STEAM POWER PLANT ENGINEERING

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PREFACE TO FIFTH EDITION

Although the first edition of this work was published less than a decade ago, the development of the Steam Power Plant has been so rapid that nearly all of the descriptive matter and a considerable portion of the data of this early edition became obsolete shortly after publication. Revisions in 1909, 1911, and in 1913 failed to keep pace with the art, and the task of recording correct practice appeared to be a hopeless one. Fortunately radical changes during the past two years have been less marked and many of the elements entering into the modern Steam Power Plant have virtually reached the limit of efficiency, and some degree of stability may be expected from now on. It is quite unlikely that the Steam Power Plant of the immediate future will differ radically from the latest type already in operation, though increased boiler pressure, forced boiler capacity and the use of powdered low-grade fuel may effect minor changes.

The same treatment of the subject has been followed in this edition as in earlier issues, but the book is to all intents and purposes a new one. Particular stress has been laid upon the subject of Fuels and Combustion and supplementary chapters on Elementary Thermodynamics, Properties of Steam, and Properties of Dry and Saturated Air have been added at the request of practicing engineers. Numerous examples have been incorporated in the text, and the addition of typical exercises and problems may prove of value to the instructor. The scope of the work has been greatly enlarged, and with the exception of a few minor sections the entire text has been rewritten and reset. An extended study of certain portions of the work is facilitated by numerous references to current engineering literature.

G. F. G.

Chicago, Illinois,
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STEAM POWER PLANT ENGINEERING

CHAPTER I

ELEMENTARY STEAM POWER PLANTS

1. General. — By far the greater part of the mechanical and electrical energy generated for commercial purposes is furnished by the steam power plant. Despite the tremendous progress of the internal combustion engine and the rapid development of water power the steam plant is more than holding its own.

The most efficient plant, thermally, in the conversion of energy from one form to another, is not necessarily the most economical commercially, since the various items involved in effecting this conversion may more than offset the gain over a less efficient plant. There is no question as to the low operating cost of power generated by hydro-electric plants, but when the cost of transmission and the overhead charges are taken into consideration the economy is not so evident and may be completely neutralized. From a purely thermal standpoint the Diesel engine electric plant is superior to the best steam-electric plant for power purposes, but the fuel item is only one of the many involved in the total cost. It is the *commercial efficiency* which enables the steam power plant, with its extravagant waste of fuel, to compete successfully with the gas producer, internal-combustion engine and hydro-electric plant.

A station which distributes power to a number of consumers more or less distant, is called a *Central Station.* When the distances are very great, electrical current of high tension is frequently employed, and is transformed and distributed at convenient points through *Sub-stations:* A plant designed to furnish power or heat to a building or a group of buildings under one management is called an *Isolated Station.* For example, the power plant of an office building is usually called an isolated station.

When the exhaust steam from the engines is discharged at approximately atmospheric pressure the plant is said to be operating *non-condensing.* When the exhaust steam is condensed, reducing the back pressure on the piston by the partial vacuum thus formed, the plant is said to operate *condensing.*

When the exhaust steam may be used for manufacturing, heating, or other useful purposes, as is frequently the case in various manufacturing establishments, and in large office buildings, it is usually more economical to run non-condensing, while power plants for electric lighting and power, pumping stations, air-compressor plants, and others, in which the load is fairly constant and the exhaust steam is not required for heating, are generally operated condensing.

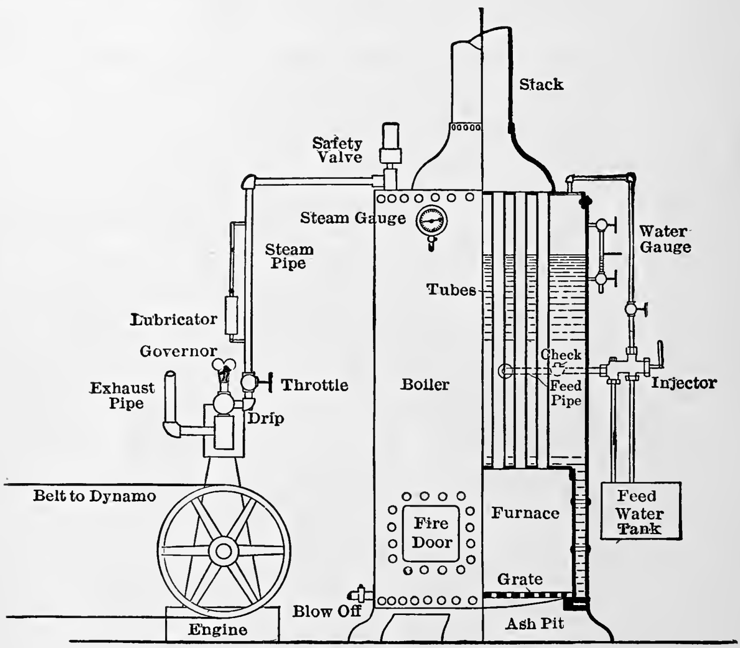


Fig. 1. Elementary Non-condensing Plant.

2. **Elementary Non-condensing Plant.** — Fig. 1 gives a diagrammatic outline of the essential elements of the simplest form of steam power plant. The equipment is complete in every respect and embodies all the accessories necessary for successful operation. Where a small amount of power is desired at intermittent periods, as in hoisting systems, threshing outfits and traction machinery, the arrangement is substantially as illustrated. The output in these cases seldom exceeds 50 horsepower and the time of operation is usually short, so the cheapest of appliances are installed, simplicity and low first cost being more important than economy of fuel.

Such a plant has three essential elements: (1) The furnace, (2) the boiler, and (3) the engine. Fuel is fed into the *furnace*, where it is burned. A portion of the heat liberated from the fuel by combustion is absorbed by the water in the *boiler*, converting it into steam under pressure. The steam being admitted to the cylinder of the *engine* does work upon the piston and is then exhausted through a suitable pipe to the atmosphere. The process is a continuous one, fuel and water being fed into the furnace and the boiler in proportion to the power demanded.

In such an elementary plant, certain *accessories* are necessary for successful operation. The *grate* for supporting the fuel during combustion consists of a cast-iron grid or of a number of cast-iron bars spaced in such a manner as to permit the passage of air through the fuel from below. The solid waste products fall through or are "sliced" through the grate bars into the *ash pit*, from which they may be removed through the *ash door*. The latter acts also as a means of regulating the supply of air below the grate. Fuel is fed into the furnace through the *fire door*, and when occasion demands, air may be supplied above the bed of fuel by means of this door. The *combustion chamber* is the space between the bed of fuel and the boiler heating surface, its office being to afford a space for the oxidation of the combustible gases from the solid fuel before they are cooled below ignition temperature by the comparatively cool surfaces of the boiler. The *chimney* or *stack* discharges the products of combustion into the atmosphere and serves to create the draft necessary to draw the air through the bed of fuel. Various forced-draft appliances are sometimes used to assist or to entirely replace the chimney. The *heating surface* is that portion of the boiler area which comes into contact with the hot furnace gases, absorbs the heat and transmits it to the water. In the small plant illustrated in Fig. 1, the major portion of the heating surface is composed of a number of fire tubes below the water line, through which the heated gases pass. The *superheating surface* is that portion of the heating surface which is in contact with the heated gases of combustion on one side and steam on the other. The volume above the water level is called the *steam space.* Water is forced into the boilers either by a *feed pump* or an *injector*. In small plants of the type considered, steam pumps are seldom employed; the injector answers the purpose and is considerably cheaper. A *safety valve* connected to the steam space of the boiler automatically permits steam to escape to the atmosphere if an excessive pressure is reached. The water level is indicated by *try cocks* or by a *gauge glass*, the top of which is connected with the steam space and the bottom with the water space. *Try cocks* are small valves placed in the water column or boiler shell, one at normal water level, one above it, and one below. By opening the valves from time to time the water level is approximately ascertained. They are ordinarily used in case of accident to the gauge glass. *Fusible plugs* are frequently inserted in the boiler shell at the lowest permissible water level. They are composed of an alloy having a low fusing point which melts when in contact with steam, thus giving warning by the blast of the escaping steam if the water level gets dangerously low. The *blow-off cock* is a valve fitted to the lowest part of the boiler to drain it of water or to discharge the sediment which deposits in the bottom. The steam outlet of a boiler is usually called the *steam nozzle.*

The essential accessories of the simple steam engine include: A *throttle valve* for controlling the supply of steam to the engine; the *governor,* which regulates the speed of the engine by governing the steam supply; the *lubricator,* attached to the steam pipe, which is usually of the "sight-feed" class and provides for lubrication of piston and valve. Lubrication of the various bearings is effected by *oil cups* suitably located. *Drips* are placed wherever a water pocket is apt to form in order that the condensation may be drained. The apparatus to be driven by the engine may be *direct connected* to the crank shaft or *belted* to the *flywheel* or *geared.*

In small plants of this type no attempt is made to utilize the exhaust steam except in instances where the stack is too short to create the necessary draft, in which case the exhaust may be discharged up the stack. If the draft is produced by convection of the heated gases in the chimney, the fuel is said to be burned under *natural draft;* if the natural draft is assisted by the exhaust steam, the fuel is said to be burned under *forced draft.* The power realized from a given weight of fuel is very low and seldom exceeds $2\frac{1}{2}$ per cent of the heat value of the fuel. The distribution of the various losses in a plant of, say, 40 horse power is approximately as follows:

	B.t.u.
Heat value of 1 pound of coal	14,500
Boiler and furnace losses, 50 per cent	7,250
Heat equivalent of one horsepower-hour	2,546
Heat used to develop one horsepower-hour (50 pounds steam per horsepower-hour, pressure 80 pounds gauge, feed water 67 deg. fahr.)	57,500
	Per cent.
Percentage of heat of the steam realized as work, $\frac{2,546}{57,500}$	4.4
Percentage of heat value of the coal realized as work, $\frac{2,546}{57,500 \div 0.50}$	2.2

In Europe small non-condensing plants are developed to a high degree of efficiency. Through the use of highly superheated steam, specially designed engines and boilers, plants of this type as small as 40 horsepower are operated with over-all efficiencies of from 10 to 12 per cent.

The power plant of the modern locomotive is very much like that illustrated in Fig. 1, the main difference lying in the type of boiler and engine. The entire exhaust from the engine is discharged up the stack through a suitable nozzle, since the extreme rate of combustion requires an intense draft. The engine is a highly efficient one compared with that in the illustration, and the performance of the boiler is more economical. In average locomotive practice about 5 per cent of the heat value of the fuel is converted into mechanical energy at the draw bar.* In general, a non-condensing steam plant in which the heat of the exhaust is wasted is very uneconomical of fuel, even under the most favorable conditions, and seldom transforms as much as 7 per cent of the heat value of the fuel into mechanical energy.

3. Non-condensing Plant. Exhaust Steam Heating. — Fig. 2 gives a diagrammatic arrangement of a simple non-condensing plant differing from Fig. 1 in that the exhaust steam is used for heating purposes. This shows the essential elements and accessories, but omits a number of small valves, by-passes, drains, and the like for the sake of simplicity. The plant is assumed to be of sufficient size to warrant the installation of efficient appliances. Steam is led from the boiler to the engine by the *steam main.* The moisture is removed from the steam before it enters the cylinder by a *steam separator.* The moisture drained from the separator is either discharged to waste or returned to the boiler. The exhaust steam from the engine is discharged into the *exhaust main* where it mingles with the steam exhausted from the steam pumps. Since the exhaust from engines and pumps contains a large portion of the cylinder oil introduced into the live steam for lubricating purposes, it passes through an *oil separator* before entering the heating system. After leaving the oil separator the exhaust steam is diverted into two paths, part of it entering the *feed-water heater* where it condenses and gives up heat to the feed water, and the remainder flowing to the heating system. During warm weather the engine generally exhausts more steam than is necessary for heating purposes, in which case the surplus steam is automatically discharged to the *exhaust head* through the *back-pressure valve.* The back-pressure valve is, virtually, a large weighted check valve which remains closed when the pressure in the heating system is below a certain prescribed amount, but which opens automatically when the pressure is greater than this amount. During cold weather it often happens that the engine ex-

* Best modern practice gives about 8 per cent as a maximum.

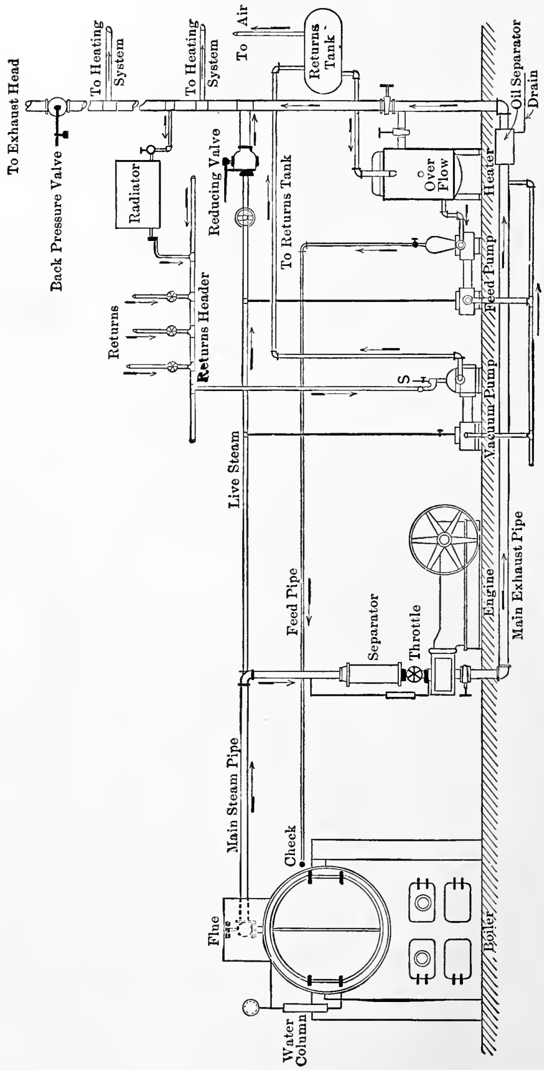


FIG. 2. Elementary Non-condensing Plant with Heating System (Reciprocating Engine).

haust is insufficient to supply the heating system, the radiators condensing the steam more rapidly than it can be supplied. In this case live steam from the boiler is automatically fed into the main heating supply pipe through the *reducing valve.*

The condensed steam and the entrained air which is always present are automatically discharged from the radiators by a *thermostatic valve* into the *returns header.* The thermostatic valve is so constructed that when in contact with the comparatively cool water of condensation it remains open and when in contact with steam it closes. The *vacuum pump* or vapor pump exhausts the condensed steam and air from the returns header and discharges them to the *returns tank.* The small pipe S admits cold water to the vacuum pump and serves to condense the heated vapor, and at the same time supply the necessary *make-up water* to the system. The returns tank is open to the atmosphere so that the air discharged from the vacuum pump may escape. From the returns tank the condensed steam gravitates to the *feed-water heater* where its temperature is raised to practically that of the exhaust steam. The feed water gravitates to the *feed pump* and is forced into the boiler. There are several systems of exhaust steam heating in current practice which differ considerably in details, but, in a broad sense, are similar to the one just described. The more important of these will be described later on.

During the summer months when the heating system is shut down, the plant operates as a simple non-condensing station and practically all of the exhaust steam, amounting to perhaps 80 per cent of the heat value of the fuel, is wasted. The total coal consumption, therefore, is charged against the power developed. During the winter months, however, all, or nearly all, of the exhaust steam may be used for heating purposes and the power becomes a relatively small percentage of the total fuel energy utilized. The percentage of heat value of the fuel chargeable to power depends upon the size of the plant, the number and character of engines and boilers, and the conditions of operation. It ranges anywhere from 50 to 100 per cent for the summer months and may run as low as 6 per cent for the winter months. This is on the assumption, of course, that the engine is debited only with the difference between the coal necessary to produce the heat entering the cylinder and that utilized in the heating system.

4. Elementary Condensing Plant. — Under the most favorable conditions a non-condensing plant cannot be expected to realize more than 10 per cent of the heat value of the fuel as power. In large non-condensing power stations the demand for exhaust steam is usually limited to the heating of the feed water, and as only 12 or 15 per cent can be utilized in this manner, the greater portion of the heat in the exhaust is lost. Non-condensing engines using saturated steam require from 20 to 60 pounds of steam per hour for each horsepower developed. On the other hand in condensing engines the steam consumption may be reduced to as low as 10 pounds per horsepower-hour. The saving of fuel is at once apparent.

Fig. 3 gives a diagrammatic arrangement of a simple condensing plant in which the back pressure on the engine is reduced by condensing the exhaust steam. A different type of boiler from that in Fig. 1 or Fig. 2 has been selected, for the purpose of bringing out a few of the characteristic elements. The products of combustion instead of passing directly through *fire tubes* to the stack as in Fig. 1 are deflected back and forth across a number of *water tubes,* by the *bridge wall* and a series of *baffles.* After imparting the greater part of their heat to the heating surface the products of combustion escape to the chimney through the *breeching* or *flue.* The rate of flow is regulated by a *damper* placed in the breeching as indicated.

The steam generated in the boiler is led to the engine through the *main header.* The steam is exhausted into a *condenser* in which its latent heat is absorbed by *injection* or cooling water. The process condenses the steam and creates a partial vacuum. The condensed steam, injection water, and the air which is invariably present are withdrawn by an *air pump* and discharged to the *hot well.* In case the vacuum should fail, as by stoppage of the air pump, the exhaust steam is automatically discharged to the *exhaust head* by the *atmospheric relief valve,* and the engine will operate non-condensing. The *atmospheric relief valve* is a large check valve which is held closed by atmospheric pressure as long as there is a vacuum in the condenser. When the vacuum fails the pressure of the exhaust becomes greater than that of the atmosphere and the valve opens.

The feed water may be taken from the hot well or from any other source of supply and forced into the *heater.* In this particular case it is taken from a cold supply and upon entering the heater is heated by the exhaust steam from the *air* and *feed pumps.* From the heater it gravitates to the feed pump and is forced into the boiler. Various other combinations of heaters, pumps, and condensers are necessary in many cases, depending upon the conditions of operation. Feed pumps, air pumps, and in fact all small engines used in connection with a steam power plant are usually called *auxiliaries.*

A well-designed station similar to the one illustrated in Fig. 3 is capable of converting about 12 per cent of the heat value of the fuel

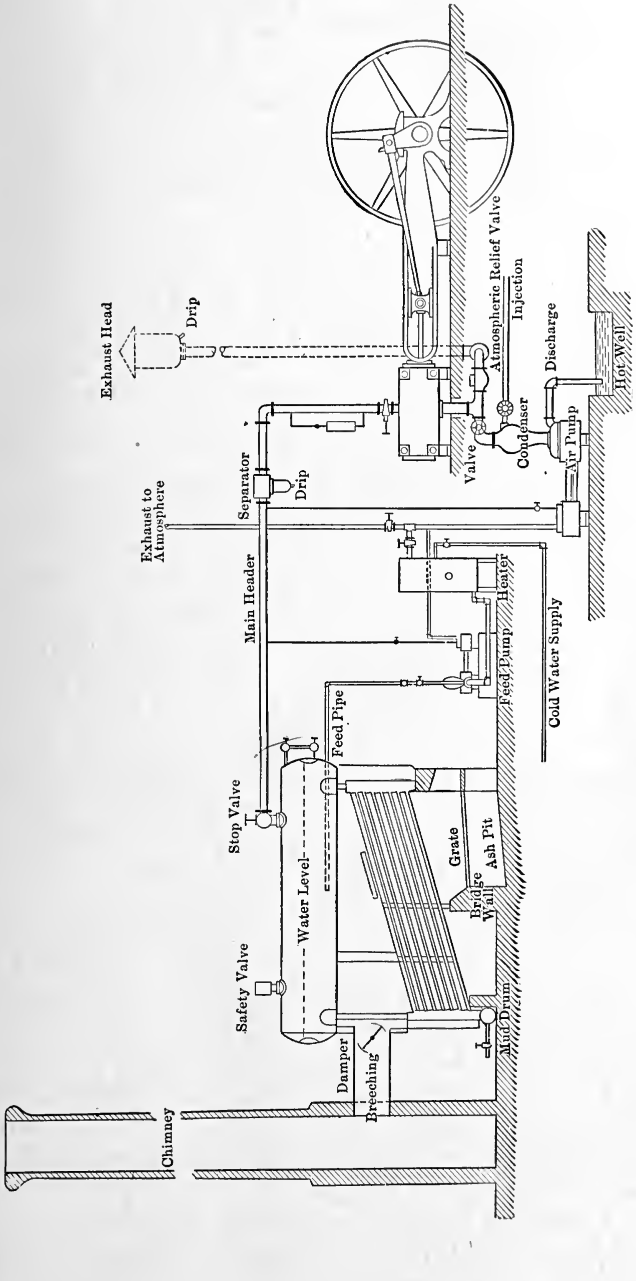


Fig. 3. Elementary Condensing Plant (Reciprocating Engine).

into mechanical energy. The various heat losses under average conditions are approximately as follows:

BOILER LOSSES.

	Per cent.
Loss due to fuel falling through the grate	4
Loss due to incomplete combustion	2
Loss to heat carried away in chimney gases	20
Radiation and other losses	9
Total	35
	B.t.u.
Heat used by engines and auxiliaries (16 pounds of steam per i.h.p-hour, pressure 150 pounds, feed water 210 deg. fahr.	16,250
Engine and generator friction, 5 per cent	812
Leakage, radiation, etc., 2 per cent	325
Total	17,387
Heat equivalent of one electrical horsepower	2,546
	Per cent.
Percentage of the heat value of the steam converted into electrical energy	14.7
Percentage of heat value of fuel converted into electrical energy $\frac{2546 \times 0.65}{17,387}$	9.5

One of the best recorded American performances of a reciprocating engine steam-electric power plant is that of the Pacific Light and Power Company at Redondo, Cal. When operating under regular commercial conditions approximately 14 per cent of the available heat of the fuel (crude oil) is realized as power at the switchboard. This includes all standby losses. For a detailed description of the plant and the results of the acceptance tests, see Jour. of Elec. Gas and Power, Aug. 22, 1908.

Fig. 4 gives a diagrammatic arrangement of one section of a modern large turbo-alternator central station. Each section is to all intents and purposes an independent plant. It will be noted that the essential elements are practically the same as in the reciprocating station engine plant, Fig. 3, differing only in size and design.

The power house, coal storage pile, storage and switch tracks, overhead bunkers, and coal and ash conveyors have been omitted for the sake of simplicity, though the fuel supply and distributing system is an important factor in the design and operation of the plant. In the very latest designs the entire coal and ash handling equipment is electrically operated from a centralized board control. Assuming the coal bunkers over the boilers to be supplied with fuel, the operation is as follows: Coal descends by gravity to the stokers which, in this particular case, are of the under-feed, sloping fire-bed type. Ash and clinkers are removed by clinker grinders located in a pit and are discharged into the *ash hopper*. Motor-driven blowers supply the air required for com-

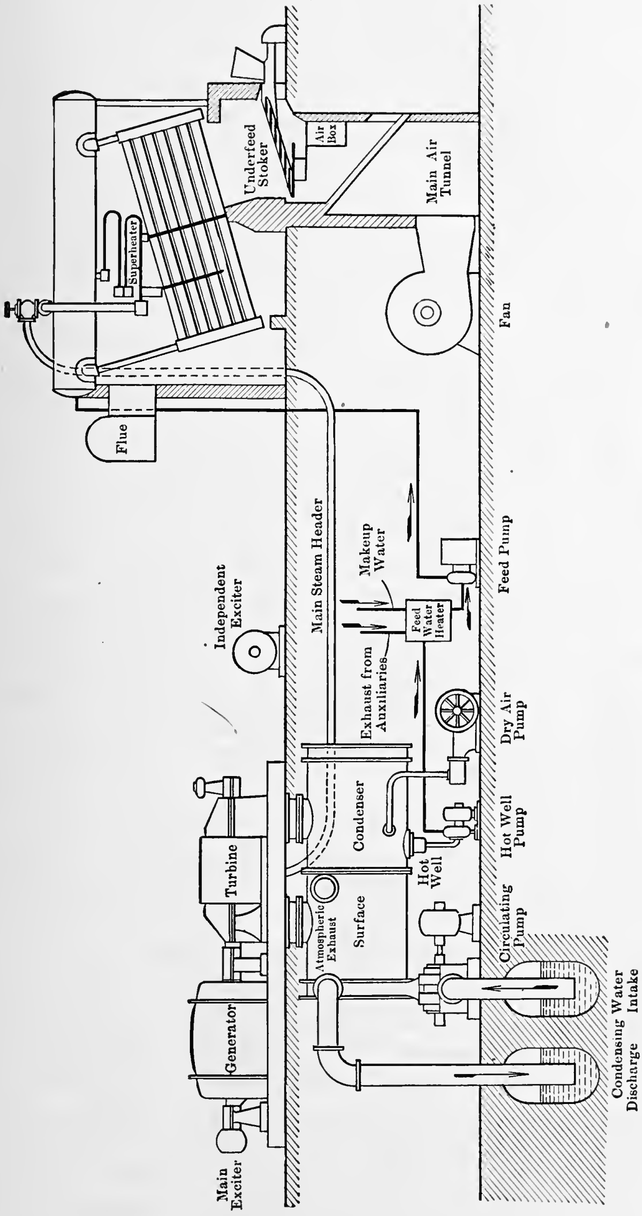


FIG. 4. Elementary Turbo-alternator Station.

bustion. This air, in some installations, is preheated by the recovery of radiation and electrical losses in the turbine room and by radiation from the steam pipes.

The boilers are much larger individually and fewer in number than in the old style reciprocating-engine plant and generate steam at 250 to 300 pounds pressure, superheated to approximately 650 deg. fahr. When operating the turbines at full load the boilers are driven at 175 to 200 per cent or more of their commercial rating. Reserve or spare boilers are conspicuous by their absence. When a boiler is cut out for repairs the rest of the battery is operated at from 225 to 275 per cent rating or more in order to evaporate the required amount of water. Each battery is designed to furnish steam directly to one particular turbine but by means of a *cross-over main* the steam from any battery of boilers may flow to any turbine.

The *prime movers* are horizontal steam turbines direct connected to alternators. The bearings are water cooled and lubrication is automatically effected by means of a pump connected to the governor shaft. Each turbine is normally excited by the *main exciter* mounted on an extension of the generator shaft. The generator field may also be excited from an independently driven exciter or from the station storage battery. Air, washed and conditioned if necessary, is drawn into the generator by the revolving member and absorbs the electrical heat losses. The efficiency of the generator is very high (96 per cent) and yet because of the great amount of energy transformed in the generator this 4 per cent loss represents a large amount of heat and forced ventilation is necessary to prevent overheating. This preheated air may be discharged to waste or carried through conduits to the forced draft fan.

The condenser is ordinarily of the surface condensing type and is attached directly below the low pressure end of the turbine. A much higher vacuum is maintained in the condensers than in reciprocating engine practice since the turbine gives its best efficiency at low back pressures. Condensing water is circulated through the tubes of the condenser by motor-driven or steam-driven centrifugal pumps and the condensed steam or *condensate* collected in the hot wells is withdrawn by a turbine-driven or motor-driven *hot-well pump.* Air and non-condensable vapors are removed by a reciprocating dry air pump, steam or electrically driven. Rotary air pumps and so called turbo-air pumps are also used for this purpose. The hot-well pump discharges the condensate into a feed-water heater which receives the steam exhausted from the steam driven auxiliaries. The steam turbine-driven centrifugal boiler feed pump takes its supply from the feed-water heater and delivers it to the boiler.

A modern station similar to the one illustrated in Fig. 4, equipped with 30,000 kilowatt units, is capable of converting over 18 per cent of the heat value of the fuel into electrical energy when operating at its most economical load. Under commercial conditions of operation with its attendant standby losses the average overall efficiency ranges from 12 to 16 per cent.

5. Condensing Plant with Full Complement of Heat-saving Appliances. — When fuel is costly it frequently becomes necessary for the sake of economy to reduce the heat wastes as much as possible. The chimney gases, which in average practice are discharged at a temperature between 450 and 550 deg. fahr., represent a loss of 20 to 30 per cent of the total value of the fuel. If part of the heat could be reclaimed without impairing the draft the gain would be directly proportional to the reduction in temperature of the gases. Again, in some types of condensers all of the steam exhausted by the engine is condensed by the circulating water and discharged to waste. If provision could be made for utilizing part of the exhaust steam for feed-water heating, the efficiency of the plant could be correspondingly increased. In many cases the cost of installing such heat-saving devices would more than offset the gain effected, but occasions arise where they give marked economy.

Fig. 5 gives a diagrammatic arrangement of a condensing plant in which a number of heat-reclaiming devices are installed. The plant is assumed to consist of a number of engines, boilers, and auxiliaries. Coal is automatically transferred from the cars *to coal hoppers* placed above the boiler, by a system of buckets and conveyors. These hoppers store the coal in sufficient quantities to keep the boiler in continuous operation for some time. From the hoppers the coal is fed intermittently to the *stoker* by means of a *down spout.* The stoker feeds the furnace in proportion to the power demanded and automatically rejects the ash and refuse to the *ash pit.* The ashes are removed from the ash pit, when occasion demands, and are transferred to the *ash hopper* by the same system of buckets and conveyor which handles the coal. The ash hopper is usually placed alongside the coal hoppers and is not unlike them in general appearance and construction.

The products of combustion are discharged to the stack through the flue or breeching. Within the flue is placed a feed-water heater called an *economizer,* the function of which is to absorb part of the heat from the gases on their way to the chimney. The heat reclaimed by the economizer varies widely with the conditions of operation and ranges between 5 and 20 per cent. Since the economizer acts as a resistance to the passage of the products of combustion it is sometimes necessary

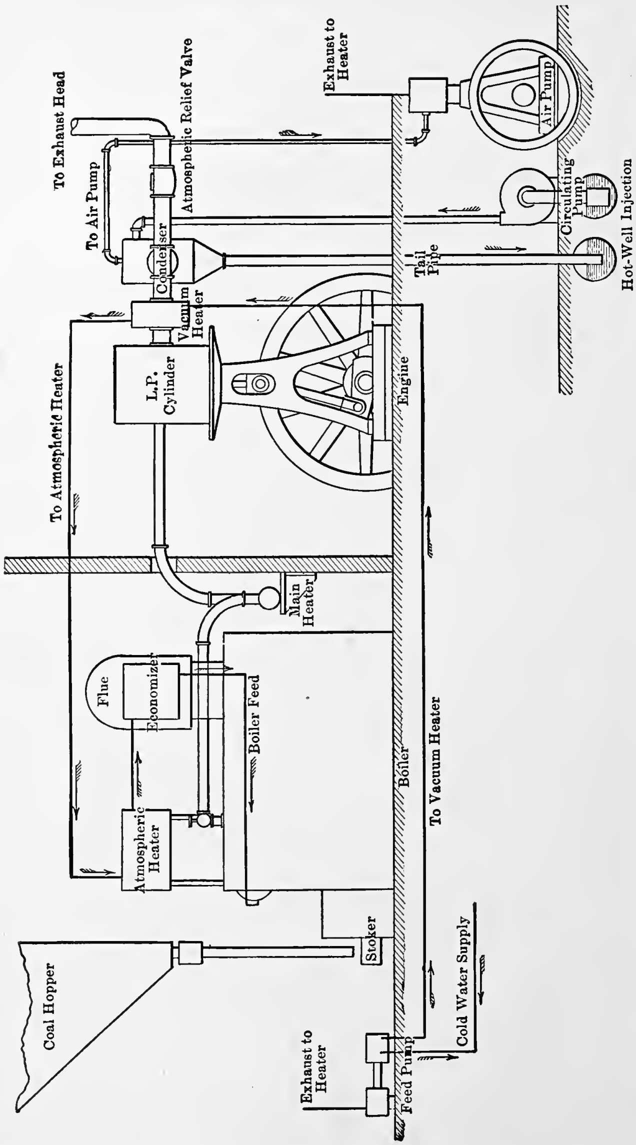


Fig. 5. Elementary Condensing Plant with Full Complement of Heat-saving Appliances (Reciprocating Engine).

to increase the draft either by increasing the height of the chimney or, as is the usual practice, by using a forced-draft system.

Part of the heat of the exhaust steam is reclaimed by a *vacuum heater* which is placed in the exhaust line between engine and condenser. For example, if the feed water has a normal temperature of 60 deg. fahr. and the vacuum in the condenser is 26 inches, the vacuum heater will raise the temperature of the feed to, say, 120 deg. fahr., thereby effecting a gain in heat of approximately 6 per cent. If the feed supply is taken from the *hot well* the vacuum heater is without purpose, as the temperature of the hot well will not be far from 120 deg. fahr.

Referring to the diagram, the path of the steam is as follows: From the boiler it flows through the *boiler lead* to the *main header* or equalizing pipe. From the main header it flows through the *engine lead* to the high-pressure cylinder. The exhaust steam discharges from the low-pressure cylinder through the vacuum heater and into the condenser. Part of the exhaust steam is condensed in the vacuum heater and gives up its latent heat to the feed water. The remainder is condensed by the injection water which is forced into the condenser chamber by the *circulating pump.* The condensed steam and circulating water gravitate through the *tail pipe* to the hot well. The air which enters the condenser either as leakage or entrainment is withdrawn by the air pump. The steam exhausted by the feed pump, air pump, stoker engine, and other steam-driven auxiliaries is usually discharged into the *atmospheric heater,* which still further heats the feed water.

Referring to the feed water, the circuit is as follows: The pump draws in cold water at a temperature of, say, 60 deg. fahr., and forces it in turn through the vacuum heater, the atmospheric heater, and the economizer into the boiler. The vacuum heater raises the temperature to 120 deg. fahr., the atmospheric heater increases it to 212 deg. fahr., and the economizer still further to about 300 deg. fahr. The heat reclaimed by this series of heaters is evidently the equivalent of that necessary to raise the feed water from 60 deg. fahr. to 300 deg. fahr., or approximately 24 per cent of the total steam supplied. In some plants the economizer only is installed; in others the economizer and atmospheric heater are deemed desirable; still others utilize all three. The distribution of the heat losses in a plant of this type using saturated steam and operating under favorable conditions is approximately as follows:

	Per Cent.	B.t.u.
Delivered to engine, 15 pounds steam per i.hp-hour; pressure 150 pounds, feed 60 deg. fahr.	100	17,482
Delivered to feed pump	1.5	262
Delivered to circulating pump	1.5	262
Delivered to air pump	2	349
Delivered to small auxiliaries	1.5	262
Loss in leakage and drips	0.5	87
Engine and generator friction	5	874
Radiation and minor losses	1	175
Total		19,753

	Per Cent.	B.t.u.
Returned by vacuum heater	5.5	1,086
Returned by atmospheric heater	7.9	1,560
Returned by economizer	9.7	1,916
Total	23.1	4,562
Net heat delivered to engine in the form of steam to produce one electrical horsepower, 19,753 − 4,562		15,191
Percentage converted to electrical power $\frac{2,546}{15,191}$	16.7	
Boiler efficiency	70	
Percentage of heat value of fuel necessary to produce one electrical horsepower at switchboard $\frac{2546 \times 0.70}{15,191}$	11.7	

The preceding figures refer to reciprocating engine plants only and give the results of very good practice. So much depends upon the size and character of the prime movers, the nature of the fuel, and the conditions of operation that no definite figure can be given for the percentage of heat converted to power in a given type of station. Six per cent represents good average practice in a non-condensing plant and 10 per cent in a condensing plant using saturated steam. Pumping stations operating continuously under full load have realized as much as 15 per cent of the total heat value of the fuel, but such performances are practically unobtainable in connection with reciprocating engine steam-driven electrical power plants with the usual peak loads and low yearly load factor.

Figure 6 gives a diagrammatic arrangement of the essential elements of a modern steam turbine plant including the various heat-reclaiming devices described in the preceding paragraph. Turbo-generator No. 3 of the Northwest Station, Commonwealth Edison Company, Chicago, Illinois, is a well-known example of this arrangement of prime mover and auxiliaries. In the Northwest Station the prime mover is a horizontal turbine-generator of 30,000 kilowatt rated capacity at 100 per cent power factor, with high and low pressure cylinders mounted in tandem on the same shaft. The exciter is direct connected to the main generator and is rated at 110 kilowatts. Approximately 60,000 cubic feet per minute of cooling air are required to ventilate the generator and carry away the electrical heat losses. All bearings are water cooled and the oil supply is automatically maintained by a pump connected to the shaft of the governor. The boilers, five in number, are rated at 1220 horsepower each, and are equipped with traveling chain grates. When operating at full capacity they are capable of delivering about 400,000 pounds of steam per hour. Steam is generated at a pressure of 250 pounds gauge and superheated to a final temperature of approxi-

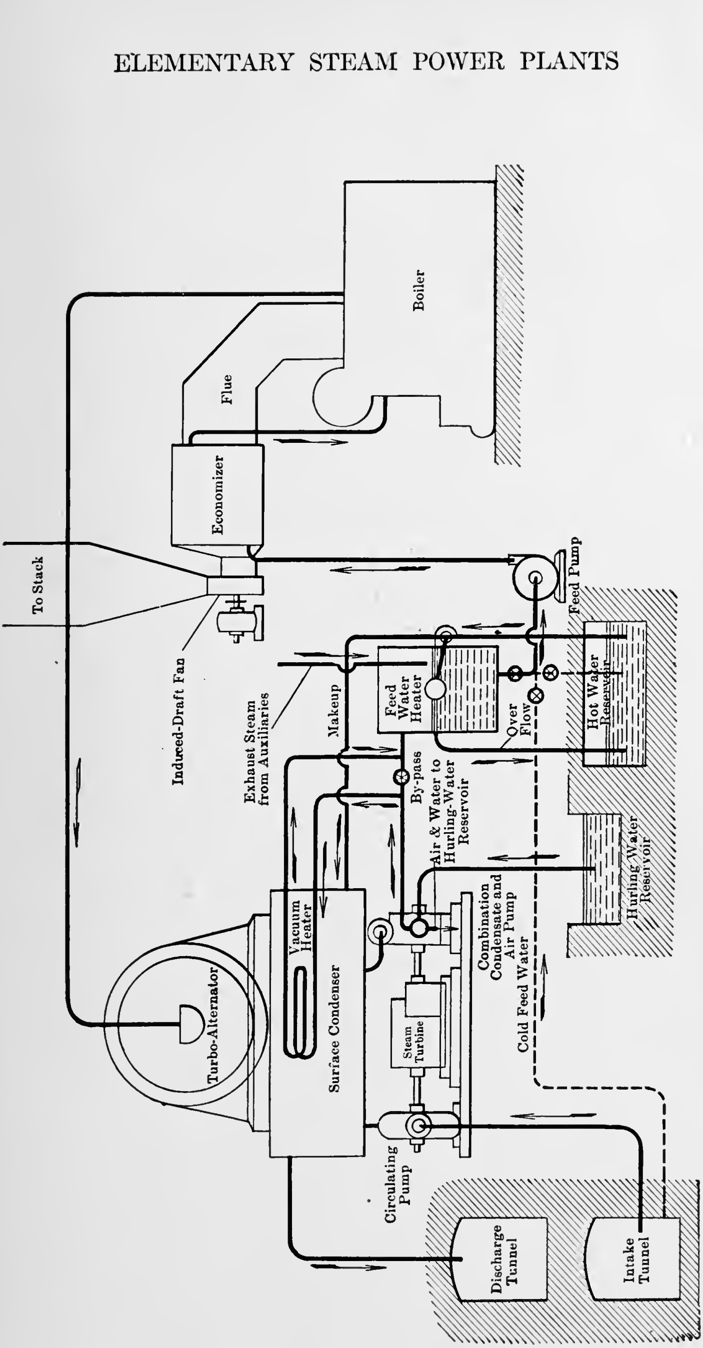


FIG. 6. Elementary Steam Turbine Plant with Full Complement of Heat-reclaiming Devices.

mately 625 deg. fahr. The main header is 20 inches in diameter and delivers the steam to the turbine at a velocity of about 8000 feet per minute at rated load. The condenser is of the surface type, contains 50,000 square feet of cooling surface and is attached directly below the low pressure turbine. Circulating water is obtained from the river through an intake tunnel fitted with revolving screens. A double suction centrifugal pump of 52,000 gallons per minute capacity delivers the water to the condenser against a static head of 15 feet. This pump is driven by a 650 horsepower non-condensing turbine at 1500 r.p.m. In passing through the condenser the temperature of the circulating water is raised approximately 15 degrees. The air and condensate are withdrawn from the condenser through a single pipe which connects with the separating chamber of a combination condensate and air pump of the "turbo-air" type. This pump is mounted on the same shaft with the turbine-driven circulating pump. The condensate at a temperature of 85 deg. fahr. is removed from the separating chamber by the "condensate" end of the combination pump and delivered to the preheater or vacuum heater. The air is removed by the "hurling water" end of the combination pump and delivered to the hurling water reservoir. The hurling water is used over and over again and serves only for the ejection of air.

The condensate in passing through the preheater, which is located in the condenser near the opening of the exhaust of the low pressure turbine, has its temperature raised to 100 deg. fahr. After passing through the preheater the condensate is discharged into the atmospheric heater where its temperature is increased to 160 deg. fahr. by the exhaust steam from the steam-driven auxiliaries. The overflow from the heater is discharged into the hot water reservoir from which a certain amount is drawn into the condenser through the agency of an automatically controlled float located in the heater. This system of drawing make-up and overflow water into the condenser becomes inoperative when the distance through which the water is to be lifted is sufficiently great to cause "vapor binding." The higher the temperature of the water in the reservoir the lower will be the permissible lift. The object of this arrangement is to maintain a continuous supply to the boilers irrespective of the fluctuation in the amount of condensate and to conserve the overflow. The boiler feed pumps are turbine-driven three-stage centrifugal pumps, and at a speed of 2500 r.p.m. are capable of delivering 400,000 pounds of condensate into the economizers at a pressure of 315 pounds gauge. Each boiler has its own economizer and independently driven induced draft fan. Each economizer contains 7300 square feet of heating surface and is inserted between the boiler and the breaching. The feed water is raised from a temperature of 160 deg. fahr. to 270 deg. fahr. in passing through the economizers. The five 100 horsepower motor-driven induced-draft fans maintain a draft in the boiler uptakes of 2.4 inches and are capable of handling 90,000 cubic feet of gases per minute.

The thermal efficiency of these very large steam turbines is higher than that of the gas engine and is excelled only by engines of the Diesel type. Allowing an average steam consumption of 12 pounds per kilowatt-hour for the turbine and all its auxiliaries and a boiler and economizer efficiency of 80 per cent, the over-all efficiency from switchboard to coal pile is approximately 20 per cent. The over-all efficiency measured over the period of one year is somewhat less than this because of great variation in load and the accompanying standby losses.

In Europe a combined plant efficiency of 15 per cent is not uncommon. Even small semi-portable plants of 40 horsepower are operated with over-all efficiencies as high as 14 per cent. In these small plants the engine, boiler, and auxiliaries are combined, permitting a high degree of superheat with minimum heat losses. A 40-horsepower plant tested by Professor Josse of the Royal Technical School, Germany, gave the following results: coal consumed per brake hp-hour, 1.23 pounds, corresponding to an over-all efficiency of 14.2 per cent. Steam consumption, 9.5 pounds per i.hp-hour. Boiler and superheated efficiency, 77.7 per cent. (See Zeit. des Ver. Deut. Ingr., March 18 and 25, 1911, and Power, Sept. 27, 1910, p. 1714.)

The remarkable economy which is being effected in Europe with this type of plant is still further marked by the performance of a 100 horsepower Wolf tandem compound locomobile which is credited with a performance of one brake horsepower per 0.86 pound of coal, corresponding to an over-all efficiency of 20 per cent. (Zeit. des Ver. Deut. Ingr., June, 1911.)

The percentage of the heat value of the fuel realized as energy at the point of consumption is considerably less than the over-all efficiency from "coal-pile to switchboard" because of the transmission, distribution and service losses. These losses vary within wide limits, depending upon the size and type of plant, character of equipment, length of transmission lines and various other influencing factors. Figure 7 illustrates the approximate losses for a large plant such as the Northwest Station of the Commonwealth Edison Company of Chicago.

For a description of the Ford Gas-Steam Plant see Power, Nov. 21, 1916, p. 706; Jan. 16, 1917, p. 70.

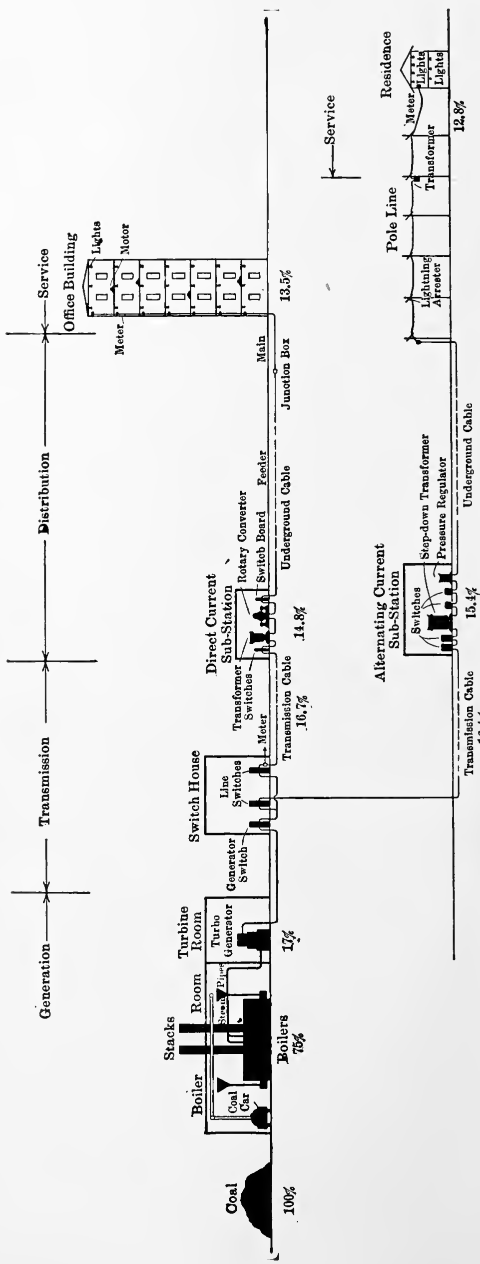


FIG. 7. Approximate Distribution of Energy Losses in a Modern Turbo-alternator Central Station.

EXERCISES.

1. Make a diagrammatic outline of a simple non-condensing plant correctly locating all the essential elements entering into its composition. Indicate by means of arrow points the direction of flow of the feed water and steam.

2. Same instruction as in Problem 1, except that a non-condensing plant with exhaust steam heating system is to be considered.

3. Enumerate the character and extent of the heat losses from "coal-pile to switchboard" in a simple non-condensing piston engine plant.

4. Beginning with the cold water supply trace the path of the feed water and steam through the various essential elements in a condensing plant equipped with a full complement of "heat-saving" appliances.

5. Make a skeleton outline of a modern turbo-alternator plant correctly locating and designating by name all the essential elements entering into its composition.

CHAPTER II

FUELS AND COMBUSTION

6. General. — The cost of fuel is by far the greatest single item of expense in the production of power and ranges from 40 per cent to 70 per cent of the total operating costs. Furthermore, all fuels are slowly but surely increasing in price and larger investments for fuel saving equipment are justified. In localities where a specific fuel is plentiful the problem resolves itself merely into a study of the best methods of burning this fuel, but in situations where various kinds of fuel are available the selection of the one best suited for a given or proposed equipment includes a careful consideration of such items as composition of the fuel, size, cost per ton, heating value, refuse incident to combustion, initial waste products such as ash and moisture, storage requirements, and transportation facilities.

The fuels used for steam making are coal, coke, wood, peat, mineral oil, natural and artificial gases, refuse products such as straw, manure, sawdust, tan bark, bagasse, and garbage.

In most cases that fuel is selected which develops the required power at the lowest cost, taking into consideration all of the circumstances that may affect its use. Occasionally the disposition of waste products is a factor in the choice, but such instances are uncommon. The boilers and furnaces are designed to suit the fuel selected.

7. Classification of Fuels. — Fuels may be divided into three classes as follows:

1. Solid fuels.
   a. Natural: straw, wood, peat, coal.
   b. Prepared: charcoal, coke, peat, and briquetted fuels.
2. Liquid fuels.
   a. Natural: crude oils.
   b. Prepared: distilled oils.
3. Gaseous fuels.
   a. Natural: natural gas.
   b. Prepared: coal gas, water gas, producer gas, oil gas.

8. Solid Fuels. — Solid fuels are of vegetable origin and exist in a variety of forms between that of a comparatively recent cellulose growth and that of nearly pure carbon as anthracite coal. They owe their forms to the conditions under which they were created or to the geological changes which they have undergone. With each succeeding stage the percentage of carbon increases and the oxygen content decreases. The chemical changes are approximately as given in Table 1.

TABLE 1.

PROGRESSIVE CHANGE FROM PURE CELLULOSE TO ANTHRACITE.

Substance.	Carbon.	Hydrogen.	Oxygen.
	Per Cent.	Per Cent.	Per Cent.
Pure cellulose	44.44	6.17	49.39
Wood	52.65	5.25	42.10
Peat	59.57	5.96	34.47
Lignite	66.04	5.27	28.69
Brown coal	73.18	5.58	21.14
Bituminous coal	75.06	5.84	19.10
Semi-bituminous coal	89.29	5.05	6.66
Anthracite	91.58	3.96	4.46
Graphite	100.00	....	....

Of all the various grades of solid fuels coal is by far the most important.

Origin of Coal: Bulletin No. 491, U. S. Geological Survey, p. 705.

9. Composition of Coal. — All coals when separated into their ultimate chemical constituents are composed principally of varying proportions of carbon, hydrogen, oxygen, sulphur and refractory earths. Carbon and hydrogen are the only desirable elements from a combustion standpoint and the others may be considered impurities. The various combinations into which the carbon, hydrogen and oxygen are united are extremely complex and greatly influence the physical characteristics of the fuel. All of the carbon and hydrogen is not available for combustion since part of the carbon may be present as a carbonate and part of the hydrogen as water. A knowledge of the physical and chemical characteristics of a fuel as determined in the laboratory is of great importance since it enables the engineer to determine in advance the fuels best suited for a given or proposed equipment.

Proximate Analysis. — This analysis enables the engineer to predict to a certain extent the behavior of the fuel in the furnace by giving the percentages of moisture, ash, fixed carbon and volatile matter.

Moisture as obtained from this analysis is purely an arbitrary quantity based upon the loss in weight of a sample when maintained for approximately one hour at a temperature of 220 deg. fahr. The material driven off in this manner is not all water since some of the volatile combustible may distill off; furthermore, all of the water may not be evaporated by this treatment. It is intended and does bring the material to a condition which can be duplicated closely and represents a fixed basis for comparison. Moisture not only increases the cost of transporting and handling the fuel but also absorbs heat in the furnace which might otherwise be available for generating steam. Coal free from "moisture" is known as *"dry coal."* *

The residue which remains after the coal has been completely burned is classified as *ash.* It is derived from the inorganic matter in the coal, such as sand, clay, shale, "slate" and iron pyrites, and is composed largely of compounds of silica, alumina, iron and lime, together with small quantities of magnesia. A large percentage of ash is undesirable since it reduces the heat value of the fuel, increases the cost of transportation and handling, necessitates disposal of refuse and often produces troublesome clinker. Coal free from moisture and ash is commonly designated as *combustible* though the nitrogen and oxygen included are not combustible.

That portion of the carbon combined with hydrogen, and other gaseous compounds which are driven off the dry coal by the application of heat, constitutes the *volatile combustible matter,* or simply *volatile* matter. The term "volatile combustible" is a misnomer since a considerable fraction of the distilled gases consists of water vapor, carbon dioxide, nitrogen and other inert, non-combustible dilutants. The pertinence of the "volatile matter" to the engineeer is obvious, since a high percentage indicates that special care must be observed in effecting smokeless combustion.

The uncombined carbon or that portion which remains after the volatile matter has been driven off is known as *fixed carbon.* Fixed carbon, however, is not pure carbon since the carbonized residue contains in addition to the ash forming constituents, small amounts of hydrogen, oxygen, nitrogen and approximately half the original sulphur content. "Fixed carbon" is a measure of the relative coking properties of coals though in the commercial manufacture of coke or gas the yield of coke is several per cent higher than that obtained in the laboratory. In

* "Moisture" as determined from the proximate analysis must not be confused with "air-drying loss." The primary purpose of air-drying is to reduce the moisture content to such a condition that there will not be rapid changes in the weight of the sample during the course of analysis; it simply shows the amount of moisture removed in order to bring the sample to a condition of equilibrium with respect to the moisture in the air of the room. "Air-drying loss" is the amount of moisture driven off when the sample, as received, is subjected to a temperature of 86 to 95 deg. fahr. The drying process is continued until the loss in weight between two successive weighings made 6 to 12 hours apart does not exceed 0.2 per cent. See "Analysis of Coal in the United States," Bulletin 22, 1913, Bureau of Mines.

the proximate analysis of coal the sulphur is included in the volatile matter, fixed carbon and ash. Sulphur occurs in coal as pyrites, sulphate of iron, lime, and alumina, and in combination with the coal substance as organic compounds. Although classed as an impurity, sulphur has a heating value, when in the form of iron pyrites, of almost one-half that of the coal it replaces. For steaming purposes sulphur is objectionable only when its presence produces a badly clinkering ash.

Proximate Analysis: Journal of the American Chemical Society, Vol. 21, p. 116; U. S. Bureau of Mines, Bulletins No. 22, 1913, and No. 85, 1914.

Ultimate Analysis. — In the ultimate analysis the composition of the fuel is expressed in terms of its elementary constituents of carbon, hydrogen, oxygen, nitrogen and sulphur, and ash. The ultimate analysis is of considerable importance in determining the more important heat losses incident to combustion, but an accurate analysis requires considerable time for its consummation and necessitates the services of a competent chemist. For that matter an accurate proximate analysis requires even more skill than the ultimate analysis, since the determination of hydrogen, carbon and nitrogen is not subject to the arbitrary conditions that must be maintained in the proximate analysis. But as ordinarily made the latter requires little apparatus and is within the skill of the average engineer.

Both the ultimate and the proximate analyses may be expressed in terms of

(1) "Coal as received" or *coal as fired.*
(2) "Coal, moisture free" or *dry coal.*
(3) "Coal, moisture and ash free" or *combustible.*
(4) "Coal, moisture, ash and sulphur free."

In the various fuel publications issued by the Bureau of Mines and the U. S. Geological Survey, the quoted terms are used almost exclusively, whereas in the Boiler Code advocated by the American Society of Mechanical Engineers and in most engineering literature the italicised terms are given preference. Engineers prefer to have the results based on coal as fired, since this represents the condition of the fuel as fed to the furnace. For convenience in comparing analyses the results are usually based on dry coal and combustible, but occasionally, as will be shown later, the "coal, moisture, ash and sulphur free" basis is of service. Analyses are readily converted from one basis to another as will be seen from the following example.

Example 1. Given the proximate and ultimate analyses of a sample of coal as received. Transfer these analyses to the "moisture free" and "moisture and ash free" basis. Also transfer the ultimate analysis as received to the "moisture, ash and sulphur free" basis.

ILLINOIS COAL.

(Carterville District.)

	Coal as Received or Coal as Fired.	Coal, Moisture Free or Dry Coal.	Coal, Moisture and Ash Free, or Combustible.
	A.	B.	C.
Fixed carbon	50.19	54.42	61.49
Volatile matter	31.44	34.08	38.51
Ash	10.61	11.50	....
Moisture	7.76	....	....
	100.00	100.00	100.00

Column B = column A ÷ (1 − proportional weight of moisture)
= column A ÷ 0.9224.
Column C = column A ÷ [1 − (proportional weight of moisture + ash)]
= column A ÷ 0.8163.

For the ultimate analysis:

	Coal as Received.		Coal, Moisture Free.	Coal, Moisture and Ash Free.	Coal, Moisture, Ash and Sulphur Free.
	A.	A_1.	B.	C.	D.
Carbon	66.55	66.55	72.15	81.52	83.54
Hydrogen	5.14	4.28	4.64	5.24	5.37
Nitrogen	1.32	1.32	1.43	1.62	1.66
Oxygen	14.41	7.51	8.14	9.21	9.43
Sulphur	1.97	1.97	2.14	2.41	....
Ash	10.61	10.61	11.50	....	....
Free moisture	....	*7.76	....	....	....
	100.00	100.00	100.00	100.00	100.00

* From the proximate analysis.

In the ultimate analysis of the coal as received (Column A) the free moisture of "Moisture" is included in the hydrogen and oxygen. Since the water is composed of one part hydrogen and eight parts oxygen, one-ninth of the moisture should be subtracted from the hydrogen and eight-ninths from the oxygen in order to include free moisture as a separate item, thus:

Hydrogen (column A_1) = hydrogen (column A) − $\frac{1}{9}$ × per cent moisture
$= 5.14 - \frac{1}{9} \times 7.76$
$= 4.28.$

Oxygen (column A_1) $= $ oxygen (column A) $- \frac{8}{9} \times$ per cent moisture

$= 14.41 - \frac{8}{9} \times 7.76$

$= 7.51.$

Column B = column $A_1 \div (1 -$ proportional weight of moisture)

$=$ column $A_1 \div 0.9224.$

Column C = column $A_1 \div [1 -$ proportional weight of (moisture + ash)]

$=$ column $A_1 \div 0.8163.$

Column D = column $A_1 \div [1 -$ proportional weight of (moisture + ash + sulphur)]

$=$ column $A_1 \div 0.7966.$

The term *free hydrogen* or *available hydrogen* is based on the assumption that all of the oxygen in the coal is combined with hydrogen in the proper ratio to form water, or,

$$\text{Free hydrogen} = \text{Total hydrogen} - \frac{\text{Oxygen}}{8} = H - \frac{O}{8}.$$

All of the oxygen $+ \frac{O}{8}$ is the weight of the *combined moisture,* and the sum of the free moisture and combined moisture is designated as the *total moisture.*

Example 2. Determine the free hydrogen, combined moisture and total moisture for coal as fired, the analysis of which is given in Example 1.

$$\text{Free hydrogen} = 4.28 - \frac{7.51}{8}$$
$$= 3.34.$$
$$\text{Combined moisture} = 7.51 + \frac{7.51}{8}$$
$$= 8.45.$$
$$\text{Total moisture} = 7.76 + 8.45$$
$$= 16.21.$$

For most engineering purposes extreme accuracy is not necessary in determining the ultimate analysis, since the average commercial heat balance is in itself only approximate at the best, consequently recourse may be had to empirical formulas for approximating the weight of the chemical constituents from the proximate analysis, thus:*

$$\text{For hydrogen, } H = V\left(\frac{7.35}{V + 10} - 0.013\right), \tag{1}$$

in which

H = the per cent of hydrogen in the combustible.
V = the per cent of volatile matter in the combustible.

* "Experimental Engineering," Carpenter & Diederichs, 1915, p. 507.

For nitrogen,

$$N = 0.07\ V \text{ for anthracite and semi-anthracite}$$
$$= 2.10 - 0.012\ V \text{ for bituminous and lignite.} \qquad (2)$$

For total carbon (fixed carbon + volatile carbon),

$$\left.\begin{aligned} C &= F + 0.02\ V^2 \text{ for anthracite} \\ &= F + 0.9\ (V - 10) \text{ for semi-anthracite} \\ &= F + 0.9\ (V - 14) \text{ for bituminous coals} \\ &= F + 0.9\ (V - 18) \text{ for lignites} \end{aligned}\right\} \qquad (3)$$

in which

C = per cent of total carbon in the combustible.

F = per cent of fixed carbon as determined from the proximate analysis.

V = as above.

Sulphur in the coal increases the value of V, hence the calculated value of C is too high by practically the sulphur content of the combustible.

Example 3. Calculate the ultimate analysis from the proximate analysis of the coal given in Example 1.

$$H = 38.51\left(\frac{7.35}{38.51 + 10} - 0.013\right) = 5.33 \text{ per cent.}$$ (Analysis gives H = 5.24.)

$$N = 2.10 - 0.012 \times 38.51$$
$$= 1.64 \text{ per cent.}$$ (Analysis gives N = 1.62.)

$$C = 61.49 + 0.9\ (38.51 - 14)$$
$$= 83.55 \text{ per cent.}$$ (Analysis gives C = 81.52.)

The ultimate analysis of the *coal as received*, neglecting the sulphur, is:

	Calculated Values, Per Cent.	Actual Values, Per Cent.
H = 5.33 } × 0.8163	4.35	4.28
N = 1.64 } × 0.8163	1.33	1.32
C = 83.55 } × 0.8163	68.20	68.52*
Ash (by analysis)	10.61	10.61
Moisture (by analysis)	7.76	7.76
O (by difference)	7.75	7.51
	100.00	100.00

* Carbon + sulphur = 66.55 + 1.97 = 68.52.

It will be seen that the agreement is fairly close with the exception of that for total carbon. As previously stated this is largely due to the fact that the sulphur content is practically all added to the total carbon. If the sulphur content of the coal is known, as in this case (2.41 per cent), correction can be made so that the final computed value for the total carbon is 83.55 − 2.41 = 81.14 per cent per lb. of combustible.

This method of calculating the ultimate from the proximate analysis gives fairly accurate results for most coals but with some grades of bituminous coals the results for H and C may be in error as much as 5 per cent for each constituent.

The average plant is not equipped with the necessary apparatus for making the proximate analysis, not alone the ultimate analysis, so that the preceding calculations are of little value to the engineer in charge. The proximate analysis is too cumbersome even for the large plant when a number of heat balances are required in a short time, as when trying out new fuels. In such cases the following method enables the engineer to approximate the ultimate analysis with sufficient accuracy for most practical purposes, provided the source of coal supply is known:*

Bulletins Nos. 22 and 85 issued by the Bureau of Mines, contain a large number of ultimate analyses of coals from all parts of the country. A study of the data will show that *coals from any given locality have practically the same analysis when expressed on a "free from moisture, ash and sulphur" basis;* hence, it is principally a question of determining the amount of free moisture and ash in the sample (a comparatively simple test) and in assuming the sulphur content. Since the percentage of sulphur is not uniform some error may be introduced in making this assumption but it is negligible as far as the average commercial heat balance is concerned. This method of obtaining the ultimate analysis is best illustrated by an example.

Example 4. Assume that a sample of Illinois coal (analysis as per Example 1) is available and that the ash and moisture determinations only have been made. Approximate the ultimate analysis from the average "moisture, ash and sulphur free" analysis of Illinois coals.

The average of a number of Illinois coals † as recorded in the Government bulletin referred to is:

Combined Moisture.	Free Hydrogen.	Carbon.	Nitrogen.
11.94	4.14	82.4	1.52

Assuming the per cent of sulphur in the coal under consideration to be the average of Illinois coals as recorded in the Government bulletins (S = 2.84 per cent), the total free moisture, ash and sulphur would be $7.76 + 10.61 + 2.84 = 21.2$ per cent; and the "free from moisture, ash and sulphur" content, $100 - 21.2 = 78.8$ per cent. The ultimate analysis of the coal as received may then be calculated as follows:

* P. W. Evans, *Armour Engineer*, May, 1915, p. 301.

† Moisture, ash and sulphur free.

	Calculated Values, Per Cent.	Actual Values, Per Cent.
Combined moisture, 11.94 × 0.788	9.40	8.45
Free moisture (by test)	7.76	7.76
Free hydrogen, 4.14 × 0.788	3.26	3.34
Total carbon, 82.3 × 0.788	64.98	66.55
Nitrogen, 1.52 × 0.788	1.19	1.32
Ash (by test)	10.61	10.61
Sulphur	*2.80	1.97
	100.00	100.00

* By assumption.

The agreement between calculated and actual values for most Illinois coals is much closer than in this particular example. The splendid work of the U. S. Bureau of Mines will soon place at the disposal of the public complete analyses of all the coal fields in the country and the error in assuming the average values of an entire state, as in the preceding example, may be greatly reduced by taking the average values for the particular field in which the coal under consideration is mined.

Ultimate Analysis: See references under "Approximate Analysis."

Methods of Determining Sulphur Content of Fuels: U. S. Bureau of Mines, Technical Paper 26, 1912.

Methods of Analyzing Coal and Coke: U. S. Bureau of Mines, Technical Paper 8, 1913.

The Coking of Coal at Low Temperatures: University of Illinois Bulletin, Vol. XII, No. 39, 1915.

The Analysis of Coal with Phenol as a Solvent: University of Illinois Bulletin, No. 76, 1915.

10. Classification of Coals. — Coals and allied substances have been variously classified according to

1. Oxygen-hydrogen ratio, or Gruner's classification.
2. Fixed carbon and volatile combustible matter.
3. Fuel ratio, or the ratio of the fixed carbon to the volatile combustible matter.
4. Calorific value.
5. Fixed carbon.
6. Total carbon.
7. Hydrogen.
8. Carbon-hydrogen ratio, or the ratio of the total carbon to the hydrogen.

Gruner's classification is as follows:

(Eng. and Min. Jour., July 25, 1874.)

	Ratio $\frac{O}{H}$.		Ratio $\frac{O}{H}$.
Anthracite	1 to 0.75	Peat	6 to 5
Bituminous	4 to 1	Wood	7
Lignite	5	Cellulose	8

Kent's classification, according to the constituents of the combustible, is as follows (Steam Boiler Practice):

	Per Cent of Dry Combustible.	
	Fixed Carbon.	Volatile Matter.
Anthracite	97 to 92.5	3 to 7.5
Semi-anthracite	92.5 to 87.5	7.5 to 12.5
Semi-bituminous	87.5 to 75	12.5 to 25
Bituminous — Eastern	75 to 60	25 to 40
Bituminous — Western	65 to 50	35 to 50
Lignite	Under 50	Over 50

Gruner's, Kent's, and other schemes of classification outlined above, with the exception of the carbon-hydrogen ratio, are more or less unsatisfactory, since the groups are not as clearly defined as indicated and overlap to a considerable extent.

