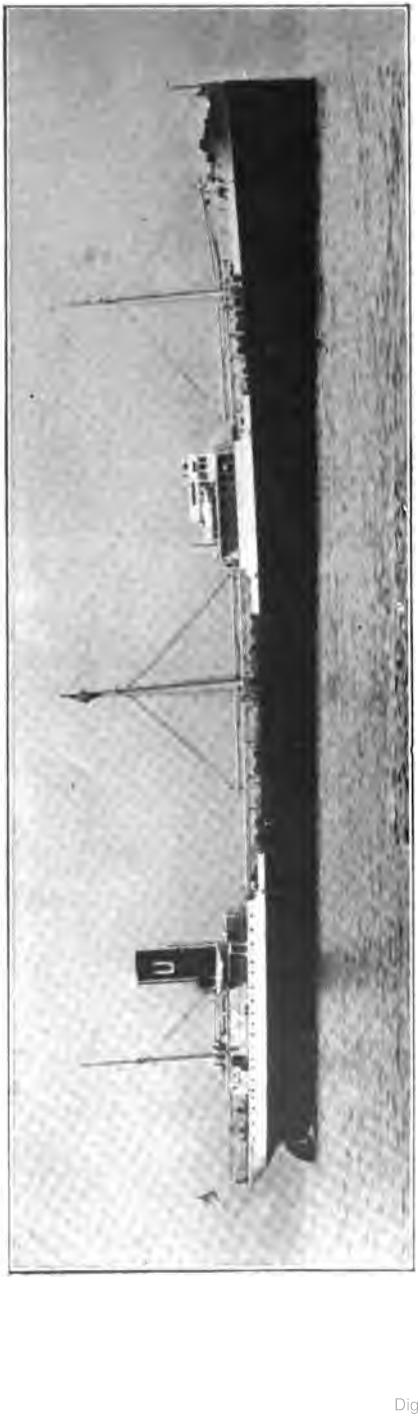

FIG. 244.—S. S. La Brea in ballast trim.

This is the first oil tank steamer equipped with reduction gear turbines. A product of a Western port, its features are representative of typical fuel oil and steam engineering practice that will be encountered in future service on the Pacific Coast. By means of such efficient tankers as this much of the foreign export trade in petroleum on the Pacific Coast is accomplished.

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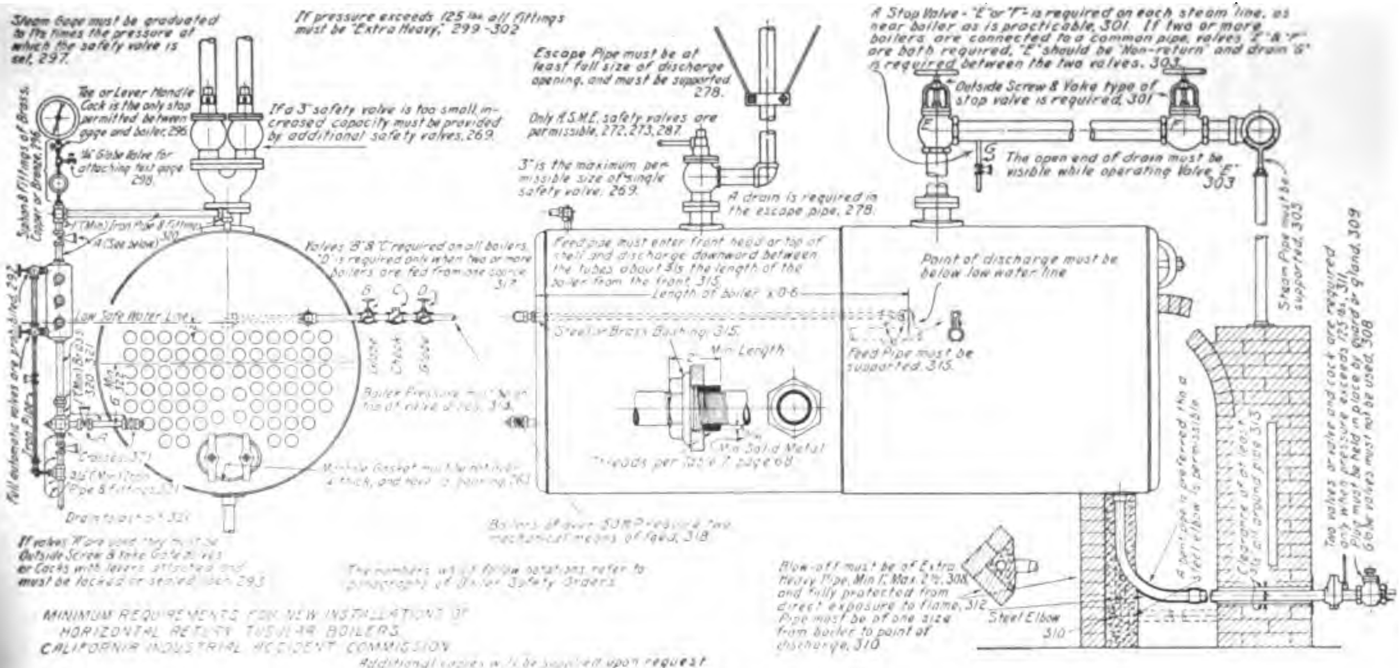

FIG. 245.—Suggestions for safety in boiler installation.

Helpful suggestions for safety in boiler installation have been put into concrete form by the California Industrial Accident Commission. **Here is a reproduction of a drawing which has recently been made up by the Commission to serve as a guide in the installation of boilers. This commission is aiding in more efficient fuel oil uses in steam** generation by making the working conditions of the operator more secure.

In regard to current aids in the technical press, the only paper in existence that devotes a regular department to the technical discussion of fuel oil and steam engineering is the Journal of Electricity published in San Francisco. This journal is now in its thirty-third year of publication and is the recognized authority on this line of discussion. Electrical World and POWER published in New York City also publish much timely and helpful data along the line of fuel oil and steam engineering.

The work of the Pacific Coast Section, N.E.L.A. along lines of fuel oil economy in steam power generation has proven most helpful. Through this means of expression the great power companies of the West which use oil as a fuel are contributing noteworthy aids to efficient uses of this product.

The American Society of Mechanical Engineers and the American Institute of Electrical Engineers constitute the two great national societies of professional status that are exceedingly helpful in fuel oil study.

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APPENDIX III

RULES AND REQUIREMENTS OF THE NATIONAL BOARD OF FIRE UNDERWRITERS FOR THE STORAGE AND USE OF FUEL OIL AND FOR THE CONSTRUCTION AND INSTALLATION OF OIL BURNING EQUIPMENTS

CLASS A

LARGE SUPPLY OR STORAGE TANKS FOR OILS HAVING A FLASH POINT ABOVE 150°F.

(Abel-Pensky Flash Point Test)

This flash point corresponds closely to 160°Fahr. (Tagliabue Open Cup Tester) which may be used for rough estimations of the flash points. These storage tanks are generally used for storage in oil fields, oil refineries or distributing stations and are in most cases installed above ground. The hazards of such systems of storage depend upon the distance from burnable property and upon the topography of the surrounding land.

TABLE 1

Capacity in gallons	Minimum distance of tanks	
	To line of adjoining property which may be built upon, feet	To any other tank, feet
1,000	10	2
2,000	20	2
16,000	25	2
24,000	30	2
36,000	40	3
48,000	50	3
60,000	60	3
96,000	75	3
150,000	85	3
200,000	100	15
300,000	150	25
500,000	250	35
1,000,000	300	50
2,000,000	350	75
Unlimited	400	200

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1. Capacity and Location of Tanks.

(*a*) Tanks to be so located as to avoid undue exposure of adjacent burnable property. The distances specified in Table No. 1 are for plants or storage tanks located outside of fire limits.

(*b*) Tanks to be located at lowest point available and so placed as to avoid possible danger from high water. When near a stream without tide, tanks to be located down-stream from the water front of any adjacent town. On tide water tanks to be located well away from shipping districts.

(*c*) When it is impossible to locate tanks as specified in Rule 1*b*, each tank to be surrounded with an embankment or dike not less than 4 feet in height and having a capacity not less than fifty per cent. greater than the tank to be protected.

(*d*) Embankments or dikes to be made of earth, reinforced concrete or brick. If made of earth, embankments to be firmly and completely built of earth from which stones, vegetable matter, etc., have been removed, and to have a crown of not less than three feet and a slope of at least 2 to 1 on both sides. If made of reinforced concrete or brick to be designed to provide protection equivalent to an earth embankment with a sufficient factor of safety to allow for the effect of fire on the concrete or brick facing.

(*e*) Embankments or dikes to be continuous, with no openings for piping or roadways. Piping to be laid well below the foundation of the embankments and, at points where it is necessary to pass over the embankment, properly built steps or concrete roadway to be provided.

2. Height of Tanks.

Vertical tanks must not exceed 30 feet in height.

3. Material and Construction of Tanks.

(*a*) Tanks must be constructed of iron or steel plates of a gauge depending upon the capacity as specified in the following table.

TABLE 2—THICKNESS OF METAL FOR ABOVE GROUND TANKS.

HORIZONTAL

Maximum diameter	Minimum thickness	
	Heads, in.	Shell, in.
Not over 5 feet	3/16	9/64
5 feet to 8 feet	1/4	3/16
8 feet to 11 feet	3/8	1/4

VERTICAL

Capacity 5,000 gallons or less, diameter less than 40 feet.

Bottom	No. 8 U. S. gauge.
Bottom ring	No. 8 U. S. gauge.
Other rings	No. 10 U. S. gauge.
Top	No. 12 U. S. gauge.

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Capacity 10,000 gallons or less, diameter less than 40 feet.

Bottom	No.	8 U. S. gauge.
Bottom ring	No.	7 U. S. gauge.
Other rings	No.	8 U. S. gauge.
Top	No.	12 U. S. gauge.