The U. S. Geological Survey proposes the following classification according to the carbon-hydrogen ratio which appears to apply satisfactorily to all grades of coal.

(Compiled from Report of Government Coal Testing Plant, Professional Paper No. 48, 1906.)

Group.	Class.	Example.	Carbon-hydrogen Ratio.
A	Graphite		—
B	Anthracite	*Buck Mountain, Pa	— to 30
C	Anthracite	*Scranton, Pa	30 to 26
D	Semi-anthracite	*Bernice Basin, Pa	26 to 23
E	Semi-bituminous	Spadra Bed, Ark.	23 to 20
F	Bituminous	New River, W. Va	20 to 17
G	...do	Connelsville Field, Pa	17 to 14.4
H	...do	Marion County, Ill	14.4 to 12.5
I	...do	Red Lodge, Mont	12.5 to 11.2
J	Lignite	Gallup Field, N. M	11.2 to 9.3
K	Peat		9.3 to
L	Wood		7.2

* Not included in Government's Report.

In its various bulletins the U. S. Bureau of Mines uses the following arbitrary classification which is virtually based on the fuel ratio:

Anthracite	Bituminous
Semi-anthracite	Sub-bituminous
Semi-bituminous	Lignite

Classification of Coals: U. S. Geographical Survey Bulletin 541, 1914; Prac. Engr. U. S., Jan., 1910: Mines and Minerals, Feb., 1911.

11. Anthracites. — These are the best coals and consist almost entirely of carbon; they contain very little hydrocarbon and burn with little or no smoke, are slow to ignite, burn slowly, and break into small pieces when rapidly heated. They require a very large grate of about twice the surface necessary for bituminous coal. Large sizes may be burned in almost any kind of a furnace and with moderate draft.

TABLE 2.

COMPOSITION OF TYPICAL AMERICAN ANTHRACITE COALS.

	Treverton, Pa.	Wilkesbarre, Pa.	Drifton, Pa.	Lehigh, Pa.	Scranton, Pa., Culm, Air Dried.	Lykens Valley.
Proximate analysis:	*	*	†	†	‡	†
Water	0.84	3.45	1.37	1.97	2.08	1.50
Volatile matter	6.67	2.75	3.59	4.35	7.27	7.84
Fixed carbon	85.66	87.90	89.11	86.49	74.32	81.07
Ash	6.83	5.90	5.93	7.19	16.33	9.59
	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:						
Carbon	90.66	88.86	87.70	85.66	75.21	83.20
Hydrogen	1.73	2.04	2.56	2.78	2.81	3.29
Nitrogen	....	0.90	1.03	0.77	0.80	0.95
Oxygen	0.78	1.95	2.26	2.87	4.08	2.45
Sulphur	....	0.35	0.56	0.64	0.77	0.50
Ash	6.83	5.90	5.89	7.28	16.33	9.61
	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:						
Calorimeter	13,980	13,950	....	....	12,472	....
Dulong's formula	14,194	14,103	14,217	14,038	12,426	14,003
Classification:						
Carbon-hydrogen ratio	52.5	42.5	34.4	30.9	26.7	25.
Fuel ratio	12.9	32.0	29.9	11.0	10.2	10.4

* Authority not stated. † H. J. Williams. ‡ U. S. Geological Survey.

For smaller sizes a thinner bed has to be carried unless a strong draft is used. There is difficulty in keeping a thin bed free from air-holes.

When possible the coal should be at least six inches deep on the grate. On account of the large percentage of ash in the smaller sizes, the fire requires frequent cleaning. Anthracites do not require "slicing" and should be disturbed only when cleaning is necessary. Nearly all anthracites, with some unimportant exceptions, come from three small fields in eastern Pennsylvania. On account of the limited supply and the great demand for domestic purposes, sizes over "pea coal" are prohibitive in price for steam power plant use. Table 2 gives the composition and classification of a number of typical American anthracite coals, and Table 3, one of the standard divisions of mesh according to which they are classed and marketed. Specific gravity, 1.4 to 1.6; fuel ratio, not less than 10.

Burning No. 3 Buckwheat: Power, Dec. 27, 1901; Mar. 21, 1911. *Burning Anthracite Culm of Poor Quality:* Trans. A.S.M.E., 7-390. *Anthracite Culm Briquets,* Am. Inst. Min. Engrs., Bulletin, Sept., 1911. *Calorific Value of Anthracite:* Mines and Minerals, Sept., 1911. *Preparation of Anthracite:* Am. Inst. Min. Engrs., Bulletin, Oct., 1911. *Stoking Small Anthracite Coal:* Power, Oct. 19, 1916, p. 540.

TABLE 3.

SIZES OF ANTHRACITE COAL.

A.S.M.E. Code of 1915.

Size.	Diameter of Opening Through or Over which Coal Will Pass, Inches.	
	Through.	Over.
Broken	$4\frac{1}{2}$	$3\frac{1}{4}$
Egg	$3\frac{1}{4}$	$2\frac{5}{16}$
Stove	$2\frac{5}{16}$	$1\frac{5}{8}$
Chestnut	$1\frac{5}{8}$	$\frac{7}{8}$
Pea	$\frac{7}{8}$	$\frac{9}{16}$
*No. 1 Buckwheat	$\frac{9}{16}$	$\frac{5}{16}$
*No. 2 Buckwheat	$\frac{5}{16}$	$\frac{3}{16}$
*No. 3 Buckwheat	$\frac{3}{16}$	$\frac{3}{32}$
Culm	$\frac{3}{32}$	....

* The terms "Buckwheat," "Rice," and "Barley," respectively, are used in some localities instead of No. 1, No. 2, and No. 3 Buckwheat.

12. Semi-anthracites. — These coals kindle more readily and burn more rapidly than the anthracites. They require little attention, burn freely with a short flame and yield great heat with little clinker and ash. They are apt to split up on burning and waste somewhat in falling through the grate. They swell considerably, but do not cake. They have less density, hardness and metallic luster than anthracite, and can generally be distinguished by their tendency to soil the hands, while pure anthracite will not. Semi-anthracites are not of great importance in the steam power plant field on account of the limited supply and high cost. They are found in a few small areas in the western part of the anthracite field. Specific gravity, 1.3 to 1.4. Fuel ratio 6 to 10.

13. Semi-bituminous. — These coals are similar in appearance to semi-anthracite, but they are somewhat softer and contain more volatile matter. They have a very high heating value, have a low moisture, ash and sulphur content, are readily burned without producing objectionable smoke and rank among the best steaming coals in the world. The supply is limited and on account of high cost, except in the immediate vicinity of the mines, they are not generally used for power purposes. The centers of production are the Pocahontas and New River fields of Virginia and West Virginia, Georges Creek field of Maryland, Windber field of Pennsylvania and the western end of the Arkansas field. Table 4 gives the composition and classification of a number of typical American semi-anthracite and semi-bituminous coals. Fuel ratio 3 to 6 or 7.

TABLE 4.

COMPOSITION OF TYPICAL AMERICAN SEMI-ANTHRACITE AND SEMI-BITUMINOUS COALS.

	Bernice Basin, Pa.	Coalhill, Ark.	Pocahontas, W. Va.	Kanawha Series, W. Va.	New River, W. Va.	Clearfield, Pa.
	*	*	†	‡	†	§
Proximate analysis:						
Water	1.57	2.36	4.07	1.42	1.53	0.44
Volatile matter	9.40	12.68	16.34	20.72	21.54	18.76
Fixed carbon	83.69	72.88	68.47	70.05	71.88	73.15
Ash	5.34	12.08	11.12	7.81	5.05	7.65
	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:						
Carbon	85.46	76.44	76.51	81.95	82.87	80.32
Hydrogen	3.72	3.82	4.27	4.30	4.76	4.88
Nitrogen	1.12	1.37	1.00	1.29	1.68	1.46
Oxygen	3.45	4.30	6.59	3.68	4.99	4.69
Sulphur	0.91	1.99	0.51	0.97	0.65	1.00
Ash	5.34	12.08	11.12	7.81	5.05	7.65
	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:						
Calorimeter	......	13,259	13,509	14,686	14,807	......
Dulong's formula	14,552	13,273	13,329	14,363	14,691	14,432
Classification:						
Carbon-hydrogen ratio‖	23.0	20.7	19.6	19.0	17.8	16.5
Fuel ratio	8.5	5.7	4.2	3.4	3.3	3.9

* Authority not stated. † U. S. Geological Survey. ‡ W. Va. Geological Survey. § H. J. Williams. ‖ Based on air-dried sample.

14. Bituminous. — These coals are the most widely distributed and the most extensively used fuel in steam power plant engineering. They contain a large and varying amount of volatile matter and burn freely with the production of considerable smoke unless carefully fired. Their physical properties vary widely and they are commonly classified as

1. Dry, or free-burning bituminous.
2. Bituminous caking.
3. Long-flaming bituminous.

1. Dry bituminous coals are the best of the bituminous variety for steaming purposes. They are hard and dense, black in color, but somewhat brittle and splintery. They ignite readily, burn freely with a short clean bluish flame and without caking. Specific gravity, 1.25 to 1.40.

2. Bituminous caking coals swell up, become pasty and fuse together in burning. They contain less fixed carbon and more volatile matter than the free-burning grades. Caking coals are rich in hydrocarbon and are particularly adapted to gas making. The flame is of a yellowish color. Specific gravity, about 1.25.

3. Long-flaming bituminous coals are similar in many respects to the caking coals but contain a larger percentage of volatile matter. They burn freely with a long yellowish flame. They may be either caking, non-caking or splintery. They are very valuable as a gas coal, and are little used for steaming purposes. Specific gravity, about 1.2.

Table 5 gives the composition and classification of a number of typical American bituminous coals.

For sizes of bituminous coal see paragraph 38.

Mineral Resources of the United States: U. S. Geological Survey, 1911.
Analyses of Coals: Bul. No. 22, U. S. Bureau of Mines, 1913.
Analyses of Mine and Car Samples: Bul. No. 85, U. S. Bureau of Mines, 1914.
Report of the United States Fuel-Testing Plant at St. Louis, Mo.: Bul. No. 332, U. S. Geological Survey, 1908.
Index of Mining Engineering Literature: W. R. Crane, John Wiley & Sons.
Coal Mines of the United States: Peabody Atlas, A. Bement, Chicago, Ill.
Coking and Caking Coal: Power, March 28, 1916, p. 432.
Dry Preparation of Bituminous Coal at Illinois Mines: Univ. Ill. Bul. No. 43, June 26, 1916.
Fuels for Steam Boilers: Power, Mar. 28, 1916, p. 454.

15. Sub-bituminous Coals. — The term sub-bituminous has been adopted by the U. S. Geological Survey and the Bureau of Mines for what has generally been called "black lignite." These coals are not lignitic in the sense of being woody and many of them approach the lowest grade of bituminous coals for fuel purposes. It is difficult to separate

TABLE 5.

COMPOSITION OF TYPICAL AMERICAN BITUMINOUS COALS.*

	Freeport, W. Va.	Hocking Valley, Ohio.	Pere Marquette, Mich.	Brazil, Ind.	Eastern Field, Kentucky.	Horsecreek, Ala.	Fleming, Kansas.	Laddadale, Iowa.	Rich Hill, Mo.	Coffeen, Ill.	Cambria, Wyo.	Chariton, Iowa.
Proximate analysis:		†	†	†								
Water	1.48	6.65	12.15	8.98	3.10	2.34	4.99	8.24	8.33	14.43	9.44	15.39
Volatile matter	28.58	34.14	31.14	34.49	36.12	31.84	32.68	30.74	33.58	29.48	35.02	30.49
Fixed carbon	61.55	49.54	53.95	50.30	56.39	53.28	49.36	45.02	38.73	42.81	34.82	41.49
Ash	8.39	9.67	2.76	6.33	4.39	12.54	12.97	16.00	19.36	13.28	20.72	12.63
	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:												
Carbon	77.82	70.86	79.44	70.50	77.37	71.58	67.34	59.82	57.00	54.59	51.46	55.81
Hydrogen	4.89	4.70	5.29	4.76	5.43	5.01	4.98	4.81	4.97	5.49	5.00	5.74
Nitrogen	1.48	1.53	1.56	1.36	1.83	1.65	1.08	0.94	0.94	1.11	0.74	1.14
Oxygen	6.52	11.45	9.84	15.66	9.76	8.50	9.35	13.40	12.48	21.52	18.17	21.49
Sulphur	0.90	1.79	1.11	1.39	1.22	0.72	4.28	5.03	5.25	4.01	3.91	3.19
Ash	8.39	9.67	2.76	6.33	4.39	12.54	12.97	16.00	19.36	13.28	20.72	12.63
	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:												
Calorimeter	14,069	13,053	....	12,417	14,148	12,850	12,242	11,027	10,586	10,064	9650	10,242
Dulong's formula	13,927	12,444	14,160	12,084	13,955	12,927	12,217	10,881	10,656	9,866	9362	10,173
Classification:												
Carbon-hydrogen ratio‡	16.10	15.10	15.00	14.80	14.60	14.50	13.90	13.40	12.90	12.30	12.20	11.20
Fuel ratio	2.16	1.46	1.73	1.46	1.56	1.83	1.51	1.47	1.15	1.45	0.99	1.36

* Compiled from Report of Government Coal Testing Plant, U. S. Geological Survey.

† Not included in government report. ‡ Based on air-dried sample.

this class from bituminous coals and lignites by any of the classifications outlined at the beginning of paragraph 10. They are not woody in texture and are black in color, which enables them to be readily distinguished from the lignites. When exposed to the weather they slack considerably, a feature which distinguishes them from the bituminous coals. Sub-bituminous coals are found in most of the western fields.

16. Lignite, or Brown coal, is a substance of more recent geological formation than coal and represents a stage in development intermediate between coal and peat. Its specific gravity is low, 1.2, and when freshly mined contains as high as 50 per cent of moisture. It is non-caking, and on exposure to air, slackens or crumbles. The lumps check and fall into small irregular pieces with a tendency to separate into extremely thin plates. It deteriorates greatly during storage or long transportation. Lignite, as mined, is a low-grade fuel with a calorific value of about one-half that of good coal. When properly prepared and compressed into briquettes, lignite becomes an excellent fuel, resists weathering satisfactorily, permits handling and transportation without excessive deterioration and is practically smokeless. The superiority of briquettes over raw lignite is shown in Table 6:

TABLE 6.

IMPROVEMENT OF HEAT VALUE BY BRIQUETTING.*

Source	Moisture			Heat Value per Pound.		
	In Raw Lignite.	In Briquettes.	Removed.	Raw Lignite.	Briquettes.	Increase.
	Per Cent.	Per Cent.	Per Cent.	B.t.u.	B.t.u.	Per Cent.
Texas	33.0	9.0	24.0	6840	9336	36.5
North Dakota	40.0	12.0	28.0	6241	9354	50.0
North Dakota	42.0	10.0	32.0	6079	9355	54.0
California	40.0	10.0	30.0	6080	9264	52.4

* Bulletin No. 14, U. S. Bureau of Mines, p. 48.

The most extensive lignite deposits are situated long distances from fields of high-grade coal, and their use is at present limited to these regions.

North Dakota Lignite as a Fuel for Power Plant Boilers: Bul. No. 2, 1910, U. S. Bureau of Mines. *Briquetting Tests of Lignite:* Bul. No. 14, 1911, U. S. Bureau of Mines. General data pertaining to lignite fuels, Engr. U. S., Jan., 1910.

17. Peat, or Turf, is formed by the slow carbonization under water of a variety of accumulated vegetable materials. It is unsuitable for fuel until dried. Peat, as ordinarily cut and dried, is too bulky for commercial competition with coal, and is used only where coal is prohibitive in price. When properly prepared and compressed into briquettes peat is an excellent fuel. In Russia, Germany, and Holland peat briquettes have passed the experimental stage and several millions of pounds are manufactured annually. Peat is used but little in this country at present, though the deposits are extensive and widely distributed, but its possibilities are beginning to attract the attention of engineers. The proportion in which the various primary constituents exist in dried peat is approximately as follows:

	Per Cent.
Fixed carbon	35
Volatile matter	60
Ash	5

Peat: Prac. Engr. U. S., Jan., 1910, p. 21; Bul. No. 16, U. S. Bureau of Mines, 1911; Power, Sept. 6, 1910; Eng. and Min. Jour., Nov. 22, 1902; Feb. 7, 1903, Jour. Am. Peat Soc., July, 1911; Elec. Rev., Mar. 22, 1912; Min. and Eng. Wld., Nov. 28, 1911.

TABLE 7.

COMPOSITION OF TYPICAL AMERICAN SUB-BITUMINOUS COALS AND LIGNITES.*

(Run of Mine.)

	Red Lodge, Montana. (Black.)	Gallup, New Mexico. (Black.)	Texas. (Brown.)	Colorado. (Black.)	North Dakota. (Brown.)	Wyoming. (Black.)
Proximate analysis:						
Water	11.05	12.29	33.71	18.68	36.78	22.63
Volatile matter	35.90	34.58	29.25	34.88	28.16	35.68
Fixed carbon	42.08	46.14	29.76	40.45	29.97	37.19
Ash	10.97	6.99	7.28	5.99	5.09	4.50
	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis:						
Hydrogen	5.37	5.82	6.79	6.07	6.93	6.39
Carbon	59.08	63.31	45.52	57.46	41.87	54.91
Nitrogen	1.33	1.03	0.79	1.15	0.69	1.02
Oxygen	21.52	22.22	42.09	28.78	44.94	32.59
Sulphur	1.73	0.63	0.53	0.55	0.48	0.59
Ash	10.97	6.99	7.28	5.99	5.09	4.50
	100.00	100.00	100.00	100.00	100.00	100.00
Calorific value:						
Calorimeter	10,539	11,252	7348	10,143	7002	9734
Dulong's formula	10,355	11,153	7177	9,948	6944	9478
Classification:						
Carbon-hydrogen ratio†	11.50	11.20	10.90	9.80	9.60	9.40
Fuel ratio	1.17	1.09	1.02	1.16	1.06	1.05

* Compiled from Government Report, U. S. Geological Survey.

† Based on air-dried analysis.

18. Wood, Straw, Sawdust, Bagasse, Tanbark. — In certain localities cordwood is still used as a fuel, but the steadily increasing values of even the poorest qualities are rapidly prohibiting its use for steam-generating purposes. Sawdust, shavings, tanbark and other waste products of wood are burned under boilers in situations where such disposition nets the best financial returns. Recent progress, however, in industrial chemistry shows that ethyl and wood alcohols and other valuable by-products can be cheaply made from sawdust, shavings, slashings and similar waste material, and it is not unlikely that their use for steaming purposes will be unheard of in a comparatively few years. Table 8 gives the physical and chemical characteristics of a number of woods.

TABLE 8.

PHYSICAL AND CHEMICAL PROPERTIES OF WOODS, STRAW AND TANBARK.

(Prac. Engr. U. S., Jan., 1910.)

	Weight per Cubic Foot. Pounds.	Weight per Cord. Pounds.	Equivalent Weight of Coal. 13,500 B.T.U.	Carbon. Per Cent.	Hydrogen. Per Cent.	Oxygen. Per Cent.	Nitrogen. Per Cent.	Ash. Per Cent.	Calorific Value, B.T.U. per Pound. †	Authority.
Ash	46	3520	1420	......	......	......	......	......	5450	Hutton
Beech	43	3250	1300	49.36	6.01	42.69	0.91	1.06	5400	Sharpless
Birch	45	2880	1190	50.20	6.20	41.62	1.15	0.81	5580	Hutton
Cherry	42	3140	1260	......	......	......	......	......	5420	"
Chestnut	41	2350	940	......	......	......	......	......	5400	Sharpless
Elm	35	2350	940	......	......	......	......	......	5400	"
Hemlock	25	1220	580	......	......	......	......	......	6410	Hutton
Hickory	53	4500	1800	......	......	......	......	......	5400	Sharpless
Maple, Hard	49	3310	1340	......	......	......	......	......	5460	Hutton
Oak, Live	59	3850	1560	......	......	......	......	......	5460	"
" White	52	3850	1540	49.64	5.92	41.16	1.29	1.97	5400	Rankine
" Red	45	3310	1340	......	......	......	......	......	5460	Hutton
Pine, White	25	1920	970	......	......	......	......	......	6830	"
" Yellow	36	2130	1050	......	......	......	......	......	6660	"
Poplar	36	2130	1050	49.37	6.21	41.60	0.96	1.86	6660	"
Spruce	25	1920	970	......	......	......	......	......	6830	"
Walnut	35	3310	1340	......	......	......	......	......	5460	"
Willow	25	1920	970	49.96	5.96	39.56	0.96	3.37	6830	Rankine
Average	....	.....	......	49.70	6.06	41.30	1.05	1.80		
Straw:	6 to 8*		Water							
Wheat		.....	16.00	35.86	5.01	37.68	0.45	5.00	......	Clark
Barley		.....	15.50	36.27	5.07	38.26	0.40	4.50	......	"
Average			15.75	36.06	5.04	37.97	0.42	4.75	5155	
Tanbark Dry	....	.....	......	51.80	6.04	40.74	......	1.42	6100	Myers

* Compressed. † Green Fuel.

Wood as Fuel: Prac. Engr. U. S., Jan., 1910, p. 805; Power & Engr., June 30, 1908, p. 1015; Power, Dec., 1908, p. 772.

Burning Sawdust: Prac. Engr. U. S., Jan., 1910, p. 48; Power & Engr., April 7, 1908, p. 536; Oct. 13, 1908, p. 613; Jour. of Elec., Oct., 1905.

TABLE 9.

HEAT VALUES OF BAGASSE AND VARIATION WITH DEGREE OF EXTRACTION.

Per Cent Extraction on Weight of Cane.	Per Cent Moisture in Bagasse.	Fiber.		Sugar.		Molasses.		Total Heat Developed. B.T.U.	Heat Required to Evaporate the Water Present. B.T.U.	Heat Available. B.T.U.	Lb. Bagasse Required to Equal 1 lb. Coal of 14,000 B.T.U. Calorific Power.	Coal Equivalent per Ton of Cane. Pounds.	Temperature of Fire. Fahr.
		Per Cent in Bagasse.	Fuel Value. B.T.U.	Per Cent in Bagasse.	Fuel Value. B.T.U.	Per Cent in Bagasse.	Fuel Value, B.T.U.						
90	0.00	100.00	8325	....	....	....	....	8325	....	8325	1.68	119	2465°
85	28.33	66.67	5552	3.33	240	1.67	116	5900	339	5561	2.52	119	2236
80	42.50	50.00	4160	5.00	361	2.50	174	4697	509	4188	3.34	120	2023
75	51.00	40.00	3330	6.00	433	3.00	209	3972	611	3361	4.17	120	1862
70	56.67	33.33	2775	6.67	482	3.33	232	3489	679	2810	4.98	120	1732
65	60.71	28.57	2378	7.15	516	3.57	248	3142	727	2415	5.80	121	1612
60	63.75	25.00	2081	7.50	541	3.75	261	2883	764	2119	6.61	121	1513
55	66.12	22.22	1850	7.78	562	3.88	270	2682	792	1890	7.40	121	1427
50	68.00	20.00	1665	8.00	578	4.00	278	2521	815	1706	8.21	122	1350
45	69.55	18.18	1513	8.18	591	4.09	284	2388	833	1555	9.00	122	1284
40	70.83	16.67	1388	8.33	601	4.17	290	2279	849	1430	9.79	123	1222
25	73.67	13.33	1110	8.67	626	4.33	301	2037	883	1154	12.13	124	1077
15	75.00	11.77	980	8.82	637	4.41	307	1924	899	1025	13.66	124	1002
0	76.50	10.00	832	9.00	650	4.50	313	1795	916	879	15.93	126	906

Bagasse, or megass, is refuse sugar cane and is used as a fuel on the sugar plantations. Its heat value depends upon the proportions of fiber, molasses, sugar and water left after the extraction. The heat furnished by the different constituents is about as follows: Fiber, 8325 B.t.u. per pound; sugar, 7223 B.t.u. per pound; and molasses, 6956 B.t.u. per pound. Table 9 gives the heat value of bagasse and variation with the degree of extraction. A typical furnace for burning bagasse is shown in Fig. 108.

Bagasse as Fuel: Prac. Engr. U. S., Jan., 1910; Engng., Feb. 18, 1910.
Bagasse Drying: E. W. Kerr, Louisiana Bul. No. 128, June, 1911.

Tanbark is usually quite moist; the amount of moisture varies with the leaching process used and averages around 65 per cent. In this condition it has a heat value of about 4300 B.t.u. per pound. If perfectly dry its heating power is approximately 6100 B.t.u. per pound. As in the case of all moist fuels, tanbark must be surrounded by heated surfaces of sufficient extent to insure drying out the fresh fuel as thrown on the fire. A successful furnace for burning tanbark is shown in Fig. 109.

Tanbark as a Boiler Fuel: Jour. A.S.M.E., Feb., 1910, p. 181; Jour. A.S.M.E., Oct., 1909, p. 951; Prac. Engr. U.S., Jan., 1910.
Burning Coke Breeze: Power, July 4, 1916, p. 2.

19. Combustion. — To the engineer *combustion* means the chemical union of the combustible of a fuel and the oxygen of the air at such a rate as to cause rapid increase in temperature. The depreciation in heat value of bituminous coal subjected to "weathering" is due to combustion, but the rate at which the combustible unites with the oxygen is so slow that the heat is dissipated and there is practically no increase in temperature. When the combustible elements unite with oxygen they do so in definite proportions, which are always the same, and the union liberates a fixed quantity of heat independent of the time occupied. Theoretically combustion is a simple process as it is only necessary to bring each particle of fuel previously heated to the kindling temperature in contact with the correct amount of oxygen and the combustion will be complete, the fuel oxidizing to the highest possible degree. In practice, however, the size and character of fuel, type of furnace, draft, impurities in the fuel, and the mechanical difficulties affect combustion to such an extent as to render oxidation more or less incomplete.

When heat is applied to coal, combustion takes place in three separate and distinct stages:

1. Absorption of heat. A fresh charge of fuel when thrown on a fire must first be brought to the kindling point in order that chemical action may take place. The temperatures necessary to cause this union of oxygen and fuel are approximately as follows:

	Deg. Fahr.		Deg. Fahr.
Lignite dust	300	Cokes	800
Sulphur	470	Anthracite lump	750
Dried peat	435	Carbon monoxide	1211
Anthracite dust	570	Hydrogen	1100
Lump coal	600		

(Stromeyer, Marine Boiler Management and Construction, p. 93.)

The amount of heat required to realize the kindling temperature is greatly increased by the water content of the fuel since practically all of the free moisture must be evaporated before this temperature is reached.

2. Vaporization of the hydrocarbon portion of the fuel and its combustion, the hydrocarbons consisting principally of ethylene gas, C_2H_4, methane gas, CH_4, tar, pitch, naphtha and the like. As these gases are driven off they become mixed with the entering air, and the carbon and hydrogen unite with the oxygen, forming carbon dioxide, CO_2, and water vapor, H_2O, respectively, and give up heat in doing so. If volatile sulphur is present it unites with oxygen, forming sulphur dioxide, SO_2, and also gives up heat. If insufficient oxygen is present for complete oxidation, the carbon may burn to carbon monoxide, CO, and only a small portion of the available heat be liberated.

3. Combustion of the solid or carbonaceous portion of the fuel. After the hydrocarbons have been driven off and oxidized the remaining solid matter is composed chiefly of carbon and ash. The carbon unites with the oxygen, forming carbon dioxide, carbon monoxide, or both, depending upon the completeness of combustion. The ash, of course, remains unconsumed.

In commercial practice the requirements for perfect combustion are a surplus of air, a thorough mixture of the fuel particles with the air, and a high temperature. The surplus of air above theoretical requirements should be kept to a minimum, but even in the most scientifically designed furnace some excess is essential on account of the difficulty of properly mixing the gases, since the currents of combustible gases and air are apt to be more or less stratified. The products of combustion must be maintained at the kindling temperature until oxidation is complete, otherwise the carbon will be wasted as carbon monoxide or as smoke. The final products of combustion as exhausted by the chimney should consist only of carbon dioxide, water vapor, oxygen, nitrogen, and the oxides of impurities in the fuel.

As previously stated when the combustible elements unite with oxygen they do so in definite proportions, called the *combining weights*, which are always the same, for a given reaction, and the union produces a fixed quantity of heat. Thus in the complete combustion of carbon, 12 pounds of carbon unite with 32 pounds of oxygen, forming 44 pounds of carbon dioxide; hence, one pound of carbon will form

$$\frac{C + O_2}{C} = \frac{12 + 2 \times 16}{12} = 3\tfrac{2}{3} \text{ pounds of } CO_2$$

and the heat of combustion will be about 14,540 B.t.u. per pound of carbon thus consumed. (The heat value for carbon appears to depend upon the method of preparation and ranges according to various authorities from 14,220 to 14,647 B.t.u. per pound.)

If combustion is incomplete and the carbon burns to carbon monoxide, one pound of carbon will form

$$\frac{2\,C + O_2}{2\,C} = \frac{24 + 32}{24} = 2\tfrac{1}{3} \text{ pounds of CO and liberates 4380 B.t.u.}$$

Similarly, in burning to H_2O one pound of hydrogen will form

$$\frac{2H_2 + O_2}{2\,H_2} = \frac{2 \times 2 + 32}{2 \times 2} = 9 \text{ pounds of } H_2O.$$

(The exact figures, based upon the relative molecular weights, as adopted by the International Committee on Atomic Weights, are

$$\frac{2 \times 2.016 + 32}{2 \times 2.016} = 8.94 \text{ pounds.}$$

TABLE 10.

DATA RELATIVE TO ELEMENTS MOST COMMONLY MET WITH IN CONNECTION WITH COMBUSTION OF FUEL.

Substance.	Molecular Formula.	Relative Molecular Weight, Oxygen =32.	Chemical Reactions.	Weight per Pound of Substance in First Column.	
				Oxygen	Air.
Acetylene	C_2H_2	26.02	$2\,C_2H_2+5\,O_2=4\,CO_2+2\,H_2O$	3.08	13.35
Air	......	....	......	....	....
Ash	......	....	......	....	....
Carbon	*C	*12.0	$2\,C+O_2=2\,CO$	1.33	5.78
Carbon	*C	*12.0	$2\,C+2\,O_2=2\,CO_2$	2.66	11.58
Carbon dioxide	CO_2	44.0	......	....	....
Carbon monoxide	CO	28.0	$2\,CO+O_2=2\,CO_2$	0.57	2.47
Hydrogen	H_2	2.016	$2\,H_2=O_2=2\,H_2O$	8.0	34.8
Methane	CH_4	16.03	$CH_4+2\,O_2=CO_2+2H_2O$	4.0	17.4
Nitrogen	N_2	28.02	......	....	....
Ethylene	C_2H_4	28.03	$C_2H_4+3\,O_2=2\,CO_2+2\,H_2O$	3.43	14.9
Oxygen	O_2	32.0	......	....	....
Sulphur	*S	*32.07	$S+O_2=SO_2$	1.0	4.32
Water vapor	H_2O	18.02	......	....	....

Substance.	Mean Specific Heat.	Density and Specific Volume at 32 deg. fahr., and 14.7 Lb. per Sq. In.†		Heat of Combustion (Total Heat Value) B.t.u.‡	
		Weight per Cu. Foot.	Cu. Feet per Pound.	Per Pound.	Per Cu. Foot at 32 deg. fahr. and 14.7 Lbs.
Acetylene	See Par. 24 and Fig. 10.	0.0725	13.79	21,430	1582
Air		0.0807	12.39	......	......
Ash		......	......	......	......
Carbon		145 (solid)	......	4,380	......
Carbon		145 (solid)	......	14,540	......
Carbon dioxide		0.1227	8.15	......	......
Carbon monoxide		0.0781	12.80	4,380	342
Hydrogen		0.0056	177.9	62,000	345
Methane		0.0447	22.37	23,840	1067
Nitrogen		0.0783	12.77	......	......
Ethylene		0.0795	12.80	21,450	1685
Oxygen		0.0892	11.21	......	......
Sulphur		125 (solid)	......	4,000	......
Water vapor		......	......	......	......

* Atomic. † Smithsonian tables. ‡ Compiled from various sources.

For all practical engineering purposes the use of the exact values of the molecular weights is an unnecessary refinement and the decimal factors may well be omitted. In the ensuing calculations only the approximate values will be considered.) If the products of combustion are condensed and their temperature lowered to the initial temperature of the constituent gases the heat liberated will be 62,000 B.t.u. This is known as the *total heating* value. If the products of combustion are not condensed, which is the usual case in practice, the latent heat of vaporization of the water vapor is not available. The difference between the higher heating value and the unavailable heat is called the *net heating value.* The unavailable portion of the heat depends upon the temperature at which the products of combustion are discharged. This varies with practically every installation. Thus, one pound of water vapor escaping uncondensed in the products of combustion at temperature t_1 deg. fahr. will carry away approximately $(1090.6 + 0.46\ t_1 - t)$ B.t.u. above initial temperature t deg. fahr. of the constituent gases. (See paragraph 30.) Since one pound of hydrogen burns to approximately 9 pounds of water vapor, the lower heating value h' will be

$$h' = 62{,}000 - 9\ (1090.6 + 0.46\ t_1 - t)\ \text{B.t.u.} \tag{4}$$

Many attempts have been made to adopt a standard lower heating value, but the results have been far from harmonious. The U. S. Bureau of Standards recommends "that the quantity to be subtracted from the gross value to give the net value be taken as the latent heat of vaporization at zero degrees centigrade, of the water formed during combustion, and of that contained in the fuel."

This would give the net or lower heat value of hydrogen as

$$62{,}000 - 9\ (1073.4) = 52{,}340\ \text{B.t.u.}$$

For $t_1 = t = 0$ deg. cent. $= 32$ deg. fahr., formula (4) gives the same result.

Combustion of Bituminous Coals. — H. Kreisinger, Prac. Engr., Apr. 15, 1917, p. 347.

20. Calorific Value of Coal. — The heat liberated by the complete combustion of unit weight of fuel is called the *heating value* or *calorific value* of the fuel. The only accurate method of determining this quantity for a solid fuel is to burn a weighed sample in an atmosphere of oxygen in a suitable calorimeter. An alternative method is to calculate the heating value from the ultimate analysis. Approximate results may be obtained from empirical formulas based upon the proximate analysis.

Dulong's formula is the generally accepted rule for calculating the heating value of coal. It is based on the assumption that all the oxygen in the fuel and enough hydrogen to unite with it is inert in the form of water and that the remainder of the hydrogen and all of the carbon and sulphur are available for oxidation thus:

$$h_d = 14{,}600\,\mathrm{C} + 62{,}000\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) + 4000\,\mathrm{S}^* \tag{5}$$

in which h_d = heating value in B.t.u. per pound of fuel.

C, H, O and S refer to the proportion by weight of carbon, hydrogen, oxygen and sulphur in the fuel.

Heating values calculated by means of Dulong's formula fail to check with calorimetric determinations because

(1) The heating values of the elements, carbon, hydrogen and sulphur are not accurately established and the true values may depart somewhat from those given in the formula.

(2) The heating value of an element in the free state is not necessarily the same as when a component of a chemical compound, because of absorption or evolution of heat during formation of the compound.

(3) The oxygen content in the ultimate analysis is determined by difference. This method throws the summation of all the errors incurred in the other determinations upon the oxygen. Furthermore, the assumption that all of the oxygen is combined with hydrogen to form water is not true since some of the oxygen may be combined with carbon.

However, in spite of these objections, extensive investigations show that Dulong's formula gives results which agree substantially with calorimetric determinations for all ordinary coals. With lignite, wood and other fuels high in oxygen and with some fuels high in hydrogen such as cannel coal, the results are not reliable and may be considerably in error.

Numerous attempts have been made to establish empirical formulas for calculating the heat value from the proximate analysis but the results have been decidedly discordant. Many of these rules give consistent results when applied to certain classes of fuels or to fuels from a given district, but as general laws they may lead to serious error.

In this connection may be mentioned the investigations of Mahler,† Lord and Haas,‡ Parr and Wheeler,§ Goutal,|| and Kent.¶

* In the fuel bulletins of the U. S. Geological Survey and the Bureau of Mines, Dulong's formula is stated:

$$h_d = 14{,}544\,\mathrm{C} + 62{,}028\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) + 4050\,\mathrm{S}.$$

† Kent, Steam Boiler Economy, 1915, p. 143.

‡ Transactions of American Soc. of Mechanical Engineers, Vol. 27, 1897, p. 259.

§ Illinois University Engineering Experiment Station, Bulletin 37, 1909.

|| Comptes Rendus de L'Academie des Sciences, Vol. 135, p. 477.

¶ Transactions of American Soc. of Mechanical Engineers, Vol. 36, 1914, p. 189.

When a series of tests is being made with a view of improving efficiency it is of considerable importance to have the results of each test immediately after completion of the run in order that the information gained may be used in the succeeding tests. For this reason it is particularly desirable to determine the heating value of the coal and "cinders" with as little delay as possible. If the source of the coal supply is known the simplest, and a fairly accurate method, is to assume a fixed heat value for the combustible. This may be obtained from results of previous tests or from results published by the Bureau of Mines. For example, the average heat value of the combustible for a number of Illinois coals as compiled from Government reports and other sources, is 14,300 B.t.u. per pound. With the exception of a very few samples the actual heating value varied less than 2 per cent from this average and the maximum departure did not exceed 3 per cent. Extensive experiments conducted in the power plant laboratory of Armour & Company, Chicago, Illinois, show that the heat value of the combustible in the refuse or clinkers is practically that of the combustible in the fuel, averaging 14,100 B.t.u. per pound for Illinois coals.

The heating value of any fuel may be determined from the proximate analysis with a fair degree of accuracy by calculating the ultimate analysis, as shown in the preceding paragraphs, and applying Dulong's formula.

Calorimetric determinations are necessary in all cases where accuracy is required.

Example 5. Approximate the heat values for the Illinois coal (analysis as in Example 1) from the calculated ultimate analysis.

	B.t.u. per Pound of Coal as Received.	Departure from Calorimeter Determinations, Per Cent.
1. Assuming a fixed heat value for the combustible $h = 14{,}300 \times 0.8163$	11,674	−2.36
2. Calculated from Dulong's formula:		
* (*a*) $h = 14{,}600 \times 0.65 + 62{,}000 \times 0.0326 + 4000 \times 0.028$	11,623	−1.96
* (*b*) $h = 14{,}600 \times 0.682 + 62{,}000 \left(0.0435 - \frac{0.0775}{8}\right)$	12,053	+0.80
* (*c*) $h = 14.600 \times 0.6655 + 62{,}000 \left(0.0428 - \frac{0.0751}{8}\right) + 4000 \times 0.0197$	11,869	−0.76
3. Actual value from calorimeter test	11,957	0.00

* (*a*) Ultimate analysis calculated from average analysis of Illinois coals. See Example 4.
(*b*) Ultimate analysis calculated from proximate analysis (Equations (1) to (3)).
(*c*) Ultimate analysis from chemical tests.

TABLE 11.

VARIATION IN CALORIFIC VALUE OF FUELS.

(As Mined.)

	B.t.u.
Air-dried wood	6,000 to 7,500
Air-dried peat	About 7,500
Lignite	5,200 to 7,500
Sub-bituminous coal	5,500 to 11,500
Bituminous coal	10,000 to 14,500
Semi-bituminous coal	13,500 to 14,900
Anthracite coal	11,000 to 13,800
California crude oil	17,000 to 19,300
Pennsylvania heavy crude oil	About 20,700

21. Air Theoretically Required for Complete Combustion. — The combustible portion of all commercial fuels consists chiefly of carbon and hydrogen and a small percentage of volatile sulphur. Based upon the approximate molecular weights the carbon, hydrogen and sulphur require the following weights of oxygen for complete combustion:

1 lb. carbon requires $\frac{O_2}{C} = \frac{32}{12} = 2.66 +$ lb. oxygen.

1 lb. hydrogen requires $\frac{O_2}{2\,H_2} = \frac{32}{4} = 8.00$ lb. oxygen.

1 lb. sulphur requires $\frac{O_2}{S} = \frac{32}{32} = 1.00$ lb. oxygen.

In the ordinary furnace the oxygen is obtained from the atmosphere which, neglecting moisture and a few minor elements, contains oxygen and nitrogen mechanically mixed as follows:

PROPORTION OF NITROGEN AND OXYGEN IN DRY ATMOSPHERIC AIR.

	Exact Value.		Approximate Value.	
	By Volume.	By Weight.	By Volume.	By Weight.
Nitrogen	79.09	76.85	79.0	77.0
Oxygen	20.91	23.15	21.0	23.0
$N \div O$	3.782	3.32	3.76	3.34
$(N + O) \div O$	4.782	4.32	4.76	4.35

Hence the dry air requirements are:

1 lb. of carbon requires $2.66 \times 4.35 = 11.58$ lb. dry air.
1 lb. of hydrogen requires $8.00 \times 4.35 = 34.8$ lb. dry air.
1 lb. of sulphur requires $1.00 \times 4.35 = 4.35$ lb. dry air and for a compound fuel

$$A_1 = 11.58\ C + 34.8\left(H - \frac{O}{8}\right) + 4.35\ S, \tag{5}$$

in which

A_1 = weight of dry air required.
C, H, O, and S = proportional part of the carbon, hydrogen, oxygen and volatile sulphur in the fuel.
$\frac{O}{8}$ = proportional part of the hydrogen supplied with oxygen from the fuel itself.*

It should be borne in mind that these values are based on the approximate molecular weights of the various elements and the assumption that the air is composed of 23 parts oxygen and 77 parts of nitrogen, by weight. Using the exact molecular weights, as fixed by the International Committee on Atomic Weights, and taking the air as composed of 23.15 per cent oxygen and 76.85 per cent nitrogen, equation (5) becomes

$$A_1 = 11.5\ C + 34.2\left(H - \frac{O}{8}\right) + 4.3\ S. \tag{6}$$

In using equation (6) in connection with the determination of heat losses, to be consistent, all calculations should be made with the exact molecular weights and the true ratio of nitrogen to oxygen in atmospheric air. The theoretical weights of air as calculated from equations (5) and (6) differ by approximately one per cent as a maximum.

Example 6. Required the theoretical weight of dry air supplied per pound of coal as fired with analysis as follows:

	Per Cent.		Per Cent.
Carbon	65	Ash and Sulphur	13
Hydrogen	5	Water	8
Oxygen	8	Total	100
Nitrogen	1		

* This term $\left(H - \frac{O}{8}\right)$ does not contain a proper correction for the hydrogen contained in the moisture, for not all of the oxygen in coal is combined with hydrogen. Part of the oxygen is probably combined with nitrogen in organic nitrates and part may be present in carbonates in mineral matter caught in the coal. The error of this assumption, however, lies within the accuracy of the average boiler observations.

Substituting the value of C, H, and O in equation (5)

$$A_1 = 11.58 \times 0.65 + 34.8\left(0.05 - \frac{0.08}{8}\right) = 8.92 \text{ pounds},$$

the theoretical weight of dry air necessary to burn one pound of coal *as fired.*

Since the coal contains 8 per cent of moisture the weight of dry air required per pound of *dry coal* is

$$\frac{8.92}{0.92} = 9.69 \text{ pounds.}$$

The water and ash only are treated as incombustible, therefore the air required per pound of *combustible* is

$$\frac{8.92}{0.79} = 11.29 \text{ pounds.}$$

Similar calculations for different fuels will show that the theoretical air requirements per pound of fuel or combustible vary within wide limits. When expressed in terms of theoretical air requirements per 10,000 B.t.u., however, there is a close agreement between all fuels. Several hundred fuels ranging from peat to crude oil rated on this basis gave an average value of 7.5 pounds of air per 10,000 B.t.u. with a maximum departure not exceeding 2 per cent. The calorific value of the coal in the preceding example is 15,150 B.t.u. per pound of combustible; on the B.t.u. basis this gives

$$\frac{15{,}150}{10{,}000} \times 7.5 = 11.35 \text{ pounds,}$$

which checks substantially with the calculated value. See also Table 13.

22. Products of Combustion. — A knowledge of the constituents of the solid and gaseous products resulting from the combustion of a fuel offers a means of determining the losses incident to such combustion. For maximum efficiency complete combustion with theoretical air requirements is necessary and the resulting products should consist only of CO_2, N_2, H_2O, ash, and the oxides of other combustible elements in the fuel. The dry gaseous products, such as appear in the commercial flue gas analysis, will consist of CO_2 and N_2 only, since the SO_2, if there is any, is partly absorbed by the water in the sampling apparatus (see paragraph 415) while some of it probably goes into the CO_2 pipette and appears in the analysis as CO_2. It is difficult to determine the exact distribution but since the maximum error due to this source does not exceed 0.2 per cent it is common practice to disregard the SO_2 entirely. If combustion is complete but air is used in excess of theoretical requirements the gaseous products will include free oxygen. If combustion is incomplete CO will also be present in the gaseous products and perhaps small quantities of hydrocarbons. The following examples illustrate some of the accepted methods for determining the constituents of the products of combustion.

Example 7. Required the character and amount of the products of combustion if one pound of coal, as per following ultimate analysis, is completely burned with theoretical air requirements.

Carbon	65	Ash	12
Hydrogen	5	Water	8
Oxygen	8	Sulphur	1
Nitrogen	1	Total	100

TABULATED CALCULATIONS.

	Pounds of Substance per Pound of Coal as Fired.						
	C_2	H_2	O_2	N_2	CO_2	H_2O	Ash.
The carbon will produce:							
Carbon	0.65						
$0.65 \times \frac{44}{12}$					2.38		
$0.65 \times \frac{32}{12}$			1.73				
$0.65 \times \frac{32}{12} \times \frac{77}{23}$				5.80			
The available hydrogen will produce:							
Hydrogen		0.04					
$\left(0.05 - \frac{0.08}{8}\right) 9$						0.36	
$\left(0.05 - \frac{0.08}{8}\right) 8$			0.32				
$\left(0.05 - \frac{0.08}{8}\right) 8 \times \frac{77}{23}$				1.07			
The oxygen and inert hydrogen will produce:							
Hydrogen		0.01					
Oxygen			0.08				
$0.08 + \frac{0.08}{8}$						0.09	
The nitrogen in the fuel* is considered inert				0.01			
The moisture will appear as vapor						0.08	
Ash plus sulphur †							0.13
Total	0.65	0.05	2.13	6.88	2.38	0.53	0.13

* This is not strictly true since a portion of the nitrogen content of the fuel appears in the flue gas in combination with other elements, but the amount is so small compared with that supplied in the air that no appreciable error arises from the assumption that it remains inert and passes through the furnace without change.

† The sulphur content is ordinarily so small that no attempt is made to separate the volatile and non-volatile constituents and the whole is treated as ash. If the volatile portion is to be considered the influence of the SO_2 or SO_3 in the flue gas should be included in the heat balance. Some engineers treat one-half the sulphur as volatile and the balance as ash.

$$\text{Total gaseous products} = CO_2 + N_2 + \text{total } H_2O$$
$$= 2.38 + 6.88 + 0.53 = 9.79 \text{ lb.}$$

Or, separating the compounds into their elementary constituents,

$$\text{Total gaseous products} = C + H_2 + O_2 + N_2 + \text{free } H_2O$$
$$= 0.65 + 0.05 + 2.13 + 6.88 + 0.08$$
$$= 9.79 \text{ lb.}$$

Total dry gaseous products = 9.79 − 0.53 (total H_2O) = 9.26 lb.

$$\text{Dry air} = 9.26 - C + 8\left(H - \frac{O}{8}\right) - N_2 \text{ (in the fuel)}$$

$$= 9.26 - 0.65 + 8\left(0.05 - \frac{0.08}{8}\right) - 0.01 = 8.92 \text{ lb.},$$

which checks with the results as calculated from equation (5). Since the dry air consists of all the nitrogen supplied by the air and the oxygen for the combustion of the carbon and hydrogen we have as an additional check,

$$\text{Dry air} = 6.87 + 0.65 \times \frac{32}{12} + 8\left(0.05 - \frac{0.08}{8}\right)$$

$$= 6.87 + 1.73 + 0.32 = 8.92 \text{ lb.}$$

If the coal in the preceding problem is completely burned with $33\frac{1}{3}$ per cent air excess the products of combustion will be the same as before with the exception of the addition of $\frac{1}{3} \times 8.92 = 2.97$ lb. dry air. The gases, by weight, will consist of CO_2 = 2.38 lb., $N_2 = 1\frac{1}{3} \times 6.87 + 0.01 = 9.17$ lb., free $O_2 = \frac{1}{3} \times 2.05 = 0.68$ lb., H_2 = 0.53 lb. or a total of 12.76 lb. Weight of dry gases per lb. of coal = 12.76 − 0.53 = 12.23 lb.

The free oxygen comes from the air supplied and not used. This oxygen is accompanied by $\frac{79.01}{21.09} = 3.78$ times its volume of nitrogen. (N − 3.78 O) represents the nitrogen content of the air actually required for the combustion represented by the flue gas analysis. Hence $\frac{N}{N - 3.78\,O}$ is the ratio of the air supplied to that theoretically required to burn the coal to CO and CO_2. For the example under consideration

$$\frac{N}{N - 3.78\,O} = \frac{81.2}{81.2 - 3.78 \times 5.4} = 1.335.$$

100 × 1.335 − 100 = 33.5 per cent = air excess, which agrees substantially with results as previously determined.

If all the carbon had burned to CO_2 the ratio of the total air supply to that theoretically required for complete combustion is

$$\frac{N}{N - 3.78\left(O - \frac{1}{2}CO\right)}.$$

N in this case represents the nitrogen incident to the complete combustion of the carbon; $(O - \frac{1}{2}CO)$ represents the equivalent volume of oxygen due to air excess since carbon combines with one volume of oxygen to form two volumes of CO.

It will be noted that *dry air* only has been considered in the foregoing calculations. *Atmospheric air* is never dry, hence the weight of volume of atmospheric air will differ from the amounts as calculated above. For most engineering purposes atmospheric air may be considered dry.

For methods of determining the weight of dry air in atmospheric air see paragraph 470.

In the preceding calculation the products of combustion have been expressed on a weight basis, which, as will be shown later, is most convenient for calculating the various heat losses. However, in determining the gaseous constituents of the products of combustion the measurements are made volumetrically. The transfer from one basis to the other is readily effected by the following adaptation of Avogadro's law: *

$$\text{Lb. per cu. ft. of any gas} = 0.00278\, m. \tag{7}$$

$$\text{Conversely, cu. ft. per lb. of any gas} = 358.6 \div m, \tag{8}$$

in which

m = molecular weight of the gas referred to oxygen as 32. Volumes measured at 32 deg. fahr. and atmospheric pressure, 29.92 inches of mercury.

Thus, the volume of one pound of CO_2 at 32 deg. fahr. and 29.92 inches of mercury = V = 358.6 ÷ 44 = 8.15 cu. ft.

Similarly the volumes of one pound of oxygen and nitrogen are 11.21 and 12.77 cu. ft. respectively.

Example 8. Transfer the flue gases in Example 7 from a weight to a volume basis.

In Example 7 it was shown that for complete combustion with theoretical air requirements the dry gaseous products of combustion consisted of 2.38 lb. CO_2 and 6.88 lb. N_2.

$$\text{Per cent } CO_2 \text{ by vol.} = 100 \times \frac{2.38 \div 44}{2.38 \div 44 + 6.88 \div 28} = 18.1.$$

For complete combustion with $33\frac{1}{3}$ per cent air excess the dry gaseous products consisted of 2.38 lb. CO_2, 9.16 lb. N_2, and 0.69 lb. free O_2. Transferring to the volumetric basis:

$$\text{Per cent } CO_2 \text{ by vol.} = \frac{2.38 \div 44}{2.38 \div 44 + 9.16 \div 28 + 0.69 \div 32} \times 100$$

$$= 100 \times \frac{0.054}{0.054 + 0.327 + 0.022} = \frac{5.4}{0.403} = 13.4.$$

$$\text{Per cent } N_2 \text{ by vol.} = \frac{32.7}{0.403} = 81.1.$$

$$\text{Per cent free } O_2 \text{ by vol.} = \frac{2.2}{0.054} = 5.5.$$

* Equal volumes of all gases contain the same number of molecules when at the same temperature and pressure.

When chemical reactions are expressed in terms of molecules the coefficients of the molecule symbols represent relative volumes, thus, the reaction $C + O_2 = CO_2$, shows that one volume of oxygen combined with carbon forms one volume of CO_2, both being measured at the same temperature and pressure. Therefore, the volume of CO_2 after combustion is precisely the same as that of the oxygen before it was combined with the carbon. The volume of one pound of CO_2 as determined above is 8.15 cubic feet and since one pound of carbon unites with $2\frac{2}{3}$ pounds of oxygen to form $3\frac{2}{3}$ pounds of CO_2 we have

Substance.	Weight per Lb. of Carbon, Lb.	Spec. Volume, Cu. Ft. per Lb.	Actual Volume Resulting from the Combustion of 1 Lb. of Carbon, Cu. Ft. *A.*	Per Cent by Volume. *B.*
For Theoretical Air Requirement.				
CO_2	$3\frac{2}{3}$ ×	8.15 =	29.89	20.91
Free O	0			
N	8.85 ×	12.77 =	113.01	79.09
Total			142.90	100.00
For 50 Per Cent Air Excess.				
CO_2	$3\frac{2}{3}$ ×	8.15 =	29.89	13.91 } 20.91
Free O	0.5 × $2\frac{2}{3}$ ×	11.21 =	14.94	7.00 } 20.91
N	1.5 × 8.85 ×	12.77 =	169.56	79.09
Total			214.39	100.00
For 100 Per Cent Air Excess.				
CO_2	$3\frac{2}{3}$ ×	8.15 =	29.89	10.45+ } 20.91
Free O	$2\frac{2}{3}$ ×	11.21 =	29.89	10.45+ } 20.91
N	2 × 8.85 ×	12.77 =	226.08	79.09
Total			285.86	100.00

It will be noted that the actual volume of CO_2 is always the same irrespective of the excess of air supplied, while the percentage by volume decreases as the excess of air increases. In each case $CO_2 + O = 20.9$ is constant. (The approximate value 21 is ordinarily taken instead of the exact quantity, 20.9.)

The actual volume of oxygen and the percentage by volume increase with the amount of excess air, therefore either the CO_2 or O content of the products of combustion is a true index of the air excess. This applies to the complete combustion of pure carbon only. Assuming an average theoretical air requirement of 11.5 pounds of air for the complete combustion of pure carbon the resulting air requirements for different percentage of CO_2 are given in Table 12. Although the actual volume of nitrogen increases with the air excess its volume percentage remains the same after combustion as before. The nitrogen performs no useful function in combustion and passes through the furnace without change. It simply dilutes the oxygen for combustion and its presence in the flue gases represents a large percentage of the heat lost in the chimney.

CO produced by incomplete combustion of carbon will occupy twice the volume of oxygen entering into its composition as is evidenced from the molecular reaction $2C + O_2 = 2CO$.

Therefore, with pure carbon as fuel, the sum of the percentages by volume of CO_2, O_2 and $\frac{1}{2}$ CO must be in the same ratio to the nitrogen in the flue gas as is oxygen to the nitrogen in the air supplied; viz., 20.91 to 79.09. When burning coal, however, the percentage of nitrogen is obtained by subtracting the sum of the percentages by volume of the other gases from 100.

In commercial furnace practice CO_2 is used as the index to efficiency of combustion because of the ease with which it is obtained. For fuels high in volatile matter the per cent of CO_2 in the products of combustion is less than 20.91 for complete combustion, since the oxygen which combines with hydrogen to form H_2O does not appear in the sample as ordinarily tested: thus for heavy crude oil the corresponding maximum content of CO_2 is approximately 16 per cent. The air requirements and resulting CO_2 content for complete combustion of a number of typical fuels are given in Table 13.

TABLE 12.

WEIGHT OF AIR PER POUND OF CARBON AS INDICATED BY THE PERCENTAGE OF CO_2 IN THE FLUE GAS.

Per Cent of CO_2.	Pounds of Air.	Per Cent of CO_2.	Pounds of Air.	Per Cent of CO_2.	Pounds of Air.
20.9	11.5	14	17.1	7	34.3
20	12.0	13	18.5	6	40.0
19	12.6	12	20.0	5	48.0
18	13.3	11	21.8	4	60.0
17	14.1	10	24.0	3	80.0
16	15.0	9	26.7	2	120.0
15	16.0	8	30.0	1	240

TABLE 13.

THEORETICAL AIR REQUIREMENTS FOR VARIOUS FUELS AND THE RESULTING MAXIMUM PER CENT CO_2 IN THE FLUE GAS FOR COMPLETE COMBUSTION.

Fuel, Moisture and Ash Free.	Ultimate Analysis.*					Air, Pounds.		CO_2, Per Cent by Volume.
	C.	H.	N.	O.	S.	Per Pound of Fuel.	Per 10,000 B.t.u.†	
Pure carbon	100.00	......	......	......	......	11.58	....	20.91
Anthracite	94.39	1.77	0.71	2.13	1.00	11.39	7.4	20.06
Semi-anthracite	89.64	3.97	0.63	3.23	2.53	11.59	7.5	20.00
Semi-bituminous	86.39	4.84	1.46	5.50	1.81	11.41	7.6	18.65
Bituminous	79.71	5.52	1.52	9.87	3.38	10.70	7.4	18.46
Sub-bituminous	78.06	5.70	1.35	13.10	1.79	10.24	7.3	18.56
Lignite	70.64	4.61	1.22	22.67	0.86	8.75	7.3	19.68
Peat	59.42	5.50	1.50	33.33	0.25	7.30	7.6	20.79
Crude oil	84.90	13.7	0.60	0.80	......	14.45	7.5	15.90

* Compiled from Bulletins No. 22 and No. 85, U. S. Bureau of Mines.

† 200 samples of various fuels gave an average theoretical air requirement of 7.5 pounds per 10,000 B.t.u. (Bomb calorimeter). The maximum variation did not depart more than 2 per cent from the average value.

In coal-burning practice, from 15 to 16 per cent of CO_2 is all that can be expected under the very best conditions, with an average range for general practice between 10 per cent and 14 per cent. Anything less than 12 per cent shows an excessive amount of air supplied. Traveling grates, unless carefully operated, are apt to show as low as 5 per cent of CO_2.

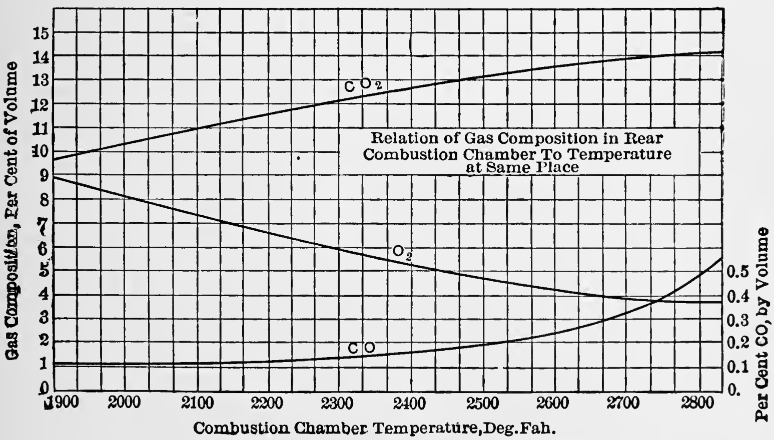


FIG. 8. Relation of Gas Composition in Combustion Chamber to Temperature.

It must not be assumed that a high percentage of CO_2 in the flue gas is necessarily a true indication of good combustion and hence of high efficiency. As the percentage of CO_2 increases there is a tendency for the CO to increase also (see Fig. 8) and the thermal gain due to minimum air excess as indicated by the high percentage of CO_2 may be more than offset by the loss due to incomplete combustion (see paragraph 27). Determinations of the CO content of the flue gas are necessary for an accurate heat balance and particularly so if the CO_2 content is high.

23. Air Actually Supplied for Combustion. — In practice the amount of air supplied is measured directly in situations where such measurements can be readily made, as in connection with mechanical draft, or where the entire air supply is forced to flow through a conduit. In most cases, however, physical measurements of flow are not feasible and the amount of air supplied is calculated from the flue-gas analysis.* The latter offers a fairly accurate method for determining air excess, provided the sample of gas is truly representative of average conditions.

Based upon the ratio of the combining weights, the weight of carbon in $CO_2 = \frac{3}{11} CO_2$, and that in $CO = \frac{3}{7} CO$. If $CO_2 + O + CO + N =$ total gas in percentage by weight the weight A_3 of the dry gas per pound of carbon actually burned is

$$A_3 = \frac{CO_2 + O + CO + N}{\frac{3}{11}CO_2 + \frac{3}{7} CO}. \tag{9}$$

Multiplying each gas by its respective molecular weight, viz., $CO_2 = 44$, $O = 32$, CO and $N = 28$, and reducing, we have

$$A_3 = \frac{11\, CO_2 + 8O + 7\,(CO + N)}{3\,(CO_2 + CO)}, \tag{10}$$

in which

A_3 = weight of dry gas per pound of carbon actually burned.

CO_2, CO, O and N = percentages by volume of the carbon dioxide, carbon monoxide, oxygen and nitrogen in the flue gas.

Since $CO_2 + CO + O + N = 100$, neglecting traces of minor constituents, $CO = 100 - CO_2 - O - N$. Substituting this value of CO in equation (10) and reducing, we have

$$A_3 = \frac{4\, CO_2 + O + 700}{3\,(CO_2 + CO)}. \tag{11}$$

Example 9. Determine the weight of dry air supplied per pound of coal as fired, analysis as in paragraph 22, if the flue gas resulting from the combustion is composed of

CO_2	12.8 per cent by volume.
CO	0.6 per cent by volume.
O_2	5.4 per cent by volume.
N_2	81.2 per cent by volume (by difference).

* For Flue-Gas see Par. 415.

Substituting the various percentages in equation (11)

$$A_3 = \frac{4 \times 12.8 + 5.4 + 700}{3\,(12.8 + 0.6)} = 18.82 \text{ lb. of dry gas per lb. of carbon}$$

actually burned.

Since the coal as fired contains 0.65 carbon, the dry gas per lb. of coal burned = 18.82 × 0.65 = 12.23 lb. If part of the coal falls through the grate, as is always the case in practice, the weight of carbon actually burned should be taken instead of the total carbon content.

The total weight of dry air actually supplied per pound of coal burned is

$$12.23 - 0.65 + 8\left(0.05 - \frac{0.08}{8}\right) = 11.90.$$

It has been previously shown (paragraph 21) that the coal under consideration requires 8.92 pounds of air for theoretical combustion, hence

$$\text{Air excess} = 100\,\frac{11.90 - 8.92}{8.92} = 33.4 \text{ per cent.}$$

The 7 N in equation (10) represents the N supplied by the air less the negligible amount furnished by the fuel itself. Since the nitrogen content of air is 77 per cent of the weight of the air, we have

$$A_4 = \frac{7\,N}{3\,(CO_2 + CO)} \div 0.77 = \frac{3.03\,N}{CO_2 + CO}, \qquad (12)$$

in which

A_4 = the weight of dry air supplied per pound of carbon burned.

N, CO_2, CO = percentages by volume of nitrogen, carbon dioxide and carbon monoxide in the flue gas.

For the example cited above

$$A_4 = \frac{3.03 \times 81.2}{12.8 + 0.6} = 18.36 \text{ pounds.}$$

For the coal under consideration

$$\text{Dry air per pound} = 0.65 \times 18.36 = 11.93.$$

This checks practically with results calculated from equation (11).

The relation between excess air and CO_2 in the flue gases for a specific case is illustrated in Fig. 9. These results were obtained from a 508 horsepower Babcock & Wilcox boiler equipped with chain grate and burning Illinois coal. (University of Illinois Bul. 32, April 12, 1915.)

Air Excess in Boiler Furnace Practice: National Engr., Feb., 1915, p. 90. The *Importance of CO_2, as an Index to Combustion and in Connection with Flue Gas Temperature, to Boiler Efficiency:* Trans. A.S.M.E., 32–1215. *Flue Gas Analysis and Calculations:* Power, Aug. 9, 1910; Eng. Review, Aug., 1910. *Real Relation of CO_2 to Chimney Losses:* Power, Dec. 7, 1909, p. 969. *Sampling and Analysis of Furnace Gas:* Power, Aug. 22, 1911, p. 282; Bulletin No. 97, U. S. Bureau of Mines.

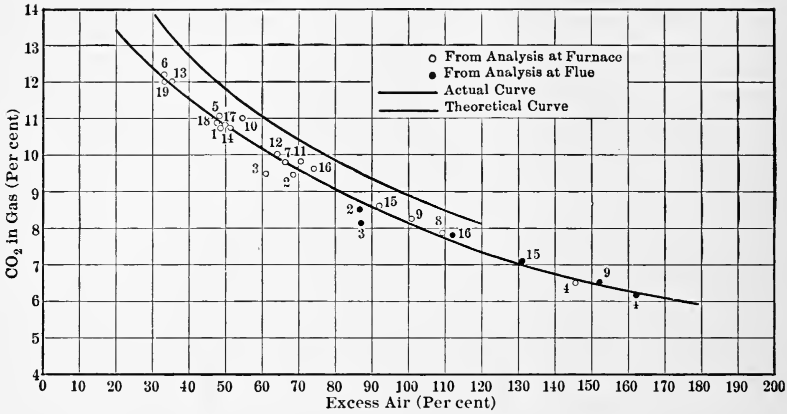


FIG. 9. Relation between Excess Air and CO_2 in Flue Gases.

24. Temperature of Combustion. — The *actual* temperature incident to the combustion of a fuel is most satisfactorily determined by means of a suitable thermometer or pyrometer. The *theoretical* temperature of combustion may be calculated from the simple relationship

$$t_1 = \frac{h}{ws} + t, \tag{13}$$

in which

t_1 = final temperature of the products of combustion, deg. fahr.
h = low calorific value of the fuel, B.t.u. per pound.
s = mean specific heat of the products of combustion.
w = weight of the products of combustion, pounds per pound of fuel.
t = initial temperature of the fuel and air supply, deg. fahr.