Other vertical tanks to be of material having a thickness of not less than indicated in the following. Figures in all columns excepting the first refer to U. S. Standard Gauge.

Diameter in feet	Top	Top ring	2d ring from top	3d ring from top	4th ring from top	5th ring from top	6th ring from top	Bottom
80	10	7	7	3	0	3–0	5–0	10
75	10	7	7	4	1	2–0	4–0	10
70	10	7	7	4	1	2–0	4–0	10
65	10	7	7	5	1	0	3–0	10
60	10	7	7	5	2	0	2–0	10
55	10	7	7	6	3	1	2–0	10
50	10	7	7	7	4	1	0	10
45	10	7	7	7	5	3	1	10
40 & less	10	7	7	7	5	3	2	10

All riveted joints to have an efficiency of at least 60 per cent.

Tanks of greater capacity than given above shall be of material of sufficient thickness to safely hold the contents and proportionately heavier.

NOTE.—For materials to be used in smaller tanks refer to Table No. 4 giving weights of material for underground storage tanks.

(*b*) All joints of tanks must be riveted and soldered, riveted and caulked, brazed or welded, or made by some equally satisfactory process. Tanks must be tight and sufficiently strong to bear without injury the most severe strains to which they are liable to be subjected in transportation or use. Tanks shipped complete must be suitably reinforced to prevent injury to the joints.

(*c*) Tanks must be provided with a vent pipe terminating in a weatherproof hood containing a non-corrodible screen. In case the vent pipe is not permanently open a suitable safety relief must be provided. When, in order to provide a means for relieving pressure, manhole covers are not provided with bolts or clamps, the openings must be protected by a non-corrodible wire mesh screen (not less than 20 × 20 meshes per square inch) which may be removable but must be normally securely held in place.

(*d*) Outside surfaces of tanks must be thoroughly protected against corrosion by a suitable rust-resisting paint.

4. Support for Tanks.

Tanks to be set upon a substantial foundation, and when elevated above the ground level, supports are to be of non-combustible material, with

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exception of suitable wooden cushions. All above-ground tanks to be thoroughly grounded electrically.

5. Means for Extinguishing Fires in Tanks.

(a) Each tank to be equipped with an independent steam pipe for use in case of fire, the outlet of which is to be inside the tank, above the surface of the oil.

This pipe to be of ample capacity, but never smaller than ½ inch.

(b) Steam which is to be supplied from conveniently located boilers to be controlled by valves outside the embankments surrounding the tanks.

NOTE.—Systems providing protection equivalent to the above may be used. Such systems generally embody the use of blanketing gas, and when the source of supply is thoroughly reliable, may be satisfactorily employed where a supply of steam is not available.

6. Pumps.

Pumps used in connection with the supply and discharge of the tank shall be located outside of the reservoir walls and at such a point that they will be accessible at all times, even if the oil in the tank or reservoir should be on fire.

7. Pipe Connections.

All oil conveying pipes to be laid underground, but under no circumstances shall they break through the reservoir walls.

The above rule does not apply to pipes passing under the reservoir wall and laid well below the surface of the ground.

8. Controlling Valves.

(a) There shall be a gate valve located at the tank in each oil conveying pipe. In case two or more tanks are cross-connected there shall be a gate valve at each tank in each cross-connection.

(b) There shall be a gate valve in the discharge and suction pipes near the pump and a check valve in the discharge pipe, located underground.

9. Indicator.

There shall be a reliable indicator to show the level of the oil in the tank. Indicator to be of such a form that its derangement will not permit escape of oil.

10. Plans and Specifications.

A complete set of plans and specifications of proposed installation shall be submitted to the Inspection Department having jurisdiction before beginning construction.

CLASS B

Individual Oil-burning Equipments for Other Than Household Purposes

Apparatus using oil for fuel, however safe-guarded, introduces a distinct increase in hazard which should be recognized.

Where used, the following rules should be rigidly observed.

All oil used for fuel purposes under these rules shall show a flash test of not less than 150°F. (Abel-Pensky Flash Point Tester.) This flash point

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corresponds closely to 160°F. (Tagliabue Open Cup Tester), which may be used for rough estimations of the flash point.

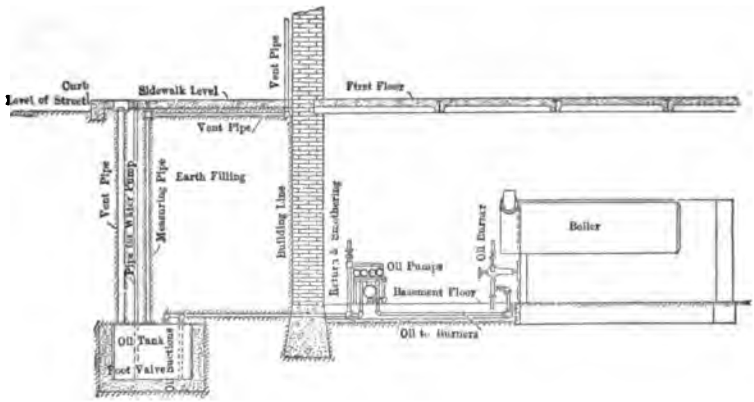

FIG. 246.—A diagrammatic sketch of an oil tank shown below the basement floor level when boiler is located in a basement where the sidewalk is not excavated.

11. Capacity and Location of Tanks.

(*a*) In closely built up districts or within fire limits tanks to be located underground with tops of tanks not less than three feet below the surface of

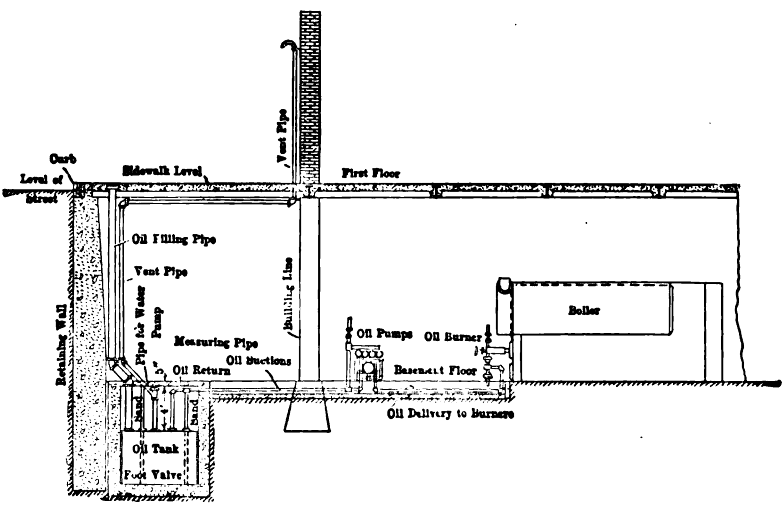

FIG. 247.—Here is shown an oil tank located 4 feet below the boiler-room floor in which the boiler is in the basement and the sidewalk has been excavated. This construction is agreeable to the rule of the Board of Fire Underwriters.

the ground and below the level of the lowest pipe in the building to be supplied. Tanks may be permitted underneath a building if buried at least

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three feet below the basement floor which is to be of concrete not less than 6 inches thick. Tanks shall be set on a firm foundation and surrounded with soft earth or sand, well tamped into place. No air space shall be allowed immediately outside of tanks. Tank may have a test well, provided test well extends to near bottom of tank, and top end shall be hermetically sealed and locked except when necessarily open. When tank is located underneath a building the test well shall extend at least 12 feet above source of supply. The limit of storage permitted shall depend upon the location of tanks with respect to the building to be supplied and adjacent buildings, as given in the following table.

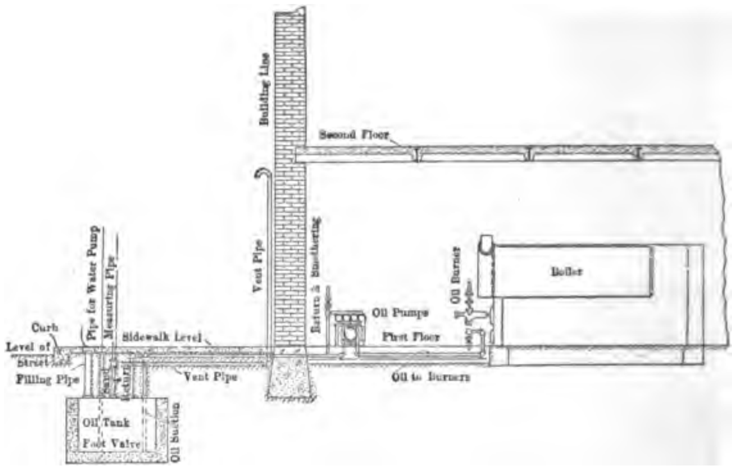

FIG. 248.—The view shows an oil tank below the level of the boiler room floor where the sidewalk has been excavated. The installation is agreeable to the Board of Fire Underwriters' specifications.

TABLE 3.—PERMISSIBLE AGGREGATE CAPACITY IF LOWER THAN ANY FLOOR, BASEMENT, CELLAR OR PIT IN ANY BUILDING WITHIN RADIUS SPECIFIED

Capacity	Radius, feet
Unlimited	50
20,000 gallons	30
5,000 gallons	20
1,500 gallons	10
*500 gallons	Less than 10

* In this case tank to be entirely encased in 6 inches of concrete.

(b) When located underneath a building no tank to exceed a capacity of 9,000 gallons and basement floors to be provided with ample means of support independent of any tank or concrete casing.

(c) Outside of closely built up districts or outside of fire limits, aboveground storage tanks may be permitted as specified in Rule 1, provided

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drainage away from burnable property in case of breakage of tanks is arranged for or suitable dikes built around the tanks. When dikes are employed the distances specified in Table 1 are to be taken as distances to nearest points of dikes.

When above-ground tanks are used all piping must be arranged so that in case of breakage of piping the oil will not be drained from tanks. This requirement prohibits the use of gravity feed from storage tanks. Above-ground tanks of less than 1,000 gallons capacity without dikes may be permitted in case suitable housings for the protection of the tanks against njury are provided.

12. Material and Construction of Tanks.

(*a*) Tanks must be constructed of iron or steel plate of a gauge depending upon the capacity as specified in the following tables:

TABLE 4.—UNDERGROUND TANKS INSIDE OF SPECIFIED FIRE LIMITS, OR WITHIN TEN FEET OF A BUILDING WHEN OUTSIDE SUCH LIMITS

Capacity, gallons	Minimum thickness of material
1 to 560	14 U. S. gauge
561 to 1,100	12 U. S. gauge
1,101 to 4,000	7 U. S. gauge
4,001 to 10,500	1/4″
10,501 to 20,000	5/16″
20,001 to 30,000	3/8″

TABLE 5.—UNDERGROUND TANKS OUTSIDE OF SPECIFIED FIRE LIMITS, PROVIDED THE TANKS ARE TEN FEET OR MORE FROM A BUILDING

Capacity, gallons	Minimum thickness of material
1 to 30	18 U. S. gauge
31 to 350	16 U. S. gauge
351 to 1,100	14 U. S. gauge
1,101 to 4,000	7 U. S. gauge
4,001 to 10,500	1/4″
10,501 to 20,000	5/16″
20,001 to 30,000	3/8″

Tanks of greater capacity than 30,000 gallons must be made of proportionately heavier metal.

(*b*) All joints of tanks must be riveted and soldered, riveted and caulked, welded or brazed together, or made by some equally satisfactory process. To be tight and sufficiently strong, to bear without injury the most severe strains to which they are liable to be subjected in practice. The shells of tanks to be properly reinforced where connections are made and all connec-

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tions should as far as practicable be made through the upper side of tanks above oil level.

(c) Tanks shall be thoroughly coated on the outside with tar, asphaltum or other suitable rust-resisting material.

NOTE.—The protection required for tanks will depend upon the condition of the soil in which they are installed. When the soil is impregnated with corrosive materials tanks should be made of heavier metal in addition to being protected as specified above.

13. Fill and Vent Pipes.

(a) Each underground storage tank having a capacity of over 1,000 gallons to be provided with at least a 1 inch vent pipe extending from the top of the tank to a point outside of building. Vent pipe to terminate at a point at least 12 feet above the level of the top of the highest tank car or other reservoir from which the storage tank may be filled. Terminal to be provided with a hood or goose neck protected by a non-corrodible screen and to be located remote from fire escapes and never nearer than 3 feet, measured horizontally and vertically, from any window or other opening. Vent pipes from two or more tanks may be connected to one upright, provided the connection is made at a point at least one foot above level of source of supply.

(b) Tanks having a capacity of less than 1,000 gallons may be provided with combined fill and vent pipes so arranged that the fill pipe cannot be opened without opening the vent pipe, these pipes to terminate in a metal box or casting provided with a lock.

(c) Fill pipes for tanks which are installed with permanently open vent pipes must be provided with metal covers or boxes which are to be kept locked except during filling operations.

(d) Fill and vent pipes for tanks located under buildings are to be run underneath the concrete floor to outside of building.

14. Indicator.

Some device for indicating the level of the oil is desirable. Where used, such attachment shall be connected through substantial fittings so as to minimize exposure of the oil, and devices the breakage of which will allow the escape of oil, must not be used.

15. Filters.

Suitable filters or strainers for the oil should be installed and preferably be located in supply line before reaching pump. Filter to be arranged so as to be readily accessible for cleaning.

16. Feed Pumps.

(a) Must be of approved design, secure against leaks.

NOTE.—Stuffing box, if used, should be provided with a removable cupped gland designed to compress the packing against the shaft and arranged so as to facilitate removal. Packing affected by the oil must not be used.

(b) To be arranged so that dangerous pressures will not be obtained in any part of system, and it is further recommended that feed pumps be interconnected with pressure air supply to burners in order to prevent flooding.

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17. Gauge Glasses and Pet Cocks.

Glass gauges, the breakage of which would allow the escape of oil, are to be avoided. If their use is necessary, they should have substantial protection or be arranged so that oil will not escape if broken. Pet cocks must not be used on oil carrying parts of system.

18. Receivers or Accumulators.

(*a*) If used, they must be designed so as to secure a factor of safety of not less than 6. Must be subjected to a pressure test of not less than twice the working pressure.

(*b*) The capacity of the oil chamber must not exceed 10 gallons.

(*c*) To be equipped with pressure gauge.

(*d*) To be provided with an automatic relief valve set to operate at a safe pressure and connected by an overflow pipe to supply tank, and so arranged that the oil will automatically drain back to the supply tank immediately on closing down the pump.

19. Standpipes.

(*a*) If used, their capacity shall not exceed 10 gallons.

(*b*) To be of substantial construction, equipped with an overflow, and so arranged that the oil will automatically drain back to the supply tank on shutting down pump, leaving not over one gallon, where necessary, for priming, etc.

(*c*) If vented, the opening should be at the top and may be connected with the outside vent pipe from storage tank, above level of source of supply.

20. Piping.

(*a*) Standard full weight wrought iron, steel or brass pipe with substantial fittings to be used and to be carefully protected against injury. Piping under pressure must be designed to secure a factor of safety of not less than 6, and after installation to be tested to a pressure not less than twice the working pressure.

(*b*) Piping to be run as directly as possible, and laid so that the pipes are pitched toward the supply tanks without traps.

(*c*) Overflow and return pipes to be at least one size larger than the supply pipes, and no pipe to be less than one-half inch pipe size.

(*d*) All connections to be perfectly tight with well-fitted joints. Unions, if used, to be of approved type having at least one face of the joint made of brass and having conically faced seats, obviating the use of packing or gaskets.

(*e*) Pipes leading to the surface of the ground to be cased or jacketed where necessary to prevent loosening or breakage, and proper allowance should be made for expansion and contraction, jarring and vibration.

(*f*) Connection to outside tanks to be laid below the frost line and not to be located near nor placed in same trench with other piping.

(*g*) Openings for pipes through outside walls to be securely cemented and made oil tight.

21. Valves, etc.

(*a*) Readily accessible shut-off valves to be provided in the supply line as near to the tank as practicable, and additional shut-offs to be installed in the main line inside building and at each oil consuming device.

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(b) Controlling valves in which oil under pressure is in contact with the stem to be provided with stuffing box of liberal size, containing a removable cupped gland designed to compress the packing against the valve stem and arranged so as to facilitate removal. Packing affected by the oil must not be used.

(c) The use of approved automatic shut-offs for the oil supply in case of breakage of pipes or excessive leakage in building is recommended.

CLASS C

Oil Conveyor or Carriers

22. Steamers, etc.

(a) Steamers, barges or vessels loading or discharging oil in bulk, shall not load or discharge at wharves other than those used by the oil company, and such wharves shall be well isolated from all burnable property or wharfage.

(b) There shall be a gate valve immediately at the point in the pipe line where connection is made with the hose leading to the ship for the purpose of shutting off the oil, and there shall be another gate valve in this line of pipe at a distance of at least 10 feet back from the wharf, where it will be readily accessible for the purpose of shutting the oil off in event of failure on the part of the valve first mentioned.

(c) A tight connection shall be made with the hose length at the wharf by means of a carefully threaded coupling, to prevent leakage and accumulation of oil around the piers.

(d) Lights. No fire nor open lights to be allowed on the vessel while at the wharf.

CLASS D

Apparatus for Cooking and Heating for Household Use

The use of oil as fuel for domestic purposes is regarded from the insurance viewpoint as more hazardous than the use of ordinary fuel, such as coal, wood and coke.

Where these systems are used their hazards should be recognized and the following rules and precautions should be observed. All oil used for fuel under these rules shall show a flash test of not less than 150°F. (Abel-Pensky Flash Point Tester.) This flash point corresponds closely to 160°F. (Tagliabue Open Cup Tester), which may be used for rough estimations of the flash point.

23. Capacity and Location of Storage Tanks. (See Rule 11.)

24. Material and Construction of Tanks. (See Rule 12.)

25. Fill and Vent Pipes. (See Rule 13.)

26. Pump.

Oil pump used in filling auxiliary tank from the main supply tank to be approved type, secure against leaks, with check valves located as close to the pump as convenient. Pumps should be rigidly fastened in place.

Note.—Stuffing box, if used, should be provided with a removable cupped gland designed to compress the packing against the shaft and

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arranged so as to facilitate removal. Packing affected by the oil must not be used.

27. Auxiliary Supply Tank.

(*a*) If used, shall not exceed five gallons in capacity, except by special consent of the Inspection Department having jurisdiction.

(*b*) Shall be located at least 10 feet, measured horizontally, from the burners.

(*c*) Shall be provided with an overflow connection draining to the supply tank and a vent pipe leading outside the building, the latter to have a weatherproof hood. To be constructed of brass, copper or galvanized plate not less than 0.050″ (No. 18 U. S. standard gauge) in thickness. Joints to be made as specified for outside storage tanks.

28. Piping.

(*a*) Standard, full weight, wrought iron, steel or brass pipe with substantial fittings to be used and to be carefully protected against injury.

(*b*) Supply pipe to be not less than one-fourth inch sizes, and overflow and return pipes to be at least one size larger.

(*c*) Pipe connections to tanks to be suitably reinforced and proper allowance to be made for expansion and contraction, jarring and vibration.

(*d*) Openings for pipes through outside walls to be securely cemented and made oil tight.

(*e*) All connections to made perfectly tight with well fitted joints. Unions, if used, to be of approved type, having at least one face of the joint made of brass and having a conically faced joint, obviating the use of packing or gaskets.

(*f*) Piping to be run as directly as possible, and to be laid so that the pipes are pitched back to the storage tank without traps.

29. Valves.

(*a*) Readily accessible valves to be provided near each burner and also close to the auxiliary tank in the pipe leading to burners.

(*b*) Controlling valves to be constructed as specified in Rule 21-b.