Thus, in the combustion of one pound of carbon with theoretical air requirements, initial temperature 62 deg. fahr., the maximum theoretical temperature will be

$$t_1 = \frac{14,540}{12.58 \times 0.29} + 62 = 4000 \text{ deg. fahr. (approx.).}$$

No such temperature has ever been obtained in practice from the combustion of carbon in air. The discrepancy between actual and calculated results is attributed to (1) difficulty of effecting complete combustion with theoretical air supply, (2) radiation losses, (3) error in the assumed value of the mean specific heat at this high temperature, and (4) uncertainty of the proportion of the calorific value of the fuel available, at this high temperature, for increasing the temperature of the products of combustion.

An inspection of equation (13) will show that the greater the weight of the products of combustion for a given weight of fuel, the lower will be the temperature of combustion. Evidently, for maximum temperature the weight of air supplied per pound of fuel should be kept as low as possible, consistent with complete combustion. A perfect union of fuel and air in theoretical proportions is almost impossible, and to insure complete combustion an excess of air is necessary. The influence of air dilution on temperature of combustion is best illustrated by a practical example:

Example 10. Required the theoretical temperature of combustion of carbon in air if 50 per cent air excess is necessary for complete combustion. Since the complete oxidation of one pound of carbon requires 11.58 pounds of air, the weight of the products of combustion will be $11.58 + 0.5 \times 11.58 + 1 = 18.37$ pounds and the final increase in temperature will be

$$t_1 = \frac{14{,}540}{18.37 \times 0.27} = 3000 \text{ deg. fahr. (approx.).}$$

Data relative to the specific heats of gases are rather discordant. The following equations are considered by the U. S. Bureau of Standards to be as nearly accurate as it is possible to give at the present time (1916).

$$\text{For } N_2 \quad s = 0.249 + 0.000'019\,t \tag{14}$$
$$CO \quad s = 0.250 + 0.000'019\,t \tag{15}$$
$$O_2 \quad s = 0.218 + 0.000'017\,t \tag{16}$$
$$H_2 \quad s = 3.40 \ \ + 0.000'27\,t \tag{17}$$
$$\text{Air} \quad s = 0.241 + 0.000'019\,t \tag{18}$$
$$CO_2 \quad s = 0.210 + 0.000'0742\,t - 0.000'000'018\,t^2 \tag{19}$$

in which

s = mean specific heat at constant atmospheric pressure and temperature range 0 deg. cent. to t.
t = maximum temperature.

For the mean specific heat of H_2O vapor see paragraph 449.

Between 1000 deg. cent. and 1500 deg. cent. the results are uncertain and dependence can be placed in only the first two significant figures in the decimal. Beyond 1500 deg. cent. the results are purely conjectural since experiments have not been made at these high temperatures. The values of the mean specific heat ($s = 0.29$ and $s = 0.27$) used in the preceding computations were calculated from these equations. The value $s = 0.27$ is probably not far from the truth, but the value $s = 0.29$ may be considerably in error.

The mean specific heat between any two temperatures t_1 and t may be determined by substituting $(t_1 + t)$ for t in above equations.

If the mean specific heats, $s_1, s_2 \ldots s_n$, and weights, $w_1, w_2 \ldots w_n$, of the constituent gases of a compound are known the mean specific heat, s, of the compound may be determined as follows:

$$s = \frac{w_1 s_1 + w_2 s_2 \cdots + w_n s_n}{w_1 + w_2 \cdots + w_n}. \tag{20}$$

The application of formulas (14)–(19) at high temperatures to equation (13) necessitates laborious calculations, and since the results are only approximate at the best, extreme refinement in calculation is without purpose. The curves in Fig. 10 are plotted from these equations and afford a means of approximating the mean specific heat without the labor of solving the equations.

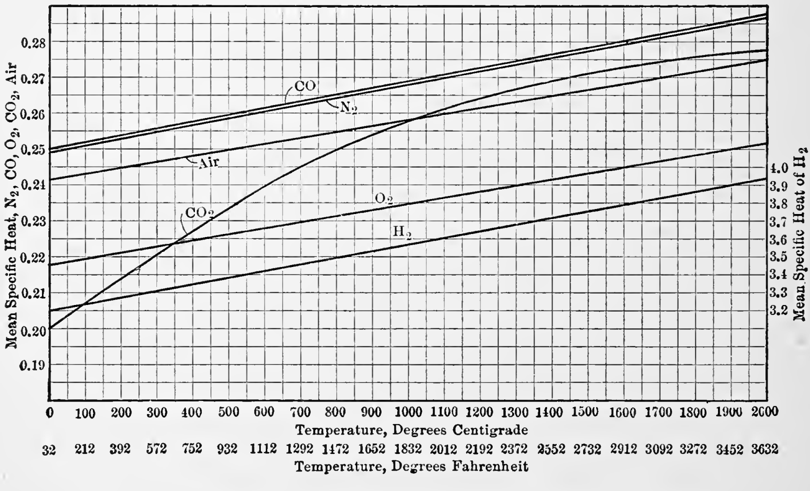


FIG. 10. Mean Specific Heats of Gases at Constant Pressure.

25. Heat Losses in Burning Coal. — A boiler in order to entirely utilize the heat of combustion of the fuel must be free from radiation and leakage losses, the fuel must be completely oxidized and the products of combustion must be discharged at atmospheric temperature. Commercially such conditions are unobtainable, hence complete utilization of the heat generated is impossible. A boiler which utilizes 83 per cent of the heat value of the fuel is exceptional and an average figure for *very* good practice is not far from 77 per cent. The various losses including the heat utilized by the boiler constitute the commercial "heat balance." The losses considered are:

1. Loss in the dry chimney gases.
2. Loss due to incomplete combustion.
3. Loss of fuel through the grate.
4. Superheating the hygroscopic moisture in the air.
5. Moisture in the fuel.
6. Loss due to the presence of hydrogen in the fuel.
7. Unburned fuel carried beyond the combustion chamber in the form of soot or smoke.
8. Radiation and minor losses.

Some of these losses are preventable. Others are inherent and cannot be avoided.

26. Loss in the Dry Chimney Gases. — This loss depends upon the type and proportion of the boiler and setting and upon the rate of driving. It is usually the greatest of all the losses. The heat carried away may be expressed:

$$h_1 = W\,(t_c - t)c, \tag{21}$$

in which

h_1 = B.t.u. lost per pound of fuel.
W = weight of dry chimney gases per pound of fuel. (See equation 10.)
t_c = temperature of the escaping gases, deg. fahr.
t = temperature of the air entering the furnace.
c = mean specific heat of the dry gases. (This may be taken as 0.24 for most purposes.)

It will be noted that the magnitude of this loss depends chiefly upon the air dilution and the temperature at which the gases are discharged. Flue temperatures less than 450 deg. fahr. are seldom experienced except in connection with economizers, and the air dilution is ordinarily in excess of 50 per cent of theoretical requirements, hence the loss from this cause may range from 8 per cent to 40 per cent of the total heat generated. In excellent practice it is not far from 12 per cent with a general average of from 20 to 25 per cent. In exceptional cases a loss from this cause as low as 9 per cent has been recorded. (Jour. A.S.M.E., Nov., 1911, p. 1463.)

Table 14 indicates the magnitude of the losses for different chimney temperatures and weights of air per pound of carbon.

TABLE 14.

HEAT CARRIED AWAY BY THE DRY CHIMNEY GASES PER POUND OF CARBON

Pounds of Air per Pound of Carbon.	Temperature of Chimney Gases. Deg. Fahr.							
	300°	350°	400°	450°	500°	550°	600°	650°
12 *	750 5.2	905 6.2	1060 7.3	1216 8.7	1370 9.5	1528 10.5	1683 11.6	1840 12.7
15	865 6	1112 7.6	1305 9.1	1498 10.3	1679 11.6	1880 13.0	2072 14.3	2262 15.6
18	1004 7.2	1321 9.1	1550 10.7	1778 12.2	2010 13.9	2235 15.4	2460 17	2692 17.9
21	1266 8.7	1530 10.5	1785 12.3	2060 14.2	2320 16	2582 17.8	2846 19.5	3118 21
24	1440 9.9	1740 12	2040 14	2340 16.1	2640 18.2	2940 20.3	3240 22.4	3540 24.4
27	1611 11.1	1950 13.5	2281 15.7	2620 18.1	2958 20.4	3291 22.7	3628 25	3962 27.4
30	1785 12.4	2160 14.9	2530 17.4	2900 20	3270 22.6	3641 25	4016 27.8	4396 30.4
33	1957 13.5	2362 16.3	2779 19.2	3180 22	3589 24.7	4000 27.6	4405 30.5	4820 33.2
36	2130 14.7	2579 17.8	3020 20.8	3461 23.9	3910 27	4350 30	4798 33	5290 36.6
39	2300 15.9	2781 19.2	3261 22.5	3743 25.8	4220 29.2	4700 32.4	5180 35.7	5670 39
42	2479 17.1	2999 20.6	3508 24.7	4023 27.7	4540 31.3	5052 34.8	5570 39.4	6100 42

* Assumed theoretical requirement.

Large type gives the loss in B.t.u. per pound of carbon.

Small type gives the per cent loss, assuming a calorific value of 14,540 B.t.u. per pound of carbon.

27. Loss Due to Incomplete Combustion. — If the volatile gases are not completely oxidized, as when the air supply is insufficient or the mixture of air and gases is not thorough, some of the carbon may escape as CO. Some of the hydrocarbons may also pass through the furnace without being burned. (See Table 15.) The presence of even a small amount of CO in the flue gas is indicative of a very appreciable loss,

TABLE 15.

ANALYSIS OF CHIMNEY GASES.

(Report of Committee for Testing Smoke-preventing Appliances, Manchester, England, 1905.)

Boiler.	Smoky.						Clear.					
	CO_2	O_2	CO	CH_4	H_2	N_2	CO_2	O_2	CO	CH_4	H_2	N_2
No. 1, hand fired	11.00 10.65	6.90 6.45	0.90 2.15	......	......	81.20 80.75	......	......	......	......	......	......
No. 1, with smoke-prevention device	......	......	......	......	......	......	7.00 9.00	13.50 9.75	0 0	......	......	79.50 81.25
No. 2, hand fired	10.25	8.60	.50	0	0	80.65	......	......	......	......	......	......
No. 3, hand fired	13.25	3.50	.05	0.25	0	82.95	......	......	......	......	......	......
No. 4, fire under caustic pot, hand fired	10.95	1.30	3.00	.70	3.23	80.82	......	......	......	......	......	......
No. 5, split bridge, hand fired	8.75	7.00	3.25	.40	1.00	79.60	......	......	......	......	......	......
No. 6, with smoke-prevention device	......	......	......	......	......	......	7.25	12.00	0	0	0	80.75
No. 7, with smoke-prevention device	......	......	......	......	......	......	7.15	12.15	0	0	0	80.70
No. 8, with smoke-prevention device	......	......	......	......	......	......	8.15	11.10	0	0	0	80.75

TABLE 16.

RELATION OF CO AND COMBUSTION-CHAMBER TEMPERATURES.

(U. S. Geological Survey).

	Per Cent of Black Smoke.						
	0	0 to 10	10 to 20	20 to 30	30 to 40	40 to 50	50 to 60
Number of tests	37	18	56	51	36	17	4
Average per cent of smoke	0	7.1	15.5	24.7	34.7	43.1	52.9
Average per cent of CO in flue gases	0.05	0.11	0.11	0.14	0.21	0.33	0.35
Average per cent unaccounted for in heat balance	9.14	10.60	9.46	10.93	11.41	13.41	13.34
Number of tests *	26	16	48	45	32	17	4
Average combustion-chamber temperature (° F.)	2180	2215	2357	2415	2450	2465	2617

* Temperatures in combustion chamber were not determined on all tests.

as will be seen from Table 17. Carbon monoxide is a colorless gas and its presence in the chimney gases cannot be detected by the fireman, consequently the absence of smoke is not an infallible guide for perfect combustion. Since the heat of combustion of C to CO is but 4380 B.t.u. against 14,540 B.t.u. for complete combustion of C to CO_2 this loss may be expressed

$$h_2 = C \frac{(14{,}540 - 4380)\,CO}{CO_2 + CO} \tag{22}$$

$$= C \frac{10{,}160\,CO}{CO_2 + CO},$$

in which

h_2 = the loss in B.t.u. per pound of fuel.

C = proportional part of carbon in the fuel which is burned and passes up the stack.

CO_2 and CO are percentages by volume.

This loss may be reduced to a negligible quantity in a properly designed and carefully operated furnace. In fact the loss from this cause is often exaggerated and seldom exceeds 1 per cent of the total heat value of the fuel except during the few moments following the replenishing of a burned-down fire with fresh fuel or when the supply of air

TABLE 17.

LOSS DUE TO INCOMPLETE COMBUSTION OF CARBON TO CARBON MONOXIDE.

Per Cent of CO in the Flue Gas by Volume.	Per Cent of CO_2 in the Flue Gas by Volume. 6	8	10	12	14	16
0.2	328 2.2	248 1.7	199 1.3	168 1.1	144 1	126 0.8
0.4	635 4.3	484 3.3	390 2.6	327 2.2	282 1.9	248 1.7
0.6	925 6.3	709 4.8	575 3.9	474 3.2	417 2.8	367 2.5
0.8	1192 8.1	923 6.3	750 5.1	635 4.3	549 3.7	495 3.4
1.0	1494 10.2	1128 7.7	923 6.3	780 5.3	676 4.6	596 4.1
1.2	1690 11.5	1321 9	1085 7.4	923 6.3	801 5.4	708 4.8
1.4	1920 13.1	1512 10.3	1248 8.5	1061 7.2	924 6.3	819 5.6
1.6	2104 14.3	1693 11.5	1400 9.5	1193 8.1	1040 7.1	924 6 3
1.8	2340 16	1865 12.7	1549 10.5	1321 9.0	1151 7.8	1025 7
2.0	2537 17.2	2030 13.8	1690 11.5	1450 9.9	1270 8.6	1129 7.7

Large type gives the loss in B.t.u. per pound of carbon. Small type gives the per cent loss, assuming a calorfic value of 14,540 B.t.u. per pound of carbon.

is checked to meet a sudden reduction in load. In improperly designed furnaces in which the volatile gases are brought into contact with the cooler boiler surface before combustion is complete, the carbon monoxide may be reduced in temperature below its ignition point and consequently will fail to combine with the oxygen. In such a case the loss may prove to be a serious one.

High efficiencies necessitate minimum air excess, hence the presence of a small amount of CO may be expected in the flue gas. In a number of recent tests of modern central station boilers operating at 150 to 250 per cent of standard rating, the loss due to the escape of CO in the flue gas ranged from 0.2 to 1.95 per cent of the heat value of the fuel (Western bituminous) with a general average, extended over several days, of 0.4 per cent. In these tests the per cent of CO_2 in the flue gas ranged from 11.95 to 15.45. The CO content appears to increase with the increase in CO_2 and furnace temperature as shown in Fig. 10, the curves of which are based on tests of a 250 horsepower Heine boiler, hand fired. (Journal Western Society of Engineers, June 1907, p. 285.) Almost complete absence of CO is to be expected with large air excess in any well designed furnace, but it is possible for a high percentage of CO and a great excess of air supply to exist at the same time, though this combination is not likely to occur in a properly designed furnace except at very low rates of combustion.

28. Loss of Fuel Through Grate. — The refuse from a fuel is that portion which falls into the pit in the form of ashes, unburned or partially burned fuel and cinders.

In steam boiler practice the unconsumed carbon in the ash pit ranges from 15 to 50 per cent of the total weight of dry refuse depending upon the size and quality of coal, type of grate and rate of driving. The loss resulting from this waste of fuel ranges from 1.5 to 10 per cent or more, of the heat value of the fuel. It is impossible to assign a minimum value because of the various influencing factors, but numerous tests of recent installations, equipped with mechanical stokers, indicate that actual loss ranges from 1.5 to 5 per cent of the heat value of the fuel at normal driving rates. Coal which necessitates frequent slicing is apt to give greater losses from this cause than a free burning coal.

Extensive tests conducted by the American Gas & Electric Company, (Reginald Trautschold, Power, Feb. 22, 1916, p. 256) show that the actual yearly loss due to combustible in the refuse is not directly proportional to the combustible content but increases as shown by the "actual loss" curve in Fig. 11. Thus, the reduction of the combustible content from 10 per cent to 5 per cent effects a yearly saving in the ratio of 12.98 to 5.83 instead of 10 to 5.

In traveling grates in which a large percentage of the fine fuel falls through the front end of the grate a special hopper is ordinarily installed in the ash pit which reclaims most of it. (See Fig. 130.)

If h_c = calorific value of combustible in the dry refuse,
y = percentage of combustible in the dry refuse,
a = percentage of ash in the coal as fired,
h_3 = heat loss in the refuse, B.t.u. per pound of coal as fired,

$$h_3 = \frac{h_c}{100}\left(\frac{ya}{100 - y}\right). \tag{23}$$

For the average boiler test the calorific value of the combustible in the dry refuse may be taken as that of pure carbon or of the combustible in the coal (see end of paragraph 20) but for accurate results calorimetric determinations are necessary.

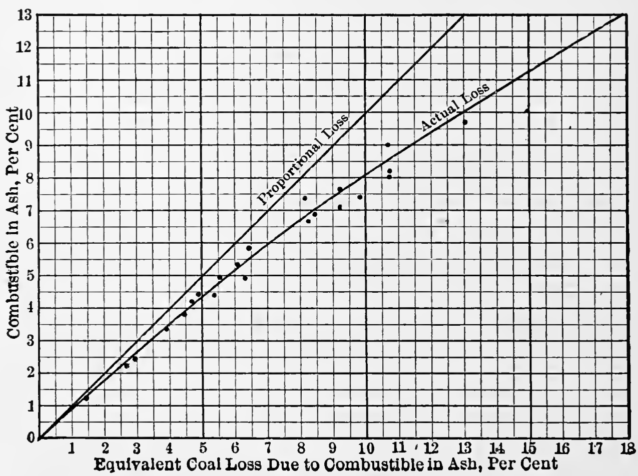


FIG. 11. Coal Loss Due to Combustible in Ash.

29. Superheating the Moisture in the Air. — The loss due to this cause is a minor one, though on hot, humid days it may be appreciable. This loss may be expressed

$$h_4 = Mc\,(t_c - t), \tag{24}$$

in which

h_4 = B.t.u. lost per pound of fuel,
M = weight of moisture introduced with the air per pound of fuel,
c = mean specific heat of water vapor, t to t_c deg. fahr.,
t = temperature of air entering the furnace, deg. fahr.,
t_c = temperature of chimney gases, deg. fahr.

$$M = zwvA, \tag{25}$$

in which

z = relative humidity (see paragraph 470),

w = weight of 1 cubic foot of water vapor at t deg. fahr. (this may be taken directly from steam tables),

v = volume of 1 pound of dry air at t deg. fahr., cubic feet,

A = weight of dry air supplied per pound of fuel burned.

30. Loss Due to Moisture in the Fuel. — Moisture in the fuel represents an appreciable loss in economy if present in large quantities, since the heat necessary to evaporate it into superheated steam at chimney temperature is lost. Firemen occasionally wet the coal to assist coking or to reduce the dust, but moisture thus added necessarily reduces the theoretical furnace efficiency. Under certain conditions wet coal may give a higher evaporation than dry coal, that is, the moisture may assist in packing the fuel and thus reduce loss through the grate, and in case of thin fires, reduce air excess. The action of the moisture is purely mechanical. (See paragraph 99.)

The loss due to evaporating the moisture may be expressed

$$h_5 = W\,[\lambda - c_1\,(t - 32) + c'\,(t_c - t')],^* \qquad (26)$$

in which

h_5 = B.t.u. lost per pound of fuel.

W = weight of free moisture per pound of fuel.

λ = total heat of one pound of saturated steam above 32 deg. fahr., corresponding to the temperature at which evaporation takes place.

c_1 = mean specific heat of water, 32 to t deg. fahr.

t = temperature of the fuel, deg. fahr.

c' = mean specific heat of the water vapor, t_c to t deg. fahr.

t_c = temperature of the chimney gas.

t' = temperature, at which evaporation takes place, deg. fahr. The temperature at which evaporation begins is low because of the low partial pressure of the vapor in the gaseous products of combustion and may range from 70 to 120 degrees, depending upon the composition of the gases and the amount of moisture evaporated. Fortunately, the term $\lambda - c't'$ is practically constant for a wide range of t' and consequently a knowledge of the actual value for each set of conditions is not necessary.

Assuming $c' = 0.46$ and taking λ from the steam tables for all values ranging from $t' = 70$ to $t' = 120$ degrees, we find that $\lambda - 0.46\,t' = 1058.6$. Substituting this value in equation (26) and reducing,

$$h_5 = w\,[1090.6 - t + 0.46\,t_c]. \qquad (27)$$

* For all engineering purposes c_1 may be taken as unity and in the following equations it has been considered as such.

31. Loss Due to the Presence of Hydrogen in the Fuel. — The hydrogen in any fuel which is not rendered inert by oxygen burns to water and in so doing liberates 62,000 B.t.u. per pound. All of this heat is not available for producing steam in the boiler, since the water formed by combustion is discharged with the flue gases as superheated steam at chimney temperature. This loss is equal to

$$h_6 = 9\,H\,(1090.6 - t + 0.46\,t_c), \tag{28}$$

in which

h_6 = B.t.u. lost per pound of fuel,
H = weight of hydrogen per pound of fuel burned.

All other notations as in equations (26) and (27).

With anthracite coal this loss is approximately 2.5 per cent of the total heat value of the combustible and with bituminous coal it runs as high as 4.5 per cent.

32. Loss Due to Visible Smoke. — Visible smoke consists of carbon in a flocculent state and ash mixed with the products of combustion. It is seldom evident in connection with anthracite coal and is generally associated with bituminous fuel. A smoky chimney does not necessarily indicate an inefficient furnace, since the losses due to visible smoke generation seldom exceed one per cent; * as a matter of fact, a smoky chimney may be much more economical than one which is smokeless. That is to say, a furnace operating with minimum air supply may cause dense clouds of smoke and still give a higher evaporation than one made smokeless by a very large excess of air. There will be some loss due to carbon monoxide, unburned hydrocarbons and soot in the former case, but this may be more than offset by the excessive losses caused by the heat carried away in the chimney gases in the latter. The amount of combustible in the soot and cinders deposited on the tubes and in various parts of the setting seldom exceeds one per cent of the calorific value of the fuel.

Smoke has become such a public nuisance, particularly in the larger cities, that special ordinances prohibiting its production have been enacted and violators are subject to heavy fines. Effective enforcement of these ordinances renders smoke production very costly and the problem of smokeless combustion becomes a momentous one.

The subject of smoke prevention and smoke-prevention devices is discussed at some length in Chapter V.

33. Radiation and Unaccounted For. — These losses are usually determined by difference. That is, the difference between the heat represented in the steam and the losses just mentioned are charged to

* See paragraph 92.

"unconsumed hydrogen and hydrocarbons, to radiation and unaccounted for." Unless accurate observations have been made in determining the various factors entering into the heat balance the radiation and unaccounted for loss may represent a large percentage of the total heating value of the coal. Careful tests on *well-designed* boiler furnaces show that the radiation loss seldom exceeds two per cent. In case of very poorly installed settings or when the rate of driving is very low the radiation loss may be considerably more than this. An examination of the data from carefully conducted tests of modern boiler furnaces will show that the "radiation and unaccounted for" items range from 2 to 6 per cent with an average of about 4 per cent. Soot deposited on the boiler tubes and throughout the setting, and cinders blown out the stack under high draft pressures may greatly increase the unaccounted for loss, unless means are available for determining these factors. For data pertaining to the loss represented by visible smoke, soot, and cinders, see paragraph 92.

34. Heat Balance. — Any chart giving the distribution of the various heat items constitutes a heat balance. The greater the number of subdivisions the more readily is it possible to locate the source of loss. The various factors entering into the commercial boiler heat balance as recommended by the American Society of Mechanical Engineers are itemized in Table 18.* According to this code the heat distribution is expressed in terms of "dry coal" or "combustible." When comparing the performance of different installations this offers a most satisfactory basis, but the operating engineer in tracing out the source of heat loss with a view of bettering operation is chiefly concerned with "coal as fired" and for this reason the heat balance is commonly expressed in terms of the latter. It is impracticable to assign specific limiting values to a general heat balance because of the wide range in the various influencing factors, such as nature and quality of fuel, type of furnace and grate, rate of driving and the like, but for a rough approximation, Table 18 may be taken as representative practice.

The heat balance in Table 18 refers to boiler in continuous operation and does not include standby losses. (See paragraph 35.)

The calculations of the various items included in the heat balance are best illustrated by a specific example.

Example 11. Calculate the various heat losses from the following data:

Heat absorbed by the boiler, 76 per cent of the calorific power of the coal as fired.

* Rules for Conducting Evaporation Tests of Boilers, A.S.M.E., Code of 1916.

Analysis of coal as fired:

	Per Cent.		Per Cent.
Carbon	65	Ash and sulphur	13
Oxygen	8	Free Moisture	8
Hydrogen	5	Nitrogen	1

Calorific value as fired, 11,850 B.t.u.

Flue-gas analysis:

	Per Cent.		Per Cent.
CO_2	12.8	CO	0.6
O_2	5.4	N_2	81.2 (by difference)

Temperature of air entering furnace, 70 deg. fahr.; temperature of flue gases, 470 deg. fahr.; temperature of the steam in the boiler, 340 deg. fahr.; relative humidity of air entering furnace, 80 per cent; combustible in the dry refuse, 20 per cent.

The heat distribution may be referred to the coal as fired, dry coal or combustible. In this problem it is referred to the coal as fired.

CALCULATION.

The combustible in the ash referred to the coal as fired is $\frac{20 \times 13}{100 - 20}$ = 3.25 per cent or 0.0325 lb. per pound of coal. Taking this as carbon the actual weight of carbon burned and appearing in the chimney gas is 0.65 − 0.0325 = 0.6175 lb. per lb. of coal as fired.

The weight of dry chimney gas per pound of carbon is equation (11)

$$A_3 = \frac{4 \times 12.8 + 5.4 + 700}{3\,(12.8 + 0.6)} = 18.82$$

For the carbon actually burned this is 18.82 × 0.6175 = 11.62 lb. per lb. of coal as fired.

The dry air supplied per pound of coal as fired is (equation 12)

$$A_4 = \frac{3.032 \times 81.2}{12.8 + 0.6} = 18.36.$$

For the carbon actually burned this is 18.36 × 0.6175 = 11.34 lb. per lb. of coal as fired.

DISTRIBUTION OF ACTUAL LOSSES PER POUND OF COAL AS FIRED.

Equation.	Loss.	Calculation.	B.t.u.	Per Cent.
......	Heat absorbed by boiler	$0.76 \times 11{,}850$	9,006	76.00
(21)	Dry chimney gas	$11.62 \times (470 - 70)\,0.24$	1,115	9.40
(22)	Incomplete combustion	$0.6175 \times 10{,}160 \times \frac{0.6}{12.8 + 0.6}$	280	2.36
(23)	Combustible in refuse	$0.0325 \times 14{,}600$	474	4.00
(27)	Moisture in the fuel	$0.08\,[1090.6 + 0.46 \times 470 - 70]$	99	0.83
(28)	Moisture from combustion of hydrogen	$9 \times 0.05[1090.6 + 0.46 \times 470 - 70]$	556	4.70
(25)	Moisture in the air	$0.08 \times 0.00115 \times 13.2 \times 11.34 \times 0.46\,(470 - 70)$	25	0.20
......	Radiation and unaccounted for	By difference	295	2.51
	Total		11,850	100.00

TABLE 18.

TYPICAL HEAT BALANCE.—BITUMINOUS COAL. BASED ON COAL AS FIRED.

	Excellent Practice.	Good Practice.	Average Practice.	Poor Practice.
	Per Cent of Calorific Value of Coal as Fired.			
Heat absorbed by the boiler	80.0	75.0	65.0	60.0
Loss due to the evaporation of free moisture in the coal	0.5	0.6	0.6	0.7
Loss due to the evaporation of water formed by the combustion of hydrogen	4.2	4.3	4.3	4.4
Loss due to heat carried away by the dry flue gas	10.0	13.0	17.5	20.0
Loss due to carbon monoxide	0.2	0.3	0.5	1.0
Loss due to combustible in the ash and refuse	1.5	2.4	4.5	5.5
Loss due to heating moisture in the air	0.2	0.2	0.3	0.4
Loss due to unconsumed hydrogen, hydrocarbons, radiation and unaccounted for	3.4	4.2	7.3	8.0
Calorfic value of the coal	100.0	100.0	100.0	100.0

35. Standby Losses.—The heat balance as ordinarily calculated refers only to the heat distribution for continuous operation over a limited period of time. It does not represent average operating conditions since the various standby losses are not considered. These include: (1) heat lost in shutting down boilers; (2) coal required to start up cold boilers; (3) coal burned in banking fires, and (4) heat discharged to waste in "blowing off" and in cleaning boilers. The magnitude of the standby losses depends upon the size and character of the boiler equipment and the conditions of operation and may range from 5 to 15 per cent or more of total heat generated (yearly basis). Thus, a continuous 24-hour full load test may show that 80 per cent of the heat of the coal is absorbed by the boiler, but when the heat represented by a month's evaporation is divided by the heat of the fuel fed to the furnace during the same period, the efficiency may drop to 70 per cent or lower. The standby losses are dependent upon so may variable factors that even average figures may be misleading unless limited to a narrow field of operation. The data in Table 19 compiled from carefully conducted tests at the central heating and power plant of the Armour Institute of Technology, serve to illustrate the extent and influence of the standby losses on the overall efficiency in a specific case.

Table 20 gives the weight of coal burned in shutting down boilers, starting up cold boilers and in banking fires for a number of Chicago plants.

TABLE 19.

INFLUENCE OF STANDBY LOSSES ON OVERALL BOILER AND FURNACE EFFICIENCY.

Period Covered by Test.	January.*	October.	July.†
Number of hours in month	744	744	744
Hours in service	708	624	153
Hours banked, or out of service	36	120	591
Per cent of rating developed, average for month	133.0	60.2	13.2
Total water:			
Fed to boiler, pounds	11,375,390	5,235,420	791,610
"Blowing off," pounds	74,800	39,870	16,150
Net evaporation	11,366,340	5,230,210	789,990
Total coal:			
Fed to furnace, pounds	1,360,370	728,360	158,960
Burned in banking, etc., pounds	3,680	13,850	37,610
Used for evaporation, pounds	1,356,690	714,510	121,350
Apparent evaporation per pound of coal fed to furnace, pounds	8.35	7.19	4.98
Actual evaporation per pound of coal used for evaporation, pounds	8.38	7.32	6.51
Gross overall efficiency of boiler and furnace, per cent	71.9	61.8	44.0
Overall efficiency, deducting standby losses, per cent	72.0	63.2	57.6

* January and October tests: 350 horsepower Stirling boiler equipped with chain grate, feed water 205 deg. fahr., pressure 100 pounds gauge, Illinois No. 3 washed nut.

† July test: 250 horsepower ditto.

TABLE 20.

COAL BURNED DURING BANKING PERIODS.*

Rated Capacity of Boiler.	Kind of Stoker.	Ratio Heating to Grate Surface	Kind of Coal.	Hours Banked.	Coal Fed to Furnace, Lb. per Boiler Hp.-hr.		*C*
					A	*B*	
250	Stationary grate	35	Buckwheat	8	0.20	0.35	......
500	Chain grate	65	Bit. scrg.	13	0.40	0.52	1000
350	Chain grate	40	Bit. No. 3	9	0.32	0.62	1600
250	Chain grate	48	Bit. scrg.	7	0.35	0.71	1450
1200	Underfeed	82	Bit. scrg.	10	0.18	0.20	2600
550	Underfeed	66	Bit. scrg.	9	0.29	0.37	1165
150	Stationary grate	40	Bit. mine run	12	0.58	0.69	560
75	Stationary grate	48	Poc. lump	12	0.81	0.95	300
400	Murphy	52	Bit. scrg.	13	0.26	0.33	1350

(*A*) Coal fired during banking period.

(*B*) Coal fed to furnace during banking period including that required to put boiler into service at end of banking period.

(*C*) Coal fed to furnace to put cold boiler into service, lb.

* These values are for specific cases only. The range in practice is so wide that average values are misleading.

The loss due to "blowing off" depends largely upon the quality of the feed water. Water containing considerable scale forming element requires frequent blowing off, the amount discharged varying from one half to two gauges of water. For example, the 350 horsepower Stirling boiler in the power plant of the Armour Institute of Technology (Table 19, Col. 1) is blown off once in 24 hours when in continuous operation, the amount averaging 3 inches as indicated by the water gauge. For one month this totals 74,800 pounds. The heat lost is 74,800 (338 − 205) = 9,200,000 B.t.u., approximately, or sufficient to evaporate 9050 pounds of water from a feed temperature of 205 deg. fahr. to steam at 100 pounds gauge. This amount should be deducted from the water fed to the boiler in calculating the net evaporation (the quality of the steam, of course, being taken into consideration). Compared with the monthly evaporation this loss is negligible, though it represents an appreciable loss *per se.*

The steam required in blowing soot from the tubes of a return tubular boiler ranges from 250 to 400 pounds of steam per cleaning with "hand blowing" and from 200 to 350 pounds with mechanically operated "soot blowers." For water tube boilers the range is considerably greater, depending upon the size of the units and the time interval between cleanings. A rough approximation is 500 to 750 pounds for hand blowing and 400 to 600 pounds for mechanical blowers incorporated within the setting.

Tests of Hand and Mechanical Soot Blowers, Power, July 13, 1915, p. 48.

36. Inherent Losses. — The heat balance as ordinarily calculated gives the distribution of the *actual* losses. Some of these losses may be considerably reduced or even entirely eliminated, while others are *inherent* and cannot be prevented. A heat balance giving the extent of the inherent losses will show at a glance where improvement may be made and where further gain is impossible. A boiler and furnace may be perfect in operation and still fail to utilize the total heat value of the fuel. For example, in the modern boiler (without an economizer or its equivalent) the flue gas cannot be lowered below the temperature of the heating surface with which it was last in contact. Since this temperature corresponds to that of the steam in the boiler, we have as the inherent losses:

1. Heat absorbed by the theoretical weight of dry chimney gases in being heated from boiler room to boiler steam temperature.
2. Heat required to evaporate and superheat the moisture in the fuel from boiler room to boiler steam temperature.
3. Heat required to evaporate and superheat the H_2O formed by

the combustion of hydrogen in the fuel from boiler room to boiler steam temperature.

4. Heat required to superheat the moisture in the air (theoretical requirements) from boiler room to boiler steam temperature.

Example 12. Determine the inherent losses from the data given in Example 11.

DISTRIBUTION OF INHERENT HEAT LOSSES PER POUND OF COAL AS FIRED.

	B.t.u.	Per Cent.
1. Inherent loss in the dry chimney gas, $9.26^* \times (340 - 70)\, 0.24$	600.0	5.06
2. Inherent loss due to moisture in coal, $0.08\,(1090.6 - 70 + 0.46 \times 340)$	94.1	0.79
3. Inherent loss due to H_2O formed by the combustion of hydrogen, $9 \times 0.05\,(1090.6 - 70 + 0.46 \times 340)$	529.6	4.47
4. Inherent loss due to "humidity" of the air, $0.8 \times 0.00115 \times 13.3 \times 8.92^* \times 0.46\,(340 - 70)$	13.5	0.11
5. Heat absorbed by ideal boiler (by difference)	10,612.8	89.57
	11,850.0	100.00

* See example 7, paragraph 22.

A comparison of the actual and inherent losses in percentages of coal as fired is as follows:

	Actual.	Inherent.
1. Dry chimney gases	9.40	5.06
2. Incomplete combustion	2.36	0.00
3. Combustible in the refuse	4.00	0.00
4. Moisture in the air	0.20	0.11
5. Moisture in the coal	0.83	0.79
6. Moisture due to combustion of hydrogen	4.70	4.47
7. Radiation and unaccounted for	2.51	0.00
8. Heat absorbed by the boiler	76.00	89.57
	100.00	100.00

The difference between the actual and inherent loss is designated as *preventable.* Although the losses due to "incomplete combustion," "combustible in the refuse," and "radiation and unaccounted for" are theoretically preventable it is almost impossible to entirely eliminate them in practice. The minimum practical loss depends upon the nature of the equipment, grade of fuel and rate of driving, and must be determined for each installation by actual test. This is also true for the "preventable" loss in the dry chimney gases and that due to the moisture in the air, moisture in the coal and moisture resulting from the combustion of hydrogen.

Since the ideal or perfect boiler under the specified conditions is able to absorb only 89.57 per cent of the calorific value of the coal it is evident that the actual boiler has a true efficiency of $76 \div 0.8957 = 84.8$ per cent.

If an economizer is used the inherent losses become less since the flue gas may be reduced to a temperature considerably lower than that of the steam but they can never be entirely eliminated unless the flue gas is discharged at the same temperature as that of the air entering the furnace.

37. Selection and Purchase of Coal.—Perhaps no single item in the operation of an existing plant or in the design of a new plant affords such an opportunity for effecting economy as the selection of fuel. Careful investigations have shown that almost any fuel can be efficiently burned in suitably designed special furnaces so that the problem of selecting a fuel for a proposed installation requires experience with the different kinds of equipment in addition to a thorough knowledge of the characteristics of various fuels. For existing plants the problem is largely a matter of testing. In many cases it has been found advisable to redesign furnaces to utilize a low-grade fuel rather than purchase an expensive coal. The following information is useful in deciding on the coal best adapted for a plant: *

a. Type and size of boilers and furnaces.
b. Load conditions, average and maximum loads.
c. Draft available and method of control.
d. Character of coals offered or available.
    1. Moisture and its effect on weight of combustible.
    2. Volatile matter and its relation to type of furnace.
    3. Ash; its amount and its fusibility and tendency to clinker.
    4. Sulphur; the amounts and how combined.
    5. Heating value, calorimeter determination.
    6. Coking qualities of the coal.
    7. Storage and tendency to spontaneous combustion.
e. Relation of the size of coal to the equipment.

After the desired grade of fuel has been decided upon the next step is to enter into an agreement with the dealer whereby the delivery of that particular fuel may be depended upon. The important items to be considered in the specifications are:

a. A statement of the amount and character of the coal desired.
b. Conditions for delivery.

* The Purchase of Coal, Dwight, T. Randall. Jour. A.S.M.E., Sept. 1911, p. 987.

c. Disposition to be made of the coal in case it is outside the limits specified.

d. Correction in price for variation in heating value and in moisture and ash content.

e. Method of sampling.*

f. By whom analyses are to be made.

In specifying the character of the coal desired for the average small plant every essential requirement of the purchaser may be fulfilled by confining them to the four following characteristics:

Moisture.
Ash.
Size of coal.
Calorific value of the coal.

Although moisture is a great and uncertain variable, and the producer can exercise no control over this factor, still the purchaser should protect himself against excessive moisture by stipulating an amount consistent with the average inherent moisture in the coal, and proper penalty should be fixed for delivery in excess of the amount allowed, a corresponding bonus being paid for delivery of less than contract amount. Considerable attention should be given to the percentage of earthy matter contained. The amount of earthy matter usually fixes the heating value of the coal, since the heating value of the combustible is practically constant. The effect of ash on the heat value of Illinois screenings as fired under a B. & W. boiler with chain grate is shown in Fig. 12. This value varies with the different types of boilers, grates, and furnaces, but is substantially as illustrated. The amount of refuse in the ash pit is always in excess of the earthy

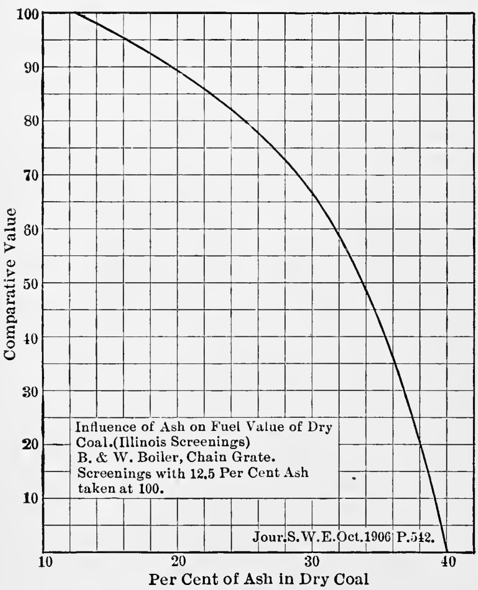


FIG. 12. Influence of Ash on Fuel Value of Dry Coal.

* See Coal Sampling Methods, Report of Committee on Prime Movers, Trans. National Electric Light Association, N. Y. City, 1916.

matter as reported by analysis, except where the amount carried beyond the bridge wall is very large.

The maximum allowable amount of sulphur is sometimes specified, since some grades of coal high in sulphur cause considerable clinkering. But sulphur is not always an indication of a clinker-producing ash, and a more rational procedure would be to classify a coal as clinkering or non-clinkering according to its behavior in the particular furnace in question, irrespective of the amount of sulphur present. An analysis of the various constituents of the ash is necessary to determine whether or not the sulphur unites with them to produce a fusible slag, and as such analyses are usually out of the question on account of the expense attached they may well be omitted. Ash fuses between 2300 and 2600 deg. fahr. and if the formation of objectionable clinker is to be avoided the furnace must be operated at temperatures below the fusing temperature. Several large concerns insert an "ash fusibility" clause in their coal specifications. For a description of ash fusion methods as practiced by various concerns consult Transactions of the National Electric Light Association, Report of the Committee on Prime Movers, 1916.

The heating value of the coal as determined by a sample burned in an atmosphere of oxygen does not give its commercial evaporative power, since this depends largely upon the composition of the fuel, character of grate, and conditions of operation. It serves, however, as a basis upon which to determine the efficiency of the furnace. In large plants where a number of grades of fuel are available it is customary to conduct a series of tests with the different grades and sizes, and the one which evaporates the most water for a given sum of money, other conditions permitting, is the one usually contracted for. In designing a new plant particular attention should be paid to the performance of similar plants already in operation, and that fuel and stoker should be selected which are found to give the best returns for the money. Where smoke prevention is a necessity the smoke factor greatly influences the choice of fuel and stoker.

The Purchase of Coal: Eng. Mag., Mar., 1911; Jour. A.S.M.E., Mar., 1911; Power, Apr. 6, 1909, p. 642.

The Purchase of Coal by the Government under Specifications: Bureau of Mines, Bull. No. 11, 1910; U. S. Geol. Survey, Bulletins No. 339, 1908; No. 378, 1909.

The Fusing Temperature of Coal Ash: Power, Nov. 28, 1911, p. 802.

The Clinkering of Coal: Trans. A.S.M.E., Vol. 36, 1914, p. 801.

Jour. A.S.M.E., April 1915, p. 205. Power, Oct. 24, 1916, p. 591.

38. Size of Coal — Bituminous. — Coal is usually marketed in different sizes, ranging from lump coal to screenings. The latter furnish by far the greater part of the stoker fuel used. The sizes and grades of bituminous and semi-bituminous coals vary so much, according to kind and locality, that there are no standards of size for these coals which are generally recognized. According to the A.S.M.E. Boiler Code (1915):

Bituminous coals in the Eastern States may be graded and sized as follows:

a. Run of mine coal; the unscreened coal taken from the mine after the impurities which can be practicably separated have been removed.
b. Lump coal; that which passes over a bar-screen with openings $1\frac{1}{4}$ inch wide.
c. Nut coal; that which passes through a bar-screen with $1\frac{1}{4}$ inch openings and over one with $\frac{3}{4}$ inch openings.
d. Slack coal; that which passes through a bar-screen with $\frac{3}{4}$ inch openings.

Bituminous coals in the Western States may be graded and sized as follows:

e. Run of mine coal; the unscreened coal taken from the mine.
f. Lump coal; divided into 6-inch, 3-inch, and $1\frac{1}{4}$ inch lump, according to the diamter of the circular openings over which the respective grades pass; also 6 by 3 lump and 3 by $1\frac{1}{4}$ lump, according as the coal passes through a circular opening having the diameter of the larger figure and over one of the smaller diameter.
g. Nut coal; divided in 3-inch steam nut, which passes through an opening 3-inch diameter and over $1\frac{1}{4}$ inch; $1\frac{1}{4}$ inch nut, which passes through a $1\frac{1}{4}$ inch diameter opening and over a $\frac{3}{4}$ inch diameter opening; and $\frac{3}{4}$ inch nut, which passes through a $\frac{3}{4}$ inch diameter opening and over a $\frac{5}{8}$ inch diameter opening.
h. Screenings; that which passes through a $1\frac{1}{4}$ inch diameter opening.

For maximum efficiency coal should be uniform in size. With hand-fired furnaces there is usually no limit to its fineness and larger sizes can be used than with stokers. As a rule the percentage of ash increases as the size of coal decreases. This is due to the fact that all of the fine foreign matter separated from larger coal, or which comes from the roof or the floor of the mine, naturally finds its way into the smaller coal. The size best adapted for a given case is dependent upon the intensity of draft, kind of stoker or grate, and the method of firing, and its proper selection often affords an opportunity to effect considerable economy. The influence of the size of screenings on the capacity and efficiency of a boiler in a specific case is illustrated in Fig. 13. The curves are plotted from a series of tests conducted with Illinois screenings on a 500-horsepower B. & W. boiler, equipped with chain grates, at the power house of the Commonwealth Edison Company.

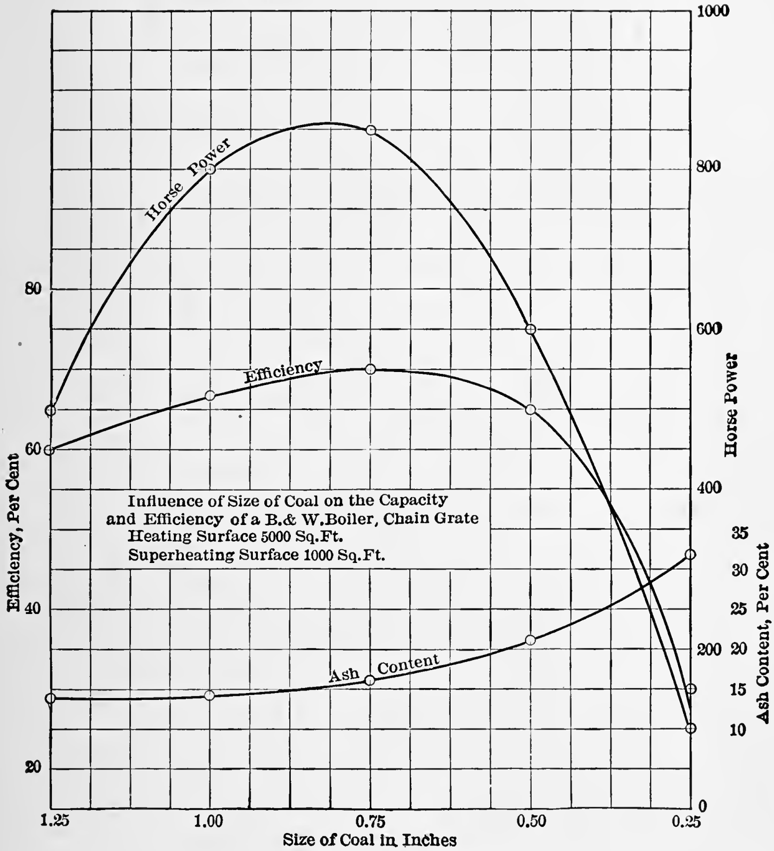


FIG. 13. Influence of Size of Coal on Boiler Capacity and Efficiency.

Influence of Thickness of Fire. — See paragraph 82.

Size of Coal: Some Characteristics of Coal as affecting Performance with Steam Boilers: Jour. West. Soc. Engrs., Oct., 1906, p. 528.

39. Washed Coal. — Coal is washed for the purpose of separating from it such impurities as slate, sulphur, bone coal, and ash. All of these impurities show themselves in the ash when the coal is burned. Screenings contain anywhere from 5 per cent to 25 per cent of ash and from 1 per cent to 4 per cent of sulphur. Washing eliminates about 50 per cent of the ash and some of the sulphur. Table 21 gives some idea of the effects of washing upon a number of grades of coal. The evaporative power of the combustible is practically unaffected by washing and the greater part of the water taken up by the coal is removed by thorough drainage. Many coals otherwise worthless as steam coals are

TABLE 21.

EFFECT OF WASHING ON BITUMINOUS COALS.

(*Journal W.S.E.*, December, 1901.)

	Before Washing. (Per Cent.)			After Washing. (Per Cent.)		
	Ash.	Sulphur.	Fixed Carbon.	Ash.	Sulphur.	Fixed Carbon.
Belt Mountain, Mont.	18.74	3.34	43.72	5.56	2.40	48.39
Wellington Colliery Co., Vancouver Island (new coal)	35.00	......	38.00	8.90	......	56.90
Alexandria Coal Co., Crabtree, Pa.	10.60	1.30	....	6.21	0.61	....
De Soto, Ill.	18.00	......	44.00	4.20	......	57.00
Northwestern Improvement Co., Roslyn, Wash.	16.30	0.57	45.90	9.70	0.40	47.86
Luhrig Coal Co., Zaleski, Ohio	15.80	1.90	....	8.00	0.87	50.90
Rocky Ford Coal Co., Red Lodge, Mont.	25.30	......	37.80	8.50	......	47.20
Buckeye Coal and Ry. Co., Nelsonville, Ohio	13.77	1.05	49.04	4.30	0.89	54.82
New Ohio Washed Coal Co., Carterville, Ill.	9.48	0.78	55.00	4.85	0.69	63.00

rendered marketable by washing. Washed coals are usually graded as follows:

Size.	Screens.	
No. 1	Over $1\frac{3}{4}$	Through 3
2	$1\frac{1}{8}$	$1\frac{3}{4}$
3	$\frac{3}{4}$	$1\frac{1}{8}$
4	$\frac{1}{4}$	$\frac{3}{4}$
5	....	$\frac{1}{4}$

Numbers 3 and 4 are excellent sizes for use in connection with stokers and No. 5 is well adapted to hand furnaces where smoke prevention is essential.

Coal Washing in Illinois. Univ. Ill. Bull. No. 9, Oct. 27, 1913.

40. Powdered Coal. — The use of powdered coal in the manufacture of cement and in other industrial processes is an established success. The steadily increasing cost of oil is stimulating the adoption of powdered coal in many situations where fuel oil is now burned. As a fuel for steam boiler furnaces, however, powdered coal is still in an experimental stage, and although published accounts of the various trials are full of promise and apparent accomplishment, but few processes have survived. In view of the high efficiencies effected with bulk coal in connection with mechanical stokers it is not likely that powdered coal will be used extensively in very large plants since the advantages incident to the combustion of the powdered product are more than offset by the cost of pulverizing, drying and handling. Some of the advantages obtained in burning powdered coal are:

a. Complete combustion and total absence of smoke. The coal in the form of dry impalpable dust is induced or forced into the zone of combustion where each minute particle is brought into contact with the necessary amount of air and complete oxidation is effected with minimum air excess.

b. A cheaper grade of coal may be burned; in fact, some grades of coal which are burned with only moderate success in bulk may be efficiently consumed in the powdered form.

c. The fuel supply may be readily controlled to meet the fluctuations in load and furnace standby losses may be greatly reduced.

d. The labor of firing is reduced to a minimum.

The practical objections which may offset these advantages are:

1. Cost of preparation.
2. Explosibility of coal dust.
3. Storage limitations.
4. Furnace depreciation.
5. Disposal of ash and slag.
6. Refuse discharged through chimney.

The various advantages and disadvantages are treated at length in the following paragraphs.

41. Types of Powdered Coal Feeders and Burners. — Powdered coal burners may be grouped into two general classes:

1. The dust-feed burner, in which the coal is supplied in the powdered form, and
2. The self-contained burner, in which the coal is crushed, pulverized, and fed to the furnace simultaneously.

The dust may be fed into the furnace by

1. Natural draft,
2. Mechanical means, or by
3. Forced draft.

The following outline gives a classification of a few of the best-known coal-dust burners:

Natural Draft	Natural Draft Feed	Pinther Wegener	Dust Feed
	Brush Feed	Schwartzkopff	
Forced Draft	Blower Feed	Rowe General Electric	
	Compressed Air	Atlas	Self-contained
	Paddle Wheel	Ideal Blake	

Since the natural draft type of feeders are not in evidence in boiler practice and are little used in industrial furnaces they will not be considered here. For an extended study the reader is referred to the accompanying bibliography.

All of the successful feeders in current practice are of the forced draft type. The Rowe coal-dust feeder, Fig. 14, manufactured by the C. O. Bartlett and Snow Company, Cleveland, Ohio, is one of the oldest examples of this type and although its application is to be found chiefly in cement kilns it has been used with some success in boiler furnaces. Referring to the illustration, powdered coal is fed from storage bin to hopper *A* and feed worm *B*. The latter forces it down spout *F* directly to the delivery tube *D* where it is caught by the air draft and fed into the furnace. The amount of feed depends upon the speed of the feed worm which is driven by the friction disk *I* pressing against the flange plate *H*. This disk is moved in or out by a suitable handle so as to get any desired speed. The air is furnished by the fan *C* the amount being controlled by the valve *E*.

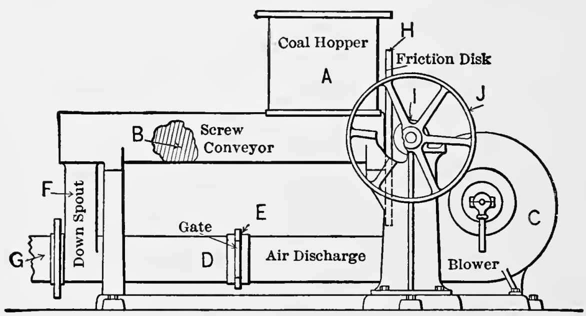


FIG. 14. Rowe Coal-dust Feeder.

Figure 15 illustrates a forced draft feeder and burner designed by A. S. Mann as applied to a water tube boiler at the Schenectady plant of the General Electric Company. This installation has been in continuous operation for some time and appears to be a successful commercial application of coal dust burning. Powdered fuel is fed from hopper *H* by a variable-speed motor-driven endless-screw *S* to down spout *D* where it is picked up by a primary current of air and induced into the suction opening of vacuum tee V. A secondary air blast at A forces the fuel into burner B whence it is discharged into the furnace. The air and fuel are thoroughly mixed in the burner and by controlling the fuel supply and the various air inlets any rate of feed may be effected without stratification. Auxiliary air jets discharging into the furnace break up the stream issuing from the burner and prevent the fuel particles from leaving the combustion chambers before oxidation is complete. (See paragraph 42 for further details of this installation.)

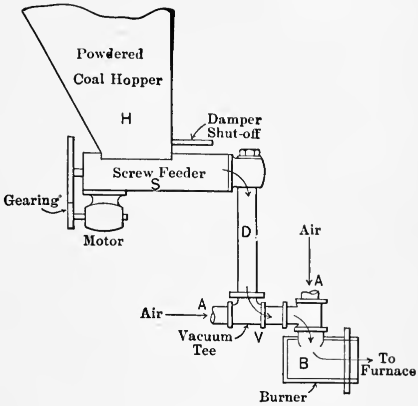


FIG. 15. Mann Powdered Coal Feeder and Burner.

Fig. 17 gives a sectional view of the Blake apparatus, and is a typical example of a self-contained system. It comprises a multistage centrifugal pulverizer, coal hopper, conveyor and fan mounted on a single bedplate. Referring to the illustration, coal previously crushed to nut size is fed to the hopper from the bottom of which it is conveyed by an endless screw to the first stage of the pulverizer. The lumps are thrown out radially by centrifugal force, due to the rapidly revolving bats, and are reduced to a dust by percussion and attrition. The largest chamber contains a fan, the function of which is to draw the pulverized material successively from one chamber to another and finally deliver it to the discharge spouts. The air is drawn into the fuel chamber with the coal through passage A, and also through opening B around the shaft. After entering the fan chamber, the mixture of coal dust and air receives an additional supply of air through opening C. The apparatus may be belt-driven or direct-connected and runs at about 1200 to 1600 r.p.m., requiring 8 to 12 horsepower for its operation.

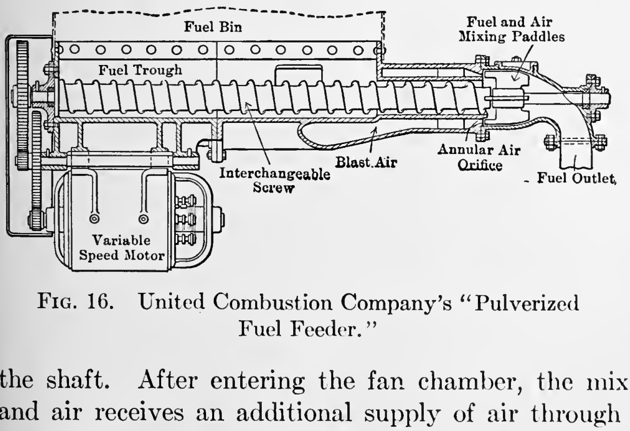


FIG. 16. United Combustion Company's "Pulverized Fuel Feeder."

Experience has demonstrated that as much as 14 per cent of moisture in the coal has little effect on the pulverization and burning. Several boiler plants equipped with this device gave smokeless combustion and high efficiency but faulty furnace design caused the system to be abandoned.

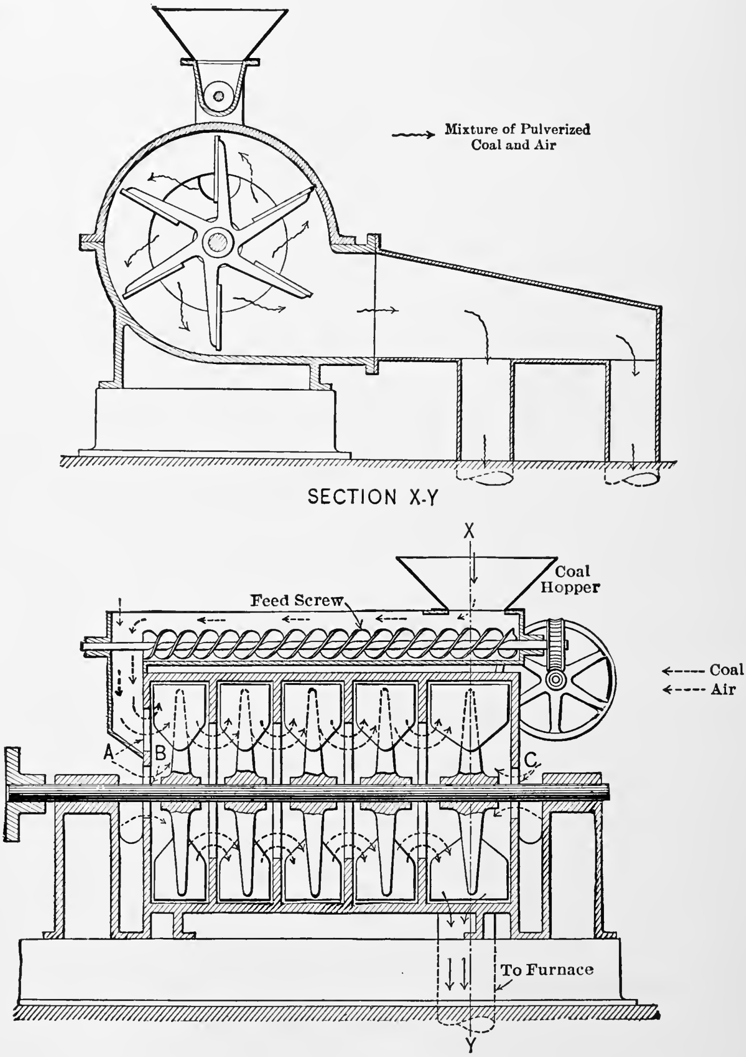


Fig. 17. Blake Coal-dust Feeder.

42. Boiler Furnaces for Burning Powdered Coal. — The main difficulty in the commercial application of powdered coal to boiler furnaces appears to lie in the correct design of furnace and in the distribution of the air supply. Several types of feeders and burners are giving the best of satisfaction in cement kilns and in other industrial furnaces, but when applied to steam boilers fail to meet requirements. The accumulation of slag and rapid deterioration of the furnace lining is the chief cause of failure. In burning bulk coal the mass of incandescent fuel stores up a quantity of heat to effect distillation and ignition of the volatile matter in the green fuel. Since powdered coal is burned in suspension a reverberatory furnace or its equivalent is necessary to bring about the same result. A large combustion chamber is necessary and the shape of the furnace and path of the flame should be such as to insure complete combustion and provide a uniform distribution of

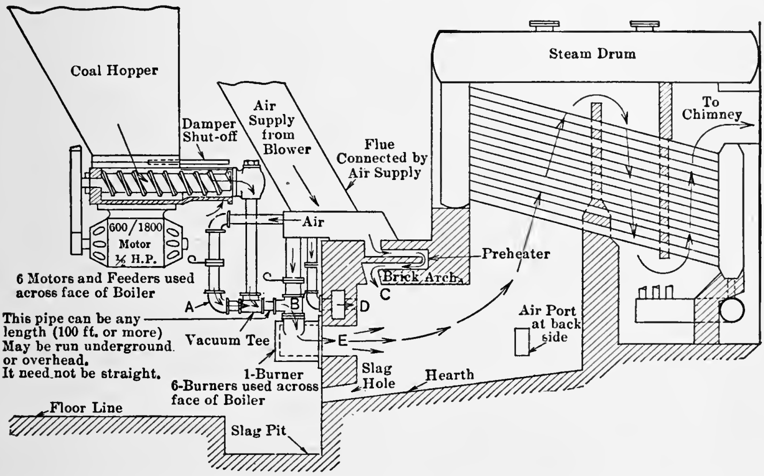


FIG. 18. General Arrangement of Powdered Coal Burning Equipment at the Schenectady Works of the General Electric Company.

heat over the boiler heating surface without direct impingement of flame. Temperature, velocity, volume and direction of current must all be considered since there is perhaps no fuel more sensitive to incorrect use than coal dust. Several types of furnaces for burning coal dust under steam boilers are described and illustrated in a "Symposium on Powdered Coal," Jour. A.S.M.E., Oct. 1914, but few have survived the experimental stage and none appears to have solved the problem, although favorable mention is made of the Bettington boiler as commercially exploited in England. An apparently successful installation is that of a 474 horsepower water tube boiler at the Schenectady Works of the General Electric Company. Fig. 18 gives a diagrammatic, longitudinal section through the boiler and furnace of this installation and Fig. 18a illustrates the system of locating burners and air passages.

Six feeders and burners, of the type illustrated in Figs. 15 and 16, are attached to the one boiler in order to effect flexibility in operation. Several auxiliary air ports distributed throughout the setting are directed at various angles against the burning currents thereby insuring perfect stirring action. Combustion is virtually complete in eight feet of travel even at 200 per cent rating. Continuous loads of 220 per cent rating have been readily maintained and a maximum load of 265 per cent rating has been carried for a short period. No difficulty whatever is experienced with slag, ash and burnt brickwork for loads under 140 per cent rating, but with heavy loads particles of slag travel with the gas current and cling to the bottom row of tubes. The accumulation of this slag is prevented by "blowing off" with a steam jet. A small amount (2 per cent) of flocculent ash in the form of a very fine powder is discharged with the chimney gases. There is no visible smoke, all soot drops in the gas chambers before reaching the stack and the slag is drawn out once during the day to a concrete pit containing water. (For detailed description

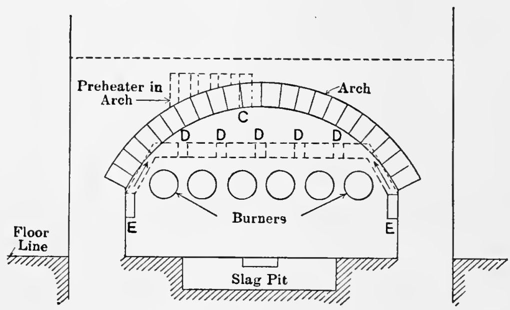


Fig. 18a. Diagram of Boiler Front Showing Location of Mann Burner and Air Passages.

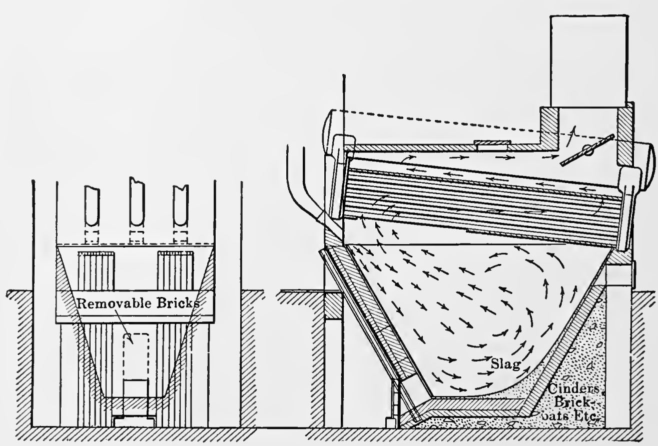


Fig. 19. Powdered Coal Furnace as Installed Under a Franklin Boiler at the American Locomotive Company's Schenectady Plant.

of this installation see "General Electric Review," September and October, 1915.)

Fig. 20 gives the general details of a powdered coal furnace as installed under a Franklin boiler at the Schenectady plant of the American Locomotive Works. This furnace is reported to have been in continued service for 18 months without repairs on the furnace walls. Protection against depreciation is effected by a coating of slag as shown in the illustration. No trouble is experienced from coke or cinder-clogged water tubes. (See Journal A.S.M.E., Dec. 1916, p. 1000.)

A number of locomotives have been recently equipped with powdered coal burners and are apparently successful in operation. (See "Pulverized Fuel for Locomotives," Proceedings N. Y. Railroad Club, Feb. 18, 1916.)