30. Installation of Burners.

(*a*) *Overflow.*—Burners shall be installed with oerflow attachment so arranged that any surplus oil will drain by gravity from the burner through a pipe into a substantially constructed reservoir having a capacity of not less than that of the auxiliary tank.

Each tank to be constructed and vented as provided for auxiliary tanks. (See Rule 27.)

(*b*) *Draughts.*—No dampers to be used in smoke pipe between burner and chimney. Any regulation of draught which is necessary is to be accomplished through the dampers in front.

31. Construction of Burners.

(*a*) The size of the orifice through which the oil is supplied to the burners should be limited to furnish only sufficient oil for the maximum burning conditions when the controlling valves are wide open.

(*b*) Valves to be arranged so as not to enlarge the orifice.

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(*c*) Burners containing chambers which allow the dangerous accumulation of gases are prohibited.

(*d*) Burners containing oil conveying pipes or parts subject to intense heat or subject to stoppage from carbonization are prohibited.

(*e*) Burners should be designed so that they can be easily cleaned, and so as not to allow leakage of oil.

32. Instruction Card.

A card giving complete instructions in regard to the care and operation of the system to be permanently placed near the apparatus.

RULES GOVERNING CONSTRUCTION AND INSTALLATION OF OIL BURNING EQUIPMENT AND STORAGE AND USE OF FUEL OILS ADOPTED BY THE BOARD OF STANDARDS AND APPEALS CITY OF NEW YORK

FUEL OIL RULES

Rule 1. Definition. Flash Point and Specific Gravity.—The term "oil used for fuel purposes" under these rules includes any liquid or mobile mixture, substance or compound derived from or including petroleum.

All oil used for fuel purposes under these rules shall show a minimum flash point of not less than one hundred and seventy-five (175) degrees Fahrenheit, in an open cup tester, or if closed cup tester be used a minimum of not less than one hundred and fifty (150) degrees Fahrenheit, and its specific gravity shall be not less than 0.933 (20°B.) at a temperature of sixty (60) degrees Fahrenheit; and must not be fed from the tank to the suction pump at a pre-heat temperature higher than its flash point.

Rule 2. Manner of Storage.—Oil to be used as fuel for commercial, heating and power purposes on the premises where stored shall be at all times contained in metal tanks with all openings or connections through the tops of the tanks, except a clean-out plug in the bottom; and, when located inside of a building, must at all times be placed in the cellar or lowest story of such building, and at least two (2) feet in a horizontal direction from any supporting portion of the structure, and if practicable shall be buried underneath the lowest floor or ground.

Rule 3. Location of Tanks. Existing Buildings.—No storage of fuel oil shall be permitted in a building of frame construction within the fire limits, or in buildings of hazardous occupancy as so defined by the fire commissioner.

If placed in buildings already erected, if not buried beneath the lowest floor or ground, such tanks shall be placed in an enlosure the floor of which shall be at least three (3) feet below the surface of the cellar or lower story; or if by reason of water or foundation conditions, or if on rock bottom, the tank may be placed above the surface of the ground, but in any case subject to the conditions as hereinafter described under Rule 5.

Rule 4. Location of Tanks—New Buildings.—In buildings hereafter erected the bottom of the fuel oil service tanks shall be located in, or below

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the floor level of the cellar or lowest story as shall be determined by the Superintendent of Buildings under the provisions of Rule 2.

Rule 5. Enclosure of Tanks.—In either existing or new buildings such fuel oil service tanks shall be enclosed in an unpierced wall and floor of approved masonry or reinforced concrete, made oilproof and waterproof, and not less than twelve (12) inches in thickness; and also of sufficient thickness to properly support any lateral pressure, and to be of lateral dimensions at least one (1) foot greater on all sides than the outside dimensions of the tank. These walls are to be carried up to a height of at least one (1) foot above the tank, or the supply and feed connections thereto, and roofed over with reinforced concrete or its equivalent at least twelve (12) inches thick and capable of sustaining a live load of at least three hundred (300) pounds per square foot; and if not buried below the ground, placed so as to leave a clear and open space (except for pipe connections) of at least two (2) feet between such roof over the enclosure and the underside of the ceiling above. The roof of every enclosure shall contain a manhole with fireproof cover properly weighted, but not fastened, placed immediately above the supply and feed connections and the manhole in top of the tank.

Where found impractical to set the bottom of the tank three (3) feet below the floor of the cellar or lowest story, the tank shall rest on steel or masonry supports, and the bottom of the tank shall be at least one (1) foot above the floor of the enclosure, and the enclosure wall and floor as above specified shall be unpierced and the space below the horizontal centre line of the tank and within the enclosure formed by the surrounding unpierced walls shall have a capacity of at least sixty (60) per cent. of the capacity of the tank.

The space within the enclosure surrounding the tank shall be at all times vented to the air outside of the building by iron or other fireproof conduit at least two and one-half (2½) inches diameter, connecting the enclosure at a point just above the floor level, and which shall finish above the street surface with proper connection at that point to permit the Fire Department to flood the enclosure.

A separate similar vent without Fire Department connection shall enter the enclosure just below its ceiling.

Rule 6. Capacity of Tanks.—In existing or new buildings of non-fireproof construction no fuel oil service tank containing over ten thousand two hundred (10,200) gallons, and in buildings of fireproof construction no tank containing over twenty thousand (20,000) gallons, shall be placed in any single portion of the cellar or lowest story unless such portion be separated from the rest of the cellar by walls of masonry or reinforced concrete with openings protected by automatic fireproof doors, with sills placed high enough above the cellar floor to contain capacity of tank located therein, in addition to the enclosure as already specified for the tank, and such portion be ventilated to the outer air. More than one such single tank may be installed if enclosed and separated as above.

When tanks are buried so that the top of the roof over the enclosure wall is level with the cellar floor, the capacity of any such tank may be increased by one hundred (100) per cent.

Rule 7. Service Tanks Located Outside of Buildings Within Fire Limits.—Within the fire limits, tanks to contain oil for use on the premises,

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and of a capacity and at distances specified below, may be placed above ground outside of the building if such tank does not exceed fifteen (15) feet in height above the surface of the ground and if completely enclosed in the same manner as provided for in Rule 5.

Distance to nearest building in feet not exceeding	Capacity in gallons
40	71,400
30	40,800
20	30,600
10	20,400
5	10,200

If such service tanks are entirely buried and roofed below the surface of the ground, the capacity in gallons may be increased by two hundred (200) per cent.

Rule 8. Outside General Storage Fuel Oil Tanks Located Above Ground Within the Fire Limits.—Such general storage tanks located within the fire limits shall not exceed twenty-five (25) feet in height, shall be built of metal, and shall be surrounded with a dike of unpierced masonry or reinforced concrete not less than four (4) feet in height, with a capacity of at least that of the tank to be protected. The walls and floor of such dikes must be continuous, and oilproof and waterproof, and must not be built within ten (10) feet of the walls of the tank. If tanks are placed in battery the dikes shall be rectangular in shape, and the dike wall separating them as well as the dike wall within one hundred (100) feet of any structure, shall be carried up as a fire stop to a height of four (4) feet above the head of the tank and coped with stone or concrete, and any openings in walls above the dike shall have automatic fireproof doors.

The capacity of any such single general storage tank within the fire limits shall not exceed one hundred thousand (100,000) gallons, and the gross capacity of storage shall not exceed the following tables:

To line of adjoining property or nearest building (feet)	Gallons
75	100,000
100	150,000
150	250,000
200	500,000

Such general storage tanks may have extra fill and emptying connections as the Fire Commissioner may determine.

Rule 9. Outside General Storage Fuel Oil Tanks Located Outside the Fire Limits.—Such general storage tanks shall be protected by dikes and fire stops as provided under Rule 8, shall not exceed thirty-five (35) feet in height above the ground, and may be constructed either of metal or of concrete reinforced with steel in order to resist the oil pressure.

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If built of concrete, the walls and floor of such tanks shall be continuous and shall be not less than eight (8) inches thick, mixed in the proportion of 1 : 1½ : 3 graded and mixed in accordance with the requirements of Chapter 5, Code of Ordinances. The walls shall be of sufficient thickness so that the tensile stress, disregarding the steel reinforcement, shall not exceed one hundred and fifty (150) pounds per square inch. The horizontal and vertical reinforcement shall be properly proportioned and placed to provide for expansion and shrinkage without leakage, and the stress in the steel shall not exceed ten thousand (10,000) pounds per square inch.

As soon as the concrete has hardened sufficiently to be self-sustaining, the forms shall be removed and all cavities filled with a one to one (1 : 1) mortar thoroughly rubbed in and all irregularities trowelled smooth.

The concrete shall harden at least twenty-eight (28) days before use, and the surface of the floor and the interior surface of the walls shall be protected by coating with a sodium silicate solution or other equally good protection to prevent oil coming in contact with the concrete.

The maximum gross capacity of any such single tank when situated outside the fire limits shall not exceed two hundred and fifty thousand (250,000) gallons, but the gross storage capacity may be double that specified in the tables under Rule 8; and when such tanks are placed at least two hundred and fifty (250) feet from the line of adjoining property or the nearest building, the gross capacity may be unlimited.

Rule 10. Material and Construction of Tanks.—1. All fuel oil storage within the fire limits shall be constructed of wrought iron, galvanized steel, basic open hearth or electric steel plates of gauge corresponding to the capacity as specified in the following tables:

TANKS PLACED UNDERGROUND

Capacity in gallons	Thickness of material U. S. gauge
500	14
1,000	12
5,000	7
10,000	¼ inch
20,000	5/16 inch
30,000	⅜ inch

TANKS PLACED ABOVE GROUND (Horizontal)

Maximum diameter in feet	Thickness of material U. S. Gauge	
	Heads	Shell
5	7	10
8	¼ inch	7
11	⅜ inch	¼ inch

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Tanks Placed Above Ground (Vertical)
Thickness of Material, U. S. Gauge

Diameter feet	Top	Top ring	2d ring from top	3d ring from top	4th ring from top	5th ring from top	6th ring from top	Bottom
40 and less	10	7	7	7	5	3	2	10
45	10	7	7	7	5	3	1	10
50	10	7	7	7	4	1	0	10
55	10	7	7	6	3	1	2–0	10
60	10	7	7	5	2	0	2–0	10
65	10	7	7	5	1	0	3–0	10
70	10	7	7	4	1	2–0	4–0	10
75	10	7	7	4	1	2–0	4–0	10
80	10	7	7	3	0	3–0	5–0	10

2. Tanks of greater capacity than above shall be proportionately heavier and of sufficient thickness to safely hold the contents.

3. All joints shall be riveted and caulked, brazed, welded, or made by some equally satisfactory process, and the tanks braced sufficiently to withstand all stresses due to transportation or use. All riveted joints shall have an efficiency of not less than sixty (60) per cent.

4. The top cover shall be of the same material as used in the construction of the tank, permanently secured to the tanks without other openings than provided for in these rules. A safety valve shall be installed on all tanks placed outside of buildings.

5. All outlets and inlets shall be through the top or cover of the tank, except for the clean-out plug as provided for under Rule 2, and in general storage tanks a water drain not exceeding one (1) inch diameter may be permitted.

6. All metal tanks shall be thoroughly coated on the outside with tar, asphaltum, or other suitable rust-resisting protection. When buried in soil impregnated with corrosive materials, steel tanks shall be entirely covered with a two-inch thickness of cement mortar or shall be of heavier metal in addition to being protected as specified.

7. All above ground storage tanks exceeding two hundred thousand (200,000) gallons capacity shall be provided with approved explosion hatches having a combined area of not less than one and one-half (1½) per cent. of the roof area of the tank.

8. All tanks shall be tested and must withstand a pressure of not less than twenty-five (25) pounds per square inch shop test.

Rule 11. Vent and Fill Pipes.—1. Each fuel oil tank shall be provided with a separate steel vent pipe and a separate steel fill pipe of at least two (2) inches diameter placed in the top of the tank. The vents for enclosure around tank shall be as specified under Rule 5.

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2. Vent pipes for fuel oil tanks located in the lower story or buried under buildings shall be run to a point outside the building, above the street surface and at least twelve (12) feet above the fill pipe and shall terminate in a weatherproof hood or a gooseneck, protected with non-corrodable screens of not less than thirty by thirty (30 × 30) nickel mesh or equivalent. Such vent shall not be located within five (5) feet either vertically or horizontally of a window or other opening or an exterior stairway or fire escape.

3. The receiver terminal of fill pipes shall be located in a metal box or casting provided with means for locking and the delivery terminal shall be connected through the top of the tank at a point furthest remote from the vent.

Rule 12. Fuel Oil Feed Systems.—1. Systems fed by gravity or force systems between tank and pump shall not be permitted.

2. Pump suction feed systems only will be approved and anti-syphon system must be provided.

Rule 13. Pumps and Piping.—1. Feed pumps for fuel oils shall be of approved design, so arranged that dangerous pressures will not obtain in any part of the system and shall be located outside of enclosure walls around storage tanks, but so placed as to be accessible at all times, and provision shall also be made for remote control. They shall be installed in duplicate when directed by the Fire Commissioner and shall be provided with a by-pass to permit the draining of the oil for repairs.

A separate hand pump shall be provided for starting purposes.

2. Oil conveying pipes shall be carried above the tank outlet; if laid underground after leaving the tank to be carried in a separate trench enclosed in fireproof or non-conducting material. They shall be of extra heavy standard wrought iron, steel or brass pipe with substantial fittings and not less than one-half (½) inch in size and if covered it shall be with asbestos or other approved fireproof material. Overflow pipes shall be at least one size larger than supply pipes and shall be carried back to the receiver terminal.

3. All connections shall be tight with well-fitted joints. Unions shall have at least one face made of brass with conically-faced seats.

4. Connections leading to outside tanks shall be laid below the frost line and shall not be located near or placed in same trench with piping other than steam lines for heating. All pipes leading to the surface of the ground shall be cased or jacketed to prevent loosening or breakage. Openings for pipes through outside walls below the ground level shall be securely cemented and made oil-tight.

5. Piping shall be run as directly as possible, without sags, and be properly supported to allow for expansion, contraction, jarring and vibration and draining.

6. Piping between any separated oil container or using parts of the equipment, should be laid as far as practicable outside of the building, underground, and inside piping in a trench with metal cover or protected by not less than three (3) inches of concrete.

7. Piping under pressure must be designed with a factor of safety of not less than six (6), and shall in every case be tested to a pressure of not less than one hundred and fifty (150) pounds after installation.

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Rule 14. Controlling Valves.—1. In fuel oil piping systems, readily accessible shut-off valves shall be provided in the supply line of fuel oils as near to tank as practicable, on both sides of any strainer which may be installed in pipe lines, in the main line inside the building, at each oil consuming device, and a gate valve in the discharge and suction pipes near the pump. Provision shall be made to insure the cessation of oil supply from tank to the burner when the pump is not in work.

Rule 15. Heating.—1. All heating to reduce viscosity of fuel oils in storage tanks in any building shall be only by means of hot water coils and the oil shall not be heated above one hundred and forty (140) degrees Fahrenheit.

2. All outside pipes subject to freezing shall be protected with a heating line of steam or hot water.

Rule 16. Fuel Oil Burners.—1. Burners containing chambers which allow dangerous accumulation of gases or containing oil-conveying pipe or parts subject to intense heat or stoppage from carbonization are prohibited.

2. Oil shall be supplied through orifices not larger than necessary to supply sufficient oil for maximum burning conditions when the controlling valves are wide open.

3. The mechanism shall be so designed that, where manual or automatic control is provided, operated at some distance from the burner, the flame cannot be extinguished except by closing the main shut-off valve in line to burner. Approved gas-pilot lights or equivalent will be acceptable.

4. A check valve of approved type shall be installed in each oil, steam and air line near the burner.

5. Smoke pipes shall be installed between the burners and chimney, and any dampers in smoke pipes shall not exceed eighty (80) per cent. of the area of the pipe. Necessary regulation of draft shall be accomplished by dampers in the fire or ash pit doors.

6. Burners shall be installed with overflow attachment so arranged that surplus oil will drain by gravity from the burner into a substantially constructed reservoir. Such reservoir shall be constructed of brass, copper or galvanized iron plate not less than No. 18 U. S. gauge in thickness and shall be provided with a vent pipe with weatherproof hood leading outside the building.

7. The supply of oil and air or steam for atomizing shall be interlocked, so that if the steam or air should fail the oil will be automatically shut off.

Rule 17. Fuel Oil Fire Extinguishing Equipment.—1. Every tank with a capacity of over ten thousand (10,000) gallons shall be equipped with a system of steam pipes, blanketing gas or other approved system for use in case of fire, so arranged and installed as to adequately protect surrounding property.

2. When steam is used, the steam supply pipe shall not be less than one-half (½) inch in size, the boilers shall be conveniently located, and shall be controlled by valves outside the tank enclosure.

Rule 18. General Devices.—All devices used in connection, with oil-burning apparatus, such as indicators, gauges and burners, shall be of such character as to minimize leakage and exposure of oil, and shall be connected

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through substantial fittings. Devices which are subject to breakage and escape of oil shall be prohibited.

Thermometers with large clear reading scales, placed in approved thermometer wells with screwed top connections, shall be installed at convenient and prominent positions in the oil supply pipe lines between the service tank and the pumps and also between the pumps and the burner, to indicate the temperature of the oil.

Rule 19. Instruction Cards.—Cards giving complete instructions for the care and operation of the fuel oil system shall be permanently fixed near the apparatus.

Rule 20. Operation of Plant.—Such fuel oil-burning plants may be operated only by a licensed engineer or by a licensed operator who shall be a citizen of the United States, who can read and write the English language, and who is familiar with the practical working of such plant, as evidenced by the certificate of the Fire Commissioner.

Rule 21. Installation.—No installation of fuel oil plants shall be commenced until after the approval of plans by the Fire Commissioner, which plans shall be submitted to him for examination, together with the certificate of the Superintendent of Buildings that the proposed construction of the enclosure and the location of tanks is in accordance with the requirements of the Building Code and of these Rules.

Adopted, Nov. 6, 1919.

JOHN P. LEO, *Chairman.*

WM. WIRT MILLS, *Secretary.*

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APPENDIX IV

FUEL OIL DATA—APPROXIMATE VALUES

1 barrel of oil .... 42 gallons
1 barrel oil at 16°Baumé .... approximately 336 lbs.
1 gallon oil at 16°Baumé ....approximately 8 lbs.

Specific heat of fuel oil .... 0.498 to 0.5

Coefficient of expansion of California oils:

0.0004 per 1°F.
0.00072 per 1°C.

Heating value of fuel oil (approximate):

Heat units in 1 pound oil........................ 18,500 B.t.u.
Heat units in 1 gallon oil (16°Baumé)............. 148,000 B.t.u.
Heat units in 1 barrel oil (16°Baumé). .... 6,216,000 B.t.u.
Latent heat of vaporization of oil............ 100–130 B.t.u. per lb.

Viscosity.

COMPARISON OF VISCOSITY SCALES. (Approximate.)

Engler, deg.	Saybolt, secs.
1	30
10	350
20	680
40	1360
60	2050
200	7000

Viscosity of water at 60°F.–1° Engler or 30 sec. Saybolt

Viscosity of Coalinga (Cal.) oil of 16°Bé. gravity.

Temp. °F.	Viscosity °Engler
110	15
120	12
125	10
140	7
180	3
212	2

Evaporation from and at 212° per lb. oil at 78 per cent. efficiency ..... 15 lb.
Actual evaporation from 100°F. feed temperature to 150 lb. pressure per lb. oil at 78 per cent. efficiency.............................. 13 lb.