The use of pulverized coal appears to be a commercial success at the power plant of the M. K. & T. Shops, Parsons, Kan. For a description of this installation together with results of the boiler tests, see National Engineer, May, 1917, p. 175.

43. Cost of Preparing Powdered Coal. — The cost of drying and grinding varies with the size and type of equipment, initial moisture content of the coal, degree of fineness required and the quantity treated per unit of time. Experience shows that the moisture content should be reduced to approximately one per cent or less for efficient grinding where screens are used and that 95 per cent of the powder should pass through a 100-mesh and 80 to 85 per cent through a 200-mesh screen. Dry coal is desired because it can be more intimately mixed with air and fed regularly to the furnace. Moist coal will clog the feeding mechanism and the screen and tends to pack in the storage bins. Machines that depend upon air separation for regulating the fineness of the coal have no screens to clog and the moisture content need not be less than 5 per cent, but the power requirements for the grinding increase with the moisture content. The average cost of drying and grinding, including maintenance and fixed charges, ranges from 25 to 75 cents per ton. In stokers of the "Blake Pulverizer" type, in which the grinding, drying and feeding are carried on simultaneously in a self-contained apparatus, the power consumed varies from 2 to 10 per cent of the total power developed by the boiler, depending upon the nature of the fuel, efficiency of the driving mechanism and the degree of fineness of the powdered coal; 5 per cent is a fair average. Powdered coal sold in the open market ranges from 50 cents to 90 cents a ton above the price of the same coal in bulk.

44. Storing Powdered Fuel. — Most cities limit the storage of powdered coal to such a small quantity as to interfere seriously with continuity of operation in case of breakdown to the pulverizing or drying apparatus. Spontaneous combustion is likely to occur with moist coal and since the dry powdered product is exceedingly hygroscopic it is necessary to store it in air-tight bins. Powdered coal in quantity should always be kept moving and should never be allowed to stand more than a day or two. Coal dust in a suspended state is dangerous and may cause a serious explosion, but this danger may be minimized by the use of equipment which prevents the leakage of the dust.

45. Efficiency of Powdered Coal Furnaces. — A comparison of a number of tests of hand fired and powdered coal furnaces with different types of feeders shows a decided gain in efficiency of the powdered coal over the hand fired where the fuel is of a low grade. The gain becomes less marked with fuel of fair quality and disappears entirely with good fuel and properly manipulated automatic stokers. Numerous tests of powdered coal installations are recorded showing boiler and furnace efficiency of 77 to 81 per cent, but these figures are readily equaled and have been frequently exceeded with bulk coal firing so that other factors than fuel economy must be considered in comparing the commercial value of the two systems.

46. Depreciation of Powdered Coal Furnaces. — For complete and efficient combustion of powdered coal high furnace temperatures are essential. On account of the high temperatures involved and the slag produced from the ash the destruction of the furnace lining is very rapid. To withstand the intense heat of combustion brick-work of the highest quality is essential since common fire brick are soon reduced to a liquid slag. A good quality of fire brick will withstand the heat for several months without renewal provided the furnace is properly enclosed, otherwise the strain of expansion and contraction due to alternate heating and cooling will crack the brick. Excellent results have been obtained from the use of bricks composed chiefly of the refuse of a carborundum slag, but the high cost has prevented their general use. Fire brick target walls are not recommended for steam boiler practice because of the localization of heat.

Pulverized Coal for Steam Making: Trans. A.S.M.E., Vol. 36, p. 123–169, 1914.
Pulverized Coal: General Bibliography; Jour. A.S.M.E., Jan., 1914, p. lv.
Some Problems in Burning Powdered Coal: Gen. Elec. Review, Sept. and Oct., 1915.
Firing Boilers with Pulverized Coal: Power, Feb. 14, 1911.
Use of Pulverized Coal under Steam Boilers: Prac. Engr. U. S., June 1, 1916, p. 490.
Tests of Pulverized Fuel: Engr. U. S., Apr. 1, 1904; Power, May, 1904, Feb. 14, 1911.
Types of Coal Dust Burners: Engr. U. S., Apr. 1, 1904; Jan. 1, 1903; Power, Mar., 1904.
Burning Low Grade Coal Dust: Power, Sept. 12, 1911, p. 393.

47. Fuel Oil. — The recent development of oil wells in the Western and Gulf States, with the consequent enormous increase in production, has given a marked impulse to the use of crude oil for fuel purposes in steam power plants. Where economic and commercial conditions permit, it is the most desirable substitute for coal. The total absence of smoke and ashes, prompt kindling and extinguishing of fires, extreme rate of combustion, and ease with which it can be handled and controlled are marked advantages in favor of fuel oil. The reduction in volume and weight over an equivalent quantity of coal for equal heating values and the increase in boiler efficiency are factors of no mean importance, particularly in connection with marine or locomotive work. In stationary work the chief objections are the difficulty in securing ample storage capacity and the increased rate of insurance. An objection sometimes raised against fuel oil is the increased depreciation of the setting, but in a well-designed setting this figure is only nominal and of secondary importance. However, in spite of the many advantages presented in the use of fuel oil for power plant purposes, the comparatively limited supply prevents its adoption as a general fuel and limits its use to the plants most favorably located.

48. Chemical and Physical Properties of Fuel Oil. — Crude oil, as pumped at the wells, consists principally of various combinations of hydrogen and carbon, together with small amounts of nitrogen, oxygen sulphur, water in emulsion and silt. The nitrogen and oxygen may be classified with the moisture and silt as inert impurities. The moisture in oil fuel should not exceed 2 per cent, since it not only acts as an inert impurity, but must be converted into steam in the furnace and thus still further reduces the heat value per pound. The sulphur, though combustible, has a low calorific value and is otherwise undesirable. From Table 22 it will be seen that the physical properties of oils from different localities in the United States differ widely, while the chemical constituents vary but slightly. For example, the oils given in the table differ greatly in volatility, specific gravity, and viscosity, but have approximately the same percentages of carbon and hydrogen. Taking hydrogen and carbon as the principal constituents it is found that oils rich in hydrogen are lighter in weight than those rich in carbon. Other things being equal, oils rich in hydrogen have a higher calorific value than those rich in carbon, but the heavier oils are usually the cheaper. The relation between heating value and specific gravity for anhydrous California oil is as shown in Table 23.

The heat value may be closely approximated by means of the following formula (Jour. Am. Chem. Soc., Oct., 1908):

$$\text{B.t.u.} = 18{,}650 + 40\,(B - 10), \tag{29}$$

in which B = degrees Baumé at 60 deg. fahr.

TABLE 22.

ANALYSES OF TYPICAL AMERICAN FUEL OILS.

Location.	Authority.	Physical Properties.					Chemical Properties.				
		Gravity.		Flash Point, Deg. Fahr.	Burning Point, Deg. Fahr.	Viscosity at 68 Deg. Fahr., Engler Scale.	C	H	O	S	B.t.u. per Lb.
		Baumé at 60 Deg. Fahr.	Specific at 60 Deg. Fahr.								
California crude:											
Coalinga	Bulletin No. 19. U. S. Bureau of Mines (1912).	17.52	0.9498	192	230	341.5	86.37	11.30	1.14	0.60	18,727
Kern River		15.16	0.9645	226	266	915.6	86.36	11.27	0.74	0.89	18,553
McKitterick		16.37	0.9566	188	207	200.0	86.51	11.41	0.58	0.74	18,508
Midway		16.34	0.9570	172	210	518.0	86.58	11.61	0.74	0.82	18,613
Sunset		14.37	0.9701	192	235	527.0	85.64	11.37	0.84	1.06	18,478
Kansas crude	B. F. McFarland	31.66	0.866	52	77	......	85.40	13.07	......	......	......
Louisiana crude	C. E. Coates	......	......	......	......	......	......	......	......	0.34	19,814
Ohio distillate	Deville	27.83	0.887	......	......	......	84.20	13.10	2.70*	......	18,718
Ohio distillate	N. W. Lord	38.25	0.838	177	212	......	......	......	......	......	......
Pennsylvania crude	Deville	39.50	0.826	......	......	......	82.00	14.80	3.20*	......	17,930
Pennsylvania distillate	Deville	27.80	0.886	......	......	......	84.90	13.70	1.40*	......	19,210
West Virginia crude	Deville	36.46	0.841	......	......	......	84.36	14.10	1.60*	......	18,400
Wyoming crude	E. E. Slosson	20.00	0.933	273	343	......	......	......	......	0.67	19,440
Texas crude	Denton	22.17	0.920	142	181	......	84.60	10.90	2.87*	1.63	19,060
Texas distillate	U. S. Naval Report	21.18	0.926	216	240	......	83.26	12.41	3.83	0.50	19,481
Oklahoma crude	Armour & Co.	25.00	0.903	264	286	92.0†	87.93	11.37	0.19	0.41	19,650

* O + N. † Relative viscosity, Saybolt scale.

TABLE 23.

APPROXIMATE RELATION BETWEEN THE HEATING VALUE AND SPECIFIC GRAVITY.

(Professor Le Conte, University of California.)

Degrees, Baumé.	Specific Gravity.	Weight per Barrel.	B.t.u. per Pound.	B.t.u. per Barrel.	Degrees, Baumé.
10	1.0000	350.035	18,280	6,398,600	10
11	0.9929	347.55	18,340	6,374,100	11
12	0.9859	345.10	18,400	6,349,800	12
13	0.9790	342.68	18,460	6,325,900	13
14	0.9722	340.30	18,520	6,302,400	14
15	0.9655	337.96	18,580	6,279,300	15
16	0.9589	335.65	18,640	6,256,500	16
17	0.9524	333.37	18,700	6,234,000	17
18	0.9459	331.10	18,760	6,211,400	18
19	0.9396	328.89	18,820	6,189,700	19
20	0.9333	326.69	18,880	6,167,900	20
21	0.9272	324.55	18,940	6,147,000	21
22	0.9211	322.42	19,000	6,126,000	22
23	0.9150	320.28	19,060	6,104,500	23
24	0.9091	318.22	19,120	6,084,400	24
25	0.9032	316.15	19,180	6,063,800	25

Oil that is to be transported or stored or used for fuel inside of buildings should be of the "reduced" variety, from which the naphtha and higher illuminating products have been distilled. The gravities of such distillates vary from 20 to 25 degrees Baumé, or close to 0.9 specific gravity, and their flash points range from 240 deg. fahr. to 270 deg. fahr.* One barrel of crude oil contains 42 gallons and weighs from 310 to 350 pounds, according to the specific gravity. Compared with coal, oil occupies about 50 per cent less space and is 35 per cent less in weight for equal heat values. The comparative heat values of coal and oil are approximately as follows:

B.t.u. per Pound of Coal.	Pounds of Coal Equal to 1 Barrel of Oil.	Barrels of Oil Equal to 1 Short Ton of Coal.
10,000	620	3.23
11,000	564	3.55
12,000	517	3.87
13,000	477	4.19
14,000	443	4.52
15,000	413	4.84

49. Efficiency of Boilers with Fuel Oil. — A coal-burning boiler which utilizes 80 per cent of the heat value of the fuel is exceptional — 77 per cent represents very good practice, and 75 per cent a fair average for good practice. The great majority of coal-burning boilers, however, operate at efficiencies less than 70 per cent. With oil fuel a

* For relationship between degrees Baumé and specific gravity see paragraph 374.

boiler and furnace efficiency of 75 per cent is quite ordinary and 80 per cent not uncommon. This increase in efficiency is partly due to the fact that the oil is readily broken up and brought into immediate contact with the necessary air for combustion and loss due to excessive air dilution is correspondingly reduced.

Table 24 gives the theoretical air requirements for different densities of fuel oils and Table 25 the air excess for various efficiencies. These tables were compiled by C. R. Weymouth (Trans. A.S.M.E., Vol. 30, p. 801).

TABLE 24.

POUNDS OF AIR PER POUND OF OIL AND RATIO OF AIR SUPPLIED TO THAT CHEMICALLY REQUIRED.

Per Cent CO_2 by Volume as Shown by Analysis of Dry Chimney Gases.	Light Oil, C, 84%; H, 13%; S, 0.8%; N, 0.2%; O, 1%; H_2O, 1%.		Medium Oil, C, 85%; H, 12%; S, 0.8%; N, 0.2%; O, 1%; H_2O, 1%.		Heavy Oil, C, 86%; H, 11%; S, 0.8%; N, 0.2%; O, 1%; H_2O, 1%.	
	Lb. of Air per Lb. of Oil.	Ratio of Air Supply to Chemical Requirements.	Lb. of Air per Lb. of Oil.	Ratio of Air Supply to Chemical Requirements.	Lb. of Air per Lb. of Oil.	Ratio of Air Supply to Chemical Requirements.
4	51.40	3.607	51.93	3.704	52.45	3.803
5	41.31	2.899	41.71	2.975	42.12	3.054
6	34.58	2.427	34.90	2.490	35.23	2.554
7	29.77	2.089	30.04	2.143	30.31	2.198
8	26.17	1.836	26.39	1.883	26.62	1.930
9	23.37	1.640	23.56	1.680	23.75	1.722
10	21.12	1.482	21.29	1.518	21.45	1.555
11	19.83	1.391	19.43	1.386	19.58	1.419
12	17.76	1.246	17.88	1.276	18.01	1.306
13	16.46	1.155	16.57	1.182	16.69	1.210
14	15.36	1.078	15.45	1.102	15.55	1.127
15	14.39	1.010	14.48	1.033	14.57	1.056

TABLE 25.

BOILER EFFICIENCY FOR EXCESS AIR SUPPLY (OIL FUEL).

Excess Air Supply, Per Cent.	10	50	75	100	150	200
Assumed temperature of escaping gases, deg. fahr.	400	450	475	490	Over 500	Over 500
Corresponding ideal efficiency of boiler, per cent	84.2	80.27	77.66	75.22	Under 70.94	Under 67.09
Possible saving in fuel due to reduction of air supply to 10 per cent excess, expressed as per cent of oil actually burned under assumed conditions	0	4.67	7.78	10.68	Over 15.76	Over 20.32

Table 26 gives the results of a series of tests made at the Redondo plant of the Pacific Light & Power Company, California, on a 604-horsepower B. & W. boiler equipped with Hammel furnaces and burners. The boiler was in regular service and under usual operating conditions.

TABLE 26.

EVAPORATIVE TESTS OF OIL-BURNING BOILER.
(Pacific Light & Power Co., Redondo, Cal.)

Date of test (1910)		Aug. 8.	Aug. 9.	Aug. 10.	Aug. 11.	Aug. 12.	Aug. 13.	Sept. 5.	Average of all Tests.
Test number	No.	1	2	3	4	5	6	7	
Duration of test	Hrs.	7	7	7	7	7	7	7	7
Steam pressure	Lbs.	184.9	184.9	186.0	184.7	183.5	184.6	184.9	184.8
Steam temperature	Deg. F.	473.3	457.4	468.8	465.1	473.8	493.8	526.7	479.8
Superheat	Deg. F.	91.8	76.0	86.9	83.7	93.0	112.5	144.3	98.3
Temperature, feed water	Deg. F.	90.6	92.6	92.7	93.4	90.8	94.6	101.2	93.7
Factor of evaporation		1.237	1.225	1.232	1.229	1.237	1.245	1.259	1.238
Average water level	Ins.	4	4	4	4	4	4	4	4
Barometer	Ins.	29.97	30.00	30.00	30.09	30.08	30.04	29.68	29.97
Temperature boiler room	Deg. F.	88.7	86.6	84.4	85.2	87.3	86.7	84.4	86.2
Temperature flue gases	Deg. F.	385.3	397.5	409.1	406.2	429.0	477.1	537.5	434.5
Draft: In ash pit	Ins.	0.025	0.035	0.055	0.044	0.071	0.127	0.230	0.084
Draft: In furnace	Ins.	0.01	0.005	0.025	0.014	0.060	0.130	0.188	0.062
Draft: In rear pass	Ins.	0.005	0.014	0.061	0.046	0.133	0.296	0.471	0.147
Carbon dioxide	Per Cent.	12.2	13.4	13.3	14.3	14.2	13.3	12.1	13.2
Oxygen	Per Cent.	3.6	2.7	2.4	1.8	1.7	2.8	6.8	3.1
Excess air	Per Cent.	28.7	17.7	18.5	10.6	11.3	18.5	43.0	21.2
Smoke		None.	None.	None.	None.	None.	None.	Lt. Haze.	
Position of ash-pit doors		Doors Wide Open Throughout all Tests.							
Temperature in first pass above third tube	Deg. F.	1,100	1,090	1,160	1,180	1,240	1,300	1,600	1,240
Temperature top first pass	Deg. F.	640	640	700	680	780	940	1,170	793
Temperature top second pass	Deg. F.	570	540	620	610	650	740	820	650
Temperature bottom second pass	Deg. F.	500	500	520	510	550	600	700	554
Temperature bottom third pass	Deg. F.	450	450	505	495	530	570	660	523
Total water actually evaporated	Lbs.	85,766	111,988	129,628	129,609	156,622	191,290	226,558	147,351
Total water evaporated from and at 212° F.	Lbs.	106,092	137,185	159,702	159,289	193,741	238,156	285,236	182,771
Water per hour evaporated from and at 212° F.	Lbs.	15,156	19,598	22,814	22,756	27,677	34,022	40,748	26,110
Steam used by burners: Pounds per hour	Lbs.	234	441	549	549	624	708	873	568
Steam used by burners: Per cent of total steam	Per Cent.	1.54	2.25	2.40	2.40	2.25	2.08	2.13	2.15
Steam pressure to burners	Lbs.	48.2	74.7	99.7	102.6	120.4	142.5	167.6	122.2
Oil pressure to burners	Lbs.	11.4	15.2	24.4	25.4	38.3	46.1	61.6	31.6
Temperature of oil to burner line	Deg. F.	131.3	134.3	133.5	142.3	140.1	142.1	141.7	137.9
Kind of oil burned		Oil from Los Angeles Fields.							
Specific gravity of oil at 60° F.	S.G.	0.9770	0.9770	0.9763	0.9770	0.9776	0.9776	0.9797	0.9776
Specific gravity of oil, Beaume, at 60° F.	S.G.	13.3	13.3	13.4	13.3	13.2	13.2	12.9	13.2
Moisture in oil	Per Cent.	0.4	0.5	0.45	0.4	0.8	0.65	0.6	0.54
Heat value of oil as fired	B.t.u.	18,280	18,256	18,131	18,253	18,214	18,171	17,985	18,184
Heat value of oil corrected	B.t.u.	18,353	18,347	18,212	18,326	18,357	18,289	18,093	18,281
Total oil as fired	Lbs.	6,913	8,758	10,322	10,115	12,602	16,580	20,205	12,213
Total oil corrected for moisture	Lbs.	6,885	8,714	10,276	10,075	12,501	16,472	20,084	12,144
Oil per hour as fired	Lbs.	987	1,251	1,474	1,445	1,800	2,369	2,887	1,745
Oil per hour, corrected	Lbs.	983	1,245	1,467	1,439	1,786	2,353	2,869	1,735
Water evaporated F. and A. 212° F. per square F.H.S.	Lbs.	2.58	3.24	3.78	3.77	4.58	5.63	6.74	4.32
Boiler horse power, builders' rating	B.h.p.	604	604	604	604	604	604	604	604
Boiler horse power developed	B.h.p.	439.3	568.0	661.3	659.6	802.2	986.2	1181.0	756.8
Per cent, builders' rating	Per Cent.	72.7	94.0	109.4	109.2	132.8	163.3	195.5	125.3
Water evaporated per pound of oil from and at 212° F.: Oil as fired	Lbs.	15.35	15.66	15.47	15.75	15.37	14.37	14.12	15.15
Water evaporated per pound of oil from and at 212° F.: Corrected for moisture	Lbs.	15.41	15.74	15.54	15.81	15.49	14.46	14.20	15.23
Boiler efficiency	Per Cent.	81.1	82.8	82.4	83.3	81.5	76.4	75.8	80.47

50. Comparative Evaporative Economy of Oil and Coal. — In determining the comparative economy of coal and oil, the fixed and operating charges must be considered in addition to the cost and efficiency of the fuel. From the market quotation on oil and coal and the comparative heating values of each the actual cost per B.t.u. is readily obtained, and by combining this with the relative efficiencies from the furnace standpoint the net cost of the fuel is obtained. The fixed charges vary with the location and size of the plant and are approximately the same per boiler horsepower for a given location in both cases. The insurance rates may be greater with the oil fuel and the depreciation of the boiler setting may be somewhat larger, but in a well-constructed furnace the latter item should be the same in both instances for average rates of combustion. The operating charges are decidedly in favor of the oil fuel, since no ash handling is necessary. Oil fuel is readily fed to the furnace, and the cost of attendance may be materially less than with coal firing, and one man may safely control from eight to ten boilers.

51. Oil Burners. — The function of the burner is to atomize the oil to as nearly a gaseous state as possible.

Classification of a few well-known burners

Mechanical Spray:
- Körting.

Vapor or Carburetor:
- Durr.
- Harvey.

Spray Burners:
- Outside Mixers.
  - *a.* Peabody.
  - *b.* Warren.
- Inside Mixers.
  - *a.* Hammel.
  - *b.* Kirkwood.
  - *c.* Branch.
  - *d.* Williams.

Oil burners for burning liquid fuel may be divided into three general classes:

1. Mechanical spray, in which the oil, previously heated to a temperature of about 150 deg. fahr., is forced under pressure through nozzles so designed as to break it up into a fine spray. The Körting Liquid Fuel Burner, Fig. 20, is an example of this type. In this design a central spindle, spirally grooved, imparts a rotary motion to the oil and causes it to fly into a spray by centrifugal force on issuing from the nozzle. The particles of oil are burned in the furnace

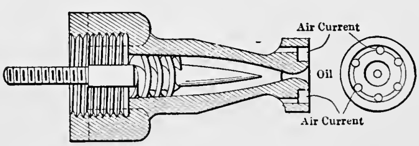


Fig. 20. Körting Fuel Oil Burner.

when they come in contact with the necessary air to effect combustion. This type of burner is little used in this country in connection with power-plant work, but is meeting with much success in Europe.

2. Vapor burners, or carburetors, in which the oil is volatilized in a heater or chamber and then admitted to the furnace, are seldom used except in connection with refined oils, as the residuals from crude oil are vaporized only at a high temperature. The Durr and Harvey gasifiers are the best known of this type.

3. Spray burners are by far the most common in use. In this type the oil is held in suspension and forced into the furnace by means of a jet of steam or compressed air. Spray burners are designed either as *outside mixers*, in which the oil and atomizing medium meet outside the apparatus, or *inside mixers*, in which the oil and atomizing medium mingle inside the apparatus.

The *Peabody burner*, Fig. 21, illustrates the principles of the "outside-mixer" type of apparatus. In this type the oil flows through a thin slit and falls upon a jet of steam which atomizes it and forces it into the furnace in a fan-shaped spray. A feature of this apparatus is its simplicity of construction.

Fig. 22 illustrates the Hammel burner as used at the power house of the Pacific Light and Power Company, Los Angeles, Cal. Oil enters the burner under pressure and flows through opening *D* to the mouth of the burner, where it is atomized by the steam jets issuing from slots *G*, *H*, and *I*. The oil is preheated to facilitate its flow through the supply system. Plates *K-K* are removable and are easily replaced when worn out or burned. The Hammel burner belongs to the "inside mixers."

A few well-known types of "inside mixers" are illustrated in Figs. 22 to 24. The operation is practically the same in all of them and they differ only in mechanical details.

The simplest and most reliable burners are of the Hammel type and are much in evidence in the Pacific States.

52. Furnaces for Burning Fuel Oil. — The efficient combustion of oil fuel depends more upon the proportions of the furnace than upon the type of burner, provided, of course, the latter is of modern design. While it is desirable to have incandescent brickwork around the flame it is impossible to do so in many cases and a satisfactory compromise is effected by using a flat flame burning close to a white-hot floor through which air is steadily flowing. A good burner will maintain a suspended flame clear and smokeless in a cold furnace. The path of the flame in the furnace must be such as to insure uniform distribution of heat over the boiler heat-absorbing surfaces without direct flame impingement. Under ordinary firing the flame should not extend into the

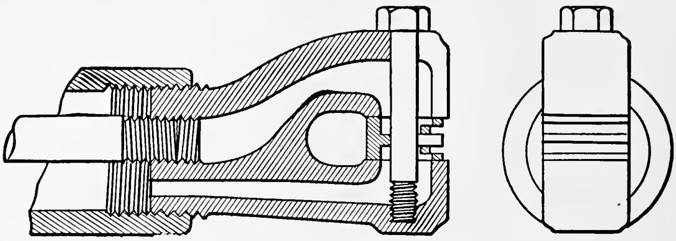


Fig. 21. Peabody Fuel-oil Burner.

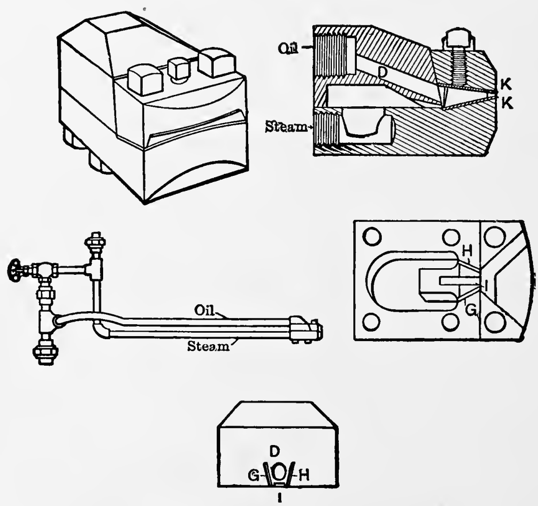


Fig. 22. Hammel Fuel-oil Burner.

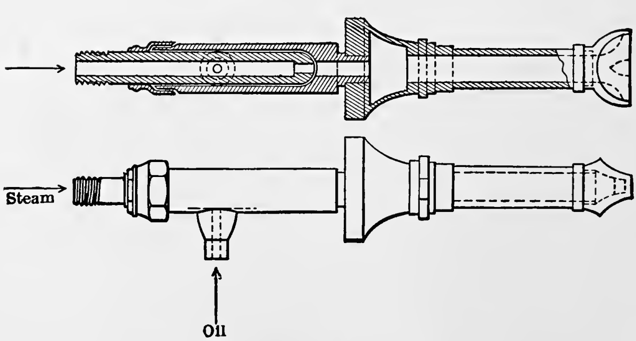


Fig. 23. Branch Fuel-oil Burner.

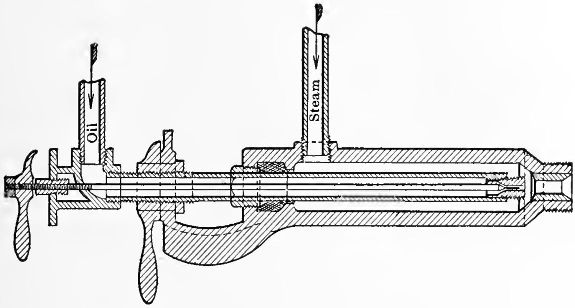


FIG. 24. Kirkwood Fuel-oil Burner.

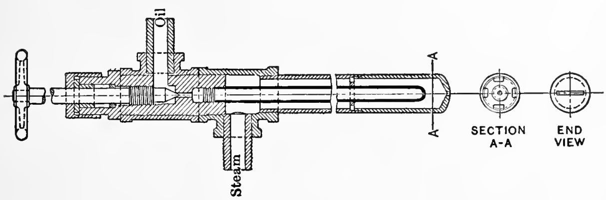


FIG. 25. Billow Type of Fuel-oil Burner.

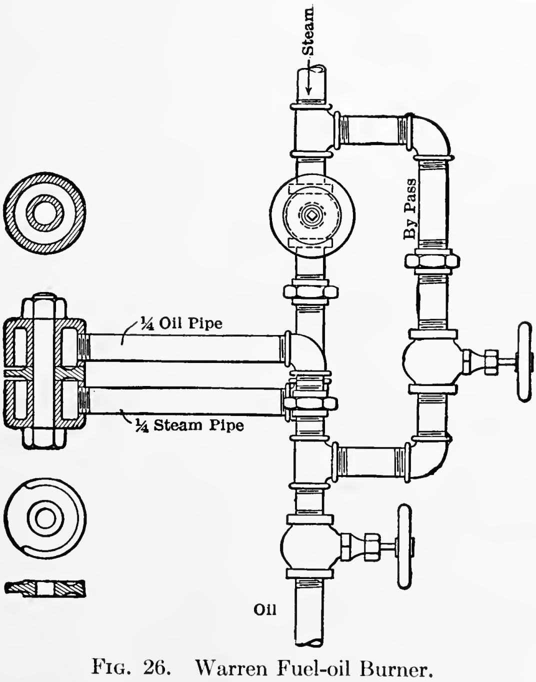


FIG. 26. Warren Fuel-oil Burner.

tubes. The first pass of the boiler should be located directly over the furnace in order that the heating surface may absorb the radiant energy from the incandescent fire brick. Fire-brick arches and target walls are not to be recommended on account of the localization of heat resulting in burning out the tubes or bagging the shell and on account of the limited overload capacity.

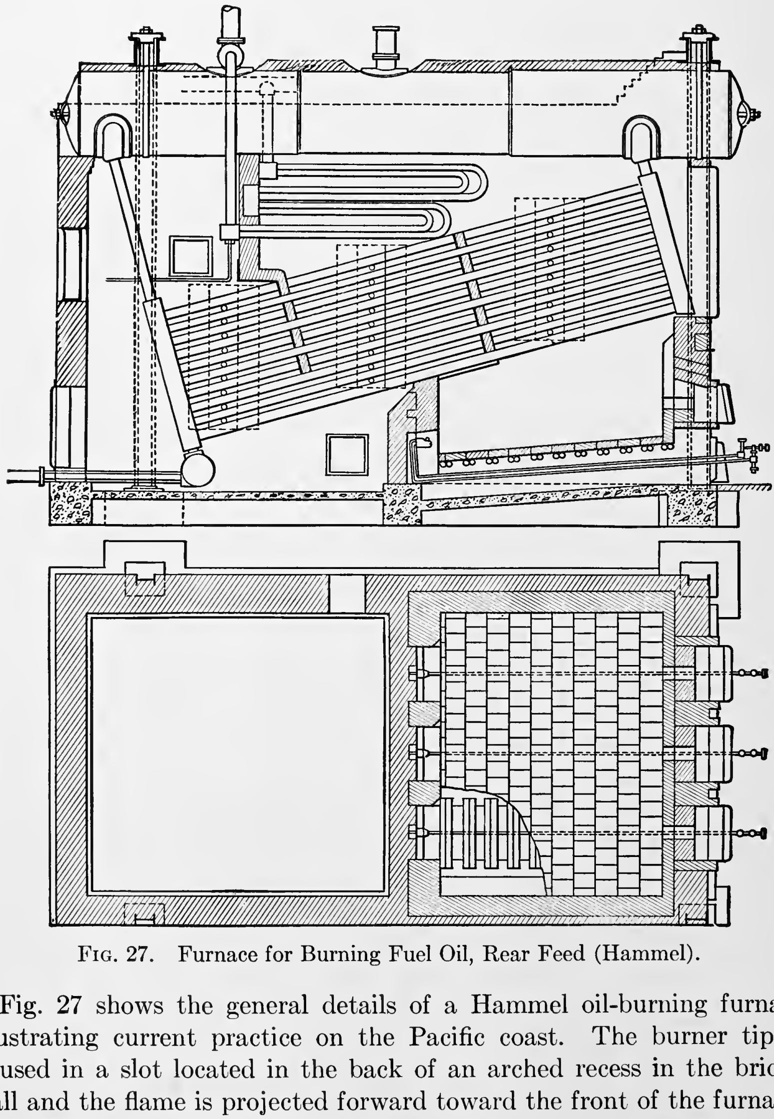


Fig. 27. Furnace for Burning Fuel Oil, Rear Feed (Hammel).

Fig. 27 shows the general details of a Hammel oil-burning furnace illustrating current practice on the Pacific coast. The burner tip is housed in a slot located in the back of an arched recess in the bridge wall and the flame is projected forward toward the front of the furnace. The furnace floor is carried on pieces of old two-inch pipe or on old rails and is solid except for narrow air slots through the deck and in front of each arch. Each burner with its accompanying recess has a separate air tunnel from the boiler front; these tunnels do not communicate with each other under the furnace floor and by closing the ashpit door any tunnel can be sealed up while the others are supplying air to their particular burners. The Hammel furnace is a modification of the well-known Peabody furnace, a section through which is shown in Fig. 28.

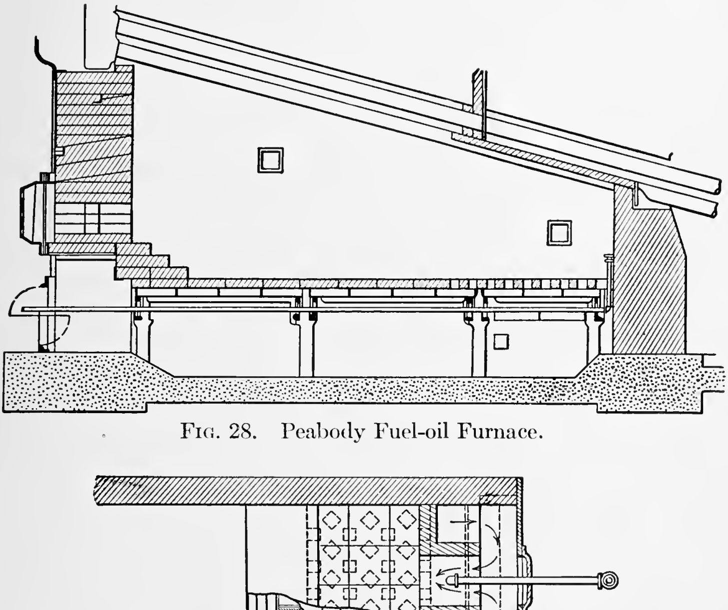


Fig. 28. Peabody Fuel-oil Furnace.

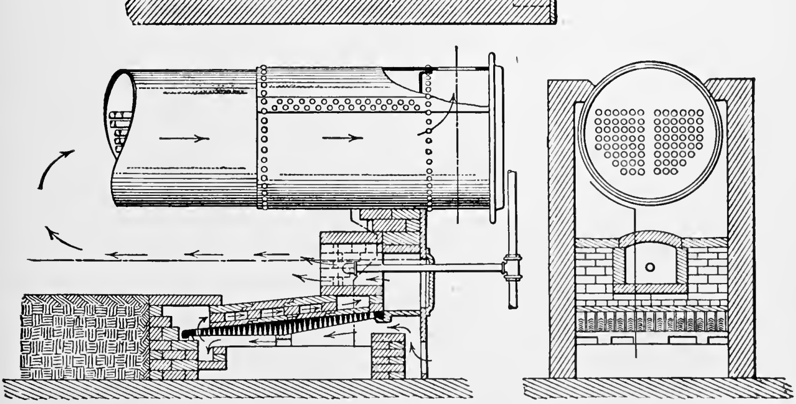


Fig. 29. Modern Furnace for Burning Fuel Oil, Front Feed.

TABLE 27.

TESTS OF FUEL-OIL BURNERS.

Number of Test.	Authority of Test.	Reference.	Type of Burner.	Steam Pressure (Pounds per Square Inch Gauge).	Pressure of Medium used for Spraying Oil.	Temperature of Spraying Medium (Deg. Fahr.).	Equivalent Evaporation from and at 212 Deg. Fahr. (Pounds).	Steam used in Spraying Oil; Per Cent of Total Steam Generated.	Conventional Efficiency of Boiler (Per Cent).
1			O. C. B. W. (Air).....	273.5	3.2	......	12.7	2.38	62.8
2		Report of U. S. Naval Liquid Fuel Board. 1902.	......do..............	273.5	4.62	......	12.18	3.89	60.3
3			......do..............	273.5	0.78	102.5	14.43	1.06	71.5
4			......do..............	273.5	3.37	122	11.73	2.88	58.1
5		Engineer U. S., June 1, 1903..	......do..............	273.5	1.41	120	14.22	1.97	70.4
6			......do..............	271.5	1.31	113.5	14.12	1.53	69.9
7	United States Naval Board.	Engineering Record, December 20, 1902.	......do..............	272.5	4.66	161	13.29	7.45	65.8
8			......do..............	276	4.68	136	10.77	4.25	53.4
9		Western Electrician, December 13, 1902.	Hayes (Steam).........	273.5	32	......	13.89	5.77	68.9
10			O. C. B. W. (Steam)..	273.1	29.9	444	13.47	3.98	66.7
11		Engineer, London, Nov. 14, 1902.	......do..............	273.7	61.4	408	13.45	4.41	66.7
12			......do..............	274.2	81	401	13.58	5.03	67.3
13			Reed (Air and Steam)	276.7	92	375	14.35	8.54	71.1
14			......do..............	277.4	89	416	14.06	6.09	69.7
15	Wallsend................	Engng., Nov. 6, 1903.........	Körting*.............	113	75	240‡	14.45	......	......
16	Denton..................	Power, February, 1902........	Williams (Steam).....	86.5	......	......	15.49	3.5	78.5
17			H. & M. (Steam)......	156	36†	109‡	15.66	2.33	80.6
18	Pacific Light and Power Co., Los Angeles, Cal.	Tests by B. & W. and Stirling Companies.	Hammel (Steam)......	156	35†	88‡	14.87	2.72	70.7
19			......do..............	156.4	36†	109‡	15.85	1.96	83
20	B. R. T. Collins.........	Jour. A.S.M.E., Aug., 1911...	Peabody (Steam).....	142	......	97	14.38	1.54	81
30	Armour & Co...........	P. W. Evans................	Billow (Steam)........	144	......	......	15.93	1.01	78.7

* Uses no steam or air. Oil under pressure. † Oil pressure. ‡ Temperature of oil.

Fig. 29 gives the general details of a modern oil-burning furnace, with front feed, as applied to a horizontal return tubular boiler.

53. Atomization of Oil. — For efficient combustion the oil should be injected into the furnace in the form of a spray. Three systems of atomization are in use in stationary practice, namely, mechanical, air, and steam. Of these, by far the greater number of installations in the United States are of the last order.

The mechanical or Körting system is not much in evidence in this country, but it is used extensively in Europe. The operation of this system is described in paragraph 51. The makers state that to operate the pumps and supply the heat to the oil takes from $\frac{3}{4}$ to 1 per cent of the steam evaporated. Mechanical atomization presents many possibilities and it is not unlikely that future development may lie along this path.

In air atomization the air is used at pressures from $1\frac{1}{2}$ to 60 pounds per square inch, depending upon the type of burner. From Table 27 it will be seen that the total steam used to compress the air varies from 1.01 to 7.45 per cent of the total generated. For air atomization and with air pressures of from 20 to 30 pounds per square inch, J. H. Hoppes (Jour. A.S.M.E., Aug., 1911, p. 902) states that from 6 to 10 cubic feet of air per minute per pound of oil burned will be required. Compressed air offers no opportunity for fuel saving over the use of steam direct in cases where steam is available. In certain industrial operations where high temperatures are essential the use of air is preferred. When it is necessary to use high-pressure air the economy decreases with the increase in pressure, since the cost of each cubic foot of compressed air increases rapidly with the pressure, but its ability to atomize the oil does not increase proportionately.

Steam is the most commonly used medium for atomizing the oil, since its use obviates complication and risk of interrupted service. The amount of steam required to atomize the oil varies from 0.15 to 0.7 pounds per pound of oil, with an average of about 0.4 pounds. The steam consumption is generally stated in per cent of the total steam generated, but the results are misleading since the percentage factor depends largely upon the efficiency of the boiler. Table 27 gives the results of a number of tests of different types of burners with air and steam as atomizing mediums.

54. Oil-feeding Systems. — Fig. 30 gives a diagrammatic arrangement of the piping commonly employed in feeding oil fuel to the burners. Steam-actuated oil pumps, installed in duplicate, draw the fuel from the supply tank and deliver it under pressure to the burners. The piping is cross-connected so that repairs can be made without inter-

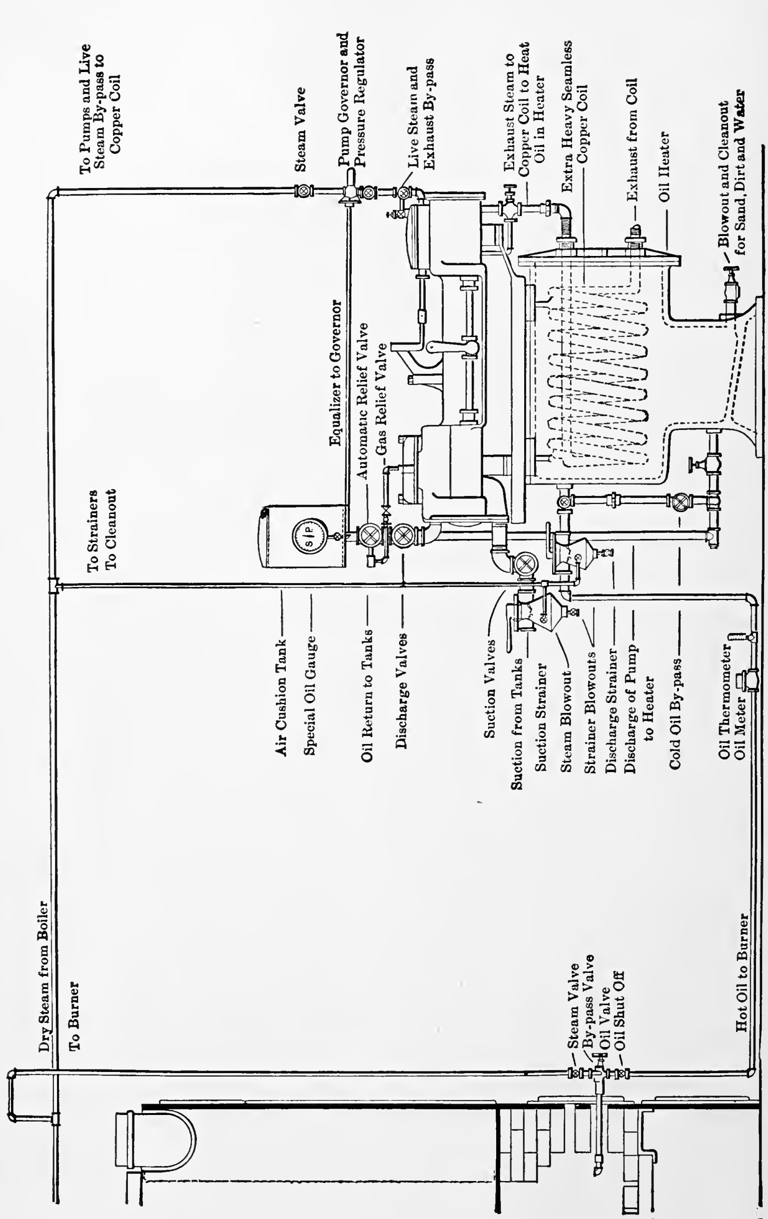


FIG. 30. Diagrammatic Arrangement of Piping for Fuel-oil System.

rupting the service. The oil is heated from the pump exhaust before it is supplied to the burners. This should not be carried beyond the flash point of the oil used or there will be danger from carbon deposits in the supply pipe. A strainer is placed in the suction line between the storage tank and the oil-pressure pump to minimize clogging of the burner. In some instances strainers are also placed in the supply pipe between the heater and burner. The relief valve between the pumps and burners is set at a definite maximum oil pressure so as to prevent excessive pressure. The oil meter is for the purpose of checking the storage tank indicator. All oil piping is installed so that it can be drained back to the storage tank by gravity in case of necessity. In many large plants the strainers, meters, heaters and piping are installed in duplicate. Arrangements are usually made for the oil to be delivered at constant pressure. The supply of steam to the burner is controlled by regulating the pressure in a separate main common to all burners, the pressure in the main bearing a certain predetermined relation to the pressure in the oil mains. In most installations the supply of steam and oil at the burner is regulated by hand to meet the requirements of the individual burners. At the Redondo plant of the Pacific Light and Power Company, Redondo, Cal., the supply of oil and steam to all burners and the supply of air for combustion to any number of boilers are automatically controlled from a central point. For a description of this system see Trans. A.S.M.E., Vol. 30, p. 808.

Low-pressure systems are ordinarily operated under standpipe pressures as in Fig. 31, which illustrates the arrangement of apparatus as advocated by the International Gas and Fuel Company. A steam pump B draws the oil from the buried tank through pipe Z and delivers it to the standpipe E. Thence it flows through pipe I to the burners under a head of about 10 feet. The pump runs constantly, the surplus oil flowing back to the tank through the pipe T. The oil is heated by the exhaust pipe Z'. The oil pump is provided with a device D having a piston connected by a chain with a cock S, which automatically opens when the boiler is not under steam pressure, so that the standpipe will be emptied, the oil flowing to the storage tank.

Fig. 32 illustrates the Hydraulic Oil Storage Company's system of storing oil and delivering it to the burners. The oil reservoirs are placed below grade, as indicated, to minimize fire risk. The operation is as follows: Water enters the "float box" and flows through a "three-way cock" to the bottom of the reservoir until all of the oil and water-pipes are filled up to the level of the float box, when the float automatically cuts off the supply. This flooding of the entire system drives

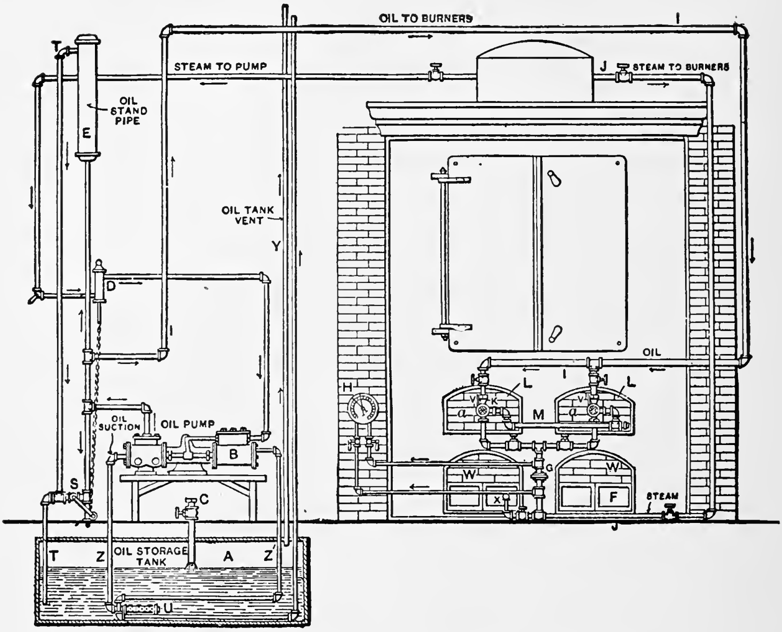


FIG. 31. International Gas and Fuel Company's Fuel-oil System.

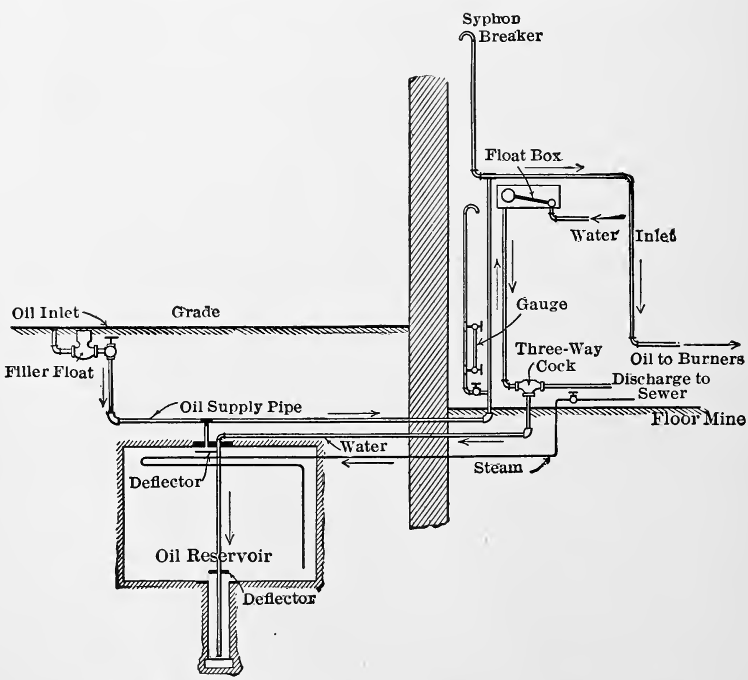


FIG. 32. Hydraulic Oil Storage Company's Fuel-oil System.

out all of the air. The three-way cock is then turned to "discharge" and part of the water flows to the sewer. The tank car or wagon is next attached to the "oil inlet" and the oil flows into the tank and displaces the water until the level of the "filler float" is reached, when the supply is automatically cut off. The inlet is so placed that the head of oil in the tank car is sufficiently great to overcome the opposing head of water. The three-way valve is next turned to the first position and the head of water forces the oil to the burners. After the oil has been withdrawn from the storage tank the water can only rise to the level of the water in the float box and therefore cannot be fed to the furnace. The small steam pipe admits steam into the tank and heats the oil, thereby making it flow more freely.

55. Oil Transportation and Storage. — Fuel oil is delivered in bulk, either in tank cars, barges or steamships, or by pipe lines, depending upon the location of the plant. It must be stored in accordance with underwriters' requirements and community ordinances. The requirements of the National Board of Fire Underwriters as regards the storage and use of fuel oil are substantially as follows:

All oil used for fuel purposes under these rules shall show a flash test of not less than 150 deg. fahr. (Abel-Pensky flash-point tester). This flash point corresponds closely to 160 deg. fahr. (Tagliabue open-cup tester), which may be used for rough estimations of the flash point.

In closely built-up districts or within fire limits, tanks to be located underground with their tops not less than 3 ft. below the surface of 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 3 ft. below the basement floor, which is to be of concrete not less than 6 in. 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. The tanks may have a test well, provided the test well extends to near the bottom of the tank, and the top end shall be hermetically sealed and locked except when necessarily open. When the tank is located underneath a building, the test well shall extend at least 12 ft. above the 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, the permissible aggregate capacity if lower than any floor, basement, cellar or pit in any building within the radius specified being as follows:

Capacity.	Radius.
Unlimited	50 ft.
20,000 gal.	30 ft.
5,000 gal.	20 ft.
1,500 gal.	10 ft.
*500 gal.	Less than 10 ft.

* In this case the tank must be entirely incased in 6 in. of concrete.

When located underneath a building no tank shall exceed a capacity of 9,000 gal., and basement floors must be provided with ample means of support independent of any tank or concrete casing.

Outside of closely built-up districts or outside of fire limits, above-ground storage tanks may be permitted provided drainage away from burnable property in case of breakage of tanks is arranged for or suitable dikes are built around the tanks.

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 the tanks. This requirement prohibits the use of gravity feed from storage tanks. Above-ground tanks of less than 1,000 gal. capacity without dikes may be permitted in case suitable housings for the protection of the tanks against injury are provided.

MATERIAL AND CONSTRUCTION OF TANKS

Tanks must be constructed of iron or steel plate of a gauge depending upon the capacity as specified in the following:

UNDERGROUND TANKS INSIDE SPECIFIED FIRE LIMITS.

Or Within 10 Ft. of a Building When Outside Such Limits.

Capacity, Gal.	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	$\frac{1}{4}$ U. S. gauge
10,501 to 20,000	$\frac{5}{16}$ U. S. gauge
20,001 to 30,000	$\frac{3}{8}$ U. S. gauge

UNDERGROUND TANKS OUTSIDE SPECIFIED FIRE LIMITS.

Provided the Tanks are 10 Ft. or More from a Building.

Capacity, Gal.	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	$\frac{1}{4}$ U. S. gauge
10,501 to 20,000	$\frac{5}{16}$ U. S. gauge
20,001 to 30,000	$\frac{3}{8}$ U. S. gauge

Tanks of greater capacity than 30,000 gal. must be made of proportionately heavier metal. All joints of tanks must be riveted and soldered, riveted and calked, welded or brazed together, or made by some equally satisfactory process. The shells of tanks must be properly reinforced where connections are made and all connections so far as practicable made through the upper side of tanks above the oil level. Tanks shall be thoroughly coated on the outside with tar, asphaltum or other suitable rust-resisting material.

FILL AND VENT PIPES

Each underground storage tank having a capacity of over 1,000 gal. must be provided with at least a 1-in. vent pipe extending from the top of the tank to a point outside the building, and to terminate at a point at least 12 ft. above the level of the top of the highest tank car or other reservoir from which the storage tank may be filled. The terminal must be provided with a hood or gooseneck protected by a noncorrodible screen and be placed remote from fire escapes and never nearer than 3 ft., 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 1 ft. above the level of the source of supply.

Tanks having a capacity of less than 1,000 gal. 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. 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. Fill and vent pipes for tanks located under buildings are to be run underneath the concrete floor to the outside of the building.

Suitable filters or strainers for the oil should be installed and preferably be located in the supply line before reaching the pump. Filters must be arranged so as to be readily accessible for cleaning. Feed pumps must be of approved design, secure against leaks and be arranged so that dangerous pressures will not be obtained in any part of the sytem. It is further recommended that feed pumps be interconnected with pressure air supply to burners to prevent flooding.

Glass gages, 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 the system.

Receivers or accumulators, if used, must be designed so as to secure a factor of safety of not less than 6 and must be subjected to a pressure test of not less than twice the working pressure. The capacity of the oil chamber must not exceed 10 gal. A pressure gage must be provided; also an automatic relief valve set to operate at a safe pressure and connected by an overflow pipe to the supply tank, and so arranged that the oil will automatically drain back to the supply tank immediately on closing down the pump.

If standpipes are used, their capacity shall not exceed 10 gal. They must 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 the pump, leaving not over 1 gal., where necessary, for priming, etc. If vented, the opening should be at the top and may be connected with the outside vent pipe from the storage tank, above the level of the source of supply.

Piping must be run as directly as possible and pitched toward the supply tanks without traps. Overflow and return pipes must be at least one size larger than the supply pipes, and no pipe should be less than ½-in. pipe size. Connection to outside tanks should be laid below the frost line and not placed near nor in the same trench with other piping.

Readily accessible shutoff valves should be provided in the supply line as near to the tank as practicable, and additional shutoffs installed in the main line inside the building and at each oil-consuming device.

Controlling valves in which oil under pressure is in contact with the stem shall be provided with a 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. The use of approved automatic shutoffs for the oil supply in case of breakage of pipes or excessive leakage in the building is recommended.

56. The Purchase of Fuel Oil. — The following extracts from Bulletin No. 3, 1911, Bureau of Mines ("Specifications for the Purchase of Fuel Oil for the Government, with Directions for Sampling Oil and Natural Gas"), though primarily intended for the guidance of Government officials, may be of service to engineers:

1. In determining the award of a contract, consideration will be given to the quality of the fuel offered by the bidders, as well as the price, and should it appear to be the best interest of the Government to award a contract at a higher price than that named in the lowest bid or bids received, the contract will be so awarded.
2. Fuel oil should be either a natural homogeneous oil or a homogeneous residue from a natural oil; if the latter, all constituents having a low flash point should have been removed by distillation; it should not be composed of a light oil and a heavy residue mixed in such proportions as to give the density desired.
3. It should not have been distilled at a temperature high enough to burn it nor at a temperature so high that flecks of carbonaceous matter began to separate.
4. It should not flash below 140 deg. fahr. in a closed Abel-Pensky or Pensky-Martens tester.
5. Its specific gravity should range from 0.85 to 0.96 at (59 deg. fahr.); the oil should be rejected if its specific gravity is above 0.97 at that temperature.
6. It should be mobile, free from solid or semi-solid bodies, and should flow readily at ordinary atmospheric temperatures and under a head of 1 foot of oil, through a 4-inch pipe 10 feet in length.
7. It should not congeal or become too sluggish to flow at 32 deg. fahr.
8. It should have a calorific value of not less than 10,000 calories per gram (18,000 B.t.u. per pound); 10,250 calories to be the standard. A bonus is to be paid or a penalty deducted according to the method stated under section 21, as the fuel oil delivered is above or below this standard.
9. It should be rejected if it contains more than 2 per cent water.
10. It should be rejected if it contains more than 1 per cent sulphur.
11. It should not contain more than a trace of sand, clay or dirt.
12. Each bidder must 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 a crude oil, a refinery residue, or a distillate.
d. The name and location of the refinery, if the oil has been refined at all.

For sampling, analysis, etc., consult complete bulletin.

Analyses of California Petroleums: Bulletin No. 19, U. S. Bureau of Mines, 1912.
Atomization: Jour. A.S.M.E., Aug. 11, 1911, p. 883, 902; Jour. El. Power and Gas, Dec. 23, 1911.
Burners: Jour. El. Power and Gas, Dec. 23, 1911, Apr. 1, 1911; Engng., Feb. 16, 1912; Power, Jan. 27, 1914, p. 139.
Comparative Evaporative Value of Coal and Oil: Jour. El. Power and Gas, March 18, 1911; Jour. A.S.M.E., Aug. 11, 1911, p. 872.
Draft Requirements for Burning Oil Fuel: Jour. A.S.M.E., Aug., 1911; Oct., 1912.
Economy Tests with Oil Fuels: Trans. A.S.M.E., 30–1908, p. 775; Jour. A.S.M.E., Aug. 11, 1911, p. 940.
Furnaces for burning Oil Fuel: Jour. El. Power and Gas, Dec. 30, 1911, Apr. 8, 1911; Jour. A.S.M.E., Aug., 1911, p. 879; Ir. Td. Review, June 3, 1908; Power, June 16, 1908.
Oil Fuel: Prac. Engr., July 15, 1916, p. 607.
Oil for Steam Boilers: Jour. A. S.M.E., Aug., 1911, p. 931; Jour. El. Power and Gas, Dec. 16, 1911; Power, Aug., 1908, p. 943; Jan. 23, 1908, p. 980; Bulletin No. 131, Louisiana State University.
Precautions with Oil Fuel: Eng. and Min. Jour., Apr. 1, 1911, p. 653.
Purchase of Fuel Oil for the Government: Bulletin No. 3, Bureau of Mines, 1911.
Regulation of Oil Supply to Burners: Trans. A.S.M.E., 30–1908, p. 804.
Storage and Transportation: Jour. El. Power and Gas, Dec. 16, 1911, p. 564; Eng. News, Sept. 25, 1902, p. 232; Power, July 16, 1908.
Unnecessary Losses in Firing Fuel Oil: Trans. A.S.M.E., 30–1908, p. 797.

57. Gaseous Fuels. — The most commonly used gaseous fuels for steam generating purposes are natural gas, blast furnace gas and by-product coke oven gas.

Natural gas is an ideal fuel for steam generation and offers all of the advantages of solid and liquid fuels and none of the disadvantages. No storage bins or reservoirs are necessary, ashes are absent and standby losses may be reduced to a minimum. In the immediate locality of natural gas wells, gas fired furnaces may prove to be more economical than coal furnaces but the limited supply restricts its use as a general fuel. A large combustion space is essential and a volume of 0.75 cubic feet per rated boiler horsepower will be found to give good results. The best results are obtained by employing a large number of small burners, each capable of handling 30 nominal rated horsepower. The use of a number of small burners obviates the danger of stratification of the gases which might occur with the large burners. A typical burner is illustrated in Fig. 33. A satisfactory working pressure is about 8 ounces at the entrance of the burner. Table 28 gives typical analyses and calorific values of natural gas.

Although *blast-furnace gas* is used extensively in gas engines its application to steam power generation is by no means discontinued, since the first high cost of a gas engine equipment, space requirements and high maintenance and attendance charges may more than offset

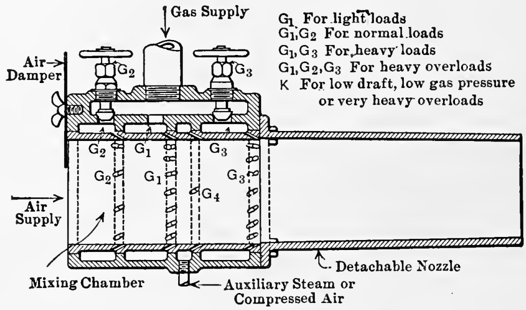


FIG. 33. Gwynne Improved Gas Burner.

the high thermal efficiency. A furnace volume of approximately 1 to 1.5 cubic feet per rated boiler horsepower gives satisfactory results. The burner illustrated in Fig. 33 may be adapted to blast furnace gas by increasing the size of gas openings. On account of the possibility of a pulsating action of the gases and the resulting puffs or explosions, settings for this class of work should be carefully constructed and thoroughly buck staved. Blast furnace gas is very dirty and ample provision should be made for removing the dust not only from the furnace but from the setting as a whole. Table 28 gives the chemical constituents and physical characteristics of a typical blast furnace gas.

By-product coke oven gas has a higher calorific value than blast furnace gas but requires about the same type of burner and furnace design as the latter. It is ordinarily burned under a pressure of four inches of water. By-product coke oven gas is saturated with water vapor as it leaves the oven and provision should be made for removing the water of condensation before it reaches the burner. Tar and other hydrocarbons, which are present in considerable amount, tend to deposit in the burners and render them inoperative. Accumulation of this deposit is ordinarily prevented by "blowing out" the burners with steam.

The Utilization of Waste Heat for Steam Generating Purposes, Jour. A.S.M.E., Nov., 1916, p. 859.

Fig. 34 shows a section through a small experimental boiler designed by Prof. Wm. A. Bone, University of Leeds, England, which involves the principle of so-called "surface combustion," and for which extravagant claims have been made as regards efficiency and capacity.

TABLE 28.

CHARACTERISTICS OF GASEOUS FUELS.

(Ulbright and Torrance; Power, Aug. 6, 1912).

Gas.	Average Constituents of Gas — in Per Cent Volume.									Theoretical Calorific Value per Cu. Ft. of Gas at 62° and Atmospheric Pressure, B.t.u.		Calorific Value per Cu. Ft. by test, B.t.u.		Theoretical Air per Cu. Ft. Gas.	B.t.u. per Cu. Ft. of Mixture.		Calculated B.t.u. per Cu. Ft. of Mixture, Theoretical Air.		Calculated B.t.u. per Cu. Ft. of Mixture — Excess Coef. as Below.				
	CO_2	CO	H	CH_4	C_2H_6	C_2H_4	C_6H_6	N	O	High.	Low.	High.	Low.		High.	Low.	High	Low.	1.1	1.2	1.3	1.4	1.5
Producer:																							
Anthracite	6.03	22.38	13.38	1.96	.....	0.36	.....	55.65	0.803	138.4	125.7	142	131.3	1.053	67.9	64.1	67.5	61.3	58.2	55.4	53	50.8	48.7
Bituminous	9.12	17.54	11.73	4.28	.....	0.36	.....	57.24	0.265	137	125	154	136	1.120	(72.7)	47.8	64.6	59.0	56	53.4	50.8	48.7	46.6
Coke	4.90	27.30	10.07	1.18	.....	.....	.....	56.60	0.550	128.7	126.3	133.7	(125)	0.985	(67.4)	(63)	65.0	63.7	60.7	57.8	55.4	53.1	51.0
Lignite	9.43	18.90	15.13	3.65	.....	0.424	.....	52.50	0.582	144	134	156.4	.....	1.145	(72.8)	.....	67.2	62.5	59.3	56.4	53.8	51.5	49.3
Oil	4.10	11.40	5.57	5.87	.....	3.10	.....	66.90	3.060	176	151	.....	.....	0.990	.....	.....	88.4	75.9	72.2	69.0	66.0	63.2	60.7
Peat	12.40	21.00	18.50	2.20	.....	0.40	.....	45.50	0.000	153	141	175	.....	1.170	(80.6)	.....	70.6	65.0	61.6	58.8	56.3	53.5	51.2
Wood	13.90	20.03	21.00	2.79	.....	0.60	.....	41.80	0.185	137	128.7	138.7	.....	1.070	(67.0)	.....	66.2	62.2	59.2	56.3	53.8	51.5	49.4
Illuminating:																							
Water	4.72	34.8	48.81	4.06	.....	1.97	.....	7.16	0.503	322.1	304	329	278	2.63	(90.6)	(76.6)	88.8	83.8	78.0	73.2	68.8	64.8	61.5
Carbureted water	2.07	24.1	32.40	23.40	.....	12.52	.....	3.75	0.510	646.4	589	677	632	6.84	99.1	90.94	82.4	75.1	69.0	63.9	59.5	55.7	52.3
Coal	1.21	6.18	43.94	37.78	.....	5.87	4.16	3.50	0.502	677	608	655	592	6.00	97.8	86.40	96.7	86.8	80.0	74.2	69.2	64.7	60.8
Natural:																							
Average	0.684	0.647	10.89	79.67	.....	1.26	6.40	5.89	0.79	948	853	965	855	9.85	96.2	87.2	95.2	85.7	78.5	72.7	67.5	63.0	59.1
High hydrogen	0.580	0.730	20.56	50.30	.....	2.07	5.88	8.80	1.13	923	834	895	810	8.66	(92.6)	(83.8)	95.5	86.3	79.2	73.1	68.0	63.5	59.5
Low hydrogen	0.800	0.525	1.92	91.40	.....	0.515	.....	3.25	0.50	967	870	971	862	9.20	96.2	87.2	94.7	85.2	78.3	72.4	67.1	62.7	58.8
Blast furnace	11.80	26.75	3.40	0.30	.....	.....	.....	58.8	0.182	98.7	95.2	98.7	.....	0.735	55.3	54.8	56.8	54.8	52.7	50.5	48.6	46.8	45.2
Coke oven	2.1	6.51	51.56	32.9	(1.4)	2.5	.....	4.0	0.36	577.0	487.0	598.0	506	4.76	96.7	84.0	100.0	84.5	78.2	72.4	67.8	63.5	59.3
Oil gas	2.79	4.96	17.70	23.60	18.78	5.53	.....	37.55	0.702	595.0	542.0	662.0	.....	4.94	(111.3)	.....	100.0	91.2	84.2	78.2	73	68.4	64.4

It consists essentially of a plain tubular boiler, having ten tubes, 3 inches in internal diameter. Each of these is bushed with a short tube, *E*, of fire clay and is filled for the rest of its length with finely broken refractory material. Mixing chambers of special design are attached

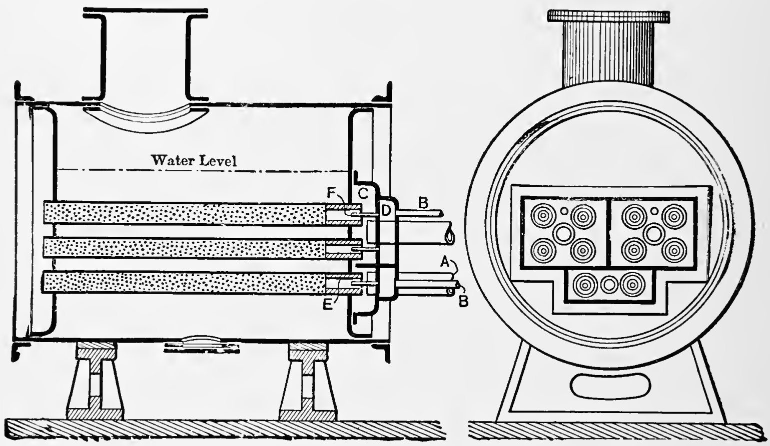


FIG. 34. Experimental Boiler Involving the Principles of "Surface Combustion."

to the front plate of the boiler as indicated. The mixture fed into the boiler tubes from these mixing chambers consists of the combustible gas with a proportion of air very slightly in excess of that theoretically required for combustion. The mixture is injected or drawn in through the orifice in the fire-clay plug. The gas burns without flame in the front end of the tube, the incandescent mass being in direct contact with the heating surface. The combustion of the mixture in contact with the incandescent material is completed before it has traversed a length of 6 inches from the point of entry of the tube. Although the core of the material at this part of the tube is incandescent the heat transference is so rapid that the walls of the tubes are considerably below red heat. The evaporation in regular working order is over 20 pounds per square foot of heating surface and this can be increased 50 per cent with a reduction in efficiency of only 5 or 6 per cent. The figures given by Prof. Bone for the boiler and economizer are as follows:

Date, Dec. 8, 1910.