1 ton of coal (2000 lb.) containing 11,500 B.t.u. per lb. is equivalent to 3 barrels of oil.

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Oil required to produce one boiler horsepower............ 2½ lb. per hr.
Boiler horsepower produced from 1 barrel oil per hr................ 134.

1 boiler horsepower = 34½ lb. water evaporated from and at 212°F. per hr.
1 boiler horsepower = 33,479 B.t.u. per hr.
1 mech. horsepower = 33,000 ft.-lb. per min.
= 2,545 B.t.u. per hr.
1 boiler horsepower = 13.14 mech. horsepower.

Air required per pound oil for complete combustion:

Theoretical requirements........................	13 lb.
15 per cent. excess air (14 % CO_2).................	15 lb.
50 per cent. excess air (10 % CO_2).................	19½ lb.

Steam required to atomize 1 lb. oil...................... 0.25—0.5 lb.
Per cent. of total steam generated required to atomize oil.. 2 %—4 %
Heat transfer in oil heaters (heated by condensing steam):

B.t.u. per hr. per sq. ft. per deg. difference in temperature.................................... 15 B.t.u.—50 B.t.u.

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TABLE 1.—OIL STANDARDS

1 barrel = 42 U. S. gallons = 5.6146 cu. ft.

Degrees Baumé	Specific gravity	Weight per barrel, lb.	Lb. per U. S. gallon	B.t.u. per lb.	B.t.u. per barrel
10	1.000	350.194	8.338	18,380	6,436,566
11	0.993	347.707	8.279	18,440	6,411,717
12	0.986	345.256	8.220	18,550	6,387,236
13	0.979	342.840	8.163	18,560	6,363,110
14	0.972	340.458	8.106	18,620	6,339,328
15	0.966	338.112	8.050	18,680	6,315,932
16	0.959	335.801	7.995	18,740	6,292,911
17	0.952	333.525	7.941	18,800	6,270,270
18	0.946	331.248	7.887	18,860	6,247,337
19	0.940	329.042	7.834	18,920	6,225,474
20	0.933	326.838	7.787	18,980	6,203,347
21	0.927	324.700	7.731	19,040	6,182,288
22	0.921	322.564	7.680	19,100	6,160,972
23	0.915	320.427	7.629	19,160	6,139,381
24	0.909	318.361	7.580	19,220	6,118,898
25	0.903	316.295	7.531	19,280	6,098,168
26	0.897	314.264	7.482	19,340	6,077,866
27	0.892	312.268	7.435	19,400	6,067,999
28	0.886	310.307	7.388	19,460	6,038,574
29	0.881	308.346	7.341	19,520	6,018,914
30	0.875	306.420	7.296	19,580	5,999,704
31	0.870	304.529	7.251	19,640	5,980,950
32	0.864	302.638	7.206	19,700	5,961,969
33	0.859	300.781	7.161	19,760	5,934,433
34	0.854	298,960	7.118	19,820	5,925,387
35	0.848	297.139	7.075	19,880	5,907,123
36	0.843	295.353	7.032	19,940	5,889,339
37	0.838	293.567	6.990	20,000	5,871,340
38	0.833	291.817	6.948	20,060	5,853,849
39	0.828	290.101	6.907	20,120	5,836,832
40	0.824	288.385	6.866	20,180	5,819,609
41	0.819	286.704	6.826	20,204	5,802,889
42	0.814	285.058	6.787	20,300	5,786,677
43	0.809	283.377	6.746	20,360	5,769,556
44	0.805	281.766	6.709	20,420	5,753,662
45	0.800	280.155	6.670	20,480	5,737,574

Sp. gr. = 140 ÷ (130 + deg. Be. at 60°F.)

Weight of oil per barrel = (sp. gr.) × (wt. per cu. ft. of water at 60°F. = 62.372 lb.) × (no. of cu. ft. per barrel = 5.6146).

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TABLE 2.—DEG. BÉ. AND CORRESPONDING SPECIFIC GRAVITIES OF OIL, POUNDS PER GALLON, AND GALLONS PER POUND[1]

Deg. Bé.	Specific gravity at 60°/60°F.	Lb. per gal.	Gal. per lb.	Deg. Bé.	Specific gravity at 60°/60°F.	Lb. per gal.	Gal. per lb.
10.0	1.0000	8.328	0.1201	15.0	0.9655	8.041	0.1244
10.5	0.9964	8.299	0.1205	15.5	0.9622	8.013	0.1248
11.0	0.9929	8.269	0.1209	16.0	0.9589	7.986	0.1252
11.5	0.9894	8.240	0.1214	16.5	0.9556	7.959	0.1256
12.0	0.9859	8.211	0.1218	17.0	0.9524	7.931	0.1261
12.5	0.9825	8.182	0.1222	17.5	0.9492	7.904	0.1265
13.0	0.9790	8.153	0.1227	18.0	0.9459	7.877	1.1270
13.5	0.9756	8.125	0.1231	18.5	0.9428	7.851	0.1274
14.0	0.9722	8.096	0.1235	19.0	0.9396	7.825	0.1278
14.5	0.9688	8.069	0.1239	19.5	0.9365	7.799	0.1282
20.0	0.9333	7.772	0.1287	26.0	0.8974	7.473	0.1338
20.5	0.9302	7.747	0.1291	26.5	0.8946	7.449	0.1342
21.0	0.9272	7.721	0.1295	27.0	0.8917	7.425	0.1347
21.5	0.9241	7.696	0.1299	27.5	0.8889	7.402	0.1351
22.0	0.9211	7.670	0.1304	28.0	0.8861	7.378	0.1355
22.5	0.9180	7.645	0.1308	28.5	0.8833	7.355	0.1360
23.0	0.9150	7.620	0.1313	29.0	0.8805	7.332	0.1364
23.5	0.9121	7.595	0.1317	29.5	0.8777	7.309	0.1368
24.0	0.9091	7.570	0.1321	30.0	0.8750	7.286	0.1373
24.5	0.9061	7.546	0.1325	30.5	0.8723	7.264	0.1377
25.0	0.9032	7.522	0.1330	31.0	0.8696	7.241	0.1381
25.5	0.9003	7.497	0.1334	31.5	0.8669	7.218	0.1385
32.0	0.8642	7.196	0.1390	38.0	0.8333	6.939	0.1441
32.5	0.8615	7.173	0.1394	38.5	0.8309	6.918	0.1446
33.0	0.8589	7.152	0.1398	39.0	0.8284	6.898	0.1450
33.5	0.8563	7.130	0.1403	39.5	0.8260	6.877	0.1454
34.0	0.8537	7.108	0.1407	40.0	0.8235	6.857	0.1459
34.5	0.8511	7.087	0.1411	40.5	0.8211	6.837	0.1463
35.0	0.8485	7.065	0.1415	41.0	0.8187	6.817	0.1467
35.5	0.8459	7.044	0.1420	41.5	0.8163	6.797	0.1471
36.0	0.8434	7.022	0.1424	42.0	0.8140	6.777	0.1476
36.5	0.8408	7.001	0.1428	42.5	0.8116	6.758	0.1480
37.0	0.8383	6.980	0.1433	43.0	0.8092	6.738	0.1484
37.5	0.8358	6.960	0.1437	43.5	0.8069	6.718	0.1489
44.0	0.8046	6.699	0.1493	50.0	0.7778	6.476	0.1544
44.5	0.8023	6.680	0.1497	50.5	0.7756	6.458	0.1548

[1] United States Bureau of Standards, United States standard tables for petroleum oils: Circular 57, Jan. 29, 1916, p. 57.

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TABLE 3.—TEMPERATURE CORRECTIONS TO READINGS OF SPECIFIC GRAVITY HYDROMETERS IN AMERICAN PETROLEUM OILS AT VARIOUS TEMPERATURES[1]

[Standard at 60°/60°F.]

Observed temperature, °F.	Observed specific gravity 0.650	0.700	0.750	0.800	0.850	0.900	0.950
	Subtract from observed specific gravity						
30	0.016	0.015	0.014	0.012	0.011	0.011	0.011
32	0.015	0.014	0.013	0.012	0.011	0.010	0.010
34	0.014	0.013	0.012	0.011	0.010	0.010	0.010
36	0.013	0.012	0.011	0.010	0.009	0.009	0.009
38	0.012	0.011	0.010	0.009	0.008	0.008	0.008
40	0.0105	0.0095	0.0090	0.0080	0.0075	0.0070	0.0070
42	0.0095	0.0085	0.0080	0.0070	0.0065	0.0065	0.0065
44	0.0085	0.0075	0.0070	0.0065	0.0060	0.0060	0.0055
46	0.0075	0.0065	0.0060	0.0055	0.0050	0.0050	0.0050
48	0.0065	0.0060	0.0055	0.0050	0.0045	0.0045	0.0040
50	0.0050	0.0050	0.0045	0.0040	0.0035	0.0035	0.0035
52	0.0040	0.0040	0.0035	0.0030	0.0030	0.0030	0.0030
54	0.0030	0.0030	0.0025	0.0025	0.0020	0.0020	0.0020
56	0.0020	0.0020	0.0020	0.0015	0.0015	0.0015	0.0015
58	0.0010	0.0010	0.0010	0.0005	0.0005	0.0005	0.0005
	Add to observed specific gravity						
60	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
62	0.0010	0.0010	0.0010	0.0005	0.0005	0.0005	
64	0.0020	0.0020	0.0015	0.0015	0.0015	0.0015	
66	0.0030	0.0030	0.0025	0.0025	0.0020	0.0020	
68	0.0040	0.0040	0.0035	0.0030	0.0030	0.0030	
70	0.0050	0.0050	0.0045	0.0040	0.0040	0.0035	
72	0.0060	0.0055	0.0050	0.0045	0.0045	0.0040	
74	0.0070	0.0065	0.0060	0.0055	0.0050	0.0050	
76	0.0080	0.0075	0.0070	0.0065	0.0060	0.0055	
78	0.0090	0.0085	0.0080	0.0070	0.0065	0.0065	
80	0.010	0.009	0.008	0.008	0.007	0.007	
82	0.011	0.010	0.009	0.008	0.008	0.007	
84	0.012	0.011	0.010	0.009	0.009	0.008	
86	0.013	0.012	0.011	0.010	0.009	0.009	
88	0.014	0.013	0.012	0.011	0.010	0.010	
90	0.015	0.014	0.013	0.012	0.011	0.010	
92	0.016	0.015	0.013	0.012	0.011	0.011	
94	0.017	0.016	0.014	0.013	0.012	0.012	
96	0.018	0.016	0.015	0.014	0.013	0.013	
98	0.019	0.017	0.016	0.015	0.014	0.013	
100	0.020	0.018	0.017	0.015	0.014	0.014	
102	0.021	0.019	0.018	0.016	0.015	0.015	
104	0.022	0.020	0.018	0.017	0.016	0.015	
106	0.023	0.021	0.019	0.017	0.016	0.016	
108	0.024	0.022	0.020	0.018	0.017	0.017	
110	0.025	0.023	0.021	0.019	0.018	0.017	
112	0.026	0.024	0.022	0.020	0.019	0.018	
114	0.027	0.025	0.022	0.020	0.019	0.019	
116	0.028	0.026	0.023	0.021	0.020	0.019	
118	0.029	0.026	0.024	0.022	0.021	0.020	
120	0.030	0.027	0.025	0.023	0.022	0.021	

[1] This table is calculated from the same data as Table 1, Circular 57, Bureau of Standards.

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TABLE 4.—TEMPERATURE CORRECTIONS TO READINGS OF BAUMÉ HYDROMETERS IN AMERICAN PETROLEUM OILS AT VARIOUS TEMPERATURES[1]

[Standard at 60°F.; modulus 140.]

Observed temperature, °F.	Observed deg. Bé.							
	20.0	30.0	40.0	50.0	60.0	70.0	80.0	90.0
	Add to observed deg. Bé.							
30	1.7	2.0	2.4	3.0	3.7	4.3	5.0	5.7
32	1.6	1.9	2.3	2.8	3.4	4.0	4.7	5.3
34	1.5	1.8	2.1	2.6	3.1	3.7	4.3	4.9
36	1.4	1.6	2.0	2.4	2.9	3.4	4.0	4.6
38	1.3	1.5	1.8	2.2	2.6	3.1	3.6	4.2
40	1.2	1.4	1.6	2.0	2.4	2.8	3.2	3.8
42	1.1	1.2	1.5	1.8	2.2	2.5	2.9	3.4
44	0.9	1.1	1.3	1.6	2.0	2.2	2.6	3.0
46	0.8	0.9	1.1	1.4	1.7	1.9	2.3	2.7
48	0.7	0.8	0.9	1.2	1.4	1.6	2.0	2.3
50	0.6	0.7	0.8	1.0	1.2	1.4	1.6	1.9
52	0.5	0.6	0.7	0.8	1.0	1.1	1.3	1.5
54	0.3	0.4	0.5	0.6	0.8	0.9	1.0	1.1
56	0.2	0.3	0.3	0.4	0.5	0.6	0.6	0.7
58	0.1	0.1	0.1	0.2	0.3	0.3	0.3	0.4
	Subtract from observed deg. Bé.							
60	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
62	0.1	0.1	0.1	0.2	0.2	0.3	0.3	0.4
64	0.2	0.3	0.3	0.4	0.4	0.6	0.6	0.7
66	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
68	0.5	0.6	0.6	0.7	0.9	1.1	1.3	1.4
70	0.6	0.7	0.8	0.9	1.1	1.4	1.6	1.7
72	0.7	0.8	0.9	1.1	1.3	1.6	1.9	2.1
74	0.8	0.9	1.1	1.3	1.6	1.8	2.2	2.5
76	0.9	1.1	1.3	1.5	1.8	2.1	2.5	2.8
78	1.0	1.2	1.4	1.7	2.0	2.4	2.8	3.1
80	1.1	1.3	1.5	1.8	2.2	2.6	3.1	3.5
82	1.2	1.4	1.7	2.0	2.5	2.9	3.4	3.9
84	1.3	1.5	1.8	2.2	2.7	3.2	3.7	4.3
86	1.4	1.7	2.0	2.4	2.9	3.4	4.0	4.6
88	1.6	1.8	2.1	2.6	3.1	3.7	4.2	4.9
90	1.7	2.0	2.3	2.7	3.3	3.9	4.5	5.2
92	1.8	2.1	2.4	2.9	3.5	4.2	4.8	5.6
94	1.9	2.2	2.6	3.1	3.8	4.4	5.1	5.9
96	2.0	2.3	2.7	3.3	4.0	4.6	5.4	6.3
98	2.1	2.4	2.9	3.4	4.2	4.9	5.7	6.6
100	2.2	2.6	3.0	3.6	4.4	5.1	6.0	6.9
102	2.3	2.7	3.2	3.8	4.6	5.4	6.3	7.2
104	2.4	2.9	3.3	4.0	4.8	5.7	6.6	7.5
106	2.5	3.0	3.5	4.2	5.0	5.9	6.9	7.9
108	2.7	3.1	3.6	4.3	5.2	6.2	7.2	8.2
110	2.8	3.2	3.7	4.4	5.4	6.4	7.5	8.5
112	2.9	3.3	3.9	4.6	5.6	6.7	7.7	8.8
114	3.0	3.4	4.0	4.7	5.8	6.9	7.9	9.1
116	3.1	3.6	4.1	4.9	6.0	7.1	8.2	9.4
118	3.2	3.7	4.3	5.1	6.2	7.3	8.5	9.8
120	3.3	3.8	4.4	5.3	6.4	7.5	8.8	10.1

[1] This table is calculated from the same data as Table 2, Circular 57, Bureau of Standards.

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Table 5.—Temperature Corrections to Apparent Specific Gravities of Petroleum Oils[1]

[This table gives the correction to be added to apparent specific gravities of heavy petroleum oils (fuel oils, lubricating oils, etc.), at temperatures of 60° to 210°F. to give the true specific gravity of the oil at 60°/60°F. It is assumed that the hydrometer or pycnometer used is of glass having a coefficient of cubical expansion of 0.000023 per degree centigrade, and is correct at 60°F.]

Observed temperature, °F.	Observed specific gravity											
	0.850	0.860	0.870	0.880	0.890	0.900	0.910	0.920	0.930	0.940	0.950	0.960
	Add to observed specific gravity to give true specific gravity at 60°/60°F.											
60	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
62	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001
64	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002
66	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002
68	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003
70	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004
72	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004
74	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
76	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006
78	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006
80	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007
82	0.008	0.008	0.008	0.008	0.008	0.007	0.007	0.007	0.007	0.007	0.007	0.007
84	0.009	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008
86	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009
88	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010
90	0.011	0.011	0.011	0.011	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010
92	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011
94	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012
96	0.013	0.013	0.013	0.013	0.013	0.013	0.012	0.012	0.012	0.012	0.012	0.012
98	0.014	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013
100	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014
105	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016
110	0.018	0.018	0.018	0.018	0.017	0.017	0.017	0.017	0.017	0.017	0.017	0.017
115	0.020	0.020	0.020	0.020	0.019	0.019	0.019	0.019	0.019	0.019	0.019	0.019
120	0.022	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021
125	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.022	0.022	
130	0.025	0.025	0.025	0.025	0.025	0.024	0.024	0.024	0.024	0.024	0.024	
135	0.027	0.027	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	
140	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.027	0.027	
145	0.030	0.030	0.030	0.030	0.030	0.030	0.029	0.029	0.029	0.029	0.029	
150	0.032	0.032	0.032	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.031	
155	0.034	0.033	0.033	0.033	0.033	0.033	0.033	0.033	0.033	0.033		
160	0.035	0.035	0.035	0.035	0.035	0.035	0.034	0.034	0.034	0.034		
165	0.037	0.037	0.037	0.037	0.036	0.036	0.036	0.036	0.036	0.036		
170	0.039	0.039	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.037		
175	0.040	0.040	0.040	0.040	0.040	0.040	0.039	0.039	0.039	0.039		
180	0.042	0.042	0.042	0.041	0.041	0.041	0.041	0.041	0.041	0.041		
185	0.044	0.044	0.043	0.043	0.043	0.043	0.043	0.043	0.042			
190	0.045	0.045	0.045	0.045	0.045	0.044	0.044	0.044	0.044			
195	0.047	0.047	0.047	0.047	0.046	0.046	0.046	0.046	0.046			
200	0.049	0.049	0.048	0.048	0.048	0.048	0.048	0.048	0.047			
205	0.051	0.050	0.050	0.050	0.050	0.050	0.049	0.049	0.049			
210	0.052	0.052	0.052	0.051	0.051	0.051	0.051	0.051	0.051			

[1] For more complete oil tables, see Circular 57, Bureau of Standards.

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TABLE 6.—TEMPERATURE CORRECTIONS TO APPARENT DEG. BAUMÉ OF PETROLEUM OILS[1]

This table gives the corrections to be subtracted from the apparent deg. Bé. of heavy petroleum oils (fuel oils, lubricating oils, etc.) at temperatures from 60° to 210°F. to give the true deg. Bé. at 60°F. (modulus, 140). It is assumed that the hydrometer is of glass having a coefficient of cubical expansion of 0.000023 per deg. C., and is correct at 60°F.]