Pressure of mixture entering boiler tubes, inches of water	17.3
Pressure of products entering economizer, inches of water	2.0
Steam pressure, pounds per square inch gauge	100.0
Temperature of steam in boiler, deg. fahr.	334.0
Temperature of gases leaving boiler, deg. fahr.	446.0
Temperature of gases leaving economizer, deg. fahr.	203.0
Temperature of water entering economizer, deg. fahr.	41.9

Temperature of water leaving economizer, deg. fahr.	136.4
Evaporation per square foot of heating surface per hour, pounds	21.6
Gas consumption, cubic feet per hour, at 32 deg. fahr. and 14.7 pounds per square inch	996.0
B.t.u. per standard cubic foot (lower heat value)	562.0
Water evaporated per hour from and at 212 deg. fahr., pounds	550.0
Efficiency of boiler and economizer (on basis of low heat value), per cent	94.3

For further details of Prof. Bone's experiment see American Gas Light Journal, Dec. 4, 1911; Engineering, April 14, 1911; Engineer (London), April 14, 1911. See also editorial, Industrial Engineering, Jan., 1912, p. 59.

At this writing (1917) practically nothing has been done in this country toward applying Prof. Bone's principles to steam boilers. For results of recent work in surface combustion see Power, Feb. 13, 1917, p. 225.

Burning Natural Gas under Boilers: Power, Oct. 22, 1912, p. 897.
Coke Oven Gas as a Fuel: Power, Aug. 29, 1916, p. 310.

PROBLEMS.

1. The following analyses were obtained from a sample of Illinois coal "as received":

Proximate Analysis.

	Per Cent.
Moisture	12.39
Volatile matter	36.89
Fixed carbon	41.80
Ash	8.92
	100.00

Ultimate Analysis.

	Per Cent.
Hydrogen	5.85
Carbon	61.29
Nitrogen	1.00
Oxygen	19.02
Sulphur	3.92
Ash	8.92
	100.00

a. Transfer these analyses to the "moisture free" and "moisture and ash free" basis.

b. Transfer the ultimate analysis to the "moisture, ash and sulphur free" basis.

c. Determine the free "hydrogen," "combined moisture" and "total moisture."

d. Calculate the ultimate analysis from the proximate analysis.

2. If the moisture and ash contents of an Illinois coal are 8 per cent and 12 per cent respectively, approximate the ultimate analysis by Evans' method. (See Example 4.)

3. Using Dulong's formula calculate the calorific value of the dry coal as per analysis given in Problem 1.

4. Using Dulong's formula approximate the calorific value of the coal as received considering the calculated values of the ultimate analysis. (See Example 5.)

5. Using the data in Problem 1, calculate the theoretical air requirements per pound of coal as fired.

6. Required the character and amount (by weight) of the products of combustion resulting from the complete combustion of the coal designated in Problem 1 with theoretical air requirements.

7. Same data as in Problem 6. Determine the per cent by volume of the CO_2 in the flue gas.

8. Determine the weight of dry air supplied per pound of coal as fired, analysis as in Problem 1, if the flue gas resulting from the combustion is composed of

CO_2	13.00	O_2	5.30
CO	0.44	N_2	81.26

(Per cent by volume)

9. Calculate the theoretical temperature of combustion if the coal as fired, analysis as in Problem 1, is completely burned with 50 per cent air excess.

10. If coke breeze containing 85 per cent carbon and 15 per cent ash is completely burned under a boiler with 50 per cent air excess and the flue gas temperature is 500 deg. fahr., required the heat loss in the flue gas per lb. of fuel as fired if the temperature of the air supply is 80 deg. fahr.

11. If the flue gas resulting from the combustion of the fuel designated in Problem 10 contains 0.5 per cent CO and 12 per cent CO_2 (by volume), required the loss due to incomplete combustion of the carbon.

12. Calculate the heat loss in the refuse if the coal as fired has an ash content of 15 per cent and the combustible in the dry refuse is 20 per cent of the dry refuse. Calorific value of the combustible in the ash, 13,600 B.t.u. per lb.

13. Required the heat lost per lb. of coal as fired in evaporating the moisture from the coal designated in Problem 1 if the temperature of the flue gas is 500 deg. fahr. and that of the boiler room, 80 deg. fahr.

14. If crude oil containing 14 per cent of hydrogen and 3 per cent of oxygen is burned under a boiler, required the amount of heat lost per lb. of oil due to the formation of water by the combustion of the hydrogen. Flue gas temperature, 450 deg. fahr., temperature of the oil, 120 deg. fahr.

15. The following data were obtained from a boiler evaporation test:

Heat absorbed by the boiler, 70 per cent of the calorific value of the coal as fired.

Analysis of the coal as fired:

	Per Cent.		Per Cent.
Carbon	65	Ash and Sulphur	12
Oxygen	8	Free moisture	10
Hydrogen	4	Nitrogen	1

Calorific value as fired, 10,350 B.t.u. per lb.

Flue Gas Analysis:

	Per Cent.		Per Cent.
CO_2	14.18	CO	1.42
O_2	3.55	N_2	80.85 (by difference)

Temperature of air entering furnace, 80 deg. fahr., temperature of the flue gas, 480 deg. fahr., temperature of the steam in the boiler, 350 deg. fahr., relative humidity of the air entering the furnace, 70 per cent, combustible in the dry refuse, 20 per cent.

a. Calculate the actual losses in per cent of the coal as fired.

b. Calculate the inherent losses in per cent of the coal as fired.

c. Approximate the extent to which the actual losses may be reduced by careful operation and proper design.

16. 800,000 pounds of water are fed into a 72-inch by 20-ft. return tubular boiler during a period of 30 days; total weight of coal fed to furnace, 150,000 lb; coal used in banking fires and in starting up, 35,000 lb.; water "blown off," 1 gauge (4-in.) per day; boiler pressure, 115 lb. per sq. in. gauge; required the extent of the stand-by losses in per cent of the net actual evaporation.

CHAPTER III

BOILERS

58. General. — The modern steam boiler is substantially identical in general design to its prototype of a generation ago. Increased pressure and superheat have necessitated improvements in structural details but changes affecting safety and operation are not to be confused with changes of design. Many of the small hand-fired boilers of today are in every way identical with those of twenty years ago. Great improvements have been made in the design and construction of the furnace and setting, mechanical stokers have been perfected and the maximum size of units has been vastly increased but the boiler proper is basically unchanged.

As affecting fuel economy the boiler equipment is by far the most important part of the power plant and involves the largest share of the operating expenses. It matters little how elaborate, modern, or well designed it may be, skill, good judgment, and continued vigilance are required on the part of the operator to secure the best efficiency.

Of the various types and grades of boilers on the market experience shows that most of them are capable of practically the same evaporation per pound of coal, provided they are designed with the same portions of heating and grate surface and are operated under similar conditions. They differ, however, with respect to space occupied, weight, capacity, first cost, and adaptability to particular conditions of operation and location.

There is a tendency towards standardization of boiler construction by legislation in various communities. In view of the numerous adoptions of the "Standard Specifications for the Construction of Steam Boilers and Other Pressure Vessels" * as formulated by the American Society of Mechanical Engineers, it is not unlikely but that this code will ultimately be the required standard for all communities in the United States. †

59. Classification. — As to design and construction there is an almost endless variety of boilers and furnaces, classified as *internally* and

* Trans. A.S.M.E., Vol. 36, 1914, p. 981. Also printed in pamphlet form.

† These rules do not apply to boilers which are subject to federal inspection and control, including marine boilers, boilers of steam locomotive and other self-propelled railroad apparatus.

externally fired; water tube and *fire tube; through tube* and *return tubular; horizontal* and *vertical.*

The internally fired type includes the *vertical tubular, locomotive, Scotch-marine,* and practically all *flue boilers.* The externally fired includes the *plain cylinder,* the *through tubular, return tubular,* and nearly all stationary *water-tube* boilers. A few well-known types will be described in detail.

60. Vertical Tubular Boilers. — Figs. 1 and 35 illustrate typical fire-tube boilers of the internally fired class. The type shown in Figs. 1 and 35 are commonly used where small power, compactness, low first cost and sometimes portability are chief requirements. They are seldom constructed in sizes over 100 horsepower. The tubes are placed symmetrically with a continuous clear space between them and the spaces crossing the tube section at right angles, by means of which the tubes and tube sheet may be readily cleaned. Two styles are in common use; the *exposed tube,* Fig. 1, and the *submerged tube.* In the former the tube sheet and the upper portion of the tubes are exposed to the steam and in the latter they are completely submerged. According to the A.S.M.E. Boiler Code, not less than seven hand holes or wash out plugs are required for boilers of the exposed tube type; three in the shell at or about the line of the *crown sheet,* one in the shell at or about the fusible plug and three in the shell at the lower part of the *water leg.* In the submerged type two or more additional hand holes are required in the shell in line with the upper tube sheet. The distance between the crown sheet and the top of the grate should never be less than 24 inches even in the smallest boiler and should be as great as possible to insure good combustion.

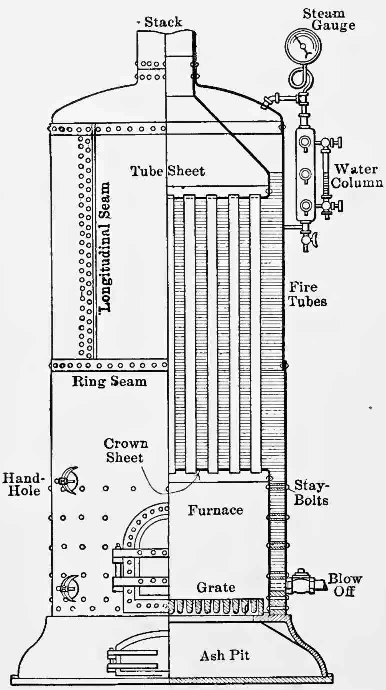


FIG. 35. Vertical Tubular Boiler with Submerged Tube Sheet.

The advantages of this type of boiler are: (1) compactness and portability; (2) requires no setting beyond a light foundation; (3) is a rapid steamer, and (4) is low in first cost. The disadvantages are: (1) inaccessibility for thorough inspection and cleaning; (2) small steam space, which results in excessive priming at heavy loads; (3) poor economy except at light loads, as the products of combustion escape at a high temperature on account of the shortness of the tubes; (4) smokeless combustion practically impossible with bituminous coals; (5) the small water capacity results in rapidly fluctuating steam pressures with varying demands for steam.

Although vertical fire-tube boilers are usually of very small size, being seldom constructed in sizes over 100 horsepower, an exception is found in the Manning boiler, Fig. 36, which is constructed in sizes as large as 250 horsepower. Many of the disadvantages found in the smaller types are obviated in the Manning boilers, which, as far as safety and efficiency are concerned, rank with any of the other first-class types. They differ from the boiler described above mainly in having the lower or furnace portion of much greater diameter than the upper part which encircles the tubes. This permits a proper proportion of grate, which is not obtainable in boilers like Figs. 1 and 35. The double-flanged head connecting the upper and lower shells allows

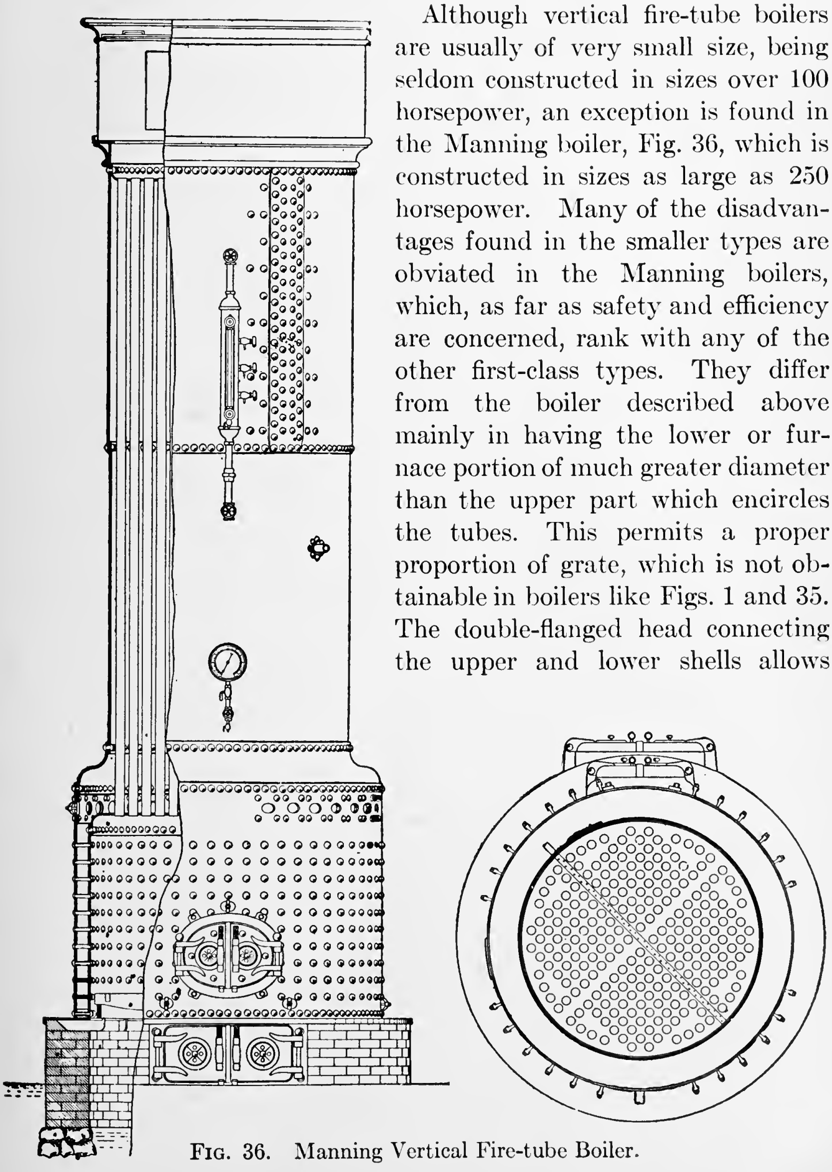


FIG. 36. Manning Vertical Fire-tube Boiler.

sufficient flexibility between the top and bottom tube sheets to provide for unequal expansion of tubes and shell. The ash pit is built of brick and the water leg does not extend below the grate level, thus doing away with dead-water space. Where overhead room permits and ground space is expensive, this boiler offers the advantage of taking up a small floor space as compared with horizontal types.

61. Fire-box Boilers. — Although vertical fire-tube boilers may be classed as fire-box boilers, yet the term "fire box" is usually associated with the locomotive types, whether used for traction or stationary purposes. The usual form of fire-box boiler as applied to stationary work is illustrated in Fig. 37. The shell is prolonged beyond the front tube

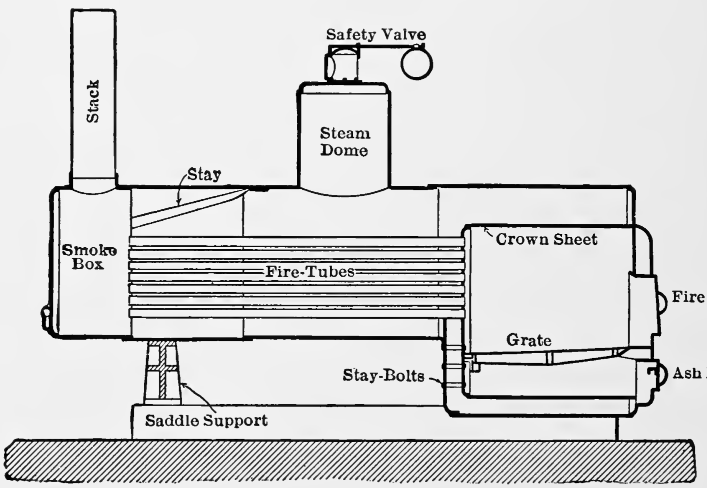


FIG. 37. Typical Fire-box Boiler — Portable Type.

sheet to form a smoke box. The front ends of the tubes lead into the smoke box and the rear ends into the furnace or fire box. The fire box is ordinarily of rectangular cross section, and is secured against collapse by stay bolts and other forms of stays. In Fig. 37 the smoke box is of cylindrical cross section and hence requires no staying except at the flat surface. Fire-box boilers are used a great deal in small heating plants where space limitation precludes other types. Their steam capacity gives them an advantage over the vertical tubular form. Being internally fired no brick setting is required. They are usually of cheap construction, designed for low pressure, and seldom made in sizes over 75 horsepower. Unless carefully designed and constructed high steam pressures are apt to cause leakage because of unequal expansion of boiler shell, tubes, and fire box. Portable fire-box boilers with return tubes are made in sizes as large as 150 horsepower and for pressures as high as 150 pounds per square inch, but being more costly than some of the other types of boilers of equal capacity are used only where portability is an essential requirement.

Fig. 38 shows a longitudinal section through a fire-box boiler of the "brick set" type. This class of boiler is much in evidence in lower pressure heating installations. The boiler proper is encased in brick work and the gases are carried along the outer surface of the shell

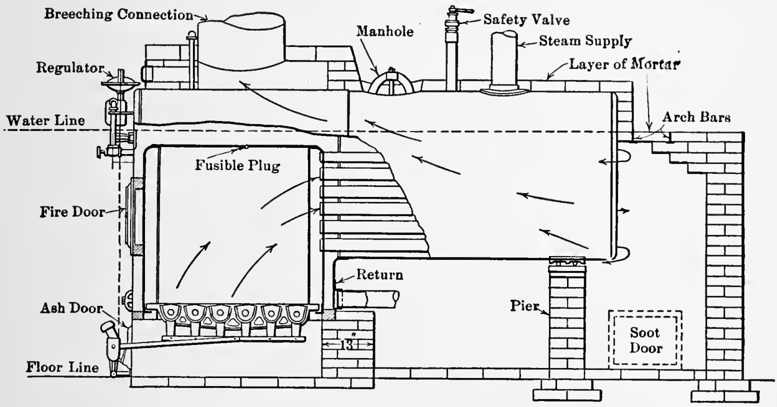


Fig. 38. Typical Fire-box Boiler, "Kewanee Brickset."

before being discharged through the breeching. For smokeless combustion the furnace is fitted with a down-draft grate (see paragraph 98), and lined with refractory material.

62. Scotch-marine Boiler. — Where an internally fired boiler is desired for large powers the Scotch-marine type is finding much favor with engineers. A number of the tall office buildings in Chicago are equipped with boilers of this class which are giving good results. They require little overhead room, no brick setting, and are excellent steamers. The Continental boiler, Fig. 39, is one of the best known of this type. The boiler is self-contained and requires no brick setting, the only fire brick used being those that form the bridge wall, baffle ring, and the layer at the back of the combustion chamber. The furnace and tubes are entirely surrounded by water, so that all fire surfaces, excepting the rear of the combustion chamber, are water cooled. The furnace is corrugated for its whole length. These corrugations, in addition to giving greater strength to the furnace, act as a series of expansion joints, taking up the strains due to unequal expansion of furnace and shell. Practically all types of mechanical stokers and grates are applicable to these boilers. The advantages of a Scotch boiler and of all

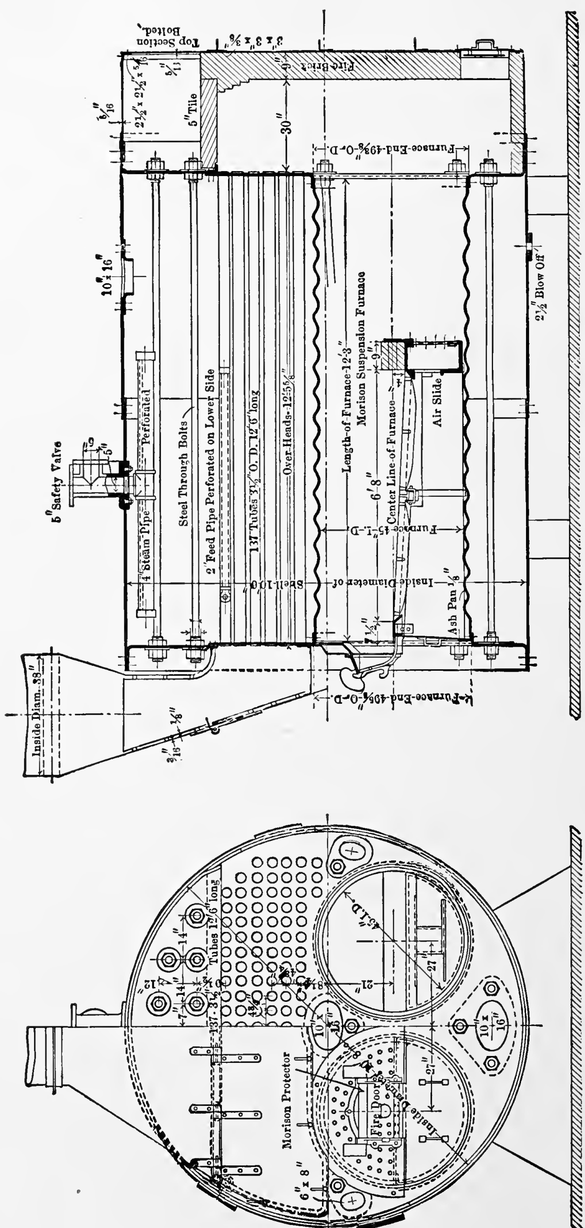


Fig. 39. Stationary Scotch-marine Boiler.

internally fired boilers are: (1) minimum radiation losses; (2) require no setting; (3) no leakage of cool air into the furnace as sometimes occurs through cracks or porous brickwork of other types; (4) large steaming capacity for the space occupied. The circulation, however, is not always positive and the water below the furnace may be considerably below the average or normal temperature, giving rise to unequal expansion and contraction which may cause leakage. The boiler proper is relatively costly, but this is offset to some extent by the absence of setting.

63. Robb-Mumford Boiler. — Fig. 40 shows a section through a Robb-Mumford boiler, which is a modification of the Scotch-marine and of the horizontal tubular type. It consists of two cylindrical

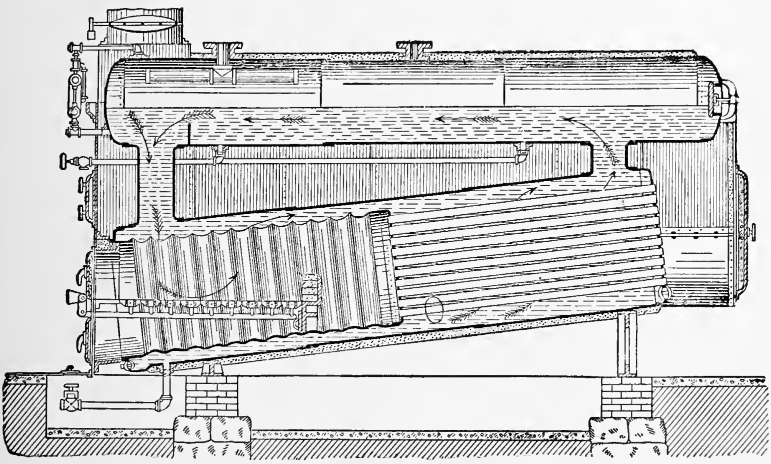


Fig. 40. Robb-Mumford Boiler.

shells, the lower one containing a round furnace and tubes and the upper one forming the steam drum, the two being connected by two necks. The lower shell has an incline of about one inch per foot from the horizontal, for the purpose of promoting circulation and draft, and also for convenience in washing out the lower shell. Combustion takes place in the furnace, which is surrounded entirely by water, and the gases pass through the tubes and return between the lower and upper shells (this space being inclosed by a steel casing) to the outlet at the front of the boiler. Mingled water and steam circulate rapidly up the rear neck into the steam drum, where the steam is released, the water passing along the upper drum towards the front of the boiler and down the front neck, a semi-circular baffle plate around the furnace causing the down-flowing water to circulate to the lowest part of the lower shell under the furnace. The outer casing, which incloses the space between the lower and upper shells, including the rear smoke box and the smoke outlet, is constructed of steel plate, with angle-iron stiffeners, the various sections being bolted together for convenient removal. The inside of the steel case, including the rear smoke chamber, is lined with asbestos air-cell blocks fitted in between the angle-iron stiffeners. The top of the upper drum and bottom of the lower shell are also covered with non-conducting material after the boiler is erected. Owing to the fact that steam and water spaces are divided between two cylindrical shells, the thickness of plates is not so great as in the Scotch-marine or horizontal return tubular types; and the rear chamber of the marine boiler is avoided.

The chief claim for this type of boiler is compactness. A battery of five 200-horsepower units occupies a floor space of but 33 feet in width by 20 feet in depth and 12.5 feet high. Each unit is entirely independent and may be isolated for cleaning, inspection, and repairs.

64. Horizontal Return Tubular Boilers. — These are the most common in use and are constructed in sizes up to 500 horsepower. They are simple and inexpensive and, when properly operated, durable and economical. Figs. 41 to 44 show various forms of standard settings, and Figs. 88, 89, and 90 different "smokeless" settings. The grate is independent of the boiler, and the products of combustion pass beneath the shell to the back end, returning through the tubes to the front, and into the smoke connection.

The tubes are from 3 to 4 inches in diameter and from 14 to 18 feet long, and are expanded into the tube sheets. The portion of the tube sheets not supported by the tubes is secured against bulging by suitable stays. Access to the interior of the boiler is obtained through manholes. The most convenient arrangement for inspection and cleaning is to have one manhole located at the top of the shell and one at the bottom of the front tube sheet. Return tubular boilers are made either with an *extended* or *half-arch front* (Fig. 41) or *flush front* (Fig. 42). The shell may be supported by lugs resting on the brickwork as in Fig. 41 or by steel beams and hangers as in Fig. 43. The latter construction permits the brickwork and shell to expand or contract independently, and settling of the brickwork does not affect the boiler alignment. According to the A.S.M.E. Boiler Code all horizontal tubular boilers over 78 in. in diameter are required to be supported by this *outside suspension* type of setting. With the side bracket support, the front lugs usually rest directly on iron or steel plates embedded in the brickwork, and the back lugs on rollers, to permit free expansion and contraction. The brackets are long enough to rest upon the outside wall, so that the inside brick

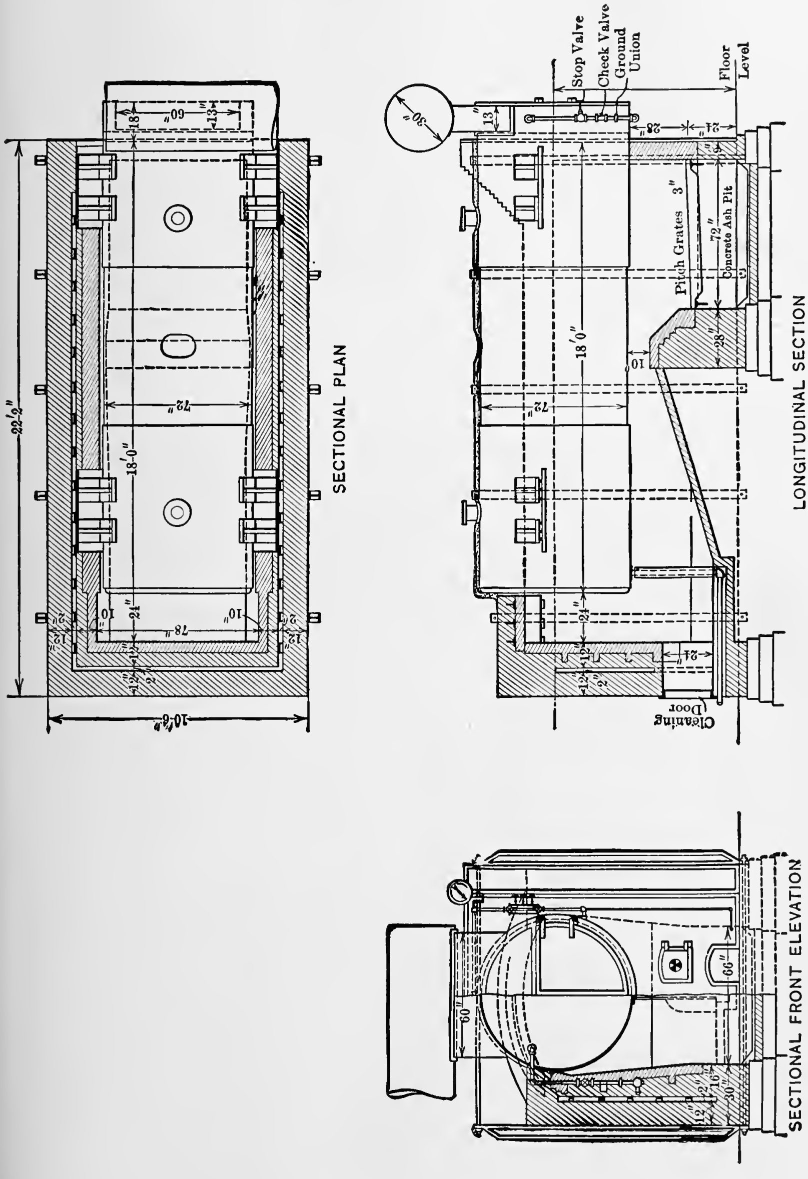


FIG. 41. Return Tubular Boiler and Standard Double-wall Setting—Extended Front.

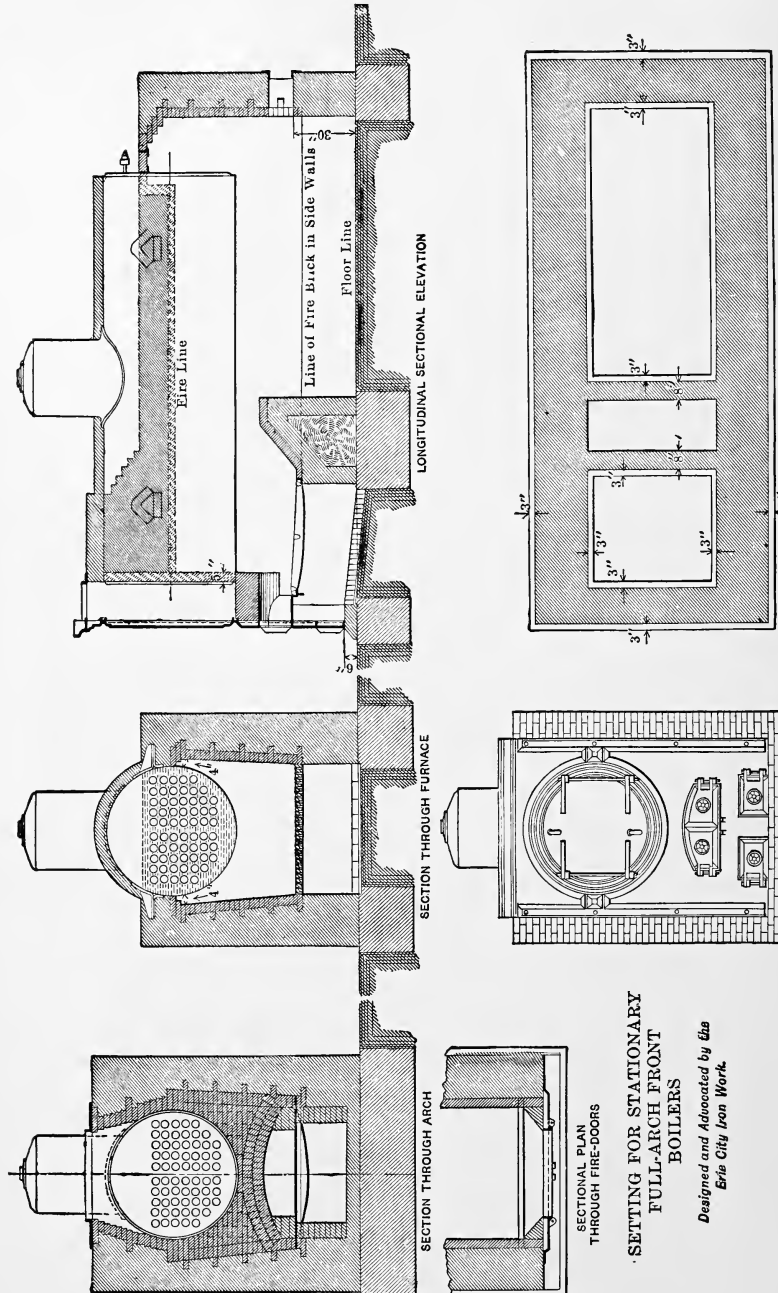


Fig. 42. Return Tubular Boiler and Standard Single-wall Setting — Flush Front.

lining can be renewed without disturbing the setting. The A.S.M.E. Boiler Code specifies four pairs of brackets (two pairs on each side) for boilers over 54 in. and up to and including 78 in. in diameter, and not less than two brackets on each side for boilers up to and including

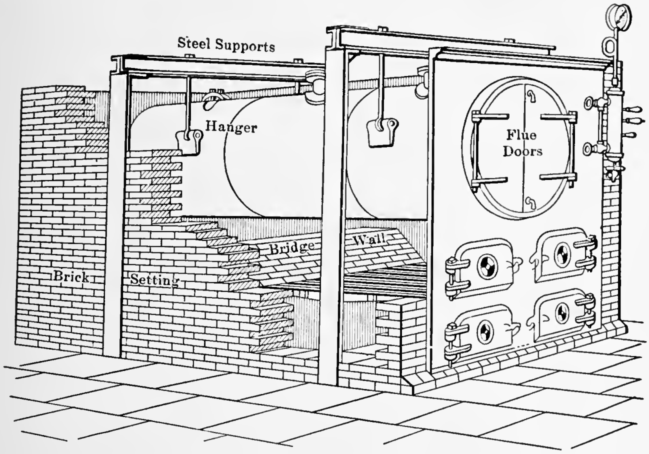


FIG. 43. Return Tubular Boiler Setting — Outside Suspension Type.

54 in. in diameter. The distance between the rear tube sheet and wall should be about 16 inches for boilers less than 60 inches in diameter and from 20 to 24 inches for larger ones. The distance between grate and boiler shell should not be less than 28 inches for anthracite coal and 36 inches for bituminous coal.* The greater this distance the more complete the combustion, since the gases will have a better opportunity for combining with the air before coming into contact with the comparatively cool surfaces of the shell. The shell should be slightly inclined toward the blow-off end so as to drain freely.

The vertical distance between the bridge wall and shell is usually between 10 and 12 inches. The lower part of the combustion chamber behind the bridge wall may be filled with earth and paved with common brick as in Fig. 44 or left empty as in Fig. 42. The shape of the bridge walls whether curved to conform to the shell or flat appears to have little influence on the economy.

The side and end walls are ordinarily constructed of common brick with an inner lining of fire brick, and may be solid as in Fig. 42 or double with air spaces as in Fig. 41. The latter construction permits the inner and outer walls to expand independently without cracking

* For smokeless combustion the setting must be modified. See furnaces illustrated and described in paragraph 93.

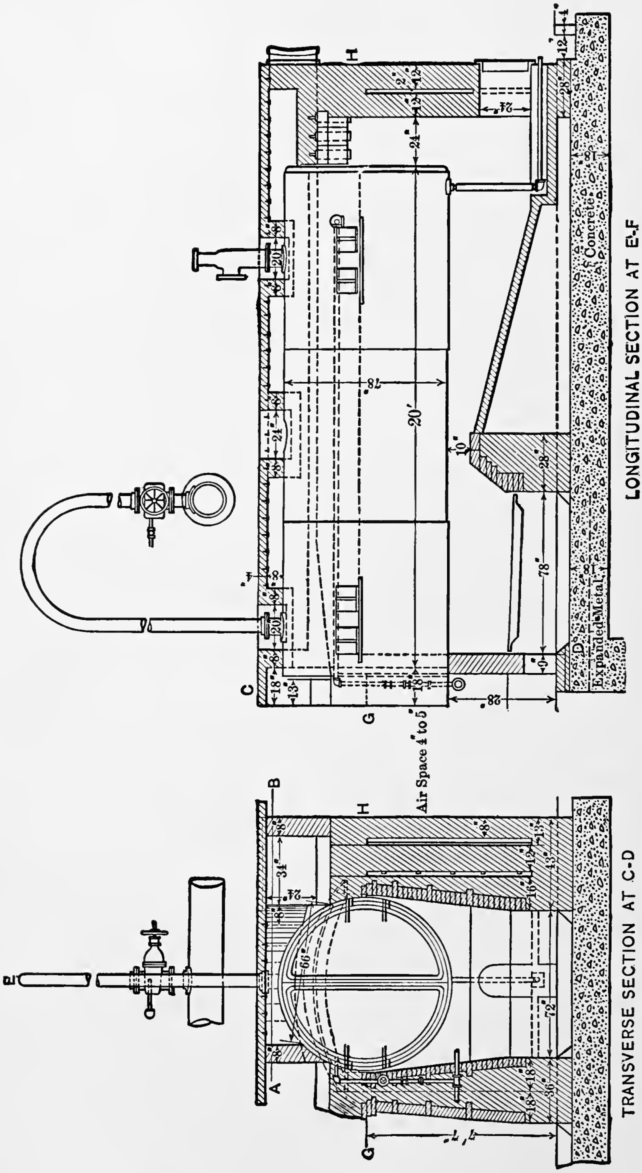


FIG. 44. Boiler Setting; "Wood" Mill of the American Woolen Company, Lawrence, Mass.

and settling. Tests conducted by Ray and Kreisinger * show that a solid wall is a better heat insulator than a wall of the same total thickness containing an air space, hence if air spaces are used they should be filled with loose non-conducting material. The side walls are braced by five pairs of buckstaves, with through rods under the paving and over the tops of the boilers. Air leakage through the setting is completely eliminated by enclosing the entire setting within a steel casing. A lining of kieselguhr or similar insulating material within the casing will greatly reduce the heat losses through the walls of the setting. See "Insulation of Boiler Settings," Joseph Harrington, Power, Mar. 27, 1917, p. 410.

The connection between the rear wall and the shell is a source of more or less trouble on account of the expansion and contraction of the boiler. Cast-iron supports of T section supporting a fire-brick arch

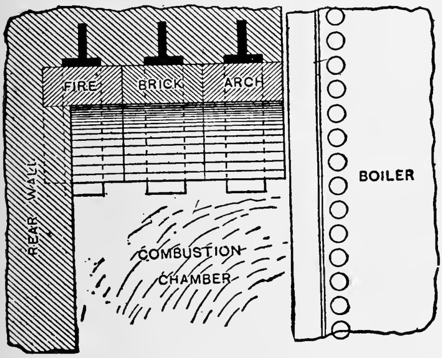


FIG. 45. Furnace Arch Bars.

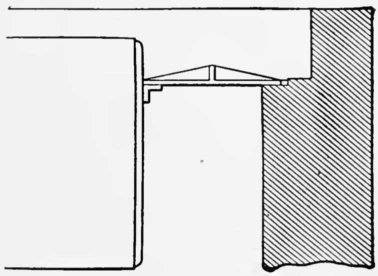


FIG. 46. Back Connection Made with Cast-iron Plate.

are usually employed as illustrated in Fig. 45, the clearance between the arch and the shell being sufficient to allow the necessary expansion. In order to avoid air leakage this clearance space is filled with asbestos fiber.

Fig. 46 shows the common method of resting one end of the arch supports on the rear wall and the other end on an angle iron riveted to the boiler. Fig. 47 illustrates the principles of the Woolson Arch connection.

The products of combustion are sometimes carried over the top of the boiler as shown in Fig. 44. This tends to superheat the steam, but the advantage gained is probably offset considerably by the extra cost of the setting and the accumulation of soot on the top of the shell. The arrangement is not common.

* Bul. No. 8, U. S. Bureau of Mines, 1911.

The steam connection is naturally made to the highest point in the boiler shell. Frequently a steam dome, to which the steam nozzle is connected, is provided as in Fig. 42. The function of the steam dome is to increase the steam space so as to permit the collection of dry steam at a point high above the water level. If a boiler is too small for its work and is forced far above its rating a steam dome is probably an advantage, though its use is less common now than formerly, since a properly designed boiler insures ample steam space without one. A dry pipe inside the boiler above the water line as in Fig. 39 or 40 is commonly used to guard against priming where the nozzle is connected to the shell.

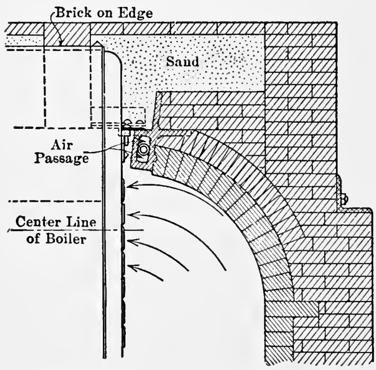


Fig. 47. Woolson's Gas Tight Back Arch Connection.

For low pressures and small powers the return tubular boiler has the advantage of affording a large heating surface in a small space and large overload capacity. It requires little overhead room and its first cost is low. On the other hand the interior is difficult of access for purposes of cleaning and inspection. Boilers of this type are constructed in various sizes ranging from a 36-in. by 8 ft., rated at 15 horsepower, to a 108-in. by 21 ft., rated at 500 horsepower, though sizes above 200 horsepower are exceptional. The working pressure seldom exceeds 150 pounds per square inch.

The standard externally fired return tubular boiler is limited in size since the damage from overheating the shell directly over the fire bed increases rapidly with the increase in thickness of the plate. The Lyons boiler overcomes this restriction through the addition of a bank of water tubes which form a roof to the furnace. These tubes protect the shell from the direct action of the gases and insure a positive and rapid circulation. They are covered with tile or split brick and form the equivalent of a "Dutch oven."

Arches—Firebrick Furnace: Jour. A.S.M.E., Jan., 1916, p. 7; Power, Feb. 20, 1912, Oct. 24, 1916, p. 598.

65. Babcock & Wilcox Boilers. — Fig. 48 shows a longitudinal section through a Babcock & Wilcox boiler, illustrating a typical horizontal water-tube type. The tubes, usually 4 inches in diameter and 18 feet in length, are arranged in vertical and horizontal rows and are expanded into pressed-steel headers. Two vertical rows are fitted to each header and are "staggered" as shown in Fig. 49. The headers are connected with the steam drum by short tubes expanded into bored holes. Each

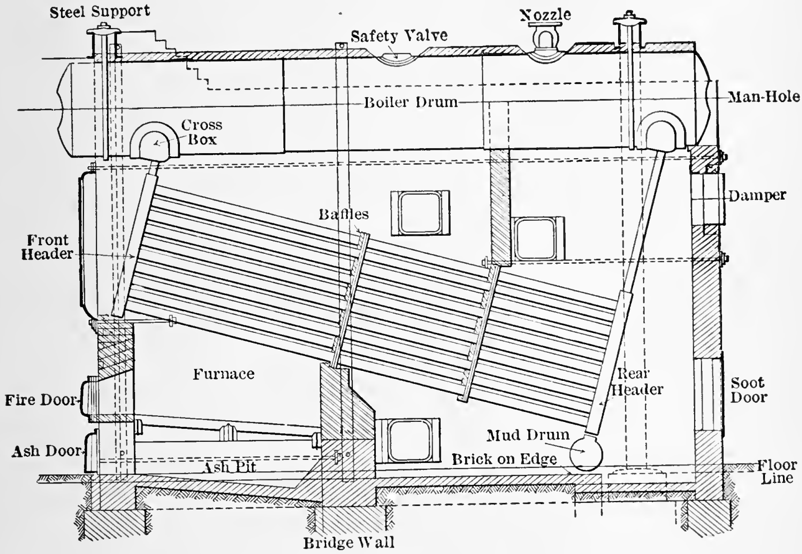


FIG. 48. Babcock & Wilcox Boiler and Standard Hand-fired Setting.

tube is accessible for cleaning through openings closed by covers with ground joints held in place by forged steel clamps and bolts. The tubes are inclined at an angle of about 22 degrees with the horizontal. The rear headers are connected at the bottom to a forged steel mud drum. The steam drum is horizontal and the headers are arranged either vertically as shown in Fig. 94 or inclined as in Fig. 50. The boiler is supported by steel girders resting on suitable columns independent of the brick setting. The grate is placed under the higher ends of the tubes, the products of combustion passing at right angles to the tubes and being deflected back and forth by fire-tile baffles. The feed water enters the front of the steam drum as shown in Fig. 50. A rapid circulation is effected by the difference in density between the solid column of water in the rear header and the mixed steam and water in the front one. Babcock & Wilcox boilers under 150 horsepower have but one steam drum, and the larger sizes have

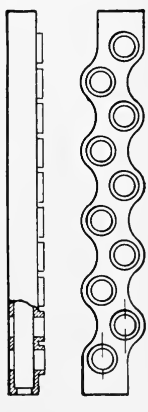


FIG. 49. Details of Header — Babcock & Wilcox Boiler.

two. The drums are accessible for inspection through manhole openings. The number of tubes varies with the size of boiler, ranging from 6 wide and 9 high in the 100 horsepower boiler to 14 high and 18 wide in the 500 horsepower boilers. The spacing of the tubes is based primarily upon the proportions of the grate. The width of the grate determines the "number of tubes wide" and the capacity of the boiler controls the "number of tubes high." Babcock & Wilcox boilers may be baffled so that the gases may pass out either at the front or rear of the top of the setting or at the rear of the bottom of the setting. The gases may be directed across the tubes as illustrated in Fig. 48 or along the tubes as shown in Fig. 53.* Large doors in the sides of the setting give full access to all parts for inspection and for removal of accumulations of soot. In the strictly modern power plant the setting is encased in steel in order to prevent air leakage, and the casing is lined on the inside with heat insulating material, such as kieselguhr, so as to reduce heat losses.

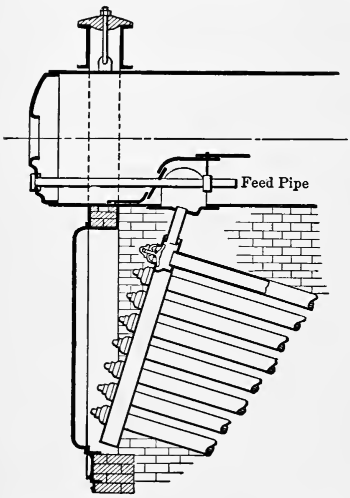


Fig. 50. Front Section — Babcock & Wilcox Boiler.

Fig. 51 shows a section through a Babcock & Wilcox marine type boiler with cross drum. Boilers of this design have been installed in units of 1200 rated horsepower and are giving eminent satisfaction as to efficiency and capacity. For smokeless settings see Chapter IV.

66. Heine Boiler. — Fig. 52 shows a longitudinal section through a Heine horizontal water-tube boiler. This boiler differs from the Babcock & Wilcox boiler in that the tubes are expanded into a single large header constructed of boiler steel. The drum and tubes are parallel with each other and inclined about 22 degrees with the horizontal. The feed water enters at the front of the steam drum and flows into the mud drum, from which it passes to the rear header. Steam is taken from the front of the steam drum and is partially freed from moisture by the dry pipe *A*. A baffle over the front header prevents an excess of water from being carried into the dry pipe. As the rear header forms one large chamber, no additional mud drum is necessary and the sediment is "blown off" from the bottom by the blow-off cock. The circulation is

* Horizontal and Vertical Baffling for B. & W. Boilers, S. H. Viall, Power, June 20, 1916, p. 874.

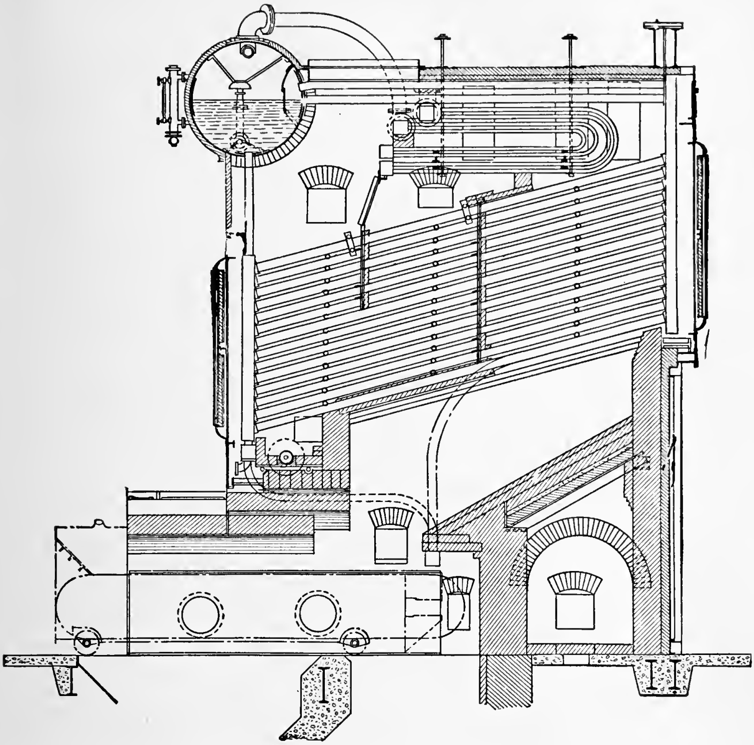


FIG. 51. Babcock & Wilcox Boiler — Cross Drum Type.

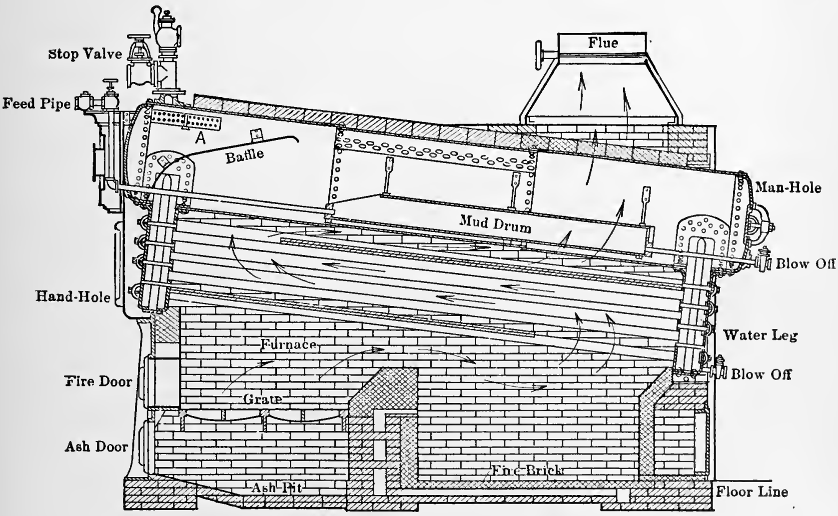


FIG. 52. Heine Boiler and Standard Hand-fired Setting.

somewhat freer than in the Babcock & Wilcox boiler on account of the large sectional area through the headers.

Circulation in Water Tube Boilers: Jour. A.S.M.E., Jan. 1916, p. 17.

67. Parker Boiler. — Fig. 53 shows a longitudinal sectional elevation and an end sectional elevation of a 1200-horsepower Parker down-flow boiler with double-ended setting. This type of boiler is finding much favor with engineers for central stations where large units are desired. The Parker boiler differs from the conventional horizontal water-tube boiler principally in circulation and flexibility.

Feed water is pumped into the economizer or feed element (1), Fig. 53, at O, O, and flows downward through a series of tubes, discharging finally into the drum through an upcast H. In a large unit, as illustrated here, there are two feed elements and two drums. The circulation in the feed element is indicated by solid lines and arrow points at the left of the end sectional elevation, the tubes having been omitted from the drawing for the sake of clearness.

The intermediate elements (2) take their water supply from the bottom of the drum through a cross-box V, the circulation being downward, as indicated by arrow points, through four tube wide elements, and finally discharge it through an upcast X into the steam space of the drum. Each element has a "down-comer" and an upcast. In the smaller-sized boilers the intermediate elements are omitted.

The evaporator elements (3) take their water supply from the bottom of the drum at V, the circulation being downward through two tube wide elements, and finally discharge it into the drum at U. The last two passes of the water are through the two bottom tubes of each element, thus assuring dry steam without the use of dry pipes. To prevent reversal of flow each element is fitted with a check valve at the admission end. Each drum is equipped with a diaphragm, as indicated, separating the steam and water spaces, thus insuring against foaming and priming.

Saturated steam is taken from the drum at A and passes by way of B to C, where it enters the superheater S. The superheated steam leaves the superheater at D and passes by way of E and R and the storage drum N, finally leaving the boiler at G. The superheater is designed to maintain an approximately constant degree of superheat for all variations in load.

All tubes are connected by malleable-iron junction boxes the interior of each tube being accessible through hand holes placed opposite the end of each tube. The hand-hole cover plates are on the inside of the box and have conical ground joints, thus dispensing with gaskets.

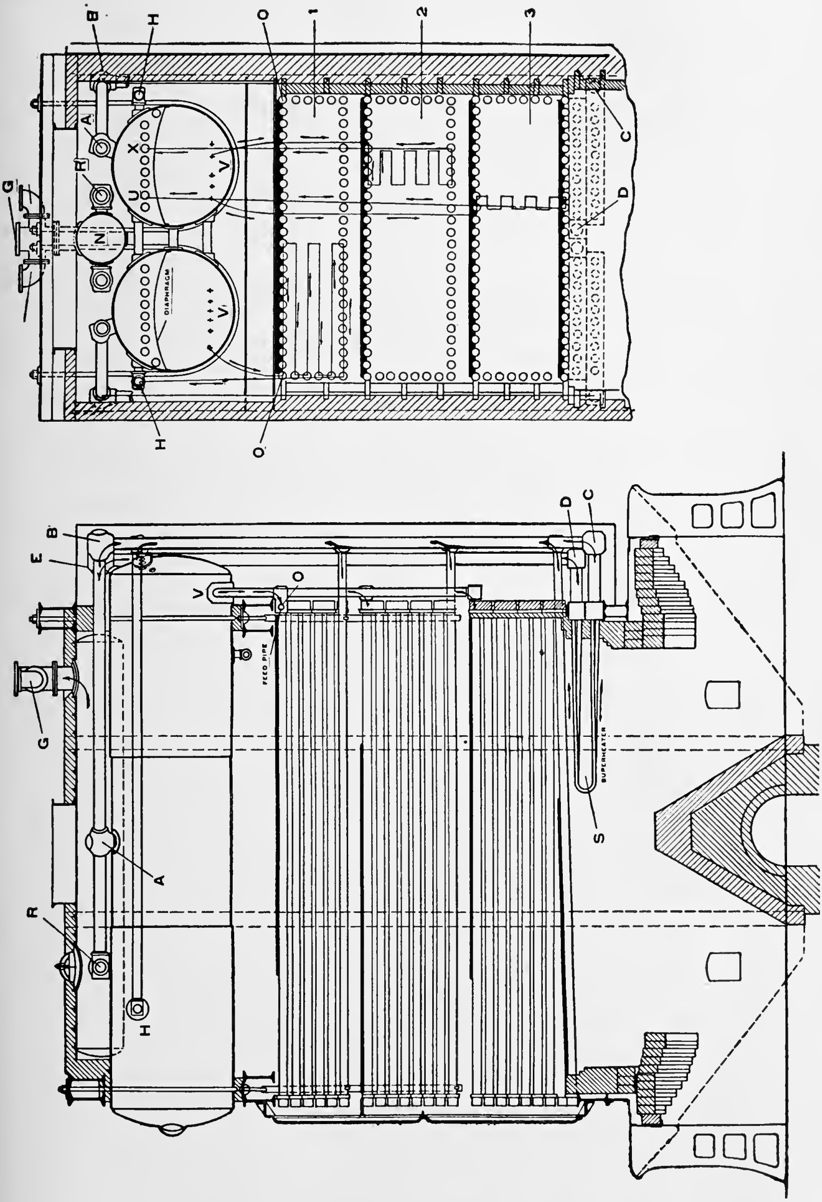


FIG. 53. 1200-Hp. Parker Down-flow Boiler with Double-ended Setting.

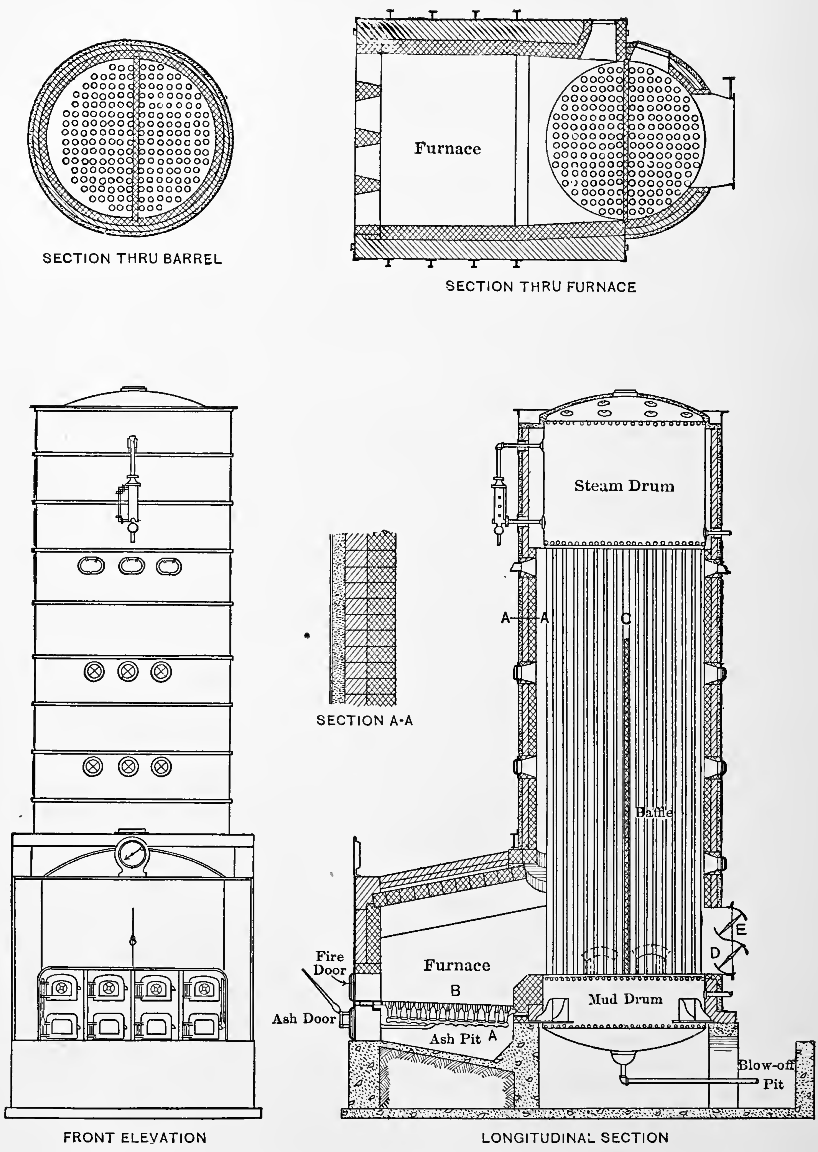


Fig. 54. Wickes Vertical Boiler with Steel Encased Setting.

The Parker boiler is built single or double ended, with or without superheater, and in sizes ranging from 50-horsepower to 2500-horsepower standard rating.

68. Wickes Boiler. — Fig. 54 shows a section through a Wickes vertical boiler, illustrating the vertical water-tube type. The steam drum and water drum are arranged one directly above the other. The tubes are expanded and rolled into both tube sheets and are divided into two sections by fire-brick tile. The water line in the steam drum is carried about two feet above the tube sheet, leaving a space of five feet between water line and top of the drum. This affords a large steam space and disengagement surface. Feed water is introduced into the steam drum below the water line and flows downward through the tubes of the second compartment. The boiler is supported by four brackets riveted to the shell of the bottom drum and is independent of the setting. The entire boiler is enclosed in a steel casing, insulated with non-conducting material and lined with fire brick. The boiler is completely surrounded by the products of combustion. The steel encased setting prevents lowering the temperature of the products of combustion by air infiltration and reduces radiation losses. The upper part of the steam drum acts as a superheating surface and tends to dry the steam. Wickes boilers are simple in design, easy to inspect and clean, low in first cost, and comparable in efficiency with any water-tube type of boiler.

69. The Bigelow-Hornsby Boiler. — Fig. 55 shows a vertical section through a Bigelow-Hornsby boiler equipped with Foster superheater and Taylor stoker. This boiler is of the vertical water-tube type and is made up of a number of cylindrical elements, each element comprising an upper and lower drum connected by straight tubes. The two front elements are inclined over the furnace at an angle of about 68 degrees, and the two rear elements are vertical. The upper drums of the elements are connected to a horizontal main steam drum by flexible tubing as indicated. Four elements constitute a section with an effective heating surface of 1250 square feet. Any number of sections may be connected together forming units of from 250 to 2500 boiler horsepower or more. All parts, both external and internal, are readily accessible. Feed water enters the top drum of the rear elements and passes twice the length of the tubes before entering into the general circulation. This arrangement permits a considerable portion of the impurities in the water to be precipitated in the rear drum from which they are readily discharged. By the time the water reaches the front of the boiler directly over the furnace, where the heat transmission is the most intense, the scale-forming elements have been practically eliminated.

The particular features of this boiler lie in the great extent of heating surface exposed to radiant heat and the height and volume of the combustion chamber. Bigelow boilers are productive of high economy and are readily forced to twice their rated capacity with little decrease

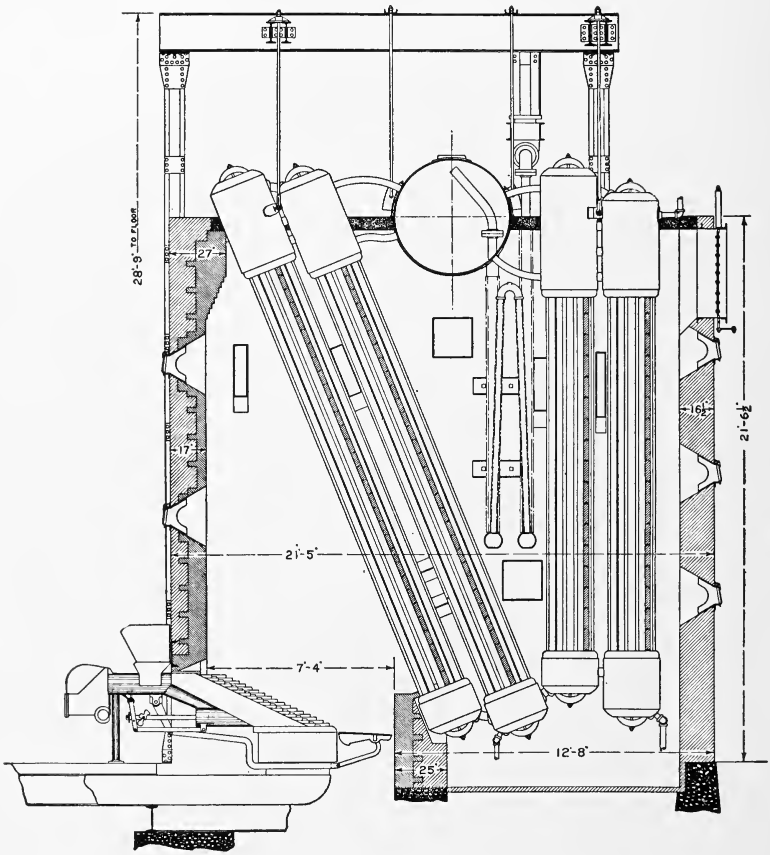


FIG. 55. Bigelow-Hornsby Boiler and Setting.

in over-all efficiency. The most notable installation of Bigelow boilers in this country is at the power plant of the Hartford Electric Light & Power Company, Hartford, Conn., where two 1250- and one 2500-boiler-horsepower units are installed.