Observed temperature, °F.	Observed degrees Bé.											
	14	16	18	20	22	24	26	28	30	32	34	36
	Subtract from observed deg. Bé. to obtain true deg. Bé. at 60°F.											
60	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
62	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
64	0.2	0.2	0.2	0.2	0.2	0.2	0.3	0.3	0.3	0.3	0.3	0.3
66	0.3	0.3	0.3	0.3	0.3	0.3	0.4	0.4	0.4	0.4	0.4	0.4
68	0.4	0.4	0.4	0.5	0.5	0.5	0.5	0.5	0.6	0.6	0.6	0.6
70	0.5	0.5	0.5	0.6	0.6	0.6	0.6	0.6	0.7	0.7	0.8	0.8
72	0.6	0.6	0.6	0.7	0.7	0.7	0.7	0.7	0.8	0.8	0.9	0.9
74	0.7	0.7	0.7	0.8	0.8	0.8	0.9	0.9	0.9	0.9	1.0	1.1
76	0.8	0.8	0.8	0.9	0.9	0.9	1.0	1.0	1.1	1.1	1.2	1.2
78	0.9	0.9	0.9	1.0	1.1	1.1	1.1	1.2	1.2	1.2	1.3	1.4
80	1.0	1.0	1.1	1.1	1.2	1.2	1.2	1.3	1.3	1.3	1.4	1.5
82	.....	1.1	1.2	1.2	1.3	1.3	1.3	1.4	1.4	1.5	1.5	1.6
84	.....	1.3	1.3	1.3	1.4	1.4	1.5	1.5	1.5	1.6	1.7	1.8
86	.....	1.4	1.4	1.4	1.5	1.5	1.6	1.6	1.7	1.8	1.8	1.9
88	.....	1.5	1.5	1.6	1.6	1.7	1.7	1.8	1.8	1.9	2.0	2.0
90	.....	1.6	1.6	1.7	1.7	1.8	1.8	1.9	2.0	2.0	2.1	2.1
92	.....	1.7	1.7	1.8	1.8	1.9	1.9	2.0	2.1	2.1	2.2	2.3
94	.....	1.8	1.8	1.9	1.9	2.0	2.0	2.1	2.2	2.2	2.3	2.4
96	.....	1.9	1.9	2.0	2.0	2.1	2.2	2.3	2.3	2.4	2.5	2.5
98	.....	2.0	2.0	2.1	2.2	2.2	2.3	2.4	2.4	2.5	2.6	2.7
100	.....	2.1	2.2	2.2	2.3	2.3	2.4	2.5	2.6	2.7	2.7	2.8
105	.....	2.4	2.4	2.5	2.6	2.6	2.7	2.8	2.9	3.0	3.1	3.2
110	.....	2.6	2.7	2.8	2.8	2.9	3.0	3.1	3.2	3.3	3.4	3.5
115	.....	2.9	2.9	3.0	3.1	3.2	3.3	3.4	3.5	3.6	3.8	3.9
120	.....	3.1	3.2	3.3	3.4	3.5	3.6	3.7	3.8	3.9	4.0	4.2
125	.....	.....	3.4	3.5	3.6	3.7	3.8	4.0	4.1	4.2	4.3	4.5
130	.....	.....	3.7	3.8	3.9	4.0	4.1	4.3	4.4	4.5	4.7	4.8
135	.....	.....	3.9	4.1	4.2	4.3	4.4	4.6	4.7	4.8	5.0	5.2
140	.....	.....	4.2	4.3	4.4	4.6	4.7	4.8	5.0	5.1	5.3	5.5
145	.....	.....	4.4	4.6	4.7	4.8	5.0	5.1	5.3	5.4	5.6	5.8
150	.....	.....	4.7	4.8	5.0	5.1	5.2	5.4	5.6	5.7	5.9	6.1
155	.....	.....	4.9	5.1	5.2	5.4	5.5	5.7	5.9	6.0	6.2	6.4
160	.....	.....	.....	5.3	5.5	5.6	5.8	6.0	6.2	6.3	6.5	6.7
165	.....	.....	.....	5.6	5.7	5.9	6.1	6.3	6.5	6.6	6.8	7.0
170	.....	.....	.....	5.8	6.0	6.2	6.3	6.5	6.7	6.9	7.1	7.3
175	.....	.....	.....	6.0	6.2	6.4	6.6	6.8	7.0	7.2	7.4	7.6
180	.....	.....	.....	6.3	6.5	6.6	6.8	7.1	7.3	7.5	7.7	8.0
185	.....	.....	.....	6.5	6.7	6.9	7.1	7.3	7.6	7.8	8.0	8.3
190	.....	.....	.....	6.8	7.0	7.2	7.4	7.6	7.8	8.1	8.3	8.6
195	.....	.....	.....	7.0	7.2	7.4	7.6	7.9	8.1	8.4	8.6	8.9
200	.....	.....	.....	.....	7.5	7.7	7.9	8.1	8.4	8.6	8.9	9.2
205	.....	.....	.....	.....	7.7	7.9	8.2	8.4	8.7	8.9	9.2	9.5
210	.....	.....	.....	.....	8.0	8.2	8.4	8.7	8.9	9.2	9.5	9.8

[1] For more complete oil tables see Circular 57, Bureau of Standards.

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AVERAGE SPECIFIC HEAT

All fuel oils are mixtures of many hydrocarbon compounds, each with its own specific heat. When the heat capacity of an oil is desired it is the practice to accept the average specific heat of a similar oil as determined by experiment. Table 7 gives average values for different petroleums.

TABLE 7.—SPECIFIC HEAT CAPACITIES[1]

	Specific gravity	Specific heat capacity
Petroleum ether at $-180°C$		0.452
Petroleum ether at $-100°C$		0.445
Petroleum ether at 0°C		0.419
Kerosene, 21–58°C		0.511
Kerosene, 18–99°C		0.498
Paraffin solid, $-20°$ to 3°		0.377
Paraffin solid, $-19°$ to 20°		0.525
Paraffin solid, 25° to 30°		0.589
Paraffin solid, 35° to 40°		0.622
Paraffin liquid, 52.4° to 55°		0.700
Crude oils:		
Japan	0.862	0.453
Pennsylvania	0.810	0.500
Russia	0.908	0.435
California	0.960	0.398
Bustenari	0.842	0.462
Campina, 0.8 per cent. paraffin	0.859	0.467
Campina, 3.2 per cent. paraffin	0.854	0.457

Mabery and Goldstein[2] give the following formula for calculating specific heats:

$$\frac{\text{Specific heat} \times \text{molecular weight}}{\text{Number of atoms in molecule}} = K$$

For the hydrocarbons of the paraffin series, C_nH_{2n+2}, the value of K is 2.28.

[1] Holde, David, The examination of hydrocarbon oils, 1915, pp. 13 and 17.

[2] Mabery, C. F., and Goldstein, A. H., On the specific heats and heat of vaporization of the paraffin and methylene hydrocarbons: Am. Chem. Jour., 1902, vol. 28, pp. 66–78.

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TABLE 8.—COMPOSITION AND CALORIFIC VALUE OF VARIOUS OILS[1]

[C = carbon; H = hydrogen; S = sulphur; O = Oxygen; M = moisture.]

Kind of oil	C	H	S	O	Specific gravity	Flash point, °F.	M	B.t.u. per lb.	Authority
	Per cent.	Per cent.	Per cent.	Per cent.			Per cent.		
California, Kern	85.6	11.89	1.09	1.42	0.9589			18,840	
California, Coalinga					0.927	134		17,117	Babcock & Wilcox.
California, Bakersfield					0.975			17,600	Wade.
California, Bakersfield			1.30		0.992			18,257	Wade.
California, Kern					0.950	140		18,845	Babcock & Wilcox.
California, Los Angeles			2.56					18,328	Babcock & Wilcox.
California, Los Angeles					0.977		[3].40	18,280	Babcock & Wilcox.
California, Los Angeles					0.957	196		18,855	Babcock & Wilcox.
California, Monte Christo					0.966	205		18,875	Babcock & Wilcox.
California, Whittier			0.98		0.944		1.06	18,507	Wade.
California, Whittier			0.72		0.936		1.06	18,240	Wade.
	83.04	11.52	2.45	[2]0.99			1.40	17,871	Babcock & Wilcox.
California	81.52	11.51	0.55	[2]6.92		230		18,667	U. S. Naval Fuel Board.
California			0.87				0.95	18,533	Blasdale.
California					0.891	257		18,655	Babcock & Wilcox.
California			2.45		0.973		[3]1.50	17,976	O'Neill.
California			2.46		0.975		1.32	18,104	Shepherd.
Texas, Beaumont	84.6	10.9	1.63	2.87	0.924	180		19,060	U. S. Naval Fuel Board.
Texas, Beaumont	83.3	12.4	0.50	3.83	0.926	216		19,481	U. S. Naval Fuel Board.
Texas, Beaumont	85.0	12.3	1.75	[2]0.92				19,060	Denton.
Texas, Beaumont	86.1	12.3	1.60		0.942			20,152	Sparkes.
Texas, Beaumont					0.903	222		19,349	Babcock & Wilcox.
Texas, Sabine					0.937	143		18,662	Babcock & Wilcox.
Texas	87.15	12.33	0.32		0.908	370		19,338	U. S. Navy.
Texas	87.29	12.32	0.43		0.910	375		19,659	U. S. Navy.
Ohio	83.4	14.7	0.60	1.3				19,580	
Pennsylvania	84.9	13.7		1.4	0.886			19,210	Booth.
West Virginia	84.3	14.1		1.6	0.841			21,240	
Mexico					0.921	162		18,840	Babcock & Wilcox.
Russia, Baku	86.7	12.9			0.884			20,691	Booth.
Russia, Novorossick	84.9	11.6		3.46				19,452	Booth.
Russia, Caucasus	86.6	12.3		1.10	0.938			20,138	
Java	87.1	12.0		0.9	0.923			21,163	
Austria, Galacia	82.2	12.1	5.7		0.870			18,416	
Italy, Parma	84.0	13.4	1.8		0.786				
Borneo	85.7	11.0		3.31				19,240	Orde.

[1] Table from Babcock & Wilcox Co.'s Handbook on steam. [2] Includes nitrogen. [3] Includes silt.

APPENDIX V

BRIEF BIBLIOGRAPHY ON FUEL OIL

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