70. Stirling Boiler. — Fig. 56 shows a longitudinal section through a Stirling water-tube boiler, which differs considerably from the types just described. Three horizontal steam drums and one horizontal mud drum are connected by a series of inclined tubes. The tubes are bent at the ends to permit them to enter the drums radially. Short tubes connect the steam spaces of all the upper drums and also the water spaces of the front and middle drums. Suitably disposed fire-tile baffles between the banks of tubes direct the gases in their proper course. The boiler is supported on a structural steel framework in-

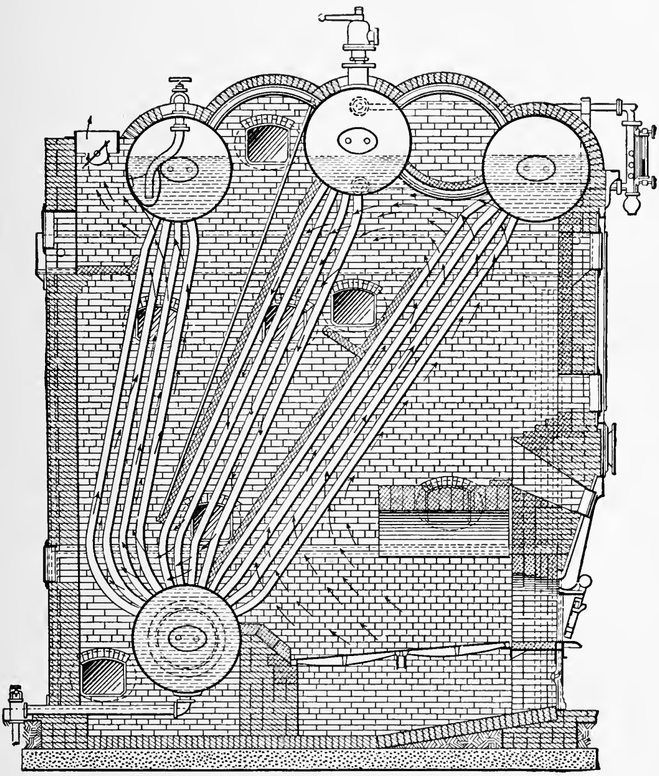


FIG. 56. Stirling Boiler and Standard Hand-fired Setting.

dependent of the setting. The feed water enters the rear upper drum, which is the cooler part of the boiler, and flows to the bottom or mud drum, where it is heated to such an extent that many of the impurities are precipitated. There is a rapid circulation up the front bank of tubes to the front drum, across to the middle drum, and thence down the middle bank of tubes to the mud drum. The interior of the drums is accessible for cleaning by manholes located in the ends. The Stirling furnace is distinctive in design. A fire-brick arch is sprung over the grates immediately in front of the first bank of tubes. The large triangular space between boiler front, tubes, and mud drum forms the combustion chamber.

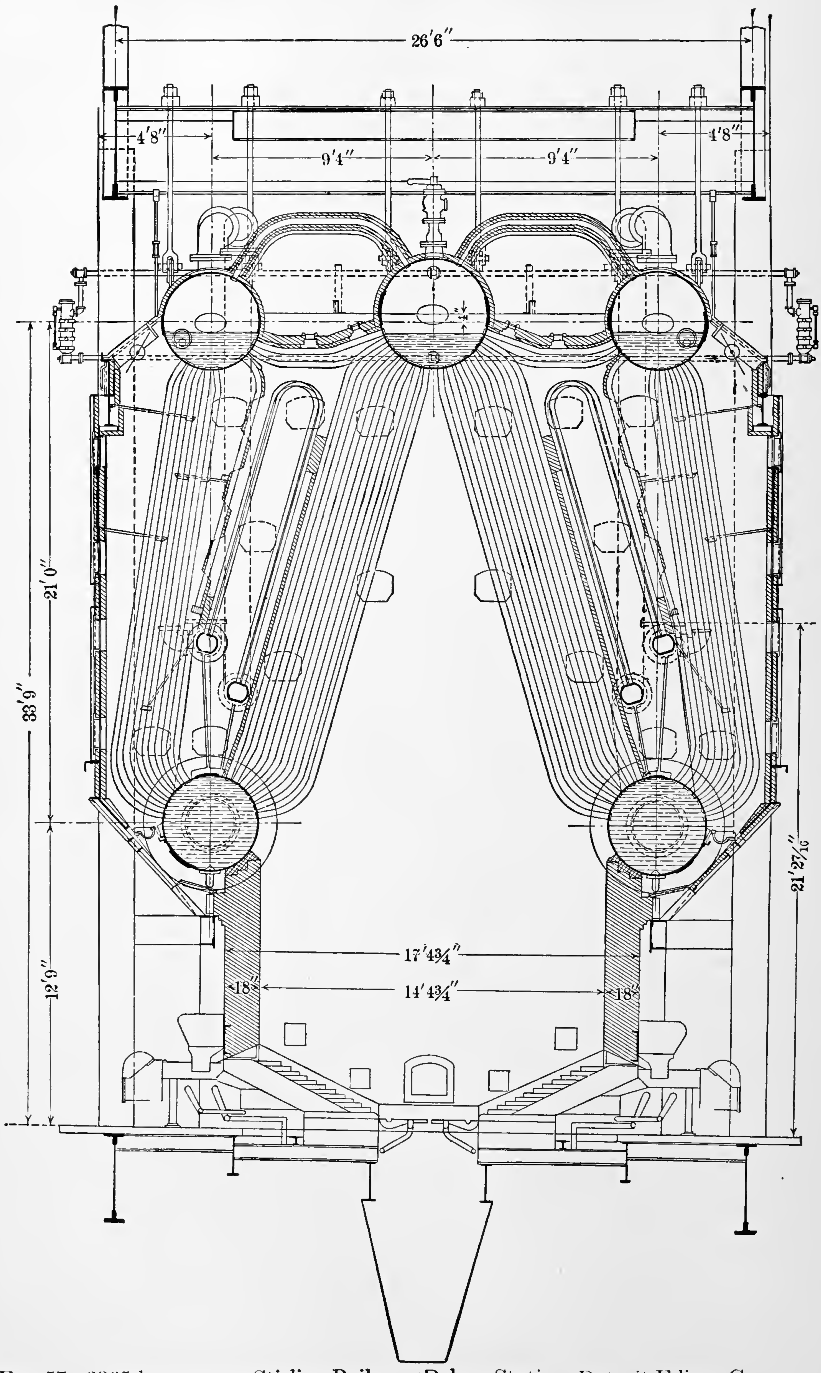


Fig. 57. 2365-horsepower Stirling Boiler — Delray Station, Detroit Edison Company.

Fig. 57 gives a sectional view through the boiler and setting of a 2365-horsepower Stirling boiler equipped with Taylor stokers as installed at the Delray station of the Detroit Edison Company. Five boilers are now in operation and it is planned to eventually install ten. Though rated at 2365 boiler horsepower they are capable of carrying continuously a load equivalent to 6000 kilowatts with a maximum of 8000 kilowatts. The overall dimensions of the boiler and setting are shown in the illustration. Each unit contains 23,654 square feet of effective heating surface and is provided with superheaters for supplying steam at 150 degrees superheat. Table 33 gives a résumé of the principal results obtained from tests of these units with Roney and Taylor stokers. The grate surface per boiler for the Roney stoker is 446 square feet and for the Taylor stoker 405 square feet, thus giving as ratios of grate surface to heating surface 1 : 53 and 1 : 58.5 respectively. For a complete description of these tests see Jour. A.S.M.E., Nov., 1911, p. 1439.

The largest boilers in this country (1917) are installed in the new Highland Park plant of the Ford Motor Company. Each unit contains 25,000 sq. ft. of effective heating surface and furnishes 4000 boiler horsepower continuously. These boilers are of the Badenhausen type and are equipped with Taylor stokers (Power, Oct. 3, 1916, p. 474).

71. Winslow High-pressure Boiler. — The standard types of boiler described in the preceding paragraph are seldom designed for pressure exceeding 250 lb. per sq. in. A few installations have been made for working pressures as high as 350 lb. per sq. in., but it is doubtful if this pressure will be exceeded without considerable modification in basic design. The weak element lies in the drum since excessive thickness of material is necessary for pressures above the limit mentioned. For example, the 60-in. drums of the Babcock & Wilcox boilers for the Joliet plant of the Public Service Company of Northern Illinois are $1\frac{3}{8}$ in. thick. With the prospect of pressures ranging as high as 1000 lb. per sq. in. (see paragraph 179) engineers are interested in types of boilers which can be built commercially to withstand these high pressures. Fig. 58 shows a section through the setting and one element of a "Winslow Safety High-pressure" boiler which may be designed to withstand working pressures considerably in excess of 1000 lb. per sq. in. The assembled boiler consists of a number of sections, similar to the one illustrated in Fig. 58, forming a closely nested mass of tubes, each section being connected to a common steam header, feed pipe and mud drum. Referring to the illustration: each section is composed of a "front section header" *A* and "rear section header" *B*, connected by a number of approximately horizontal tubes, *C*, all made of seamless steel tubing. The lower tubes are inclined, the front ends being higher than the rear. This degree of inclination gradually decreases in the upper tubes until the highest tube is practically horizontal. All tubes are slightly curved, the lower ones more than the upper. This preserves

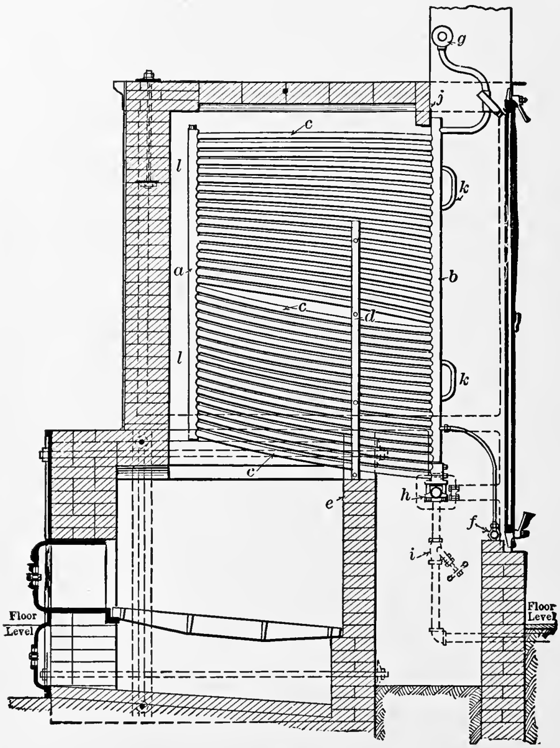


FIG. 58. Winslow "Safety High-pressure" Boiler and Hand-fired Setting.

each tube in its original plane, even if it should expand considerably under heat.

All joints between the tubes and section headers are welded. Extra material is added in the welding process and the joint is thus made stronger than the tube itself.

Each section carries a baffle D, riveted in place, in contact with each side of the section, each baffle touching the one on the adjoining section, the outside ones being in contact with the wall of the enclosure. These several baffles form a complete baffle wall, which is in contact with the fire bridge E, confining the first and most intense action of the fire to the front part of the section. These baffles are either made of cast iron or of steel channels filled with plastic refractory material.

The baffles being in metallic contact with the tubes, their temperature can never greatly exceed that of the water or steam.

All sections are supplied with water from a common feed pipe *F* closed at one end and carrying the check valve and pump connection at the other. A branch tube leads to each section entering the rear section header somewhat above its lower end. The joints at header and feed pipe are clearly shown.

From the upper end of each rear section header a steel tube leads to the steam header *G*. This tube consists of two parts, one welded to the steam header and the other to the rear section header, connected by a special joint. To insure equal distribution of the flow of steam over the length of the steam header, the steam is taken through a large number of small holes, properly distributed, in a tube located inside the steam header and passing through one of its sealed ends.

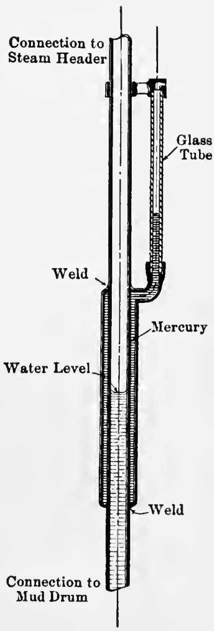


Fig. 59. Water Level Gauge — Winslow High-pressure Boiler.

The lower end of the rear section header is formed into a special joint, which is connected to the mud drum *H*. The opening into the mud drum is as large as it is possible to make it and the passage is straight and without obstructions. Connected to the mud drum is the blow-off valve *I*, shown in dotted lines, Fig. 58.

The three unions on each section at steam header, feed pipe and mud drum, are the only joints which are not welded, but these are all located in the last pass of the furnace gases and are not subjected to high temperatures. They are easily accessible and are made with metal to metal contact, without gaskets or packing.

At high pressures and temperatures the ordinary gauge glasses are not desirable. One of the best indicators for severe conditions is shown in section in Fig. 59. The water column is a steel tube, surrounded by a steel jacket, the space between being filled with mercury, visible in a vertical glass tube. That part of the mercury which surrounds steam in the column absorbs much more heat than the part which surrounds water. The average temperature of the mercury and consequently its height in the glass tube is, therefore, a positive indication of the water level, being high for low water and low for high water. There is, of course, a certain lag in the indication on account of the time necessary to transfer the heat through the metal wall of the column, but this is negligible on account of the wide range of water content which is permissible in the Winslow boiler without danger or improper influence on its operation. When superheated steam is produced a thermometer can be used as an additional indicator of the equivalent water level. This instrument is usually of the dial form, its bulb being inserted into the steam pipe at the flange connecting it to the steam collector. The dial, being connected to the bulb by a flexible tube, can be located at any point most convenient to the operator. A drop in temperature indicates a higher water level and vice versa.

Circulation is as follows: When heat is applied to the boiler its first effect is to expand the water which is contained in that part of the section forward of the baffle, thereby reducing its specific gravity. Expansion causes the water column to rise in front and the heated water to flow toward the rear in the upper tubes. Reduced specific gravity in the front part causes a forward flow in the lower tubes.

Each particle of water absorbs a certain amount of heat during circuit through the set of tubes. When its temperature has reached 212 deg. fahr. further absorption of heat causes the generation of steam, or, in other words, a sudden increase of volume and a consequent reduction of the specific gravity of the water. The immediate effect of this is to make the circulation more active. The rising column in the front section header then consists of a mixture of steam and water.

The water drains back toward the rear section header through the upper circulation tubes, and the steam naturally tends to separate from the water at this point and to flow through the front header and the top tubes, toward the point of discharge. The returning water does not completely fill the upper tubes of the "zone of circulation and evaporation," but it exposes a certain amount of surface, from which further separation takes place of such steam as is carried along with the water or as is generated within the return flow tubes. The foregoing will make it clear that the office of the upper part of the circulating tubes, which have been designated as "return flow tubes," is to intercept the rising column of water and steam in the front section header, carry the water back by gravity, and prevent its entering the uppermost tubes. It should be noted that the inclination of these return flow tubes gradually decreases toward the top, as the amount of water they carry becomes less.

The uppermost tubes practically contain steam only, and, being located in the flow of the hot gases, they effectively dry the steam. It is even possible, without any further provision, to superheat the steam somewhat in this "drying zone" before it is discharged from the section. If a higher degree of superheat is desired, the separate flue *L*, already referred to and shown in Fig. 58 is provided. It carries the desired amount of hot gases from the furnace, in front of the nest of tubes, directly over the top of the boiler, through the "drying zone," which thereby becomes a "superheating zone."

72. Unit of Evaporation. — The performance of a boiler and furnace is commonly expressed in terms of the weight of water evaporated per pound of fuel or of the weight evaporated per hour per square foot of heating surface. To reduce all performances to an equal basis so as to facilitate comparison the evaporation under actual conditions is conveniently referred to the equivalent evaporation from a feed water temperature of 212 deg. fahr. to steam at atmospheric pressure. The heat required to evaporate one pound of feed water at a temperature of 212 deg. fahr. into saturated steam of the same temperature, or from and at 212 deg. fahr. as it is commonly called, is designated as *one unit of evaporation* (U.E.). The 1915 A.S.M.E. Boiler Code stipulates the use of Mark's and Davis' value for the latent heat of steam at 212 deg. fahr. and defines the standard unit of evaporation as 970.4 B.t.u. G. A. Goodenough (Properties of Steam and Ammonia, 1915, John Wiley & Sons, Publishers) assigns a value of 971.7 to this quantity and intimates that the correct value may be even slightly greater. The ratio of the heat necessary to evaporate one pound of water under actual conditions of feed temperature and steam pressure and quality to the heat required to evaporate one pound from and at 212 deg. fahr. is called the *factor of evaporation*. Thus for dry saturated steam, using Mark's and Davis' value for the latent heat,

$$F = \frac{\lambda - q_2^*}{970.4}, \tag{30}$$

in which

F = factor of evaporation,

λ = total heat of one pound of steam at observed pressure above 32 deg. fahr.,

q_2 = total heat of one pound of feed water above 32 deg. fahr.

If the steam is wet,

$$\lambda = xr + q, \tag{31}$$

in which

x = the quality of the steam,

r = latent heat of evaporation at observed pressure,

q = heat in liquid at observed pressure.

If the steam is superheated,

$$\lambda = r + q + Ct_s, \tag{32}$$

* For most purposes q_2 may be taken at $t_2 - 32$, in which t_2 = temperature of the feed water, deg. fahr.

in which

C = the mean specific heat of the superheated steam,
t_s = the degree of superheat, deg. fahr.

73. Heat Transmission. — Fig. 60 shows a section through a boiler-heating plate and serves to illustrate the accepted theory of heat transmission. The outer surface of the plate is covered with a thin layer of soot and a film of gas, and the inner surface is similarly protected by a layer of scale and a film of steam and water. It is, therefore, reasonable to assume that the *dry* surface of the plate is located somewhere within the film of gas, and the *wet* surface within the film of water and steam.

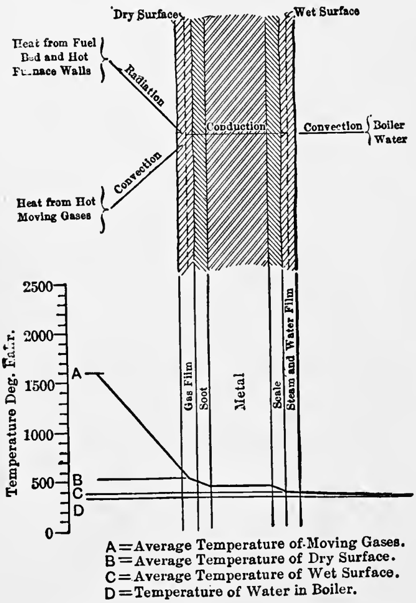


FIG. 60. Heat Transmission through Boiler Plate.

The heat is imparted to the *dry* surface by: (1) *radiation* from the hot fuel bed and furnace walls, and by (2) *convection* from the moving furnace gases. The heat is transferred through the boiler plate and its coatings purely by *conduction.* The final transfer from the *wet* surface to the water is mainly by *convection.*

Radiation depends on the temperature, and according to the law of Stefan and Boltzmann is approximately proportional to the difference between the fourth power of the absolute temperature of the fuel bed and furnace walls and the temperature of the dry surface of the heating plate. According to this law the heat transmitted by radiation increases rapidly with the increase in furnace temperature. In the ordinary boiler and setting the surface exposed to radiation is only a small portion of the total heating surface, and, since in well-operated furnaces the temperature of the furnace cannot be increased materially on account of practical considerations, there is little hope of increasing the capacity of a boiler by increasing the furnace temperature. The extent of heating surface exposed to radiation, however, may be greatly increased. Many authorities are of the opinion that the boiler of the future will depend largely upon radiation. That this prediction is being realized is evidenced by the high combustion efficiency and extremely high ratings effected by the modern duplex furnace, Figs. 57 and 103, in which a considerable portion of the boiler heating surface is exposed to direct radiation.

The amount of heat imparted by convection from heated gases to cooler metal surfaces has been the subject of a great deal of investigation both from the experimental and theoretical side. Numerous attempts have been made to correlate the experimental data with the theoretical deductions but the results have been far from harmonious. This, however, has had little effect on the practical development of the boiler and it is quite probable that a more complete understanding of the phenomena will have no radical effect on the present design.

The resistance of the metal itself is so small that it may be neglected in calculating the total heat transmission and it may be logically assumed that the plate will take care of all the heat that reaches its dry surface.

The three distinct methods of heat transfer, radiation, convection and conduction, do not exist separately in the modern steam boiler but are operating at the same time. For this reason and in view of the number of arbitrary coefficients entering into the theoretical treatment of each method of heat transfer, engineers find it simpler to consider only the total heat transfer and to use empirical or semi-empirical equations. Thus, the total heat transfer, assuming no losses in the transmission, may be expressed

$$SUd = WC_pt_m, \tag{33}$$

in which

S = square feet of heating surface,
U = mean coefficient of heat transfer, B.t.u. per sq. ft. per degree difference in temperature per hour,
d = mean temperature difference between the heated gases and the metal surface, deg. fahr.,
W = weight of gases flowing, lb. per hour,
C_p = average mean specific heat of the gases,
t_m = mean temperature drop of the gases between furnace and breeching, deg. fahr.

In practice U varies from 30 or more in the first row of tubes of a water tube boiler directly over the incandescent fuel bed to 5 or less in the last row immediately adjacent to the uptake.

Experiments conducted by Jordan and the Babcock & Wilcox Company indicate that the value of U varies approximately as follows:

$$U = K + B\frac{W}{A},^* \tag{34}$$

* Trans. Int. Eng. Congress, "Mechanical Engineering," 1915, p. 366.

in which

K = coefficient determined experimentally,

B = a function of the dimension of air passage and mean temperature difference of the gas and metal,

A = average cross sectional area of the gas passages through the boiler.

Other notations as in equation (33).

For the standard type of Babcock & Wilcox water tube boiler, the Company's investigators found the following modification of equation (34) to give satisfactory results for 100 to 150 per cent ratings.

$$U = 2.0 + 0.0014 \frac{W}{A}. \tag{35}$$

The curves in Fig. 61 may be used as a guide in approximating the heat transfer in fire tube boilers.

An examination of equation (34) shows that for a given set of conditions and within certain limits the rate of heat transfer varies directly with the weight of gases flowing per unit area of gas passage. This is not strictly true since the rate of heat transfer varies as some power of the weight less than unity. But within narrow limits it is sufficiently accurate to consider the exponent as unity.

Experiments by Professor Nicholson * and the U. S. Geological Survey † show that by establishing a powerful scrubbing action between the gases and the boiler plate the protecting film of gas is torn off as rapidly as it is formed and new portions of the hot gases are brought into contact with the plate, thereby greatly increasing the rate of heat transmission. Similarly, the faster the circulation of the water the greater will be the scrubbing action tending to remove the bubbles of steam from the wet surface and the more rapid will be the transfer from the plate to the boiler water.

Professor Nicholson found that by filling up the flue of a Cornish boiler with an internal water vessel, leaving an annular space of only 1 inch around the latter, an evaporation eight times the ordinary rate was effected at a flow of gases 330 feet per second (8 to 10 times the average flow). The fan for creating the draft consumed about $4\frac{1}{2}$ per cent of the total power.

The conclusion is that the heating surface for a given evaporation at the present rating may be reduced as much as 90 per cent for the same output, with a corresponding reduction in the size, cost, and space requirements, or with a given heating surface of standard rating the

* Proc. Inst. of Engr. & Shipbuilders, 1910.

† Bul. 18, U. S. Bureau of Mines, 1912.

output may be enormously increased; also the increase in power necessary to create the draft is by no means comparable with the advantages gained.

The modern locomotive boiler is the nearest approach to these conditions in practice. Here a powerful draft forces the heated gases

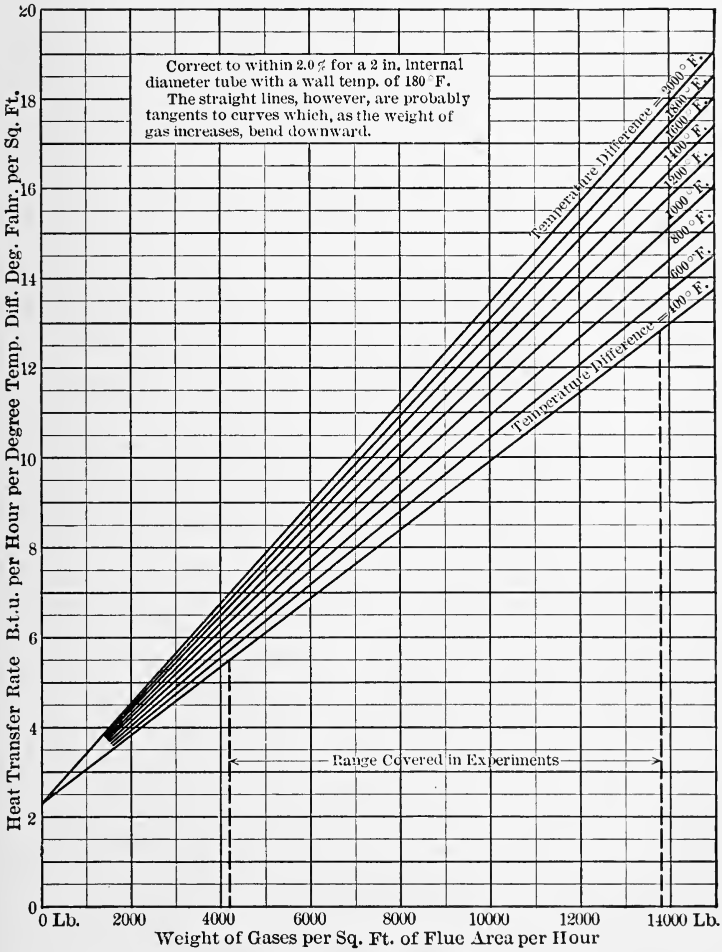


FIG. 61. Heat Transfer in Boiler Flues. Results of Experiments by the Babcock & Wilcox Company.

through small tubes at a very high velocity and an enormous evaporation is effected with a comparatively small heating surface.

These principles have been applied to a limited extent to stationary boilers already installed by making the gas passages smaller as compared to the length by means of suitable baffles (Fig. 53) and by forcing larger weight of gas through the boiler, either by forced draft or by increasing the grate area (Fig. 103).

In a general sense when the capacity of a boiler is doubled or tripled the over-all efficiency of the whole steam-generating apparatus drops, but the advantage gained usually offsets the loss in fuel economy. In the modern central station equipped with mechanical stokers of the forced draft type the boilers are operated normally on a basis of 5 to 6 sq. ft. of heating surface per boiler horsepower, and at peak loads 3 sq. ft. per horsepower is not unusual.

On the Transmission of Heat in Boilers, Hedrick & Fessenden, Jour. A.S.M.E., Aug. 1916, p. 619.

Heat Transmission in Boilers, Kreisinger and Ray: Tech. Paper 114, U. S. Bureau of Mines, 1915; Power and Engr., June 29, 1909, p. 1144; Bulletin No. 18, U. S. Bureau of Mines, 1912.

Some Notes on Heat Transmission and Efficiencies of Boilers, R. Royds, Trans. Inst. of Engr. & Shipbuilders in Scotland, Vol. 58, 1915.

The Heat of Fuels and Furnace Efficiency: W. D. Ennis, Power and Engr., July 14, 1908, p. 50.

A Study in Heat Transmission: (The transmission of Heat to Water in Tubes as Affected by the Velocity of the Water), J. K. Clement and C. M. Garland, Univ. of Ill. Bulletin No. 40, Sept. 27, 1909; Power & Engr., Feb. 7, 1911, p. 222.

Heat Transmission in Tubes, Dr. Wilhelm Nusselt: Zeit. d. ver. Deut. Ingr., 1909, p. 750.

On the Rate of Heat Transmission Between Fluids and Metal Surfaces, H. P. Jordan, Pro. Inst. Mec. Engrs., 1909.

74. Heating Surface. — All parts of the boiler shell, flues, or tubes which are covered by water and exposed to hot gases constitute the heating surface. Any surface having steam on one side and exposed to hot gases on the other is superheating surface. According to the recommendations of the American Society of Mechanical Engineers, the side next to the gases is to be used in measuring the extent of the heating surface. Thus measurements are made of the inside area of fire tubes and the outside area of water tubes. The heating surface in a boiler under average conditions of good practice is most efficient when the heated gases leave the uptake at a temperature of 75 to 150 deg. fahr. above that of the steam. Each square foot of heating surface is capable of transmitting a certain amount of heat, depending upon the *conductivity of the material, the character of the surface, the temperature difference between the gases and the metal surface, the location and arrangement of the tubes and the density and the velocity of the gases.*

Thus one square foot of heating surface in the first pass of a water-tube boiler immediately over the incandescent mass of fuel may evaporate as high as 75 pounds of water per hour from and at 212 deg. fahr., whereas the same extent of surface close to the breeching evaporates less than one pound per hour. Because of this extreme variation it is convenient to assume a uniform heat transmission for the entire surface which will give the same total evaporation as that actually obtained. For maximum economy under *average* conditions of hand fired operation this gives a *mean* evaporation of 3 to 3.5 pounds of water per square foot per hour from and at 212 deg. fahr., which is equivalent to allowing 10 to 12 square feet per boiler horsepower. By providing a large combustion chamber, increasing the extent of the first pass or the equivalent and by carrying a very thick bed of fuel a *mean* evaporation of 10 pounds per square foot per hour has been maintained with high economy. This corresponds to 3.5 square feet of heating surface per boiler horsepower.

The maximum evaporation is limited only by the amount of coal which can be burned upon the grate. For example, a *mean* evaporation as high as 23.3 pounds * per square foot per hour has been effected in locomotive work, under intense forced draft, and 20 pounds per square foot per hour is not unusual in torpedo boat practice. Such extreme, high rates of evaporation, however, are invariably obtained at the expense of fuel economy. In the very latest central stations the boiler and settings are proportioned to operate at 100 per cent above standard rating with high over-all efficiency and at 200 per cent above rating with only a small drop in efficiency, but such results are not obtainable in the ordinary hand fired boiler and setting.

Builders of return tubular and vertical fire-tube boilers allow from 11 to 12 square feet of heating surface per horsepower; water-tube boilers are rated at 10 square feet per horsepower, and Scotch-marine boilers at 8 square feet per horsepower.

See, also, paragraph 79, Effect of Capacity on Efficiency.

The following table shows approximately the relation between boiler horsepower and heating surface for different rates of evaporation:

EVAPORATION FROM AND AT 212 DEG. FAHR. PER SQUARE FOOT PER HOUR.

2	2.5	3.0	3.5	4	5	6	7	8	9	10
SQUARE FEET HEATING SURFACE REQUIRED PER HORSEPOWER.										
17.3	13.8	11.5	9.8	8.6	6.8	5.8	4.9	4.3	3.8	3.5

Efficiency of Boiler Heating Surface: Trans. A.S.M.E., 18–328, 19–571. Kent, "Steam Boiler Economy" (John Wiley & Sons), Chapter IX.

The Nature of True Boiler Efficiency: Jour. West. Soc. Engrs., Sept. 18, 1907.

Heat Transference through Heating Surface: Engineering, 77–1.

* Jour. A.S.M.E., Jan., 1915, p. 22.

75. The Horsepower of a Boiler.* — A boiler horsepower is equivalent to the evaporation of 34.5 pounds of water per hour from a temperature of 212 deg. fahr. to steam at atmospheric pressure. This corresponds to 33,479 B.t.u. per hour. Since the power from steam is developed in the engine and the boiler itself does no work, the above measure of capacity is merely conventional but unfortunately leads to much confusion. Thus one boiler horsepower will furnish sufficient steam to develop about four actual horsepower in the best compound condensing engine, but only one-half horsepower in a small non-condensing engine. Boilers should be purchased on the basis of heating surface and not on the horsepower rating, since one bidder may offer a boiler with, say, 5 square feet of heating surface per horsepower and another with 10 square feet, both being capable of the required evaporation, but the one with the small heating surface (which will, of course, be the cheaper boiler) will have considerably less reserve capacity. Manufacturers ordinarily rate their boilers on the basis of from 10 to 12 square feet of heating surface per horsepower, and the power assigned is called the *builder's rating.* As this practice is not uniform, bids and contracts should always specify the amount of heating surface to be furnished. According to the recommendations of the American Society of Mechanical Engineers, "A boiler rated at any stated capacity should develop that capacity when using the best coal ordinarily sold in the market where the boiler is located, when fired by an ordinary fireman, without forcing the fires, while exhibiting good economy. And, further, the boiler should develop at least one-third more than stated capacity when using the same fuel and operated by the same fireman, the full draft being employed and the fires being crowded; the available draft at the damper, unless otherwise understood, being not less than one-half-inch water column."

In determining the boiler horsepower required for a given engine horsepower it is convenient to estimate the steam consumption of the engine under actual conditions and then ascertain the equivalent evaporation from and at 212 deg. fahr. For example, assume a simple non-condensing engine developing 20 horsepower to use 50 pounds of steam per horsepower hour, or 1000 pounds steam per hour; steam pressure, 80 pounds per square inch; feed-water temperature 120 deg. fahr. Required the boiler horsepower necessary to furnish this quantity of steam.

* The unit "myriawatt" has been suggested by H. G. Stott as a unit of boiler capacity. For the conversion of myriawatts to other engineering units see Appendix F.

From equation (30), the factor of evaporation is

$$F = \frac{\lambda - q_2}{970.4} = \frac{1185.3 - 87.91}{970.4} = 1.131.$$

One thousand pounds of steam under the given conditions are therefore equivalent to $1000 \times 1.131 = 1131$ pounds from and at 212 deg. fahr.

The boiler horsepower necessary to furnish steam for the 20-horsepower engine will be

$$\text{Boiler horsepower} = \frac{1131}{34.5} = 32.8.$$

Example 13. A 15,000-kilowatt steam turbine and auxiliaries require 14.7 pounds of steam per kilowatt-hour at rated load; steam pressure, 200 pounds per square-inch gauge; superheat, 150 deg. fahr.; feed-water temperature, 179 deg. fahr.

Required the boiler horsepower necessary to furnish this quantity of steam.

The heat furnished to the turbine and auxiliaries per kilowatt-hour is

$$\begin{aligned} w(\lambda + C_p t_s - q_2) &= 14.7\,(1199.2 + 0.57 \times 150 - 146.88) \\ &= 16{,}724 \text{ B.t.u.,} \end{aligned}$$

$$\text{Boiler horsepower} = \frac{15{,}000 \times 16{,}724}{33{,}479} = 7500 \text{ (approx.).}$$

If the boilers are to be operated at builder's rating (10 sq. ft. of heating surface per boiler horsepower) 75,000 sq. ft. of heating surface would be necessary. In plants of this size, however, the boilers would in all probability be operated at 200 per cent rating or more when furnishing steam at full load requirements, and 37,500 sq. ft. of heating surface would suffice. Assuming 200 per cent, the ratio of installed horsepower (builder's rating) to kilowatts of rated turbine capacity would be 1 to 4 in this case. In large, modern central stations this ratio ranges between 1 to 5 and 1 to 8 (reserve boilers not included).

76. Grate Surface and Rate of Combustion. — The amount of fuel which can be burned per hour limits the amount of water evaporated per unit of time and depends upon the extent and nature of the grate surface, the character of the fuel and the draft. In locomotive and torpedo-boat practice space limitations necessitate the use of small grates and the rate of combustion is primarily a direct function of the draft. In stationary practice there is a wide permissible range in proportioning the grate surface, since a given rate of combustion may be effected with large grate surface and light draft or with small grate surface and strong draft. In a general sense the best results are obtained with a small grate and a high rate of combustion, but in the majority of installations the draft is comparatively feeble and a liberal grate area is necessary. So much depends upon the grade and size of the fuel that general rules for proportioning the grate surface are apt to lead to serious error. A liberal allowance of grate surface is desirable for hand-fired furnaces with natural draft, particularly if the ash is easily fusible, tending to choke the grate, but with forced draft and automatic stokers the best results are obtained with a thick fire and small grate surface. The maximum grate area is limited by the dimensions of the furnace. In hand-fired furnaces with stationary grates the width of the furnace is limited by that of the boiler and the length by the distance which the fireman can control the fuel bed. Anthracite requires no slicing and a much greater length of grate can be manipulated than with the caking variety of coals. Shaking and self-dumping grates may be of greater length than stationary grates since hand manipulation is largely dispensed with. The dimensions of mechanical stokers depend largely upon the type of stoking device. In practice the maximum rate of combustion, pounds per square foot of grate surface per hour, is usually assumed and the grate area and chimney height or equivalent proportioned to effect the desired rate of combustion. See Table 29.

The ratio of grate area to heating surface is sometimes used as a guide in proportioning the grate but the extent of grate surface depends upon so many other factors that this method of procedure is of little value and apt to lead to serious error. Thus, a study of several hundred boiler installations gave results as follows:

Type of Grate or Stoker.	Kind of Coal.	Ratio Grate Surface to Heating Surface.	
		Minimum.	Maximum.
Hand-fired	Anthracite	1 to 30	1 to 65
Hand-fired	Bituminous	1 to 40	1 to 78
Chain grate	Bituminous	1 to 36	1 to 72
Roney	Bituminous	1 to 30*	1 to 55
Taylor	Bituminous	1 to 50	1 to 82
Jones	Bituminous	1 to 55	1 to 68

* Double Stoker.

A number of boiler tests made by Barrus ("Boiler Tests") showed that the best economy with anthracite coal, hand-fired, was obtained with an average ratio of grate surface to heating surface of 1 to 36, and at a rate of combustion of approximately 12 pounds of coal per square foot of grate surface per hour. In these tests a variation in grate and heating-surface ratio of 1 to 36 up to 1 to 46 gave practically no difference in economy. With bituminous coal the tests showed that an average ratio of 1 to 45 gave the best results and at a rate of combustion of 24 pounds of coal per square foot of grate surface per hour.

Tests made by Christie (Trans. A.S.M.E., 19–330) gave an average combustion of 13 pounds of anthracite per square foot of grate per hour for maximum efficiency and 24 pounds of bituminous.

Table 32 gives the relation between heating and grate surface in a number of recent boiler installations using different kinds of coal, and is illustrative of current practice.

The rate of combustion depends upon the grade and size of coal, thickness of fire, percentage of air spaces in the grate, available draft through the fire and the efficiency of combustion, and can only be found accurately by experiment. For a general set of conditions the rate of combustion is primarily a function of the pressure difference between the ash pit and furnace, and is approximately as shown in Table 29.

TABLE 29.

MAXIMUM ECONOMICAL RATE OF COMBUSTION.

Pounds of Coal per Sq. Ft. of Grate Surface per Hour.

Kind of Coal.	Force of Draft between Furnace and Ash Pit, Inches of Water.								
	0.1	0.15	0.20	0.25	0.30	0.35	0.40	0.45	0.50
Anthracite, No. 3	3.5	5.0	6.2	7.2	8.2	9.0	10.0	11.0	12.0
Anthracite, No. 2	5.5	7.2	9.0	10.5	11.8	13.0	14.2	15.5	16.5
Anthracite, Pea	8.0	9.2	11.5	13.5	15.0	17.0	18.5	20.0	21.5
Semi-bituminous	9.5	13.0	16.0	19.0	23.0	25.0	27.0	30.0	32.0
Ky., Pa., and Tenn. bitum.	10.0	14.0	18.0	21.0	24.0	27.0	30.0	33.0	36.0
Ill., Ind., and Kan. bitum.	11.0	15.0	20.0	24.0	28.0	32.0	35.0	38.0	41.0

With forced draft these rates of combustion may be greatly increased. Some idea of the extreme rate of combustion in modern locomotive practice may be obtained from the following figures which give the pounds of coal burned per hour per square foot of grate surface for various conditions of operation:

Maximum rate	225	Average rate	80
Very high rate	150	Economical rate	60
Average high rate	100	Low rate	50

In proportioning the grate surface for a proposed installation the principal factor considered is the character of the fuel, a study being made of the various fuels available, and the one selected which gives the highest evaporation per dollar (all items entering into the handling and combustion of the fuel being considered). This information may usually be obtained from records of plants using the same grade of fuel and grates similar to those intended for the proposed plant.

77. Boiler, Furnace and Grate Efficiency. — A perfect boiler and furnace is one which transmits to the water in the boiler the total heat of the fuel. In order to effect this result combustion must be complete, there must be no radiation or leakage losses and the products of combustion must be discharged at the initial temperature of the fuel. No commercial form of steam boiler can fulfill these conditions, hence the amount of heat absorbed by the boiler will always be less than the calorific value of the fuel.

The efficiencies recommended by the A.S.M.E., Rules for Conducting Boiler Tests, 1915, may be expressed as

$$\text{Efficiency of boiler, furnace and grate} = \frac{\text{Heat absorbed by the boiler per pound of } \textit{coal as fired}}{\text{Calorific value of one pound of } \textit{coal as fired}}, \tag{36}$$

$$\text{Efficiency based on combustible} = \frac{\text{Heat absorbed by the boiler per pound of } \textit{combustible burned} \text{ on the grate*}}{\text{Calorific value of one pound of } \textit{combustible as fired}}. \tag{37}$$

Example 14. Calculate the various boiler efficiencies from the following data:

DATA AS OBSERVED.

Steam pressure, pounds per square inch (gauge)	151.0
Barometer, inches of mercury	28.5
Temperature of feed water, deg. fahr.	161.8
Temperature of the furnace, deg. fahr.	2100.0
Temperature of flue gases, deg. fahr.	480.0
Temperature of boiler room, deg. fahr.	60.0
Quality of steam, per cent	98.0
Water apparently evaporated, pounds per hour	86,000
Coal as fired, pounds per hour	10,000
Refuse removed from ash pit, pounds per hour	1600

COAL ANALYSIS, PER CENT OF COAL AS FIRED.

Moisture	8
Ash	12

B.t.u. per pound, 11,250.

CALCULATED DATA.

Water apparently evaporated per pound of coal as fired, pounds = 86,000 ÷ 10,000 = 8.60.

Factor of evaporation † = $[0.98 \times 856.8 + 338.2 - (161.8 - 32)] \div 970.4 = 1.08$.

* The combustible burned on the grate is determined by subtracting from the weight of coal supplied to the boilers, the moisture in the coal, the weight of ash and unburned coal withdrawn from the furnace and ash pit and the weight of dust, soot, and refuse, if any, withdrawn from the tubes, flues and combustion chambers, including soot and ash carried away in the gases.

† See footnote, par. 72.

Equivalent evaporation per pound of coal as fired, pounds = 8.6 × 1.08 = 9.288.
Heat absorbed by the boiler per pound of coal as fired, B.t.u. = 9.288 × 970.4 = 9,013.0.
Efficiency of boiler furnace and grate, per cent = (9.013 ÷ 11,250) 100 = 80.11.
Refuse in ash referred to coal as fired, per cent = (1600 ÷ 10,000) 100 = 16.0.
Combustible burned on the grate, per cent of coal as fired = 100 − (8 + 16) = 76.0.
Equivalent evaporation per pound of combustible burned, pounds = 9.288 ÷ 0.76 = 12.221.
Heat absorbed per pound of combustible burned, B.t.u. = 12.221 × 970.4 = 11,860.
Combustible as fired, per cent = 100 − (8 + 12) = 80.00.
Calorific value of the combustible as fired, B.t.u. = 11,250 ÷ 0.80 = 14,062.
Efficiency based on combustible, per cent = (11,860 ÷ 14,062) 100 = 84.34.

For oil fuel furnaces and coal furnaces equipped with stokers and forced draft appliances the *net* efficiency of the boiler and furnace may be taken as the boiler and furnace efficiency minus the equivalent heat required to feed the fuel and to create the draft.

Attempts have been made to separate the combined efficiency of boiler, furnace, and grate into two parts, viz., efficiency of the boiler alone and efficiency of the furnace and grate, but the results have been discordant and involve the use of factors which cannot be obtained with any degree of accuracy. Thus "true" boiler efficiency has been defined as the ratio of the heat absorbed to that available. The "heat absorbed" is taken as the difference between the heat generated in the furnace and that discharged into the flue, and the "available" heat is defined as the difference between the heat generated in the furnace and that discharged by the products of combustion at the temperature of the saturated steam.

If w_f, w_c = weight of the products of combustion in the furnace and passing through the uptake, respectively, lb. per hour

T_f, T_c, T_s, T = absolute temperature of the furnace gases, flue gases, saturated steam and boiler room, respectively, deg. fahr.

c_f, c_c, c_s = mean specific heat of the products of combustion for temperature ranges t *to* t_f, t_c, t_s, respectively.

Then, neglecting radiation and minor losses, the "true" boiler efficiency equals

$$E_1 = \frac{w_f c_f T_f - w_c c_c T_c}{w_f c_f T_f - w_c c_s T_s} \tag{38}$$

Assuming no leakage, $w_f = w_c$; and neglecting the difference in the mean specific heats, $c_f = c_c = c_s$. With these assumptions, equation (38) reduces to

$$E_1 = \frac{T_f - T_c}{T_f - T_s} = \frac{t_f - t_c}{t_f - t_s} \tag{39}$$

TABLE 30.

RELATION BETWEEN FUEL CONSUMPTION AND BOILER, FURNACE AND GRATE EFFICIENCY.

(Pounds of Fuel Burned per Boiler Horsepower-hour.)

Calorific Value of Fuel, B.t.u. per Pound.	Boiler, Furnace and Grate Efficiency.									
	40	45	50	55	60	65	70	75	80	85
7,500	11.17	9.91	8.94	8.12	7.45	6.87	6.37	5.95	5.58	5.25
8,000	10.45	9.30	8.37	7.60	6.97	6.43	5.98	5.58	5.22	4.92
8,500	9.84	8.75	7.87	7.12	6.56	6.05	5.62	5.25	4.97	4.63
9,000	9.30	8.25	7.45	6.76	6.20	5.72	5.31	4.96	4.65	4.36
9,500	8.80	7.83	7.05	6.40	5.87	5.41	5.02	4.69	4.40	4.14
10,000	8.37	7.44	6.70	6.09	5.58	5.15	4.79	4.46	4.18	3.94
10,500	7.98	7.09	6.39	5.80	5.86	4.90	4.56	4.26	3.99	3.76
11,000	7.60	6.79	6.09	5.52	5.06	4.67	4.34	4.05	3.80	3.59
11,500	7.28	6.49	5.83	5.29	4.85	4.47	4.16	3.88	3.64	3.45
12,000	6.97	6.22	5.58	5.06	4.65	4.28	3.99	3.72	3.48	3.28
12,500	6.69	5.97	5.35	4.86	4.46	4.11	3.82	3.57	3.34	3.14
13,000	6.44	5.74	5.15	4.68	4.29	3.96	3.68	3.43	3.22	3.02
13,500	6.20	5.52	4.96	4.51	4.18	3.81	3.54	3.31	3.10	2.91
14,000	5.98	5.33	4.79	4.35	3.99	3.68	3.42	3.19	2.99	2.81
14,500	5.77	5.15	4.62	4.20	3.84	3.54	3.30	3.08	2.88	2.72
15,000	5.58	4.96	4.47	4.06	3.72	3.43	3.19	2.98	2.79	2.64

TABLE 31.

RELATION BETWEEN RATE OF EVAPORATION PER POUND OF FUEL AND BOILER, FURNACE AND GRATE EFFICIENCY.

(Pounds of Water Evaporated per Hour from and at 212 deg. fahr. per Pound of Fuel.)

Calorific Value of Fuel, B.t.u. per Pound.	Boiler, Furnace and Grate Efficiency.									
	40	45	50	55	60	65	70	75	80	85
7,500	3.09	3.48	3.86	4.25	4.64	5.02	5.41	5.80	6.18	6.57
8,000	3.30	3.71	4.12	4.55	4.95	5.36	5.77	6.18	6.60	7.01
8,500	3.51	3.94	4.38	4.81	5.26	5.70	6.14	6.57	7.01	7.45
9,000	3.71	4.18	4.64	5.10	5.56	6.04	6.50	6.96	7.42	7.90
9,500	3.92	4.41	4.90	5.39	5.88	6.47	6.86	7.35	7.85	8.33
10,000	4.12	4.64	5.16	5.66	6.19	6.70	7.21	7.74	8.25	8.76
10,500	4.31	4.86	5.40	5.94	6.48	7.01	7.55	8.10	8.64	9.17
11,000	4.52	5.09	5.65	6.22	6.79	7.35	7.91	8.48	9.05	9.61
11,500	4.74	5.31	5.91	6.50	7.10	7.69	8.28	8.86	9.45	10.0
12,000	4.94	5.55	6.16	6.78	7.40	8.01	8.64	9.25	9.86	10.5
12,500	5.14	5.78	6.42	7.06	7.70	8.35	9.00	9.64	10.3	11.0
13,000	5.35	6.01	6.69	7.35	8.01	8.69	9.35	10.0	10.7	11.4
13,500	5.56	6.25	6.95	7.65	8.34	9.03	9.72	10.4	11.1	11.8
14,000	5.75	6.48	7.20	7.91	8.64	9.35	10.1	10.8	11.6	12.2
14,500	5.96	6.70	7.45	8.20	8.95	9.70	10.5	11.2	12.0	12.7
15,000	6.18	6.95	7.72	8.50	9.26	10.1	11.8	11.6	12.4	13.1

The maximum theoretical efficiency of the boiler or the efficiency of the *ideal* or perfect boiler, based on utilizing all the heat except the inherent losses, may be expressed as

$$E_2 = \frac{H - I}{H}, \tag{40}$$

in which

H = calorific value of the coal as fired,

I = inherent losses as analyzed in paragraph 36.

The efficiency ratio or the extent to which the theoretical possibilities are realized may be taken as

$$E_3 = \frac{E}{E_2}, \tag{41}$$

in which

E = efficiency of the boiler, furnace and grate (A.S.M.E. code),

E_2 = as in equation (40).

The furnace and grate efficiency based on heat available may be expressed

$$E = \frac{H - (I + F)}{H - F}, \tag{42}$$

in which F = furnace losses consisting of the (*a*) loss due to unburned fuel dropping through the grate or withdrawn from the furnace, (*b*) loss due to the production of CO, (*c*) loss due to escape of unburned hydrocarbons, (*d*) loss due to the combustion of carbon and moisture and production of hydrogen when fresh moist coal is thrown on a bed of white hot coke, (*e*) radiation due to the furnace and (*f*) unaccounted for losses due to the furnace. (For an analysis of these losses see paragraphs 25 to 36.)

Equation (42) does not furnish a method of finding the true efficiency because it is impossible to determine loss (*d*) and impracticable to obtain loss (*c*) with the gas testing appliances ordinarily available. It is also impossible to separate losses (*e*) and (*f*) attributed to the furnace from the boiler losses alone due to radiation and unaccounted for.

In practice the operating engineer is chiefly concerned with the combined efficiency of the boiler, furnace and grate, as defined by the A.S.M.E. Boiler Code. This factor is readily determined with the ordinary instruments found in the average modern plant. Table 32, compiled from a number of tests of different types of boilers with different kinds of stokers and grades of fuel, gives some idea of the range of efficiencies incident to general practice. In attempting to better the efficiency it is necessary to separate the various losses as described in paragraphs 25 to 35, since this procedure enables the engineer to locate the source of loss, and by comparing the actual and inherent losses

TABLE 32.

EXAMPLES OF CURRENT STEAM BOILER PERFORMANCES.

Authority.	Type of Boiler.	Builder's Rating, Horsepower.	Percentage of Builder's Rating Developed.	Method of Firing.	Kind of Coal.	Grate Surface, Sq. Ft.	Ratio Heating Surface to Grate Area.	Coal Burned per Sq. Ft. Grate per Hr.	Apparent Evaporation per Lb. of Coal as Fired.	Equivalent Evaporation per Lb. of Combustible.	Equivalent Evaporation per Sq. Ft. H. S. per Hr.	Combined Efficiency of Boiler and Grate.
1	Almy	351	129	Hand fired	Anthracite egg	86.0	40.9	18.7	8.04	10.78	4.49	67.6
2	Babcock & Wilcox	107	130	....do	Pocahontas lump	25.0	42.8	20.1	10.48	12.33	4.31	73.8
3	Do	231	173	....do	No. 3 buckwheat	......	59.5	31.8	9.25	10.29	5.98	71.0
3	Do	......	109	....do	....do	......	35.9	16.1	8.37	11.99	3.74	68.8
4	Do	250	86	....do	Anthracite pea	56.0	41.5	9.8	10.40	13.20	2.60	......
5	Do	412	105	Roney stoker	New River	74.8	55.0	21.0	9.68	12.40	3.60	74.9
6	Do	420	140	Bennis stoker	......	50.0	84.0	35.2	9.95	......	4.83	82.2
7	Do	500	220	Chain grate	No. 4 bituminous nut	90.0	55.6	43.2	7.80	11.80	7.59	72.0
8	Do	650	128	Taylor stoker	......	106.0	61.4	23.1	10.00	12.94	4.52	79.8
8	Do	650	185	....do	......	106.0	61.4	35.1	9.45	12.21	6.51	72.7
9	Berry	250	123	Hand fired	Bituminous	46.0	40.7	20.3	9.60	11.40	5.68	......
10	Bigelow-Hornsby	169	145	......	Nixon's navigation	40.0	42.0	18.6	9.58	12.18	4.57	73.4
11	Edge Moor	600	188	Taylor stoker	Clearfield, mine run	92.0	66.5	42.5	8.14	11.60	6.36	71.3
12	Climax vertical	500	......	Hand fired	Nixon's navigation	64.0	78.0	17.6	9.10	......	......	79.3
12	Do	1000	......	Wilkinson stoker	George's Creek	113.6	73.5	27.0	8.96	......	......	72.7
13	Heine	210	98	Hand fired	Bituminous, mine run	40.5	50.0	18.4	7.92	10.74	3.53	66.8
14	Do	350	109	Chain grate	Bituminous screenings	72.0	48.5	23.9	6.85	9.64	4.18	66.8
15	Italian torpedo	111.5	583	......	......	28.0	39.8	121.0	6.00	......	20.09	......
16	Keeler	250	100	Hand fired	No. 2 washed bituminous	48.0	52.0	24.0	6.68	8.28	3.50	......
17	Manning	150	147	Taylor stoker	Pocahontas, mine run	29.0	56.5	29.0	9.86	12.24	4.89	76.4
18	Do	150	129	Hand fired	Pocahontas	28.7	48.2	12.3	8.22	12.00	4.90	66.9
19	Return tubular	80	218	Hawley	Ohio bituminous, mine run	22.0	43.6	26.8	8.65	10.84	6.26	......
20	Do	100	108	Chain grate	Semi-bituminous	20.0	50.0	19.2	8.92	11.40	3.10	76.0
20	Do	100	273	....do	....do	20.0	50.0	55.0	7.67	9.68	7.86	64.0
21	Do	150	124	Hand fired	....do	39.0	46.0	20.1	7.80	9.00	4.48	60.0
21	Do	150	135	Chain grate	....do	35.0	51.5	18.0	10.56	12.00	4.81	80.0

22	Do	200	122	Hand fired	....do	39.0	59.0	20.5	9.35	11.67	3.60	70.2
23	Robb-Mumford	107	90	Rocking, hand fired	George's Creek	24.5	44.0	13.2	8.75	10.60	2.93	69.0
24	Rust water tube	335	105	Roney stoker	Bituminous, crushed lump	68.0	49.0	17.0	....	12.21	3.63	75.5
24	Do	335	210	....do	....do	68.0	49.0	38.0	....	10.86	7.26	68.9
25	Sederholm	300	......	Shaking grate	Illinois screenings	51.0	57.8	24.9	8.05	10.05	3.87	....
26	Scotch-marine	75	108	Hand fired	Bituminous, mine run	15.0	35.2	20.5	8.20	10.25	5.01	70.5
26	Do	250	89	....do	Anthracite chestnut	48.0	38.0	15.2	9.98	12.27	4.80	76.2
26	Stirling	350	98	Chain grate	No. 3 washed bituminous	82.5	42.5	17.0	8.20	11.70	3.40	70.2
27	Do	150	122	....do	New River	36.0	41.5	14.7	11.20	12.50	4.20	79.0
28	Do	542	136	Jones stoker	Bituminous nut and slack	81.2	66.7	25.6	10.68	12.92	4.70	83.0
28	Do	542	227	....do	....do	81.2	66.7	44.7	10.10	11.98	7.82	79.3
29	Do	2365	105	Roney stoker	Bituminous	446.0	53.0	16.7	9.75	10.52	3.63	77.8
29	Do	2365	197	....do	....do	446.0	53.0	33.6	8.86	10.66	6.75	75.6
29	Do	2365	107	Taylor stoker	....do	405.0	58.5	18.8	9.80	11.55	3.72	80.3
30	Wickes horizontal	150	100	Shaking grate	Bituminous, mine run	27.0	64.4	21.2	7.25	9.79	2.93	67.5
30	Do	200	99	Murphy stoker	Bellmore pea	42.0	54.3	20.0	8.42	11.27	3.00	75.4
30	Wickes vertical	225	107	....do	New River, mine run	39.0	68.6	18.0	9.46	12.68	3.11	79.5
26	Vertical tubular	40	95	Hand fired	Pocahontas	9.5	52.2	13.7	8.36	10.20	2.80	63.5
26	Do	60	104	....do	No. 3 washed bituminous	12.0	59.7	21.3	7.66	9.88	3.00	61.7
31	Edgemoor	613	113	Taylor Stoker	Illinois screenings	91.0	67.4	28.5	8.37	11.71	3.28	80.9

1. G. H. Barrus. 2. Eng. Rec., July 25, 1903, p. 102. 3. B. & W. "Steam." 4. Eng. Rec., March 9, 1901, p. 220. 5. Power, Dec., 1901, p. 26. 6. Eng. Rec., April 8, 1905, p. 404. 7. Commonwealth Edison Co. 8. N. Y. Edison, Waterside. 9. Catalogue, Berry Boiler Co. 10. Circular, Bigelow Co. 11. Power, March 21, 1911, p. 447. 12. Catalogue, Climax Boiler Co. 13. U. S. Geological Survey, Professional paper No. 48, Part II. 14. Bul. No. 22. Vol. 2, University of Illinois. 15. Eng. Mag., Jan., 1912, p. 504. 16. Chicago Stock Exchange Bldg. 17. Everett Mills, Lawrence, Mass. 18. J. E. Denton. 19. Circular, Hawley Down Draft Co. 20. N. Y. Central H. R. R. R. 21. Circular, Universal Chain Grate Co. 22. Power, Aug. 2, 1910. 23. Eng. Mag., April, 1904. 24. Eng. U. S., Feb. 15, 1908, p. 232. 25. Eng. Rec., Dec. 20, 1902, p. 584. 26. Tests conducted by the author. 27. Circular, Ironworks Co., New Jersey. 28. J. W. Hill. 29. Jour. A.S.M.E., Nov., 1911. 30. Catalogue, Wilkes Boiler Co. 31. Jour. A.S.M.E., June, 1914, p. 220.

For tabulated results of 51 boiler and underfeed stoker tests see Electrical World, Sept. 9, 1916, p. 519.

show where improvement may be effected. Although efficiencies of 80 per cent or more have been realized in several instances without the use of economizers, such performances cannot be expected for continuous operation. In pumping stations or in plants where there are no peak loads and the boiler may be operated under a constant set of conditions a continuous efficiency of 75 per cent has been realized with coal as fuel and 80 per cent with fuel oil, but these figures are exceptional. In large central stations with the usual peak loads in the morning and evening and long banking periods, over-all yearly efficiency is seldom greater than 70 per cent, though the boilers may be giving 77 to 81 per cent efficiency when operating at the most economical load. In large isolated stations with variable loads an over-all boiler and furnace efficiency on the yearly basis of 65 per cent is exceptional and a fair average is not far from 60 per cent. Small stations that show at times an efficiency as high as 75 per cent seldom average 50

TABLE 33.

PRINCIPAL DATA AND RESULTS OF TESTS ON 2365-RATED-HORSEPOWER STIRLING BOILERS AT THE DELRAY STATION OF THE DETROIT EDISON COMPANY.

Tests with Roney Stoker. Résumé of Principal Results.

No. of Test.	Length, Hr.	Per Cent Rating.	B.t.u. in Coal.	Per Cent Ash in Dry Coal.	Efficiency.	Per Cent Steam used by Stoker Engines and Steam Jets.	Per Cent Combustible in Ash.	Temp. of Flue Gases Leaving Boiler, Deg. Fahr.
1	25	105.0	14,362	5.98	77.84	....	19.6	576
2	24	80.0	14,225	6.52	79.88	....	17.9	480
3	24	113.8	14,308	7.40	77.45	0.63	24.4	542
4	30	152.4	13,756	6.54	75.78	1.58	30.8	670
5	24	94.0	13,896	6.89	81.15	1.75	31.6	483
6	24	150.7	14,037	6.13	75.28	1.45	26.7	662
16	32	98.6	14,476	9.68	80.98	1.34	34.1	460
17	16.5	193.3	14,493	8.24	76.73	1.39	24.6	636
18	24	195.7	13,689	9.81	75.57	1.32	23.2	694
2–4†	90	119.8	14,098	6.81	76.13	....	25.8	572
5–6†	55	127.3	13,977	6.84	76.23	....	29.4	575

† Including periods between tests.

Tests with Taylor Stoker. Résumé of Principal Results.

No. of Test.	Length, Hr.	Per Cent Rating.	B.t.u. in Coal.	Per Cent Ash in Dry Coal.	Efficiency.	Per Cent Steam used by Stoker Auxiliaries.*	Per Cent Combustible in Ash.	Temp. of Flue Gases Leaving Boiler, Deg. Fahr.
7	24	151.2	14,000	7.03	77.07	2.61	31.5	575
8	24	107.9	13,965	6.34	80.28	2.44	27.1	493
9	50	162.8	13,998	6.75	77.85	2.87	31.3	574
10	48	92.9	14,188	9.90	77.90	2.63	27.2	487
11	26.5	211.3	14,061	9.55	75.84	3.41	36.1	651
12	48	121.3	14,010	8.09	79.24	2.57	27.6	535
14	24	185.2	14,272	8.71	76.42	2.95	28.8	647
15†	24	123.1	14,213	8.34	74.90	2.77	30.1	561
7–9‡	109	140.0	13,983	7.22	77.66	2.68	29.9	545
10–11‡	80.5	132.8	14,095	9.58	75.66	3.04	31.1	542

* Engines driving stokers and steam-turbine driving fan.

† In test No. 15 the fires were banked for 7½ hours and the averages include this period.

‡ Including periods between tests.

per cent to the year. These figures refer to boiler installations without economizers. For influence of the latter on boiler, furnace and grate efficiency see Paragraph 285. In general the over-all efficiency is dependent primarily on the load factor. The greater the load factor the smaller will be the standby losses (see Paragraph 35) and the nearer will the over-all efficiency approach test results. The usual discrepancy between efficiency as determined by special tests and average operation is due to the fact that the efficiency test is usually conducted under ideal conditions. The boiler surfaces are cleaned, the rate of combustion carefully adjusted to maximum economy and special attention given the

TABLE 34.*

PRINCIPAL DATA AND RESULTS OF TESTS ON BOILER NO. 6, UNIT NO. 10, FISK ST. STATION. COMMONWEALTH EDISON CO., CHICAGO.

(B. & W. Boiler, " Standard " Setting.)

Water-heating Surface, 5000 Sq. Ft. Superheating Surface, 914 Sq. Ft. Chain Grate Surface, 90 Sq. Ft.

Test No.	Date, 1908.	Horse-power.	Efficiency, Per Cent.	Horse-power per Sq. Ft. Grate.	Heat Lost in Refuse, Per Cent.	Total Heating Surface per Horse-power.	Super-heat of Steam, Deg. Fahr.	Dry Coal per Sq. Ft. G. S. per Hour.
2	Mar. 9	873	67.4	9.70	2.8	6.76	197	41.2
4	" 10	873	69.0	9.52	2.8	6.89	195	39.1
6	" 11	852	67.3	9.47	2.8	6.93	189	38.9
8	" 16	836	65.3	9.29	6.4	7.06	174	39.5
10	" 17	870	68.8	9.67	5.0	6.78	180	39.3
14	" 19	920	66.2	10.22	9.2	6.42	187	43.7
16	" 23	900	69.5	10.00	4.0	6.56	181	40.5
18	" 24	916	69.1	10.18	5.5	6.44	190	41.6
20	" 26	912	69.2	10.13	4.4	6.48	179	41.2
22	" 27	906	67.7	10.07	4.1	6.52	194	42.5
24	" 30	925	69.8	10.28	2.8	6.38	179	41.6
26	" 31	894	69.4	9.93	5.2	6.60	170	40.6
28	Apr. 1	922	71.2	10.24	3.6	6.40	169	40.4
30	" 2	923	71.5	10.26	4.6	6.40	173	40.5
32	" 7	914	70.0	10.20	4.5	6.46	175	40.9
34	" 8	939	73.8	10.4	3.8	6.28	181	40.4
36	" 10	911	70.9	10.1	3.0	6.48	185	40.2
38	" 11	967	70.1	10.7	3.0	6.11	192	42.6
40	" 13	995	67.8	11.1	3.4	5.93	211	43.6
42	" 14	887	66.8	9.9	4.5	6.65	202	40.8
44	" 27	880	69.5	9.8	5.5	6.72	169	39.7
48	" 29	927	71.5	10.3	3.3	6.37	171	40.8
50	" 30	899	70.3	10.0	4.2	6.57	171	39.6
52	May 6	886	69.4	9.8	5.3	6.67	171	38.2
54	" 7	900	69.1	10.0	4.8	6.56	171	39.2
56	" 8	967	71.9	10.7	4.8	6.10	164	40.1
58	" 11	902	70.5	10.0	3.3	6.55	163	39.6
60	" 13	875	70.7	9.7	3.8	6.74	147	38.3
64	" 14	1102	72.0	12.2	4.8	5.35	180	43.2

* This unit is still (1916) in operation and while the baffling has been changed the results are approximately the same as given in the table.

TABLE 34. (*Continued.*)

PRINCIPAL DATA AND RESULTS OF TESTS ON BOILER NO. 6, UNIT NO. 10, FISK ST. STATION. COMMONWEALTH EDISON CO., CHICAGO.

(B. & W. Boiler, " Standard " Setting.)

Water-heating Surface, 5000 Sq. Ft. Superheating Surface, 914 Sq. Ft.
Chain Grate Surface, 90 Sq. Ft.

Draft		B.t.u. per Pound Dry Coal.	Ash in Dry Coal, Per Cent.	Ash in Refuse, Per Cent.	Uptake Temp. Deg Fahr.	CO_2, Per Cent.	Heat Lost up Stack (Dry Gas), Per Cent.
Over Fire.	In Uptake.						
0.87	1.34	11,634	18.46	82.33	466	6.9	....
0.78	1.25	11,759	16.81	81.36	461	6.7	....
0.83	1.25	12,039	16.08	80.03	463	7.7	15.6
0.94	1.34	11,993	15.91	67.42	477	7.6	16.8
0.84	1.24	11,909	15.71	71.32	475	7.9	16.2
0.99	1.41	11,768	16.04	63.78	479	8.5	15.4
0.77	1.17	11,846	16.68	79.04	483	9.1	14.0
0.81	1.25	11,800	16.39	71.98	484	8.3	15.8
0.77	1.21	11,846	15.51	78.53	486	9.0	14.5
0.78	1.22	11,659	17.59	80.58	494	9.2	14.6
0.68	1.28	11,800	16.22	82.97	487	8.8	15.1
0.70	1.24	11,752	16.18	76.84	484	8.8	15.1
0.62	1.21	11,862	15.38	82.99	480	9.2	14.1
0.58	1.40	11,800	16.02	78.37	480	9.1	14.4
0.73	1.24	11,815	16.84	77.84	494	9.0	14.7
0.72	1.25	11,659	18.06	82.27	504	8.9	15.3
0.65	1.13	11,831	17.15	86.92	493	9.7	13.4
0.70	1.24	12,002	16.05	84.39	502	9.0	15.1
0.71	1.23	12,469	14.87	82.14	522	9.7	13.3
0.63	1.09	12,049	15.17	78.12	500	9.5	13.3
0.71	1.26	11,801	15.75	77.21	470	8.3	15.7
0.68	1.23	11,769	18.59	84.04	472	8.7	14.2
0.66	1.27	11,955	16.11	79.30	473	7.9	16.1
0.62	1.20	12,360	13.63	74.59	476	8.8	14.5
0.66	1.31	12,298	13.62	75.19	480	9.0	14.4
0.66	1.29	12,423	13.37	75.61	474	9.4	13.3
0.92	1.18	11,956	17.45	83.24	451	9.2	12.5
0.76	0.98	11,971	17.45	80.99	443	10.0	11.2
0.68	1.15	13,126	10.24	70.90	487	10.4	12.1

firing, whereas, in most plants these refinements are seldom attempted. In our strictly modern boiler plants, refinement of design and a systematic supervision of operation have resulted in over-all efficiencies far above anything hitherto thought possible.

The boiler, furnace and grate efficiency is only one of the many factors entering into the economical operation of the boiler plant. Different fuels may give the same efficiency under actual operating conditions, but the ultimate economy in dollars and cents may vary considerably. The real criterion is the net cost of evaporation, taking into consideration the cost of handling the fuel, disposition of refuse, ability to handle peak loads and depreciation of grate and setting.

The common practice of comparing the performance of boilers on the "fuel cost to evaporate 1000 pounds of water," is apt to lead to erroneous conclusions; thus, the cost of evaporating 1000 pounds of water from and at 212 deg. fahr., per pound of cheap bituminous screenings, may be 12 cents as against 18 cents per pound of high-grade and more costly washed coal, but the freight charges, cost of handling the fuel and disposition of the ash may more than offset the gain in evaporation and the cheaper fuel may prove to be the more expensive in the end. Each installation is a problem in itself and all local influencing conditions must be considered before maximum economy can be effected. In general, for plants equipped with coal and ash handling machinery and adjacent to a railroad or to water transportation, the cheaper the fuel per pound of combustible the lower will be the ultimate cost of evaporation.

Report of the Power Test Committee, A.S.M.E. Boiler Code, 1915, Jour. A.S.M.E., Vol. 37, 1915, p. 1273. This report may be had in pamphlet form. See also Appendix A.

78. Boiler Capacity. — Boilers are ordinarily rated on a commercial basis of 10 square feet of heating surface per horsepower. This rating is absolutely arbitrary and implies nothing as to the limiting amount of water that this amount of heating surface will evaporate. It has long been known that the evaporative capacity of a well-designed boiler is limited only by the amount of fuel that can be burned on the grate. Thus in locomotive practice a boiler horsepower has been developed with two square feet of heating surface and in torpedo boat practice this figure has been reduced to 1.8 square feet. If there were no practical limitations to capacity few, if any, boilers would be operated at the rated load and the amount of heating surface for a given evaporation would be only a fraction of the present requirements. Briefly stated the limitations are:

1. *Efficiency.* — As the capacity increases beyond a certain limit the over-all efficiency drops off and a point is reached where further increase in capacity is obtained at a cost greater than that of additional heating surface.

2. *Grate Surface.* — All fuels have a maximum rate of combustion beyond which satisfactory results cannot be obtained. With this limit established the only method of obtaining added capacity is through the addition of grate surface. Since the grate surface for a given boiler is limited by the impracticability of operating economically above a certain size there is obviously a commercial limit to the maximum weight of fuel burned per unit of time.

3. *Draft.* — In order to effect a heavy rate of combustion a great increase in draft is necessary. Apart from the power required to produce the draft there is the loss of fuel carried away in the "cinders."

4. At heavy rates of driving the furnace and stoker upkeep may become excessive.

5. *Feed Water.* — For continuous high boiler overloads the feed water must be practically free from scale-forming elements and matter which tends to cause foaming and priming.

TABLE 35.

RELATION BETWEEN CAPACITY AND EFFICIENCY.

(Evaporation from and at 212 Deg. Fahr. per Square Foot of Heating Surface per Hour.)

2	2.5	3	3.5	4	5	6	8	10	12
Probable Relative Economy, Ordinary Installation.									
100	100	100	95	90	85	80	70	60	50
Probable Relative Economy, Latest Improved Installation.									
95	98	100	100	100	99	98	95	90	85

TABLE 36.

FLUE GAS TEMPERATURES CORRESPONDING TO FORCED CAPACITY OF BOILERS IN MODERN POWER PLANT INSTALLATIONS.

Plant.	Type of Boiler.	Rated Horse-power per Unit.	Heat Surface per Horsepower Developed.	Flue Temperature.	Builders' Rating, Per Cent.
Buffalo General Electric...	B. & W.	1140	2.86	705	350
Cambridge Steel Co........	B. & W.	400	5.14	485	194
Commonwealth Edison Co..	B. & W.	650	4.97	588	201
Consolidated Gas, Baltimore....................	Edgemoor	736	4.52	551	211
University of Illinois.......	Locomotive	328*	2.07	703	486
Detroit Edison Co..........	Stirling	2365	4.75	651	211
Everett Mills..............	Manning	130	6.00	599	150
Interborough Rapid Transit Co., 74th St. Station..	B. & W.	520	3.00	631	335
Narragansett Electric Lighting Co.............	B. & W.	440	5.50	544.2	180
National Museum..........	Geary, W. T.	182	6.40	430	155
N. Y. Central R.R., West Albany..................	Edgemoor	600	5.28	543	193
N. Y. Central & H. R. R. R.	Ret. Tub.	100	4.40	630	273
N. Y. Edison Waterside....	B. & W.	650	5.48	550	179
Old Colony St. Ry.........	B. & W.	687	5.25	599	190
Union Gas & Electric Co...	Stirling	542	4.43	622	227

* Assuming 10 sq. ft. of heating surface per rated horsepower.

6. *External Surfaces.* Soot is such an excellent non-conductor of heat that provision must be made for its removal at frequent intervals, and particularly so if the boiler is expected to operate efficiently at heavy loads.

These factors are treated in detail elsewhere under their respective headings.

79. Effect of Capacity on Efficiency. — Tests show that if the furnace conditions are kept constant regardless of load, the efficiency of the *boiler* alone will decrease with increasing loads. But the furnace and grate efficiency increases with the capacity up to a certain point,

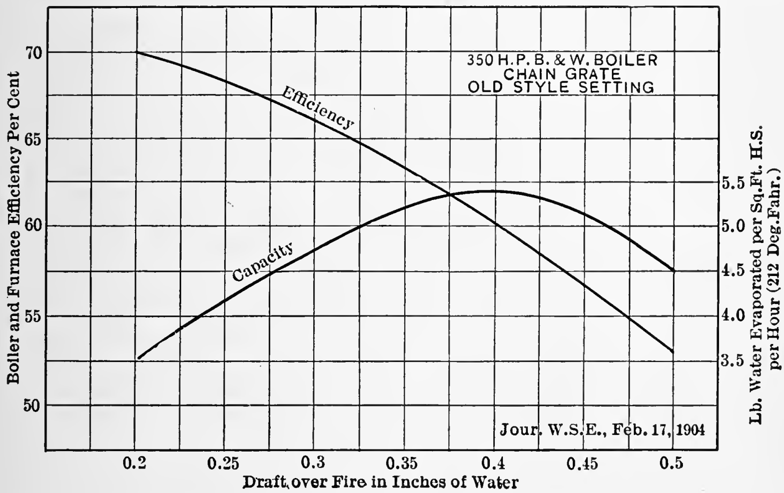


Fig. 62. Relation between Efficiency and Capacity.

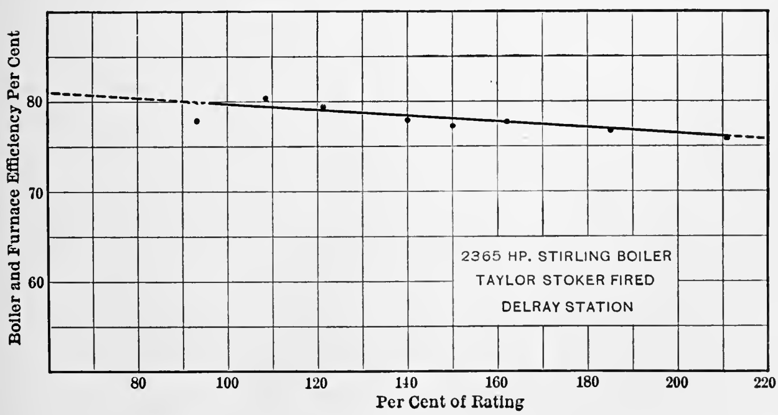


Fig. 63. Relation between Efficiency and Capacity.

beyond which it remains constant or gradually drops off. For a certain portion of the load this increase in furnace efficiency may be at a greater rate than the decrease in boiler efficiency. Consequently the maximum combined efficiency may occur at a point either side of the rated capacity or remain constant over a considerable range of ratings. In

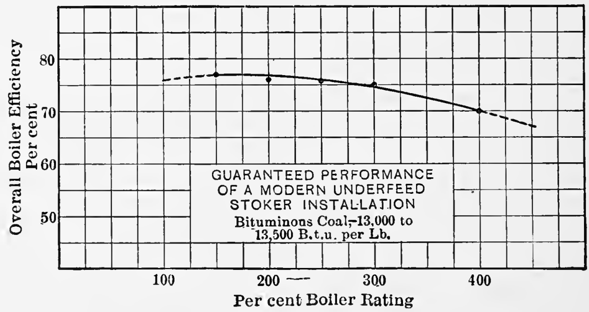


FIG. 64. Relation between Efficiency and Capacity.

general the combined efficiency of boiler, furnace and grate increases with the capacity until a maximum is reached, from which point it drops off steadily with each increment of increase in load. This point of maximum efficiency varies with the type and size of boiler, kind of grate, design of furnace, character of fuel and conditions of operation,

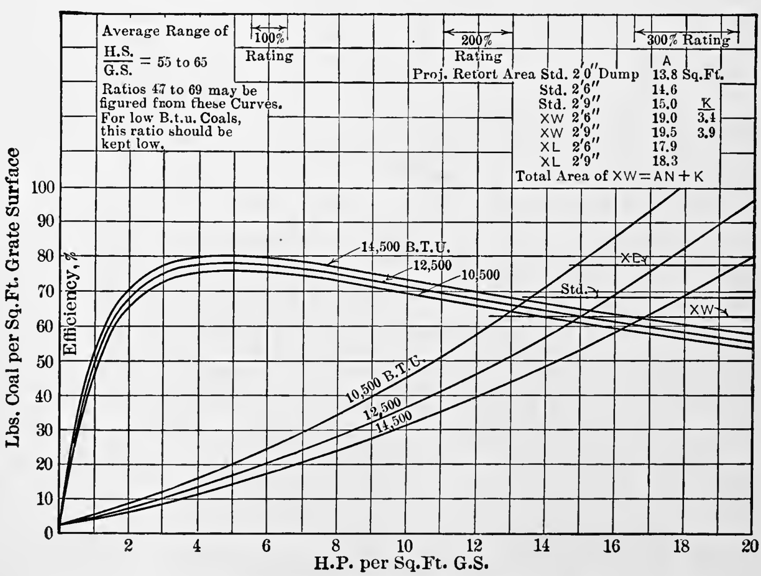


FIG. 65. Relation between Efficiency and Capacity. (Riley Stokers.)

and may range from a fraction to 200 per cent or more of the rating. With stokers of the underfeed type, other things being equal, the highest efficiency is obtained from the greatest number of retorts and the greatest effect on the over-all efficiency is the rate of driving per retort.

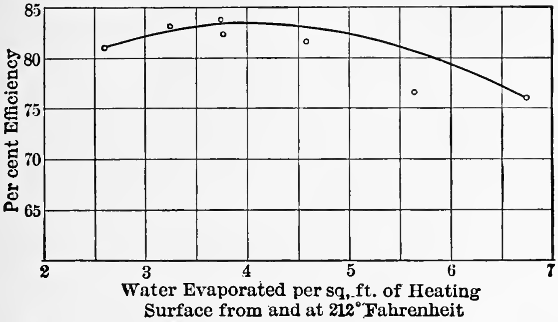


FIG. 66. Relation between Efficiency and Evaporation — Oil Fuel.

The curves in Figs. 62 to 67 are based upon authentic tests and give some idea of the effect of capacity on efficiency in specific cases. There are plants throughout the country in which boilers are developing, during periods of peak load, capacities of 400 per cent of the rating and 500 per cent has been realized in torpedo boat practice, but such loads cannot be maintained continuously with any degree of ultimate economy.

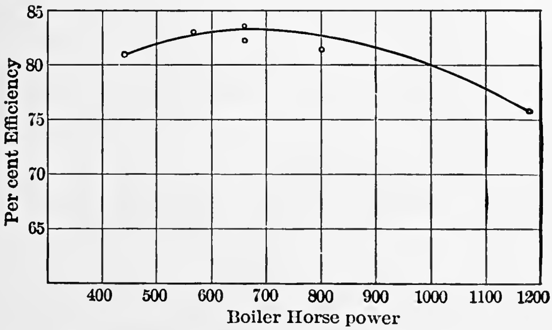


FIG. 67. Relation between Efficiency and Capacity — Oil Fuel.

It is a question if there are thirty plants throughout the country operating continuously day in and day out at 175 per cent rating. Widely varying loads are carried today in ordinary plant operation with over-all efficiencies higher than those formerly secured from constant loads and test conditions.

Oil fired boilers cannot be forced to any great extent economically because the heat is localized and intense and severe on the boiler. This is due primarily to the fact that oil involves surface combustion while coal involves a volume combustion. Increase in furnace volume will give increased over-all efficiency at overload but the efficiency will fall off at normal loads.

For influence of draft on capacity see Figs. 150 and 151.

80. Economical Loads. — The economical rating at which a boiler plant should be run depends primarily upon the load to be carried by that individual plant and the nature of such load. The most economical load from a commercial standpoint is not necessarily the most efficient load thermally, since first cost, cost of upkeep, labor, cost of fuel, capacity, and the like must all be considered along with the thermal efficiency. The controlling factor in the cost of the plant, that is the number of boiler units that must be installed, regardless of the nature of the load is the capacity to carry the maximum peak loads. While each individual set of plant operating conditions must be considered by itself the following statements * give some idea of general practice:

"For a constant 24-hour load, the operating capacity, to give the highest over-all plant economy, is between 125 and 150 per cent of the boiler's normal rating.

For the more or less constant 10- or 12-hour a day load, where the boilers are placed on bank at night, the point of maximum economy will be somewhat higher, probably between 150 and 175 per cent of the boiler's rated capacity.

The third class of load is the variable 24-hour load found in central station work.

Modern methods of handling loads of this description, to give the best operating results under different conditions of installation, are as follows:

1. The load on the plant at any time is carried by the minimum number of boilers that will supply the power necessary, operating these boilers at capacities of 150 to 200 per cent or more of their normal rating. Such boilers as are in service are operated continuously at these capacities, the variation in load being cared for by varying the number of boilers on the line, starting up boilers from a banked condition during peak load periods and banking them after such periods. This is, perhaps, at present the most general method of central station operation.

2. The variation in the load on the plant is handled by varying the capacities at which a given number of boilers are run. At low plant loads the boilers are operated somewhat below their normal

* "The Boiler of 1915," A. D. Pratt, Trans. International Eng. Congress, 1915.

rating, and during peak loads, at their maximum capacity. The ability of the modern boiler to operate over wide ranges of capacities without appreciable loss in efficiency has made such a method practicable.

3. The third method of handling the modern central station load is, perhaps, only practicable in large stations or groups of inter-connected stations. Under this method, the plant is divided into two parts. What may be considered the constant load of the system is carried by one portion of the plant, operating at its point of maximum economy. Due to the possibility of very high over-all efficiencies at high boiler capacities where the load is constant, where the grate and combustion chamber are designed for a point of maximum economy at such capacities, and where there are installed economizers and such apparatus as will tend to increase the efficiency, the capacity at which this portion of the plant is today operated will be considerably above the 150 per cent given as the point of highest economy for the steady 24-hour load for boilers without economizers.

The variable portion of the load on a plant so operated is carried by the second division of the plant under either of the methods of operation just given."

Standardization of Boiler Operating Conditions: Jour. A.S.M.E., Dec., 1916, p. 29.
Operation of Large Boilers: Trans. A.S.M.E., Vol. 35, 1913, p. 313.

81. Influence of Initial Temperature on Efficiency. — In general the higher the initial temperature of the furnace the greater will be the efficiency of the *heating surface*, since the heat transmitted increases with the difference of temperature between the water and the products of combustion. If the heating surface is properly distributed so that the final temperature of the escaping gas remains constant, the efficiency of the boiler and furnace will increase as the initial temperature increases, though not in direct proportion. This is on the assumption that the amount of heat generated per hour is the same throughout all ranges in temperatures. With a condition where the amount of heat generated remains constant and the initial temperature varies, the final temperature of the escaping gases remains practically constant, and in such cases high initial temperatures are productive of high *boiler* and *furnace efficiencies.* In practice these conditions are seldom realized and high furnace temperatures are not necessarily productive of high boiler and furnace efficiencies. Some tests show a decided gain in efficiency with the higher furnace temperatures ("Some Performances of Boilers and Chain-grate Stokers, with Suggestions for Improvements," A. Bement, Jour. West. Soc. Engrs., February, 1904), and others show little if any improvement ("A Review of the United States Geological Survey Fuel Tests under Steam Boilers," L. P. Breckenridge, Jour. Wes. Soc. Engrs., June 1907). The majority of high efficiency records, however, are associated with high furnace temperatures since the latter are realized only by minimum air excess and efficient combustion.

82. Thickness of Fire.—For a given boiler equipment, quality and size of fuel and intensity of draft, a certain depth of fuel will give maximum efficiency. Too thin a fire results in an excess of air and too

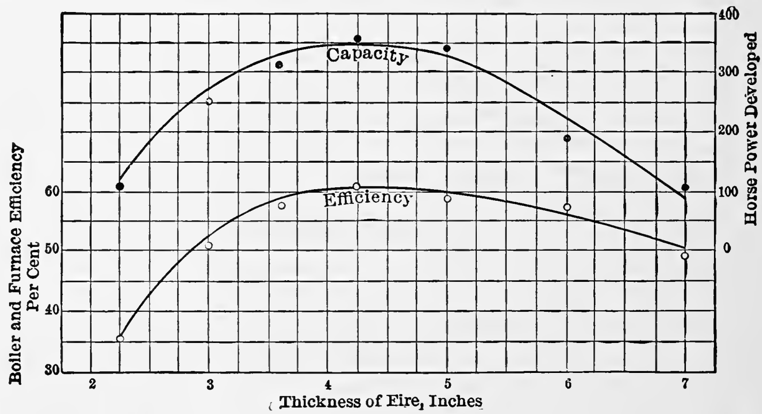


FIG. 68. Effect of Thickness of Fire on the Capacity and Efficiency of a 350-Horsepower Stirling Boiler, Equipped with Chain Grate.

thick a fire in a deficiency, the economy being lowered in either case. On account of the number of conditions upon which the proper thickness depends, it can only be determined for a particular case by actual test, the available data being insufficient for drawing conclusions.

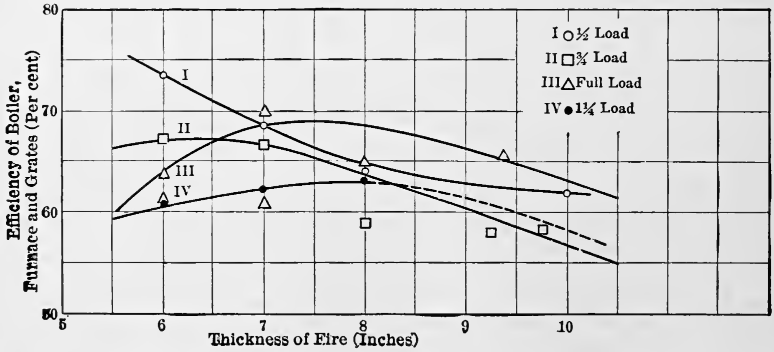


FIG. 69. Relation between Thickness of Fire and Efficiency of Boiler Furnace and Grate; 512 Horsepower B. & W. Boiler and Chain Grate.

The curves in Fig. 68 are plotted from a series of tests made on a 350-horsepower Stirling boiler equipped with chain grate at the power plant of the Armour Institute of Technology. The damper was left wide open throughout the test and the speed of the grate kept constant. Ratio of grate to heating surface, 1 to 42. Carterville washed coal No. 4 was used in all tests. The curves in Fig. 69 refer to the performance of a 512-horsepower Babcock & Wilcox boiler equipped

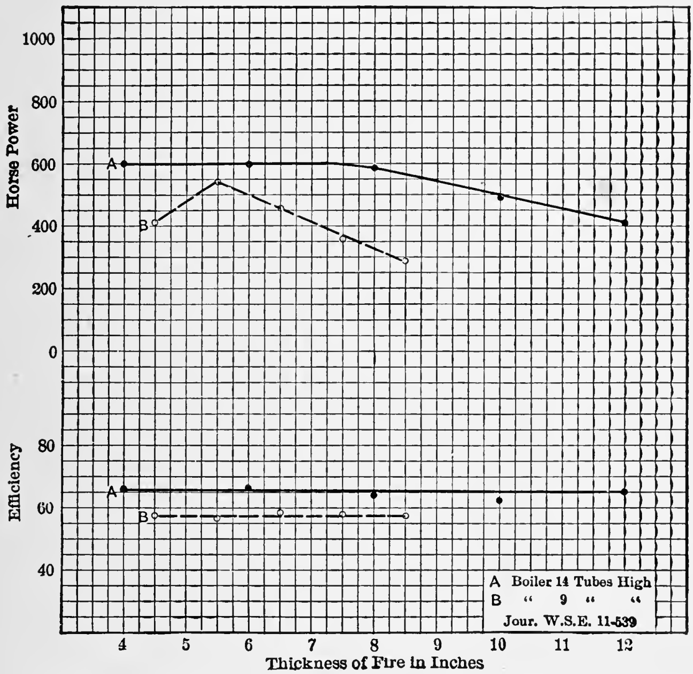


Fig. 70. Effect of Thickness of Fire on the Capacity and Efficiency of a 500-Horse-power Babcock & Wilcox Boiler.

with chain grate and located at the power plant of the University of Illinois, Urbana, Illinois. The curves in Fig. 70 are plotted from a series of tests on a 500-horsepower Babcock & Wilcox boiler equipped with chain grate at the Fisk Street station of the Commonwealth Edison Company, Chicago, Ill. In these tests the conditions of operation are not exactly comparable, but they serve to show the variation of economy with thickness of fire in each case. In general, with natural draft, fine sizes of coal necessitate thin fires, since they pack so closely as to greatly restrict the draft. Thin fires require closer attention to prevent holes being burned in spots, and respond less readily to sudden demands for steam, but have the advantage of letting the air required pass through the grate, whereas thick fires often require air to be supplied above the grate to insure complete combustion. Thick fires require less attention and hence are preferred by firemen. Where sufficient draft is available thick fires are more efficient than thin ones, as the air excess is more readily controlled.

TABLE 37.

TEMPERATURE DROP OF GASES THROUGHOUT BOILER.

(650 Hp. B. & W. Boiler, Waterside Station of N. Y. Edison Co.)

Per Cent of Rating.	Boiler and Grate Efficiency.	Temperatures, Degrees Fahr.						
		Furnace Temperature.	Middle First Pass.	Top First Pass.	Top Second Pass.	Middle Second Pass.	Middle Third Pass.	Flue.
117.3	78.5	2336	866	655	619	511	471	455
126.7	79.6	....	893	689	646	526	485	473
128.6	79.8	2420	888	681	633	521	481	468
131.0	77.1	2455	889	682	642	519	479	468
131.0	75.3	2430	913	694	655	512	486	473
137.4	77.6	....	956	723	660	546	512	492
142.2	76.5	....	939	700	634	523	493	475
185.3	72.7	2530	1051	751	700	578	541	519

83. Cost of Boilers and Settings. — The total cost of a boiler depends primarily on the cost of material and the cost of construction. The cost of material is almost a direct function of the weight but the cost of construction is relatively larger for small boilers than for large ones, so that the total cost is not a direct function of the rated horsepower. Furthermore, the difference in rating for various types of boilers (based on the extent of heating surface) has a direct influence on the cost per rated horsepower. For instance, Scotch-marine boilers are ordinarily rated at 8 square feet of heating surface per horsepower, water tube boilers at 10 square feet and small fire tube boilers at 12 square feet. Again, the rated horsepower is independent of the working pressure but the latter influences the weight of material so that costs expressed in terms of rated capacity are widely discordant and do not permit of accurate formulations. The cost of a boiler ranges from 7.5 cents per pound for the smaller sizes to 3.5 cents for very large units but even this range is subject to the market price of the raw material. A rough rule is to allow a cost of one dollar per square foot of water heating surface. The following equations (Boiler Room Economics, Patter & Simmering, Bul. 44, Kansas State Agricultural College) may be used as a guide in approximating the cost of different types of boilers with raw material based on 1915 prices.

Vertical fire-tube boilers; 100 pounds working pressure or less.

$$\text{Cost in dollars} = 51.5 + 3.62 \times \text{rated horsepower}. \tag{43}$$

Portable locomotive type fire-tube boilers; 100 pounds working pressure or less.

$$\text{Cost in dollars} = 121 + 5.68 \times \text{rated horsepower}. \tag{44}$$

Horizontal fire-tube boilers; 125 pounds working pressure for 100 horsepower or less.

$$\text{Cost in dollars} = 5.8 \times \text{rated horsepower} - 20 \text{ for } 100 \text{ to } 225 \text{ horsepower}. \tag{45}$$

$$\text{Cost in dollars} = 211 + 3.35 \times \text{rated horsepower}. \tag{46}$$

Vertical water tube boilers (100 horsepower and over); 125 pounds working pressure.

Upper limit:

$$\text{Cost in dollars} = 1032 + 7.68 \times \text{rated horsepower}. \tag{47}$$

Lower limit:

$$\text{Cost in dollars} = 797 + 6.17 \times \text{rated horsepower}. \tag{48}$$

Average:

$$\text{Cost in dollars} = 912 + 6.98 \times \text{rated horsepower}. \tag{49}$$

* Horizontal water tube boilers; 125 pounds working pressure.

$$\text{Cost in dollars} = 149 + 8.24 \times \text{rated horsepower}. \tag{50}$$

The cost of plain setting may be roughly estimated as follows:

Horizontal water tube boilers:

$$\text{Cost in dollars} = 400 + 0.8 \times \text{rated horsepower}. \tag{51}$$

Return tubular boilers:

$$\text{Cost in dollars} = 300 + 0.7 \times \text{rated horsepower}. \tag{52}$$

For other data pertaining to cost of boiler equipment and cost of operating see Chapter XVIII.

84. Selection of Type. — Boilers constructed by builders of good repute are usually designed for safety, durability, and capacity, and rigid specifications and inspection of material and workmanship on the part of the purchaser are ordinarily not necessary, as the makers' reputations are sufficient guarantee of their worth. Marked departure from standard designs must necessarily be specified, but in most cases instructions are limited to the working pressure, extent of heating and grate surface, the character of the furnace, and arrangement of

* Add 10 per cent for working pressures of 200 lbs. per sq. in. Add 30 per cent for working pressures of 300 lbs. per sq. in.

setting. Numerous tests on various types of boilers show practically the same efficiency provided the furnaces and boilers are properly designed, so that the relative merits may be considered with reference to (1) durability; (2) accessibility for repairs; (3) facility for cleaning and inspection; (4) space requirements; (5) adaptability to the type of furnace and stoker desired; (6) overload capacity; and (7) cost of boiler and setting. For high pressures 150 pounds per square inch or more, the water-tube or some form of internally fired boiler in which the shell plates are not exposed to the high temperature of the furnace are considered safer than the horizontal tubular boiler because the shell plates and the seams of the latter must be of considerable thickness in large units, and being exposed to the hottest part of the fire are likely to give trouble, especially if the water contains scale or sediment-forming elements. In the modern central station steam pressures of 200 to 250 lb. per square inch are standard practice. In a few recent installations pressures for 350 pounds have been specified and it is not unlikely but that pressures of 400, 500 and even 600 pounds may be in immediate prospect. (See paragraph 179 for a discussion of high pressures.) Return tubular and stationary locomotive boilers are seldom made in sizes over 250 horsepower and hence are not to be considered for large units. For sizes under 150 horsepower where overhead room is limited the return tubular boiler is most commonly installed, unless high pressure is essential, in which case the internally fired Scotch-marine boiler or special types of water tube boilers such as the Worthington are used. The water-tube boiler is usually employed in large central stations for high-pressure units of 300 to 2500 horsepower. The particular type of water-tube boiler is to some extent a matter of personal taste on the part of the engineer, but due consideration should be given to the special requirements as listed above. For small powers and for intermittent operation, small vertical or horizontal fire-box boilers have the advantage of low first cost. The small air leakage and radiation losses give internally fired boilers an advantage over the brick-set externally-fired fire-tube or water-tube types, but this is partly offset by the greater extent of regenerative surface in the setting of the latter. In several recent installations the brick settings are completely encased in steel and a layer of high grade insulating material is placed between the brickwork and the casing.* This reduces the leakage and radiation losses to a minimum and the setting remains effective over a long period of time. Internally fired boilers are more expensive than the exter-

* See "Insulation of Boiler Settings," Joseph Harrington, Power, Mar. 27, 1917, p. 410.

nally fired, though the extra cost of setting and foundation in the latter may bring the total cost of the entire equipment to practically the same figure. The design and installation of the boilers and furnaces should be left at the outset to a capable engineer.

Makers usually request the following information from intending purchasers:

1. Steam pressure desired.
2. The quantity of steam demanded.
3. The kind of fuel to be burned.
4. The type of furnace or stoker.
5. The nature and intensity of draft.
6. Nature of setting.
7. Quality of feed water.

85. Grates. — Grates may be divided into three general classes, namely, stationary, rocking, and traveling grates. The latter are treated in Chapter IV. Stationary grates are generally made of cast-iron sections in a variety of shapes as illustrated in Fig. 71. The bars are ordinarily from 3 inches to 4 inches deep at the center (this makes them strong enough to carry the load caused by the weight of the fuel

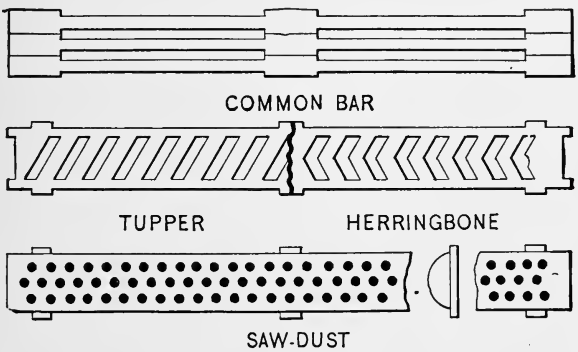


Fig. 71. Types of Grate Bars.

without sagging even when the top is red hot), $\frac{3}{4}$ inch wide at the top, and taper to $\frac{3}{8}$ inch at the bottom to enable the ashes to drop clear. The width of the air space is determined by the size of the fuel to be used, the average proportions being given in Table 38.

The "Tupper" and "Herringbone" grate bars are stiffer and less likely to warp than the common form, but are not so readily sliced and therefore not so convenient with coal that clinkers badly. Sawdust or pinhole grates are used in burning sawdust, tanbark, and very small sizes of coal. Grates are often set horizontally and the bars are held in place simply by their own weight, but long grates are best placed sloping toward the rear to facilitate firing. The front of the grate when designed for bituminous coal is often made solid, this portion being called the "dead plate." It serves to hold the green fuel until the hydrocarbons have been distilled off, when the charge is pushed back on the open grate at the time of next firing. The length of a single bar or casting should not exceed three feet. The length of grate may be made of two or three bars and should not exceed 6 feet with bituminous coal, as this is the greatest length of fire that can be readily worked by a stoker. With buckwheat anthracite furnaces 12 feet in depth are not unusual, as anthracite fires require no slicing.

TABLE 38.

AIR SPACES AND THICKNESS OF GRATE BARS.

Size and Kind of Coal.	Width of Air Spaces.	Thickness of Grate Bars.
	(Inch)	(Inch)
Screenings	$\frac{1}{4}$	$\frac{3}{8}$
Anthracite —		
Average	$\frac{1}{2}$	$\frac{3}{4}$
Buckwheat	$\frac{3}{8}$	$\frac{2}{8}$
Pea or nut	$\frac{1}{2}$	$\frac{1}{2}$
Stove	$\frac{5}{8}$	$\frac{1}{2}$
Egg	$\frac{3}{4}$	$\frac{1}{2}$
Broken	$\frac{7}{8}$	$\frac{1}{2}$
Lump	1	$\frac{3}{4}$
Bituminous, average	$\frac{5}{8}$	$\frac{3}{4}$
Wood —		
Slabs	$\frac{3}{4}$	$\frac{3}{4}$
Sawdust	$\frac{1}{4}$ to $\frac{3}{8}$	$\frac{3}{8}$
Shavings	$\frac{1}{2}$ to $\frac{5}{8}$	$\frac{1}{2}$

The disadvantage of using stationary grates is that the fire is not easily cleaned. Unless the air spaces are kept free of clinkers and ashes, combustion is hindered and the fire rendered sluggish. Frequent cleaning, however, is wasteful of fuel and reduces the furnace efficiency by letting in a large excess of air every time the fire door is opened.

86. Shaking Grates. — Shaking grates have the advantage of permitting stoking without opening the fire door and require less manual labor than stationary grates. There is a great variety of sectional shaking grates on the market and some of them are made self-dumping. One of the best-known types is illustrated in Fig. 72. Each row or section of grate bars is divided into a front and a rear series by twin stub levers and connecting rods. An operating handle is adapted to manipulate either one or both of the levers in such a manner that the front and rear series may operate separately or together. The shaking movement causes no increase in the size of the openings and hence prevents the waste of fine fuel. Ordinarily the width of the grate is made equal to two or more rows of grate bars so that the live fire may be

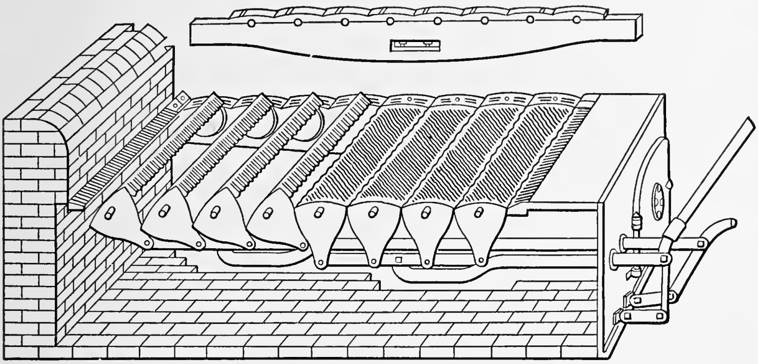


FIG. 72. A Typical Shaking Grate.

shoved sidewise from one row to the other when cleaning. A depth of fire of from 6 to 10 inches is carried according to the nature of the fuel and the available draft.

Grate Bars: Engr. U. S., Nov. 1, 1906, p. 728, Jan 1, 1907, p. 68.

87. Blow-offs. — Boilers must be provided with blow-off pipes for draining off the water and for discharging sediment and scale-forming material. The "bottom blow" is ordinarily an extra heavy pipe of suitable diameter connected to the lowest part of the boiler and fitted with a valve or cock, or both. The generally approved method of arranging the blow-off pipe is shown in Fig. 89. This method of protecting the pipe from the direct action of the heated gases by means of a V-shaped brick pier permits easy examination of the blow-off through the cleaning door in the rear wall of the setting. Where boilers are arranged in batteries the battery may have a common outlet for the blow-off pipes as illustrated in Fig. 517. Usually the blow-off pipes may discharge into the open air, but this is not permissible in large cities nor is it lawful to blow directly into the sewer. In this case the water and sediment may be

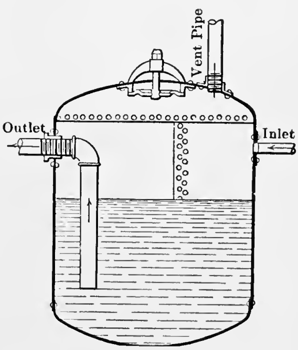


FIG. 73. Blow-off Tank and Connections.

discharged into a blow-off tank and permitted to cool before delivery to the sewer, as illustrated in Fig. 73.

Blowoff Valves and Systems: Power, July 1, 1916, p. 565.

"Surface blows" are often installed to remove scum, grease, and floating or suspended particles of dirt. The bell-mouthed shape shown in Fig. 74 permits the skimmer to accommodate itself to varying water level, and it is sometimes provided with a float and with a flexible joint, Fig. 75.

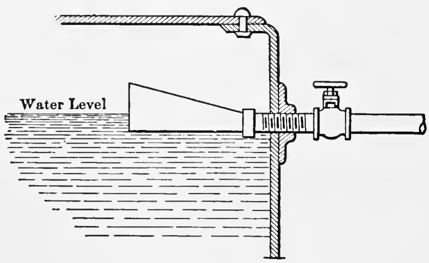


FIG. 74. Surface Blow-off.

88. Damper Regulators. — For maximum furnace efficiency the draft must be regulated to burn just enough fuel to supply the steam required. Where forced draft is employed this is done by regulating the speed of the blower. With natural draft it is the usual practice to regulate the draft by means of dampers placed in the uptake, and in order that the regulation may be effective it should be automatic. Automatic dampers are economical and useful and are particularly desirable in small plants where the demand for steam fluctuates rapidly. There are several successful types on the market, some operated by water pressure, and others by direct boiler pressure and in some of the later type by thermostats. Fig. 76 illustrates a typical hydraulic mechanism. Full boiler pressure acting at all times on a diaphragm *A* raises or lowers a weight *W* attached to arm *D* according as the steam pressure increases or decreases. Arm *D* actuates a small valve *V* which controls the supply of water to chamber *B*. A diaphragm in chamber *B* raises and lowers the damper as the water pressure varies, a drop of 0.5 pound being sufficient to open the damper to its maximum. The steam diaphragm has a movement of only 0.01 inch and the water diaphragm 0.5 inch. When properly adjusted and given proper attention automatic dampers work in a very satisfactory manner.

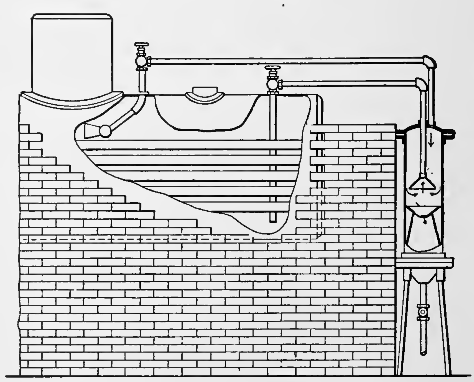


FIG. 75. Buckeye Skimmer.

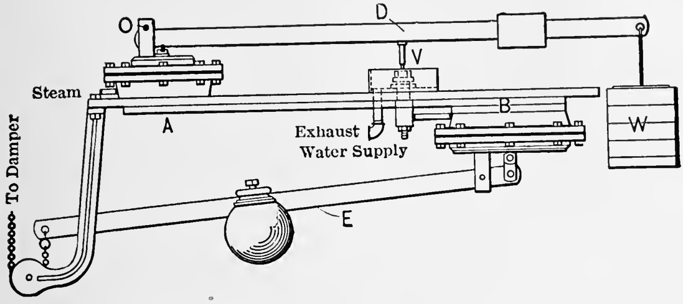


FIG. 76. Kitts Hydraulic Damper Regulator.

Fig. 77 shows a section through the Tilden damper regulator, illustrating the principles of the steam-actuated type. The device is connected directly to the boiler by pipe *A*. The pressure on piston *B* is balanced by spring *C* under normal conditions of operation. Any variation from the normal will cause the rod *R* to move up or down, so that the dampers are opened or closed in proportion to the change in pressure. The chamber *N* is separate from chamber *M*, so that steam or water cannot come in contact with the spring. Piston *D* acts as a guide only. In a recent design of this device the regulator is hydraulically actuated and simultaneously operates the damper and the stoker engine thereby automatically proportioning the air and fuel supply to the load requirements.

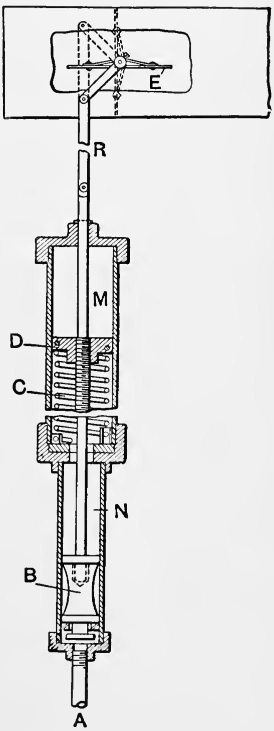


FIG. 77. Tilden Steam-actuated Damper Regulator.

89. Water Gauges. — The water level in a boiler is usually indicated either by a gauge glass, by try cocks, or both, connected directly to the boiler as in Fig. 1, or to a *water column* or *combination* as in Fig. 77. Each gauge-glass connection should be fitted with a stop valve which may be closed in case the tube breaks. In large boilers these valves, usually of the quick-closing type, are conveniently operated from the boiler-room level by means of a chain attached to the valve stem. Self-closing automatic valves have been employed, one type being illustrated in Fig. 79. If the glass breaks the outrush of steam forces the ball against a conical seat and shuts off the supply. When a new glass is inserted the ball is forced back by slowly screwing in the valve stem. Hinged valves mechanically operated from without are considered more reliable than ball valves, as scale is less likely to render them inoperative. Self-closing valves are not allowed in modern practice.

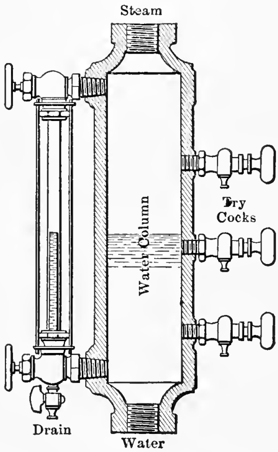


Fig. 78. Simple Water Column.

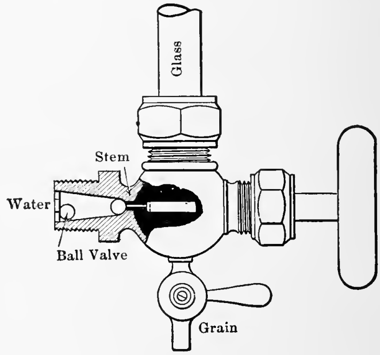


Fig. 79. Water Gauge with Self-closing Valve.

Try cocks or *gauge cocks* are set at points above and below the desired water level, preferably connected directly to the boiler shell, but sometimes to a water column as in Fig. 78. The water level is ascertained by opening the cocks in succession.

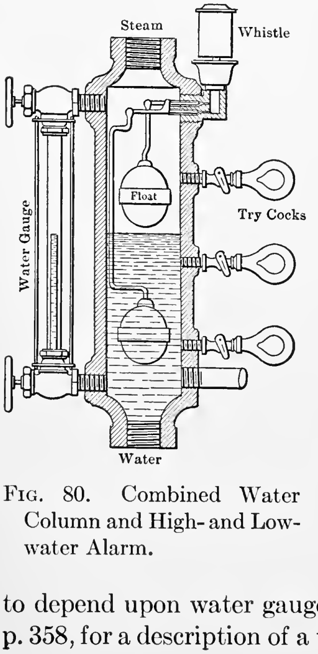


Fig. 80. Combined Water Column and High- and Low-water Alarm.

The objection to the latter arrangement is that accident to or a stoppage of the piping renders both gauge glass and try cocks useless. Water columns should be blown out once a day, and the gauge cocks opened to see that the height of the water indicated tallies with that shown by the glass. Some engineers prefer two separate columns to each boiler and no cocks, others rely solely upon cocks.

The water column shown in Fig. 80 has an alarm whistle, controlled by two floats, which gives a high- and low-water alarm. Numerous devices of this class are on the market but they are usually regarded as unreliable and most engineers are content to depend upon water gauge and try cocks. See Power, Mar. 13, 1917, p. 358, for a description of a water-level indicator for high-pressure boilers.

Water Gauges and Columns: Mach., Sept., 1905, p. 31; Power, Aug., 1905, p. 483; Am. Elecn., July, 1904, p. 359; Engr. U. S., Jan. 1, 1907, p. 58.

90. Fusible or Safety Plugs. — Fusible or safety plugs as illustrated in Fig. 81 are brass plugs provided with a fusible metal core. They are inserted in the shell or tubes at the lowest permissible water line. When covered by water the heat is conducted away sufficiently fast to keep the temperature below the fusing point, but when uncovered the low conductivity of the steam prevents the rapid withdrawal of heat,

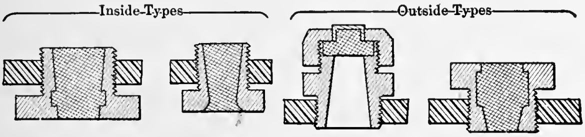


Fig. 81. Types of Fusible Plugs.

whereupon the alloy melts and the blast of escaping steam gives warning. The melting point of fusible metals being sometimes uncertain, plugs occasionally blow out without apparent cause and at other times fail to act when shell is overheated. Fusible plugs are required by law in many cities.

91. Soot Blowers, Tube Cleaners, Etc. — Aside from the assurance against burning out of tubes due to the accumulation of scale, the maintenance of clean heating surfaces is one of the most important problems in connection with recent developments with higher boiler ratings and in the operation of large boiler units. Efficiency and capacity depend to a greater extent upon cleanliness (both internal and external) of the heating surfaces than is ordinarily realized. Soot is an excellent heat insulating material and consequently any appreciable deposit on the heating surfaces will reduce the rate of heat absorption and result in high flue gas temperatures. The gain effected in economy and capacity by the removal of soot varies with depth, extent and nature of the deposit and the rate of driving. No modern plant is operated without periodically removing this deposit.

Surfaces exposed to the action of the products of combustion are customarily freed from soot and clinkers by steam lances, soot blowers incorporated within the setting, brushes, scrapers and similar appliances. Light, flocculent soot is conveniently removed at regular intervals by means of a hand-operated steam lance with which all surfaces are reached and swept clean. Under certain conditions better results are obtained by permanently installed soot-blowers. (See Figs. 82 and 83.) These consist of a series of pipes and nozzles, the latter stationary or revolving, located so that all parts of the heating surface subjected to

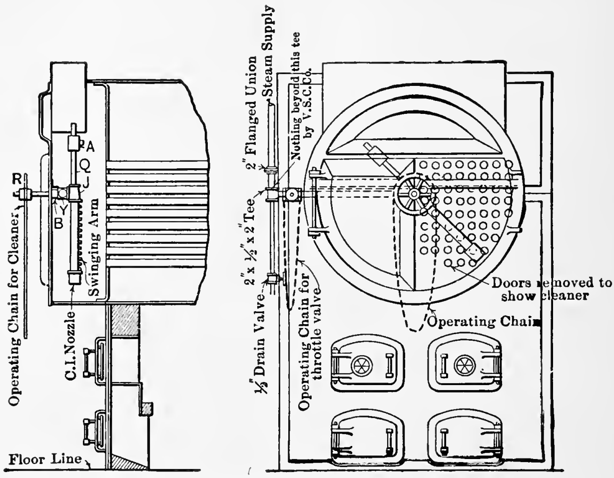


FIG. 82. Vulcan Soot Blower Installed in Front End of a Return Tubular Boiler.

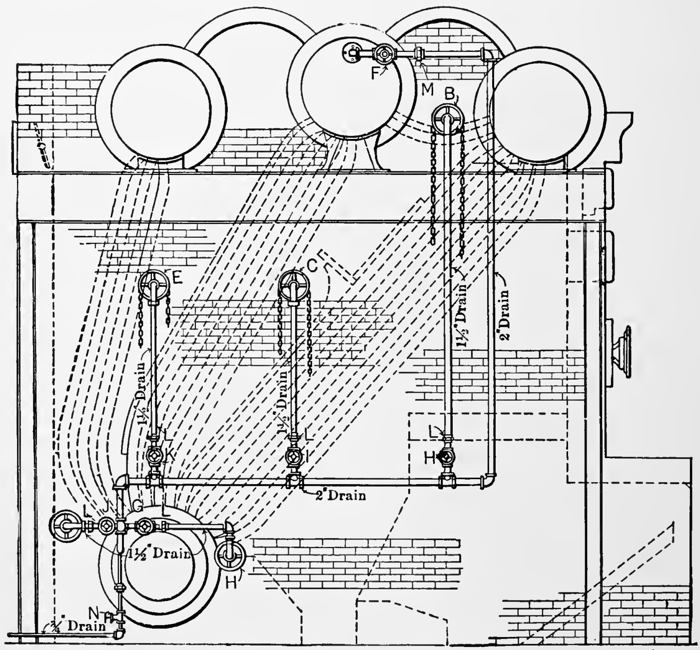


FIG. 83. Diamond Soot Blower Applied to a Stirling Boiler.

soot deposit may be swept with a jet of steam. The controlling valves are located outside the setting. (See also paragraph 35.) With certain grades of coal under heavy furnace capacity the particles of ash and slag carried along with the products of combustion are in a plastic state and adhere to the two or three lower tubes. The accumulation may eventually result in a complete choking up of the gas passages. Blowing by hand lances and machine blowing devices will not remove the accumulation and dislodging the deposit with pokers after the furnace has been partially cooled appears to be the only practical solution of the problem.

Boiler Cleaning Costs: Power, May 23, 1916, p. 741.

Keeping Boiler Heating Surface Clean: Textile Wld, Sept. 9, 1916, p. 3899; Elec. Wld., May 20, 1916, p. 1182; Power, Aug. 31, 1915, p. 314, July 13, 1915. p. 48.

The question of preventing the formation of scale by purification of the feed water and the loss in heat transmission due to scale deposit is treated at length in Chapter XII. In the average plant furnished with commercially good feed water it is customary to allow scale to deposit for a limited period of time and then remove it mechanically by tube cleaners and scrapers. The principles of construction of these

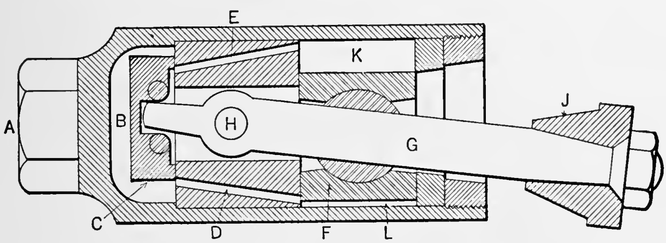


FIG. 84. Mechanical Tube Cleaner — Hammer Type.

devices vary widely according to the types of boilers in which they are used, and depend upon the nature of the duty which they must perform. Mechanical tube cleaners may be conveniently divided into two classes:

1. Those which loosen the scale by a series of rapid hammer blows, Fig. 84.

2. Those which cut out the scale by a revolving tool, Fig. 85.

The hammer device is applicable to either the water or fire-tube type of boiler, but the revolving cutter is applicable to the water-tube only. Steam, compressed air, or water under pressure may be used as the motive of power, though the latter is the most convenient and satisfactory.

Referring to Fig. 84, the hammer head *J* is given a rapid motion, which may reach 1500 vibrations per minute, and subjects the tube to repeated shocks, thereby cracking the brittle scale and jarring it loose from the water surface of the tube. The cleaner head is attached to a flexible pipe of sufficient length to enable it to be pushed from one end to the other. Even if carefully manipulated the hammer is apt to injure the tube by swaging it to a larger diameter, producing crystallization in the metal and causing leaks where the tubes are expanded into thin sheets, hence its use is not to be recommended.

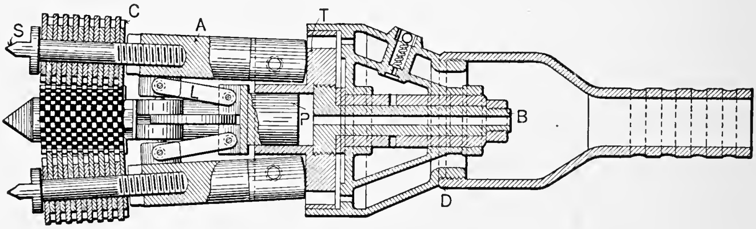


FIG. 85. Mechanical Tube Cleaner — Turbine Type.

Hydraulic turbine cutters are made in many designs, one of which is shown in Fig. 85. The cylindrical casing D contains a hydraulic turbine consisting of a fixed guide plate which directs the water at the proper angles upon the vanes of the turbine wheel T. The cutters C revolve at high speed and chip the scale into small pieces. The stream of water flowing from the turbine envelops the cutters, keeps their edges cool, and washes away the scale as fast as it is detached. Different styles of cutter wheels are furnished with each cleaner so as to adapt the device to all kinds of scale formations. In well managed plants scale is not permitted to deposit to a thickness greater than $\frac{1}{32}$ to $\frac{1}{16}$ of an inch.

American Railway Mechanics Association; Report of the Committee on Boiler Washing: Ry. Review, June 19, 1915, p. 851.

PROBLEMS.

1. Given: Initial pressure 100 lb. gauge; barometer 29.5 in.; quality 98; feed water, 82 deg. fahr. Required boiler horsepower necessary to furnish a 50 horsepower engine with steam, engine uses 45 lb. per i.hp-hr.

2. A 30,000 kw. steam turbine and auxiliaries require 12 lb. steam per kw-hr. at rated load; initial pressure 250 lb. gauge; barometer 30 in.; superheat 250 deg. fahr.; feed water 180 deg. fahr. Required the boiler horsepower necessary to furnish the turbine and auxiliaries with steam. If the boilers are operated at 250 per cent rating when supplying the turbine and auxiliaries, required the ratio of kilowatts of turbine rating to boiler rating.

3. A boiler evaporates 90,000 lb. of water per hr. from a feed temperature of 210 deg. fahr. to steam at 275 lb. absolute pressure and 200 deg. superheat. If the boiler

is being forced to 200 per cent rating when evaporating this amount of water, required the extent of heating surface assuming that the normal rating corresponds to an evaporation of 3.5 lb. water from and at 212 deg. fahr. per sq. ft. of heating surface. Allowing 10 sq. ft. of heating surface per rated boiler horsepower, required the boiler rating.

4. Determine the factor of evaporation for Problems 1 and 2.

5. The following data were taken from a boiler test:

Heating surface, 8000 sq. ft.; grate surface, 160 sq. ft.;

Coal analysis (as fired): Moisture 8 per cent; ash 12 per cent; B.t.u. per lb. 12,100.

Weight per hr.: Water fed to boiler, 32,000 lb.; coal 4,000 lb.; dry refuse removed from ash pit, 720 lb.

Temperatures: Flue gas, 480 deg. fahr.; feed water, 160 deg. fahr.; boiler room 80 deg. fahr.

Pressures: Steam pressure, 125 lb. gauge; barometer, 29.0 in., superheat, 100 deg. fahr.

Required:

a. Factor of evaporation.

b. Boiler horsepower developed.

c. Per cent of builders' rating developed (builders' rating = 10 sq. ft. of heating surface per boiler horsepower).

d. Evaporation per lb. of coal as fired:

(1) Actual.
(2) Equivalent.

e. Evaporation per lb. of dry coal.

(1) Actual.
(2) Equivalent.

f. Evaporation per lb. of combustible:

(1) Actual.
(2) Equivalent.

g. Equivalent evaporation per lb. of combustible burned.

h. Evaporation per sq. ft. of heating surface.

(1) Actual.
(2) Equivalent.

i. Heat value of the combustible as fired.

j. Heat value of the combustible as burned.

k. Efficiency of the boiler, furnace and grate.

l. Efficiency on the combustible basis.

6. The following additional data were taken during the test outlined in Problem 5. Flue gas analysis, per cent by volume:

CO_2 14.19; CO 1.42; O 3.54; N 80.85

Ultimate analysis of coal as fired, per cent by weight:

Carbon 66, hydrogen 5, nitrogen 1, oxygen 8, moisture 8, ash 12.

a. Calculate:

(1) Complete heat balance.
(2) Inherent losses.
(3) Per cent of available heat utilized.

7. If the fuel (Problem 5) cost $3.25 per ton of 2000 lb., determine the cost of evaporating 1000 lb. of water from and at 212 deg. fahr.

8. A test of an oil-fired furnace gave an actual evaporation of 12.77 lb. of water per lb. of oil with boiler and furnace efficiency of 82.8 per cent; boiler pressure 200 lb. absolute, superheat 87 deg. fahr., feed water temperature 93 deg. fahr. Required the calorific value of the oil.

CHAPTER IV

SMOKE PREVENTION, FURNACES, STOKERS

92. General. — Anthracite coal and other fuels low in volatile combustible matter can be burned in almost any type of furnace without the production of visible smoke; in fact, it is a difficult matter to produce smoke with this class of fuel. On the other hand, bituminous and other "soft" coals high in volatile matter can be burned smokelessly only in properly proportioned and carefully operated furnaces.

The problem of smoke abatement is a comparatively simple one for large plants equipped with mechanical stokers and provided with ample draft, even for widely fluctuating loads, but for hand-fired plants it depends largely upon skillful manipulation by interested and efficient firemen. The order of intelligence demanded for this work is out of all proportion to the wages paid. In many small plants — and these are usually the most obstinate smoke offenders — the fireman handles as much as a ton of coal per hour by hand, besides caring for the feed pumps and water levels, keeping the boilers clean, and removing the ash. The boiler room is frequently poorly lighted and poorly ventilated. It is, therefore, not surprising that the fireman seldom worries about the smoke problem. A better wage scale and more consideration for the fireman might do a great deal toward abating the smoke nuisance.

Since the loss in heat due to *visible* smoke is usually less than one-half per cent, and seldom greater than one per cent of the heat value of the coal (see Table 39) it is a common statement among owners of power plants that it is cheaper to smoke than to operate without smoke. This is undoubtedly true in many cases where smokeless combustion can be secured only by admitting a considerable excess of air with a consequent loss in economy frequently greater than that due to incomplete combustion and smoke, but if proper attention is given to the various factors involved practice shows that smokeless combustion can be effected with high boiler and furnace efficiency.

The amount of solids discharged from a stack has no direct relation to visibility. A stack may appear smokeless to the eye and yet discharge considerable dust into the atmosphere. Furthermore, the sulphur compounds resulting from the combustion of certain coals are eventually converted into sulphuric acid, and though invisible, are even

TABLE 39.

QUANTITY AND HEAT VALUE OF SOLIDS IN VISIBLE SMOKE.
(BITUMINOUS COAL.)

From the Report of the Chicago Association of Commerce Committee of Investigation on Smoke Abatement. (1912.)

Test Number.	Smoke Density, Per Cent.	Solids in Visible Smoke.	
		Per Cent by Weight of Fuel Fired.	Per Cent of the Heat Value of the Fuel Fired.
Fires with High Smoke Density.			
3	21.97	0.83	0.28
17	20.00	0.75	0.36
10	20.00	1.10	0.95
30	15.80	0.65	0.49
29	14.50	0.82	0.49
Average......	18.45	0.63	0.51
Fires with Low Smoke Density.			
56	0	0.51	0.21
57	0	0.30	0.08
80	0	4.07	0.74
81	0	1.81	0.48
85	0	0.47	0.11
Average......	0	0.47*	0.32

* Average of 10 plant tests not including Test No. 80.

TABLE 40.

CHEMICAL COMPOSITION OF THE SOLID CONSTITUENTS OF SMOKE.
(CHICAGO ASSOCIATION OF COMMERCE.)

Kind of Fuel.	Per Cent of Total Solids.				
	Hydro-carbons (Tar).	Combustible Solids (Carbon).	Mineral Matter (Ash).	Sulphur.	Total.
High-pressure Plants.					
Pocahontas..........	3.08	41.45	52.39	3.08	100
Bituminous..........	4.19	32.80	59.93	3.08	100
Low-pressure Plants.					
Anthracite...........	0.73	31.88	67.39	0 0	100
Pocahontas..........	11.43	54.90	33.47	0.20	100
Bituminous..........	31.43	44.06	22.12	2.39	100

more objectionable than visible smoke from a standpoint of atmospheric pollution. Sulphuric acid has a disintegrating action on building material and produces deleterious effects upon furnishings, clothing and merchandise. Evidently smokeless combustion in itself does not prevent the escape of objectionable matter from the chimney, but it is a step in the right direction. All solid matter may be removed from the products of combustion by electrical precipitation * and all solids and gaseous sulphur compounds may be completely eliminated by "washing,"† but these processes have not yet been developed to a basis where the results are compatible with the expense, at least for average power plant service.

In locomotive practice from 3 to 25 per cent of the weight of dry coal is discharged in the form of cinders, the lower figure for a pressure drop of approximately 1.5 inches of water and the higher figure for a pressure drop of 12 inches. See *Laboratory Tests of a Consolidation Locomotive,* Bulletin No. 2, Vol. XIII, University of Illinois, Sept. 13, 1915, p. 25.

In order that combustion may be smokeless and efficient, the volatile gases and separated free carbon must be brought into intimate contact with the proper quantity of air and maintained at a temperature above the ignition point until oxidation is complete before they are brought into contact with the heat-absorbing surfaces of the boiler. Mere excess of air will not effect smokeless combustion, even if the gases and air are thoroughly mixed, if the temperature is prematurely reduced below that necessary for combustion by contact with the heat-absorbing surfaces of the boiler.

Smoke may be produced, therefore, by

1. An insufficient amount of air for the perfect combustion of the volatile gases. This is primarily a function of the draft.
2. An imperfect mixture of air and combustible.
3. A temperature too low to permit complete oxidation of the volatile combustible.

Table 41 gives an idea of the distribution of smoke production by various industries in Chicago, in 1912. Rigid enforcement of the smoke ordinance has reduced the nuisance to a considerable extent, so that the distribution at this date differs somewhat from that shown in the table.

The term "smoke consumer" is a misnomer, since the combustible solids in visible smoke when once discharged from the furnace cannot be economically burned. The so-called smoke consumers are in reality smoke preventers.

* Problems in Smoke, Fume and Dust Abatement: F. G. Coterell, Publication 2307, 1914, from the Smithsonian Report for 1913.

† Washing Smoke from Locomotives: M. D. Franey, Power, Oct. 19, 1915, p. 561.

Smoke-preventing devices may be divided into two classes: (1) those which may be conveniently attached to plants already in operation without material modification of the furnace, such as steam jets and other means of mixing the air and combustible gases, admission of air through the bridge or side wall, and mechanical draft; and (2) those which are an integral part of the boiler and setting, such as mechanical stokers, Dutch ovens, down-draft furnaces, and fire-tile combustion chambers incorporated with the regular setting.

93. Hand-fired Furnaces. — Hand-fired furnaces, as a class, are most obstinate smoke producers. Although they can be operated efficiently without the production of objectionable smoke the result depends more upon the fireman than upon the design of the furnace. The chief difficulty with hand-fired furnaces lies in the intermittent nature of the firing. When a fresh charge of coal is fed into the furnace an enormous volume of volatile matter is evolved. For complete combustion a corresponding amount of air must be supplied and intimately mixed with the volatile gases before contact is made with the comparitively cool heating surface. In the average hand-fired furnace the combustion

TABLE 41.

SMOKE DISTRIBUTION IN CHICAGO PLANTS. (1912.)

(From the Report of the Chicago Association of Commerce Committee of Investigation on Smoke Abatement.)

Service.	Relative Weight of Coal and Coke* Consumed During the Year 1912, Tons.	Relative Contribution, Per Cent.				
		To Visible Smoke.	To Solid Constituents of the Smoke.	To Gaseous Products of Combustion.	To Gaseous Carbon Constituents in the Smoke.	To Sulphur Constituents in the Smoke.
Steam locomotive	13.27	22.06	7.47	10.31	10.11	18.22
Steam vessels	0.44	0.74	0.33	0.60	0.55	0.45
High-pressure steam and other stationary plants	43.13	44.49	19.34	44.96	40.68	53.70
Low-pressure steam and other stationary plants	21.91	3.93	8.60	23.00	23.06	19.73
Gas and coke plants	1.20	0.15	0.00	0.00	0.00	0.00
Furnace for metallurgical, manufacturing and other processes	20.05	28.63	64.26	21.13	25.60	7.90
Total	100.00	100.00	100.00	100.00	100.00	100.00

* Total fuel consumed 21,208.886 tons.

chamber is so small and the heating surface is so close to the grate that the partly burned gases strike the heating surface before oxidation is complete and combustion is hindered or even completely arrested.

The majority of so-called smoke preventers are merely devices for mechanically mixing the air and volatile gases. These include fire-brick piers, baffles, arches, and steam jets. There is no question as to the value of these mixing devices if properly installed, but the personal element is too variable a factor for consistent results and the ultimate solution lies in mechanical stoking. The most economical and smokeless hand-fired plants are those that approach the continuous feed of the mechanical stoker. The following rules formulated by Osborne Monnett, former Chief Smoke Inspector of the City of Chicago, apply to hand-fired furnaces burning Illinois coal which is high in volatile matter. (Power, Aug. 11, 1914, p. 207.)

Building Fires.

Cover the grate with coal to a depth of about 4 in., and build a wood fire on top, or throw live coals from another boiler into the furnace at the bridge wall. The green coal underneath will then ignite and burn. As the coal in the front gradually ignites, the volatile matter must pass over the fire already on the grates. A live, good steaming fire can be built up in this way without producing dense smoke. Keep the doors cracked and the panels wide open, giving the fire sufficient air and allowing the fresh coal to be ignited from the top. This will keep the smoke down to a minimum. When the coal has fully ignited and has been well coked, fire six or eight scoops of coal on one side, beginning at the bridge-wall and filling up the low spots all the way to the front. Do not spread the coal but allow it to lie in lumps just as it leaves the shovel. Before the fire gets too low on the other side, and after the greater part of the volatile matter has been burned off from the last charge, fire an equal amount of coal on the other side as before, always keeping the panels wide open after firing.

When enough steam has been generated, use the jets, turning them on before firing and leaving them on until the bulk of the volatile matter has been distilled off. Always use the alternate method. For smokeless operation, hand-fired boilers burning coal exclusively should not exceed a capacity of 150 hp.

Cleaning Fires.

Build up the fire on one side and let the other side burn down. Just before cleaning, "wing" over the live coal from the burned-out side and pull out the clinker and ash, cleaning the grates thoroughly. Cover the clean grates with green coal and push over live coal from the other side. When the cleaned side has become thoroughly ignited and the volatile matter has passed off, throw in coal to fill the spots not covered and pull the clinker and ash out of the other side. Cover the grates with green coal as before, winging over live coals from the opposite side. Keep the panels wide open and allow the fresh coal to ignite thoroughly. Never allow the fire to burn low before cleaning if carrying a heavy load, as there is a possibility of losing the steam pressure.

After cleaning, follow up closely with the alternate method of firing until the fuel bed is thick enough to hold the pressure.

Carry as thick a fire as the draft will permit and do not spread the coal over the grates.

Banking Fires.

Throw 15 or 20 scoops of coal on each side. Open the panel doors slightly, close the ashpit doors and partially close the damper. To break up the bank, level the fire, with the panel doors open, and start firing by the usual method, making sure the damper is wide open.

Keep the fire brick by the alternate method, using the panel doors and steam jets, and regulate the steam pressure with the ashpit doors. This insures a temperature in the furnace high enough to maintain the brickwork at the igniting point of the coal and promotes combustion of the volatile matter. At the same time, it keeps down the distillation of the volatile matter to a low rate, and by having the damper open and the panels cracked, the circulation of the gases is not retarded. Do not try to regulate the steam pressure by the damper or smoke will be produced.

The method just described is contrary to the general rules for firing, but its philosophy is explained in the following: If a shovelful of coal is thrown on a bright fire by the spreading method, every particle of that coal is immediately subjected to the intense heat of the fire and the volatile matter is rapidly driven off. If this is followed by more coal, the result is a volume of volatile matter which is beyond the capacity of the furnace to handle without dense smoke.

If, on the other hand, the fuel is fired in a lump from the shovel without spreading, there is a considerable quantity of the coal which does not immediately become subjected to the high temperature. The coal on the outside of the pile gives up its volatile and the coal within is not affected until the volatile matter has been distilled from the outside lumps. Furthermore, the volatile matter from the inner portions of the pile must pass outward through the incandescent outer layer of fuel much in the same way as in the underfeed stoker. In this way the production of smoke can be retarded, and the more coal thrown on the fire at once, the less smoke. Two shovelfuls of coal fired by the spreading method on a clean, bright fire will make more smoke than ten shovelfuls of coal fired by the lump method. In practice, it will be necessary to determine just how much coal should be fired at once, but six or eight shovelfuls on a side with the draft operated according to the method ordinarily used will be found to be about right.

When a battery of boilers is to be fired, the fires should be fed by the alternate method, as before, passing from one boiler to another until they are all charged and then repeating on the other side of the furnaces. It should be determined by experiment, however, just how many furnaces can be fired consecutively without producing dense smoke, and after this has once been made known, the fireman should adhere strictly to the rules laid down in this regard.

94. Dutch Ovens. — One of the earliest attempts at hand-fired smokeless-furnace construction consisted in placing a full-extension Dutch

Oven (Fig. 86) in front of the boiler. This provided a large combustion chamber but the setting was extravagant in floor space and the intense radiation from the incandescent furnace lining effected a too rapid distillation of the volatile matter from the green fuel. As a result

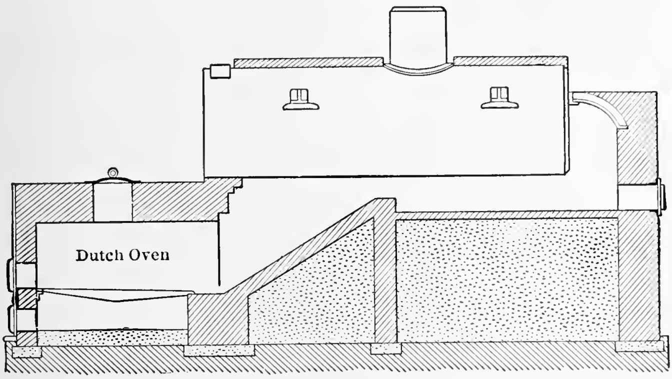


Fig. 86. Plain Dutch Oven — Full Extension.

the velocity of the gases was too high to permit complete oxidation within the furnace and visible smoke could not be prevented from forming except at light loads. Steam jets placed at the sides of the setting and blowing across the fire assisted in mixing the gaseous products

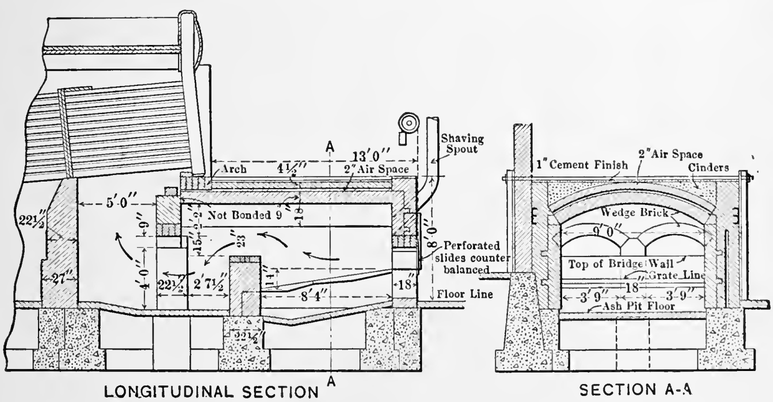


Fig. 87. Dutch Oven for Burning Wood Refuse.

but did not satisfactorily solve the problem. By placing the oven partly (semi-extension) or completely (flush front) underneath the boiler proper the extra space requirements were reduced or completely eliminated but a considerable portion of the heating surface was insulated from the fire at the expense of capacity. The next step was to remove part of the oven roof and expose the boiler surface to the direct action of the fire. This increased the economy and capacity of the setting but still failed to effect the desired result. The introduction of a deflector arch at the end of the oven made smokeless combustion possible, but the setting was rather high and expensive to install and maintain. The final development consisted in arranging the deflector arches and a double-arch bridge wall as illustrated in Fig. 89. Dutch ovens are generally used in burning wood refuse and similar fuels. See Fig. 87.

95. Twin-fire Furnace. — This arrangement, illustrated in Fig. 88 in connection with a hand-fired return tubular boiler, is a double furnace formed by longitudinal arches extending between bridge wall and fire door.

The furnaces are fed and manipulated alternately, the object being to have one furnace in a highly incandescent state, while green fuel is fed into the other. Air is admitted both below and above the grate, and the volatile gases are supplied with the necessary oxygen for combustion before they come into contact with the comparatively cool boiler surface.

The gases from both furnaces first pass into a chamber formed by a single arch sprung across the entire inner setting from the side wall, a short retarding arch being placed between this intermediate chamber and the rear of the setting. A special tile of high-grade refractory clay is used, the thickness varying from 4 to 6 inches, depending upon the size of furnace and the length of span. The furnace can readily be substituted for the ordinary types in common use under any standard tubular or water-tube boiler and may be installed either under the boiler, as indicated in the illustration, or in an extension Dutch oven. This is an excellent furnace, and when properly manipulated gives smokeless and efficient combustion.

96. Chicago Settings for Hand-fired Return Tubular Boilers. — Figs. 89 and 90 show details of settings for return tubular boilers as recommended by the Chicago Department of Smoke Inspection, and which may be considered the latest development in hand-fired smokeless settings for Illinois bituminous coals. The setting illustrated in Fig. 89, and known as the Double-arch Bridge-wall Furnace, is intended for low pressure work where steam jets are not effective and where the rate of combustion is 15 pounds of coal per square foot of grate surface per hour or less, and that shown in Fig. 90 where the rate of combustion is greater or where the plant has a regular power load. The dimensions refer to a specific set of conditions and are not general. Both settings require careful manipulation for smokeless combustion as is the case with hand-fired furnaces in general. It has been the ex-

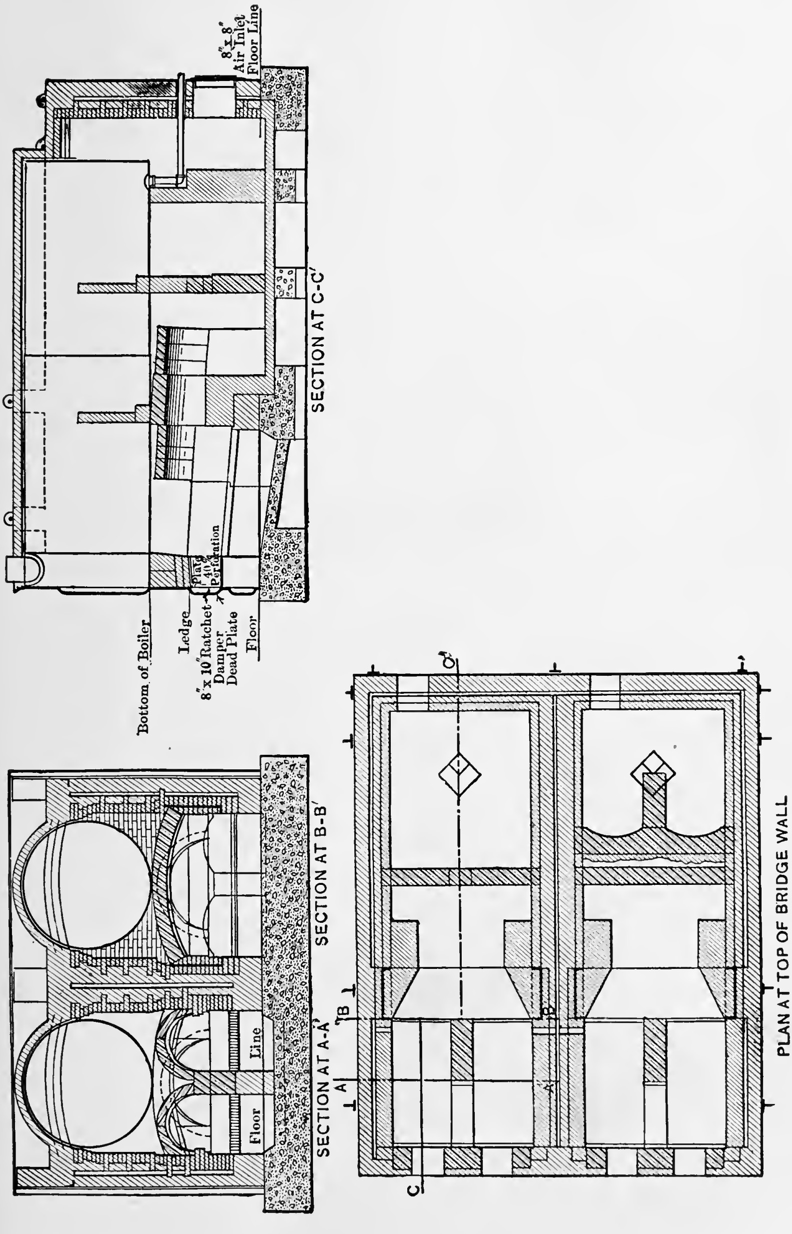


FIG. 88. "Twin Fire Arch" Applied to Two Return Tubular Boilers. (Chicago Specifications.)

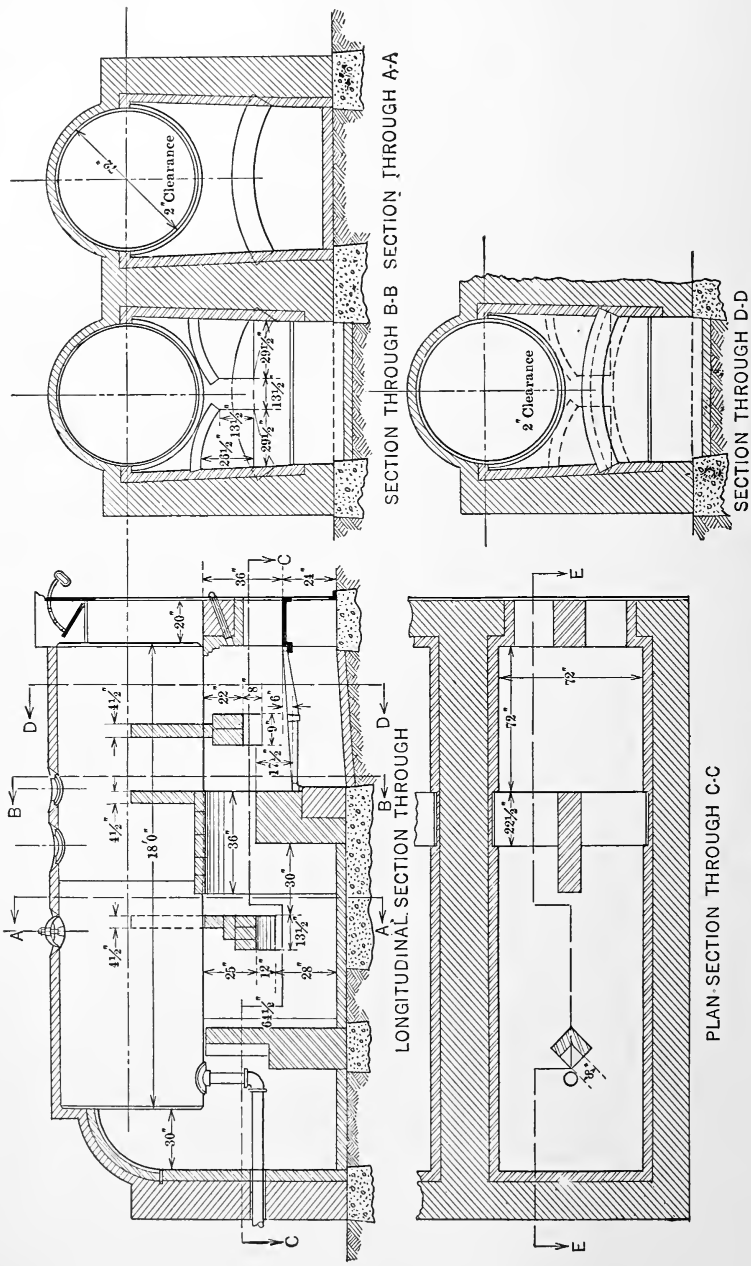


FIG. 89. Double-arch Bridge-wall Furnace; Chicago Setting.

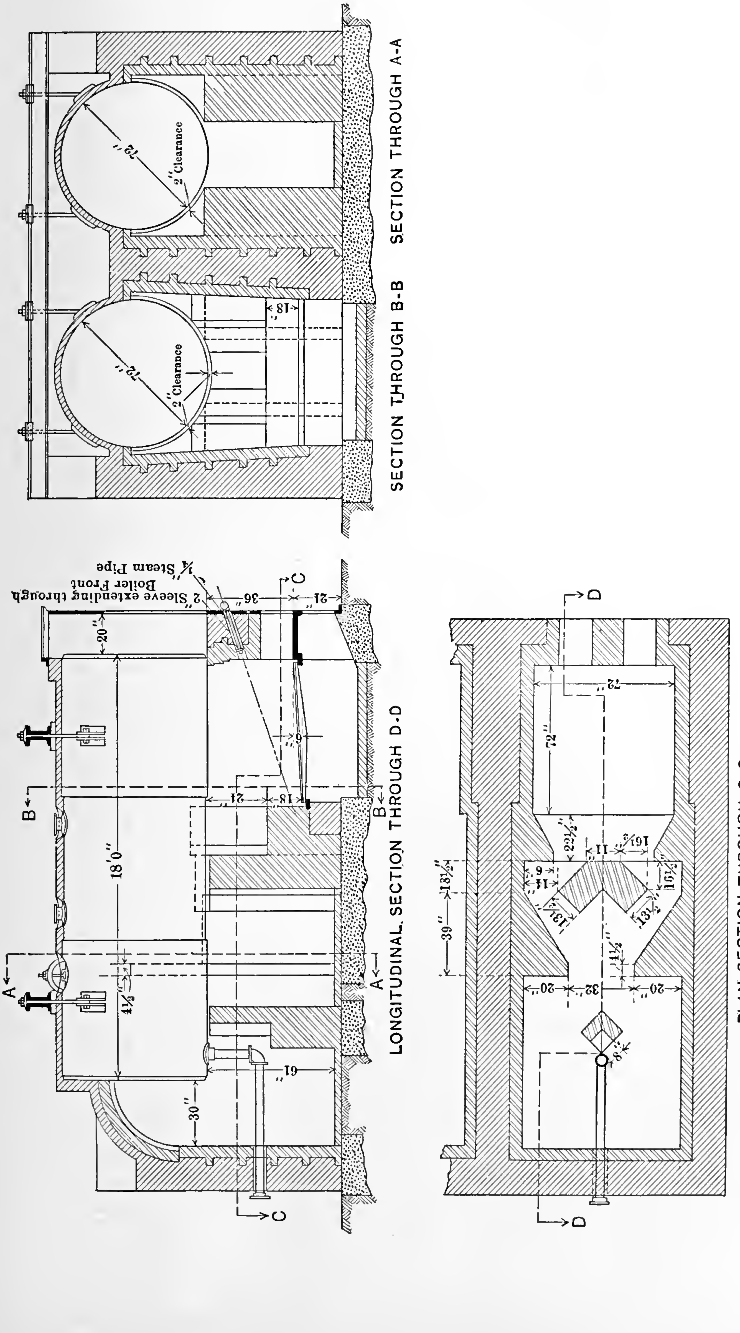


FIG. 90. Wing-wall Furnace, Chicago Setting No. 8.

perience of the Department that most violations of the smoke ordinance are due primarily to insufficient draft, the required rate of combustion being too high for the available air supply. The following note refers to the No. 8 furnace: First grade fire brick to be used throughout with the exception of the combustion chamber floor and the side walls of the combustion chamber back from a point one foot behind the rear face of the wing-walls. Wing-walls to be bonded into the side walls. No air space to be left in the setting walls. Fire doors must provide for special air admission of an area equal to four square inches for each square foot of grate surface.

97. Burke's Smokeless Furnace. — Fig. 91 shows sections through a Burke smokeless furnace as installed in a number of tall office buildings in Chicago. It amounts virtually to a Dutch oven equipped with shaking grates, and embodies an extension self-feeding coking oven of cast-iron section lined with fire brick and protected from overheating by air circulation through the sections. Natural draft is used, the

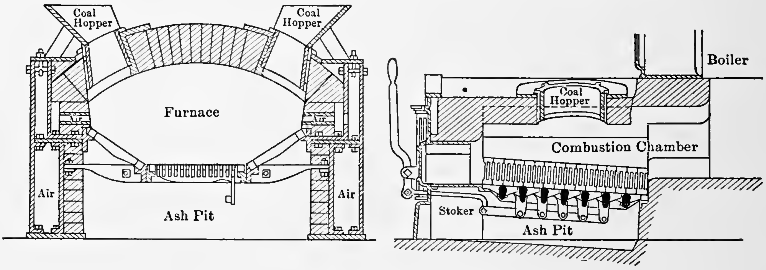


Fig. 91. Burke's Smokeless Furnace.

fire doors being closed; but air is admitted above as well as below the fire. As this stoker is manipulated by hand, more or less attention is required of the operator in keeping the fire clean. Furnaces of this type at the power plant of the Majestic Theater building, Chicago, Ill., are giving good results.

98. Down-draft Furnaces. — Fig. 92 shows the application of a Hawley down-draft furnace to a Heine water-tube boiler. In this furnace there are two separate grates, one above the other, the upper one being formed of parallel water tubes connected with the water space of the boiler through the steel headers or drums, *A* and *D*, in such a manner as to insure a positive circulation. Fuel is supplied to the upper grate, the lower one, formed of common bars, being fed by the half-consumed fuel falling from the upper grate. Air for combustion enters the upper fire door, which is kept open, and passes first through the bed of green fuel on the upper grate and then over the in-

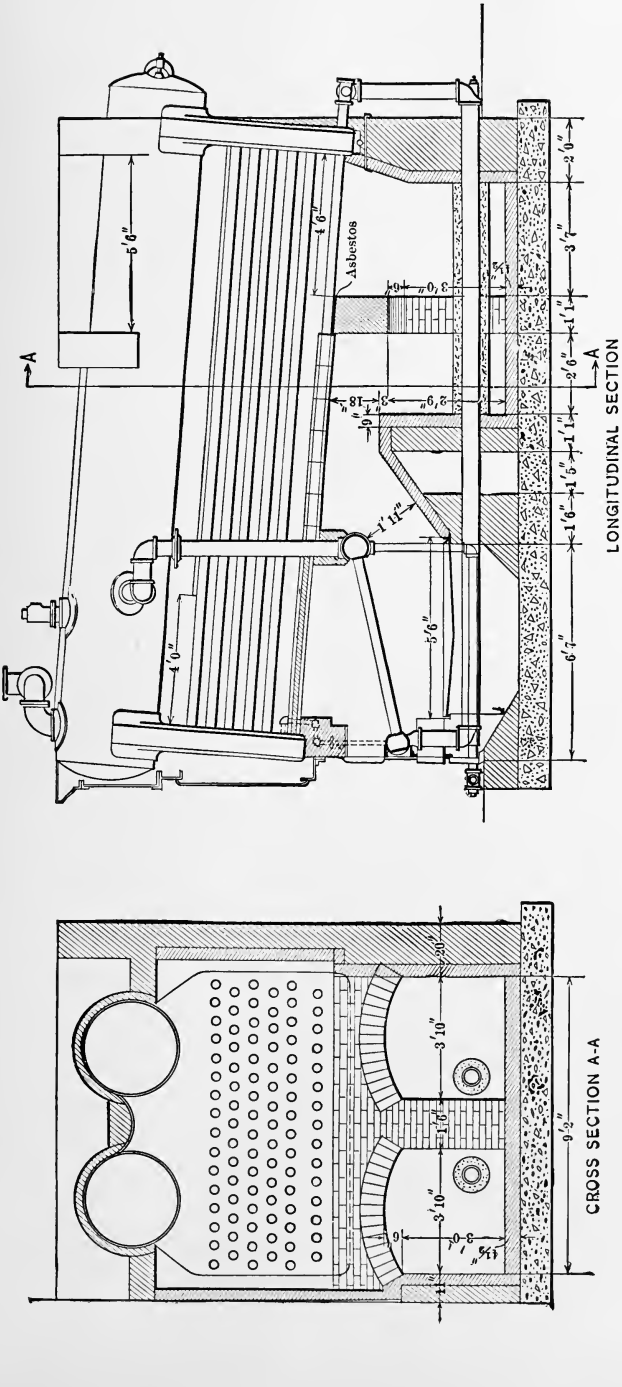


FIG. 92. Hawley Down-draft Furnace Applied to a Heine Boiler. (Chicago Setting.)

candescent fuel on the lower grate. A strong draft is required, due to the relatively small upper grate area and the correspondingly high rate of combustion. Lump coal gives better results than the smaller sizes, as the latter are apt to fall through the upper grate before being even partially consumed and when such is the case efficient results cannot be obtained. If carefully manipulated this furnace with fire-tiled tubes as illustrated in Fig. 92 gives high boiler efficiency and smokeless combustion, but its overload capacity is limited. Without the fire tiling smokeless combustion is possible only at light loads.

The down-draft furnace is remarkably successful on low rates of combustion, 10 lb. per sq. ft. per hour or less and is used extensively for heating loads.

99. Steam Jets. — The main purpose of a steam jet in connection with "smokeless furnaces" is to mix the air and gases and insure intimate mixture of the products of combustion. This action is purely mechanical, the steam in itself not being a supporter of combustion. The claims sometimes made that steam increases the calorific value of fuel are erroneous. There are conditions with certain grades of coals and refuse under which a moderate amount of steam injected into the furnace promotes complete combustion and increases the efficiency of the boiler. Such results, however, are due to increase in *available* heat and not to increase in actual calorific value. A theory advanced in this connection is that "hydrogen and CO formed by the reaction between the steam and incandescent carbon unite with the oxygen of the air passing through the grate and generate intense heat. This heat dissociates a part of the steam into hydrogen and oxygen. The hydrogen immediately recombines with oxygen of the air, while the oxygen in its nascent state effects complete combustion of the hydrocarbons which under ordinary conditions escape in the form of smoke. Although it takes as much heat to dissociate steam into its elements as is given off when the hydrogen burns back again to water vapor the gain in available heat effected by the steam lies in the combustion of the hydrocarbons which would otherwise be discharged up the stack. The heat necessary to superheat the steam to stack temperature must be charged against the coal pile but the loss may be more than offset by this increase in available heat. It takes the same amount of oxygen to burn the hydrogen as is liberated by dissociation so there is no *extra* oxygen available for combustion, but the oxygen thus liberated is in a nascent state and combines much more readily with the hydrocarbons than does atmospheric oxygen."

There is no question as to the value of properly installed steam jets in maintaining smokeless combustion in internally fired furnaces, hand-fired return-tubular boilers and improperly designed furnaces, but taking all things into consideration better results may be had with properly designed furnaces equipped with mechanical stokers. A plain setting and steam-jet equipment either manually operated or automatic will usually average from 8 to 12 per cent smoke density (see paragraph 105). The "Chicago No. 8 Furnace" properly manipulated can be operated with about 2 per cent smoke density. A smokeless stack with hand firing is not a true indication of efficient operation, since the air dilution may be excessive and the heat demands of the steam jets may be very great. Since air requirements are greatest at the moment of firing fresh coal, and the demand diminishes as distillation of the volatile matter progresses, steam jets need close regulation for best economy. If permitted to run continuously, as is often the case, they may use considerably more of the energy of the coal than they save by effecting smokeless combustion. Practically all of the so-called "smoke consumers" for hand-fired furnaces depend upon the steam jet or admission of air only above the fire for their operation. In most of these the jets are automatic and operate independently of the fireman. The most efficient jets are those based on the injector or siphon principle in which the jet induces a flow of air along with the steam. The steam nozzles are usually placed in the front wall and are charged downward toward the bridge wall, as illustrated in Fig. 90. Occasionally they are placed in the side wall or even in the bridge wall, but the front wall construction appears to be the best. The majority of the patented smokeless furnaces involving the use of the steam jet do not conform with the requirements of the Chicago Department of Smoke Inspection, chiefly because of faulty furnace design.

Theory, Practice and Design of Hand-fired Furnaces and Modern Methods of Smoke Prevention: National Engr., Nov., 1913, p. 670 (Serial).

For valuable data pertaining to smokeless combustion including brick settings for all types of hand-fired and stoker-fired furnaces see serial article by O. Monnett, Chief Smoke Inspector, Chicago, Ill., Power, May 12, 1914 to Jan. 5, 1915.

100. Mechanical Stokers. — *Uniform evolution of the volatile gases* of the fuel is the essential requisite for smokeless combustion, and it is for this reason that mechanical stokers as a class are more effective in preventing smoke than any apparatus accompanied by intermittent firing. Stokers which feed irregularly have the effect of hand-fired furnaces, and it is necessary not only to employ some powerful auxiliary mixing device but also to furnish at times an extra supply of air to take care of the enormous volume of volatile gas evolved after a fresh charge of fuel is added.

Carefully adjusted automatic stokers owe their high efficiency to:

(1) *uniformity of feed;* (2) *proper proportion of air and combustible;* (3) *absence of excessive air dilution, as when the fire doors are opened in connection with hand firing; and* (4) *self-cleaning grates.*

Daily records are essential with any type of stoker or hand firing if efficient results are expected, as only by frequent observation is it possible to determine the proper adjustment of air supply, depth of fire, rate of feed, and the like. Control of air supply is almost as important as the upkeep and effective operation. In the best firing practice the right amount of air, depth of fire, and rate of feed must be worked out by the engineer.

Stokers are often condemned by owners as inefficient and inferior to hand stoking because no particular attention has been paid to them beyond filling the hopper with coal. They should be operated in strict accordance with the principles of design.

In plants of 2000 horsepower or over, the installation of mechanical stokers and coal conveyors effects a considerable saving of labor and can usually be relied upon to solve the smoke problem if reasonable attention is given to their operation. In smaller plants interest on the investment and other considerations may make hand firing more economical, although many stoker-fired plants of capacities as small as 200 horsepower are giving satisfaction, particularly in places where a poor grade of fuel is used and smoke ordinances are rigidly enforced. A stoker of the self-cleaning, slow-running type requires much less attention than the hand-fired furnace. With hand firing one fireman can efficiently attend to the water, coal, and ashes of about 200 horsepower, or handle coal for, say, 500 horsepower, whereas with good automatic stokers equipped with overhead bunkers and down spouts he can readily take care of 4000 horsepower.

The best stokers are those which are least complicated and simplest in operation. A cheap stoker is a poor investment, since the cost of repairs and shutdowns will usually amount to more than the saving in price.

The following outline gives a classification of a few of the best-known American mechanical stokers:

Chain Grates.

Babcock & Wilcox	McKenzie	Leclede-Christy
Green	Illinois	Westinghouse

Overfeed.

Step Grates — Frontfeed.	Step Grates — Sidefeed.
Roney	Murphy
Wilkinson	Detroit
Acme	Model

Underfeed.

Jones	Taylor	Westinghouse
American	Riley	Type "E" Combustion Engineering Co.

Down Draft.	*Sprinkler*
Hawley	Swift
	Vulcan

Powdered Fuel.
See paragraphs 41–46.

101. Chain Grates. — The standard type of chain grate is one of the most popular forms of automatic stokers for burning small sizes of high ash and free-burning bituminous coals. (30 to 40 per cent volatile matter and 10 to 20 per cent ash.) It is also adapted to lignites and the very high ash coals of the West. With low ash coals at high rates of combustion the grate is apt to become overheated, and breakage with attendant high maintenance may result. The standard chain grate embodies a moving endless chain of grate bars mounted on a frame with provision for the continuous and uniform feeding of coal into the furnace, the fuel and the grate moving together. As usually installed the surface of the grate is horizontal though in some designs it is given a slight incline toward the bridge wall. The operations of feeding the coal, carrying it through the progressive stages of combustion, removing the ashes and clinkers, and maintaining a clean grate and free air supply are automatic. The driving mechanism consists of a gear train actuated by ratchet and pawls, the arms carrying the latter being given a reciprocating motion by an eccentric mounted on a line shaft. The latter may be driven by any type of engine or motor and the speed of the grate regulated by varying the stroke of the arm carrying the pawls. Fuel is fed into a hopper placed at the front end of the furnace and the depth of the fuel regulated by a guillotine damper. The front part of the furnace is provided with a flat or slightly inclined ignition arch the function of which is obvious. The entire grate and driving mechanism is mounted on a permanent truck and may readily be removed from beneath the boiler. The thickness of the fire and the speed of the grate should be so regulated that when the fuel has reached the end of the grate it shall have been completely consumed and incandescent ashes only will be discharged into the pit. With chain-grate stokers there may be considerable leakage of air between the grate and bridge wall, through the coal in the hoppers, under the coal-gate and through the fire bed at the rear where it is mostly ashes unless care is used in regulating the depth of fire and ash bed and provision is made for preventing this "short circuiting" of the air supply.

In the "Illinois" chain grate the live grate area may be varied by means of dampers placed immediately below the upper chain surface. This permits the use of a "short fire" at light loads without excessive air leakage. Chain grates as a class are seldom operated at loads exceeding 250 per cent of the rated boiler capacity.

Fig. 93 shows the general application of a B. & W. chain grate to a B. & W. boiler. The ignition arch is parallel to the grate and covers

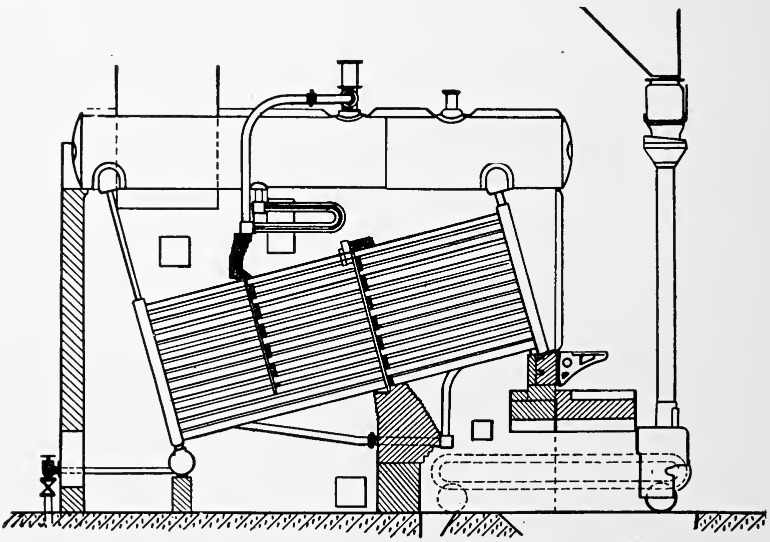


FIG. 93. Babcock and Wilcox Boiler, Chain Grate, Ordinary Setting.

a considerable portion of the grate surface. The bridge wall is fitted with a water back as indicated, to prevent the grate bars from being burned. With normal uniform loads this style of ignition arch and setting insures practically smokeless combustion, but careful manipulation is necessary with rapidly fluctuating loads to prevent the formation of objectionable smoke.

Fig. 94 shows an application of a Babcock & Wilcox chain grate to a horizontal water-tube boiler as installed at the Quarry Street Station of the Commonwealth Edison Company. It will be noted that the stoker is applied to the rear of the setting. This arrangement of stoker and Sewall baffling effects smokeless combustion, but the life of the furnace is short because of the low spring of the arch.

Fig. 95 shows another arrangement of a Babcock & Wilcox boiler and chain grate with vertical baffling as installed in units 5 and 6 of the

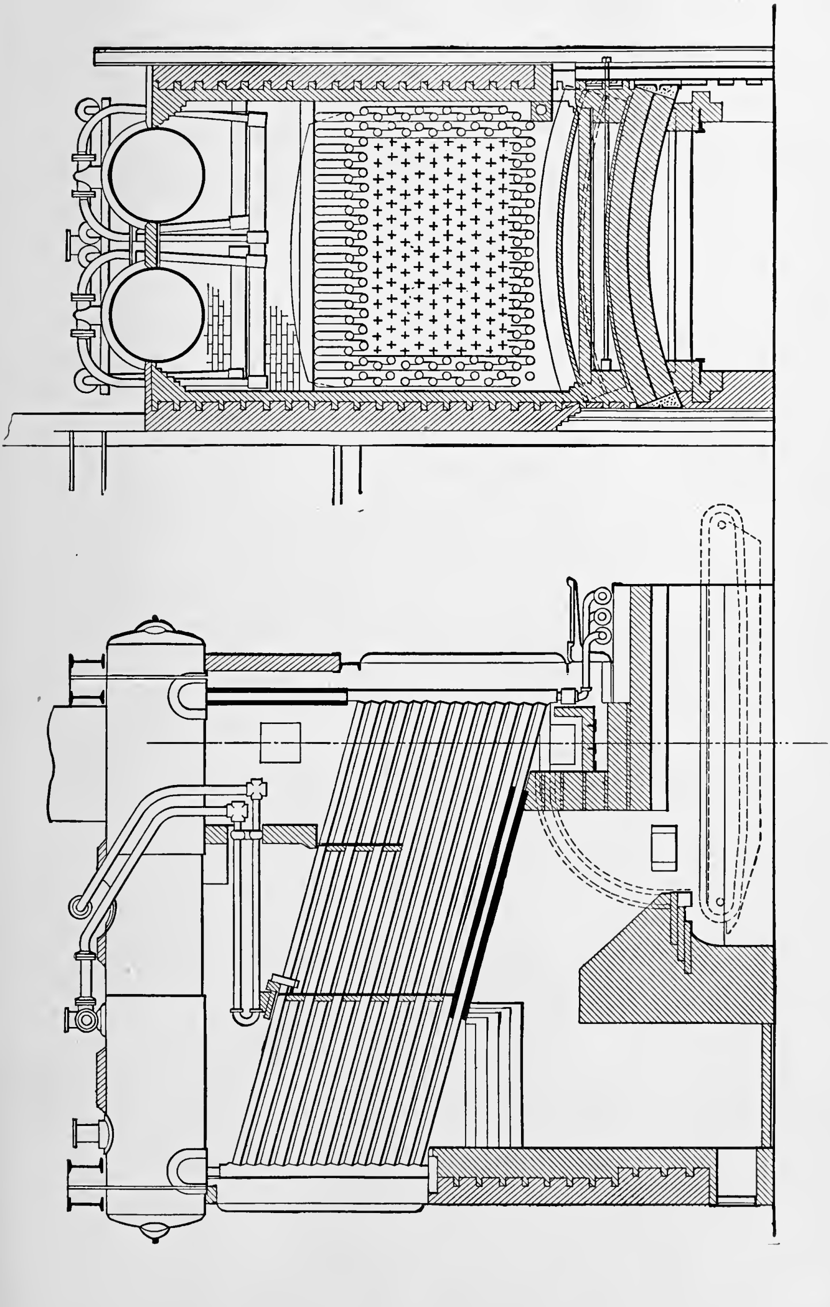


FIG. 94. Chain Grate Fired from Rear End of Setting as Installed at Quarry Street Station, Commonwealth Edison Co., Chicago.

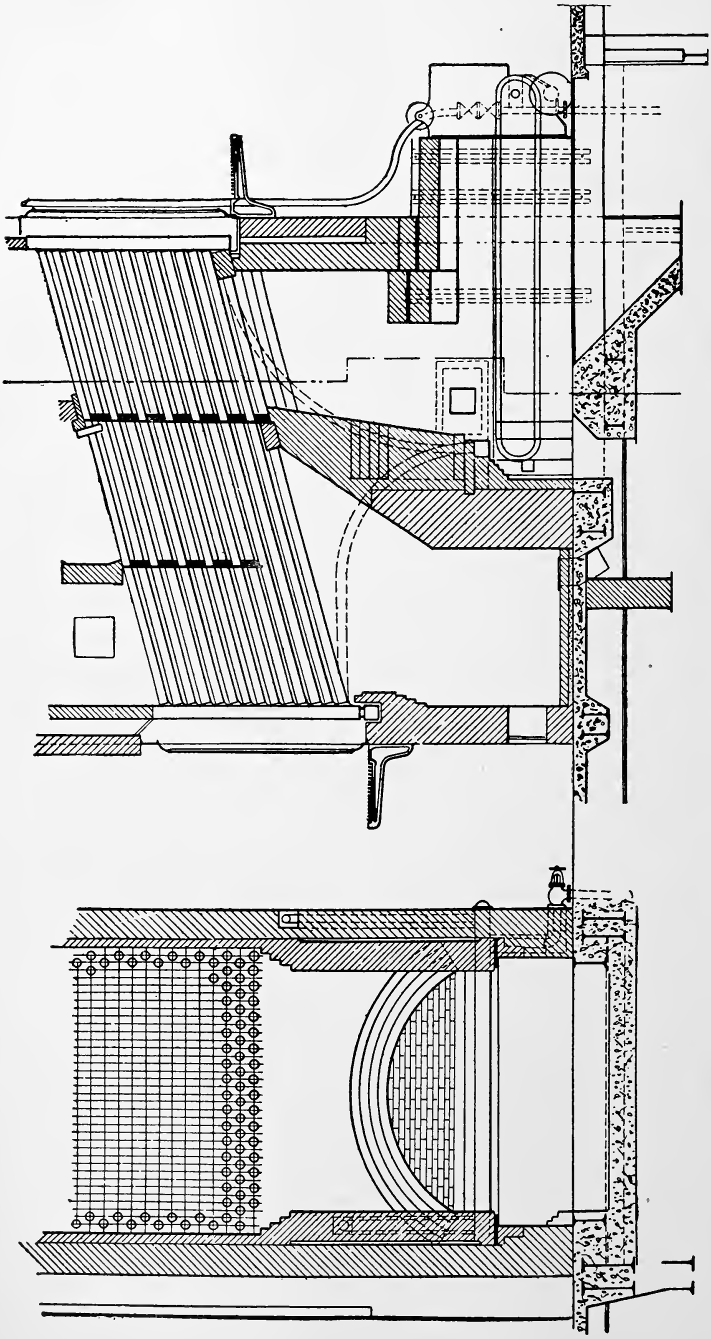


Fig. 95. Smokeless Setting, Boiler Units 5 and 6, Quarry Street Station, Commonwealth Edison Co., Chicago.

Quarry Street Station and units 1 and 2 of the Northwest Station. This is a smokeless setting up to 175 per cent of rating.

Fig. 96 gives the general details of a Green Type "K" chain grate as applied to a horizontally baffled water-tube boiler. This setting conforms with the requirements of the Chicago Smoke Department and is smokeless up to 200 per cent rating.

The standard type of chain grate is not adapted to coking coals on account of the swelling and fusing action of the fuel under the ignition arch. The chain grate may be modified to burn this class of fuel by introducing inclined coking plates immediately under the front of the ignition arch and agitating them mechanically during the period of distillation. This agitation prevents the coal from fusing together and by the time the fuel reaches the grate proper it no longer tends to cake. The Green Type "L" chain grate is an example of this modification.

Chain Grates and Smokeless Settings: Power, Oct. 13, 1914, p. 532; Oct. 20, 1914, p. 560; Nov. 3, 1914, p. 658.

102. Overfeed Step Grates. — In stokers of the overfeed step grate type coal is pushed in at the top of the slope and caked by the aid of a fire-brick arch and fed downward progressively by the movement of the grate bars aided by gravity. The upper portion of the bars is arranged to retain the uncaked part of the coal, changing to larger openings in the lower portion where the coal has fused and combustion is chiefly that of fixed carbon. The clinker collects at the bottom where it is crushed by rolls or dumped. Since the fixed carbon combustion occurs directly on the grate all overfeed stokers are subject to overheating and destruction of the grate bars may be considerable, particularly with high sulphur coals. Any of these stokers will operate efficiently with little trouble from clinker and burning if installed in properly designed furnaces and operated at their proper capacity. Very few stokers of this type are operated at loads exceeding 200 per cent of the rated boiler capacity and for this reason are not much in evidence in the modern large central station. The Roney stoker, Fig. 97, and the Wilkinson stoker, Fig. 98, are examples of the frontfeed type of step grates. The Roney stoker consists of a hopper for receiving the coal, a set of rocking stepped grates inclined at a proper angle from the horizontal, and a dumping grate at the bottom of the incline for receiving and discharging the ash and clinkers. The dumping grate is divided into several sections for convenience in handling. The coal is fed onto the inclined grate from the hopper by a reciprocating "pusher" actuated by the "agitator." The power is supplied through an eccentric operated by a small engine or motor. The normal feed is about 10 strokes per minute. The grate bars rock through an arc of 30 degrees, assuming

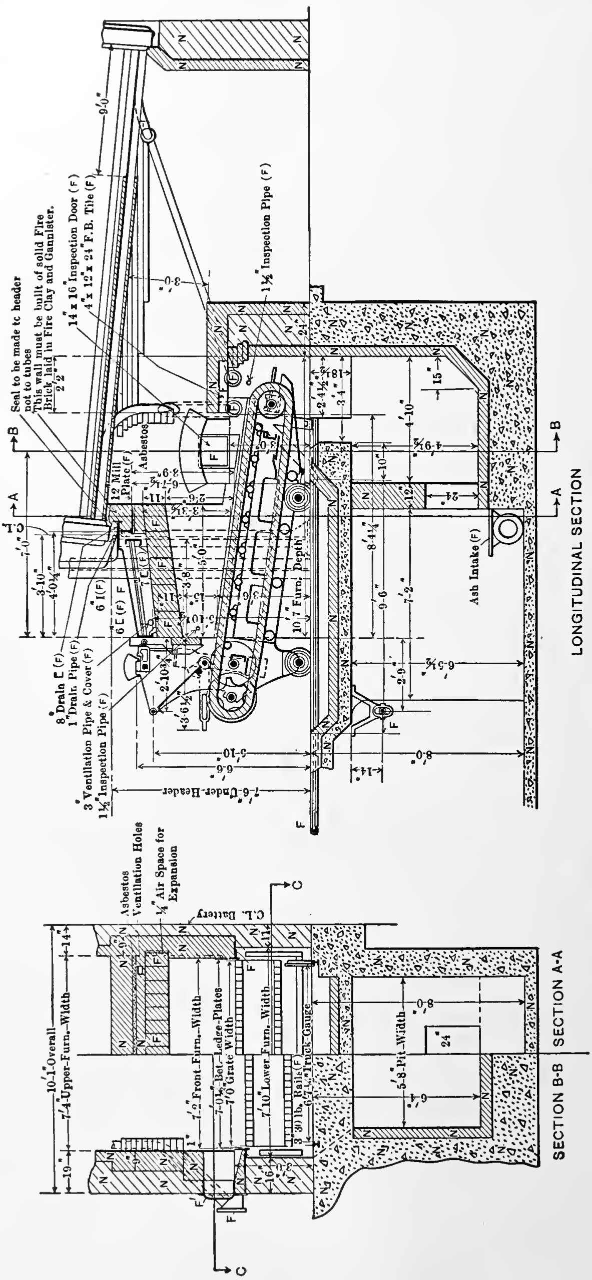
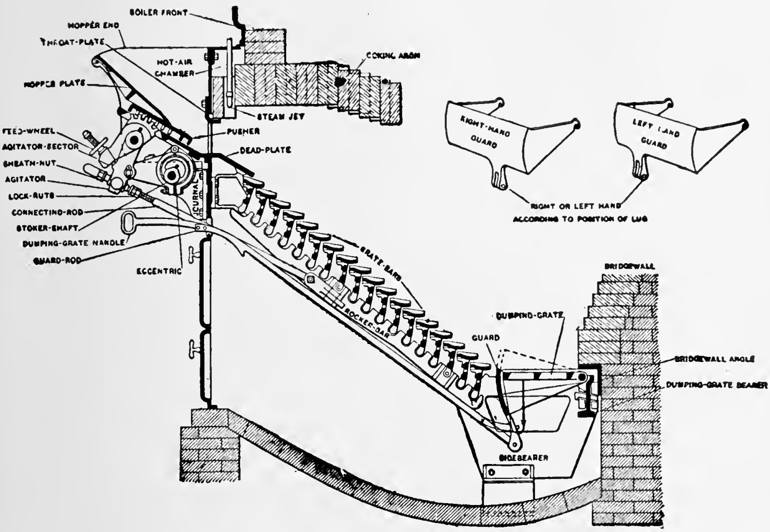


Fig. 96. Green Type K Chain Grate and Smokeless Setting.



Details of Construction of the Roney Mechanical Stoker

Fig. 97. Details of Roney Stoker.

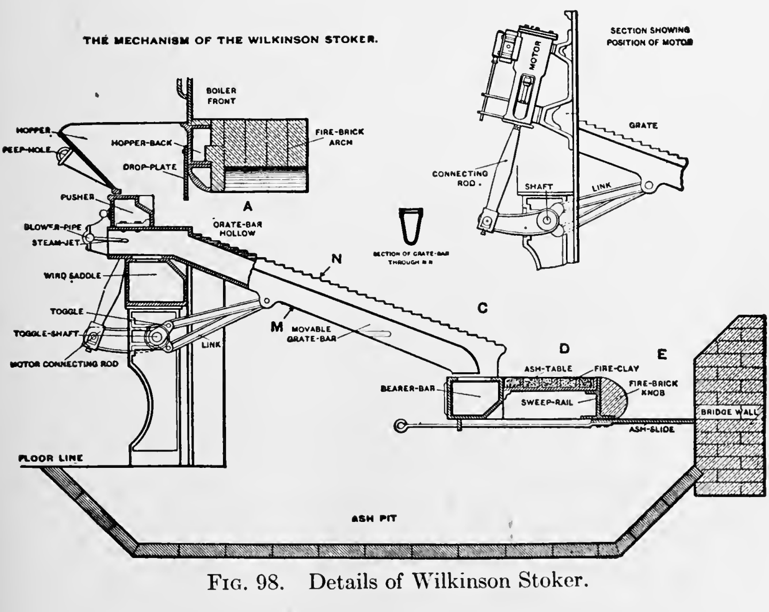


Fig. 98. Details of Wilkinson Stoker.

alternately horizontal and inclined positions. The construction permits abundance of air to pass through the fuel, with little or no possibility of coal dropping through the grate. A coking arch of fire brick is sprung across the furnace as indicated. This stoker operates with natural or forced draft and, with suitable headroom, effects complete and efficient combustion.

In the Wilkinson stoker the inclined grate bars are hollow and are arranged side by side, every alternate bar being movable. When in operation there is a constant sawing action of the grate bars, causing the fuel to flow forward and downward. A small steam jet with about $\frac{1}{16}$-inch opening is introduced into the end of each hollow grate bar, and induces the required amount of air for combustion through air openings approximately $\frac{1}{4}$ inch wide by 3 inches long. These stokers are driven by two small hydraulic motors. The water is furnished by a small pump and is used over and over again.

Front Feed Stokers and Smokeless Settings: Power, Nov. 17, 1914, p. 712.

The sidefeed step-grate stoker is represented by the "Murphy," "Detroit," and "Model." Fig. 99 shows a front vertical section through a Murphy automatic stoker and furnace. The apparatus is in effect a Dutch oven equipped with an automatic feeding and stoking device. Coal is introduced either mechanically or by hand into the magazine at each side of the furnace and above the grate and descends by gravity upon the coking plate. Reciprocating stoker boxes push the coal upon the grate bars. Every alternate grate bar is movable and pivoted at its upper end. A rocker bar driven by a small motor or engine causes the lower ends to move up and down, this action producing the required stoking effect. A device for grinding up the clinker and ash is provided as shown at the bottom of the furnace. This is hollow and is connected by a 2-inch pipe with the smoke flue, so that the cold air passing through prevents it from being destroyed by the heat. Air is supplied to the green coal through flues passing under the coking plates, and the speed of the stoker boxes and grate bars can be regulated to conform to any rate of combustion. On account of the large fire-brick combustion chamber, this stoker with careful manipulation is capable of practically smokeless combustion.

Side Feed Stokers and Smokeless Settings: Power, Nov. 8, 1914, p. 802.

103. Underfeed Stokers. — This type of stoker has practically supplanted all other types in the modern large central station burning coking bituminous coals and is adapted to all grades and sizes of free burning bituminous coal. The underfeeds are essentially forced draft stokers, since they operate with restricted air openings and very deep

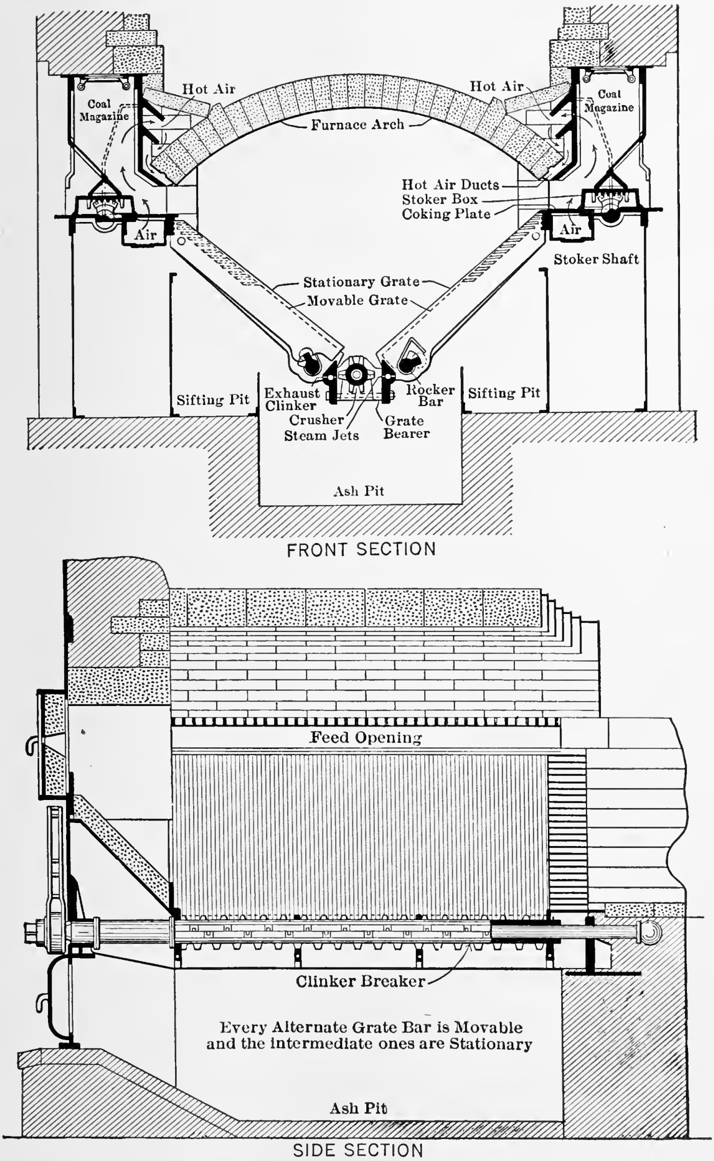


FIG. 99. Murphy Furnace.

fires. The forcing capacity is tremendous, reaching as high as 450 per cent of rating. With this class of stoker headroom is the principal factor. For smokeless combustion special brickwork is not necessary and coking arches may be dispensed with entirely. Some of the best-known underfeeds are the Jones, Taylor, Riley, Westinghouse, Combustion Engineering Company's Type "E," and American.

Fig. 100 shows the general principles of the Jones underfeed stoker. It consists of a steam-actuated ram with a fuel hopper outside of the furnace proper and a fuel magazine and auxiliary ram within. Air for

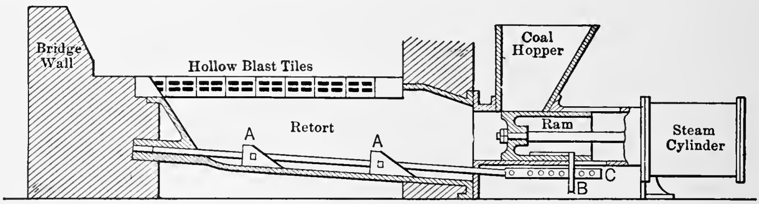


FIG. 100. Jones Underfeed Stoker.

combustion is admitted through openings in the tuyere blocks on either side of the retort. Coal is fed into hoppers and forced *under* the bed of fuel in the stoker retort, where it is subjected to a coking action. After liberation of the volatile gases the coke is pushed toward the top of the fire. The top of the fire, nearest the boiler, is always incandescent. Each charge of coal is given an upward and backward movement. Air is admitted through the tuyere blocks at the point of distillation of the gases. Grate bars form no part of the Jones system, and it is therefore impossible for the fuel to fall through. There is no ash pit. The non-combustible matter is removed from the furnace by hand. The standard size of the retort is about 6 feet in length, 28 inches in width, and 18 inches in depth, and experience has shown that other sizes are not necessary since the spaces between retort and side wall of the various furnaces may be provided for by extending the width of the dead plates. One or more stokers are installed in each furnace, depending upon the capacity of the boiler and the width of the furnace. The steam pressure automatically controls air and fuel supply, proportioning them to each other and to varying loads in the correct degree. The result is that the stoker effects complete and smokeless combustion. The only variable element in the operation of this stoker, once it is correctly installed, is cleaning of fires, but if the fireman is careful to burn down the coals before breaking them up the production of smoke may be avoided. Jones underfeed stokers are adaptable to all grades and sizes of bituminous coal, and on account of forced draft are capable of burning very low grades of coal.

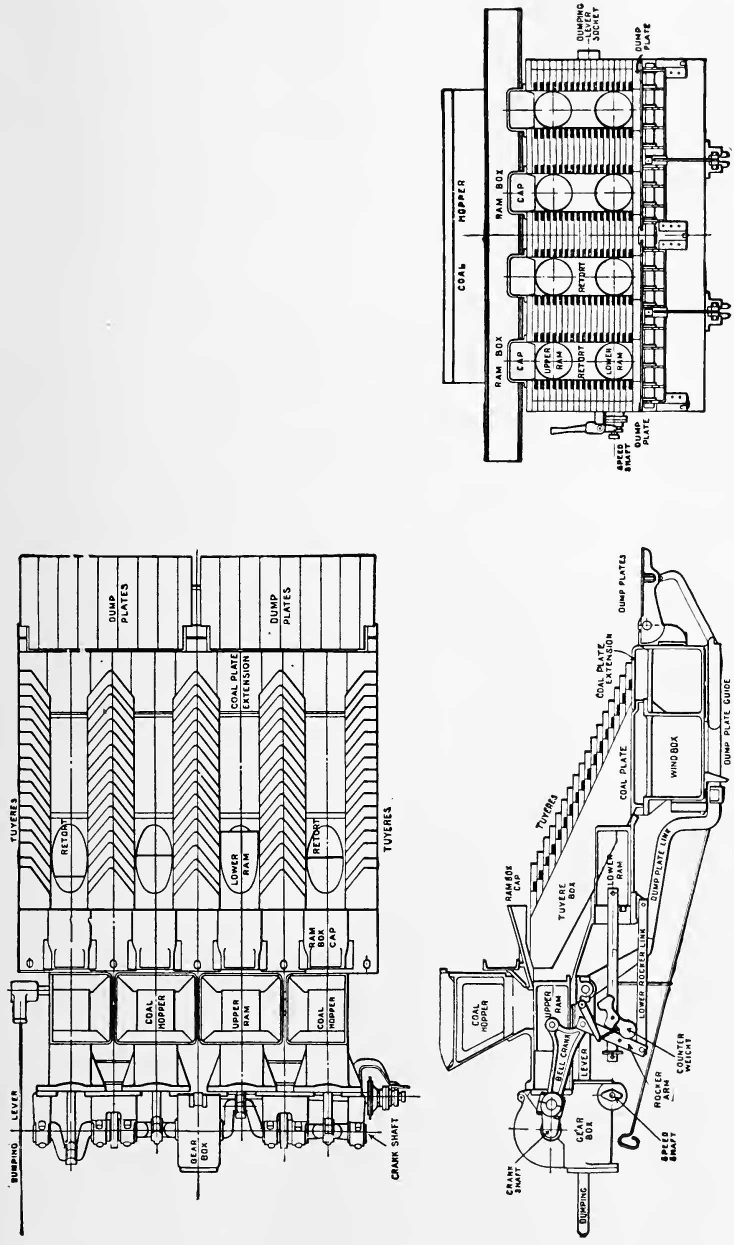


FIG. 101. General Details of a Taylor Underfeed Stoker with Hand-operated Ash Dump.

Fig. 101 shows the general details of a Taylor underfeed stoker for burning bituminous coals. The device consists essentially of a series of alternate retorts and tuyere boxes inclined as indicated. Each retort is fitted with two rams — the upper for pushing the green fuel outward and upward and the lower one for forcing the fuel bed and refuse toward the dump plates at the rear. Air is supplied by a volume blower and enters the furnace through openings in the tuyere boxes. The dump plates are hung on the rear of the wind box and are controlled from the front of the stoker. Extension grates are inserted between the mouth of the retort and the dump plates, when the nature of the fuel makes this arrangement desirable. This extension may be rocked if necessary. In the later designs of this stoker the dump plates are actuated by a steam cylinder. The valve mechanism is

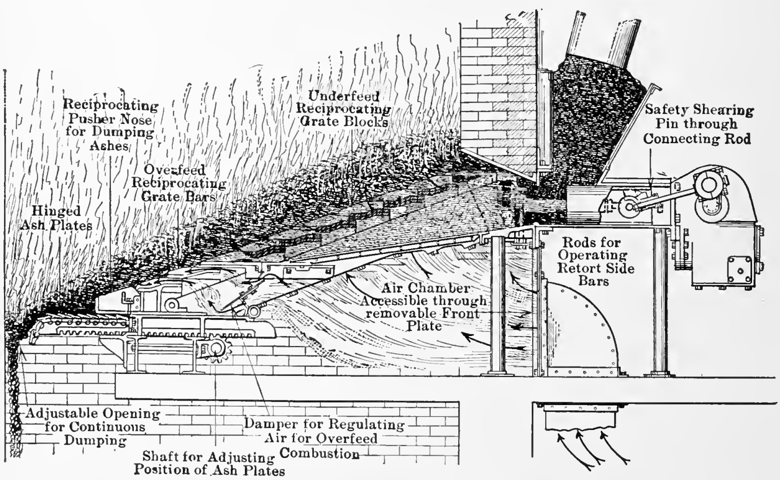


FIG. 102. General Assembly of Riley Underfeed Stoker.

placed at the side door so that the operator can manipulate the dump plates in full view of the ash and clinker. The plate is so designed that it can be rocked without dumping, hence a similar motion in the extension grates is unnecessary. The stoker and blower are operated by the same engine, the air and coal supply being automatically controlled by the variation in steam pressure. Taylor stokers may be operated smokelessly and efficiently at very heavy overloads and are much in evidence in the eastern states. The steam required to operate the blower and stoker varies from 2.5 to 5 per cent of the steam generated, depending upon the size of the installation and the percentage of rating developed.

The Riley, Fig. 102, is a multiple retort stoker with an incline of about 20 degrees. The distinctive feature of this stoker is that the sides

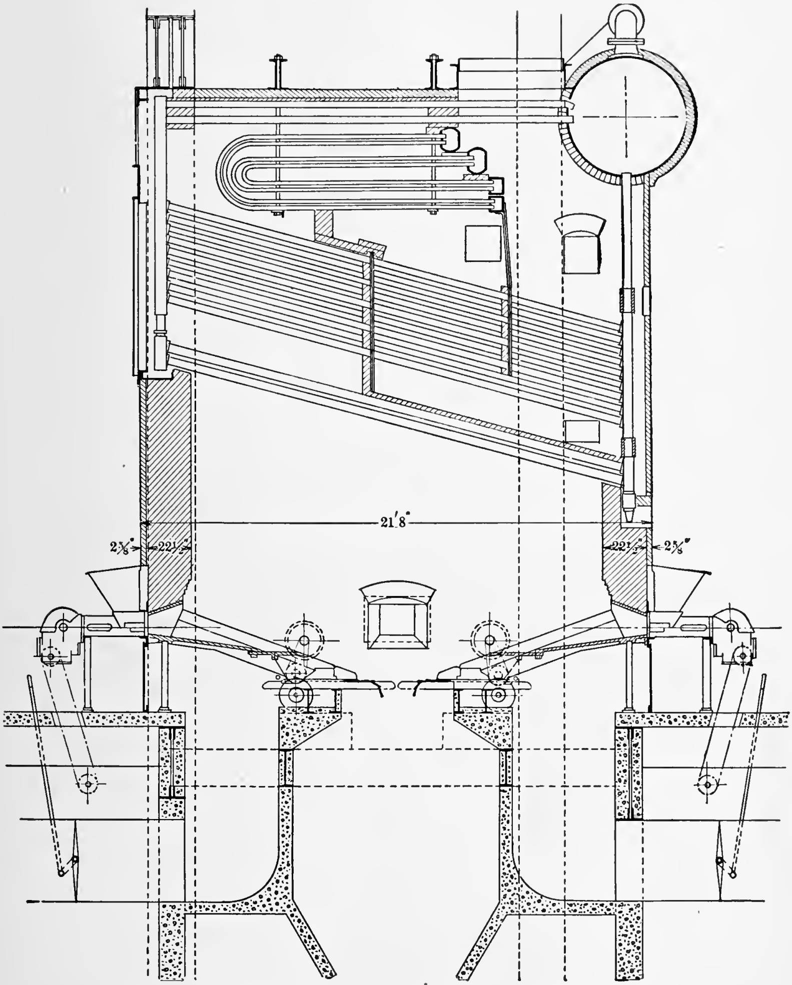


FIG. 103. Duplex Furnace with Riley Underfeed Stokers.

of the retort reciprocate relative to the bottom. This causes the fuel to be moved at a uniform rate out of the retort and across the ash-supporting plates until it is discharged through the adjustable openings next to the bridge wall. No special shape of wind box is required with this stoker since the air chamber is formed by the boiler side walls and any convenient floor The air supply may be controlled by hand or automatically. One man can operate ten or twelve stokers; siftings are negligible amounting to 0.2 or less and the wind box and retorts need be cleared but once a month. The power required to operate the stoker is approximately ½ horsepower per retort. Fig. 103 illustrates one of the latest installations of the Riley stoker for high efficiency and extremely high overload capacity.

Maintenance Costs of Two 2365-hp. Boiler Units with Taylor Stokers: Trans. A.S.M.E., Vol. 35, 1913, p. 327. *Installation Data for Underfeed Stokers:* Elec. Wld., Nov. 18, 1916, p. 1009.

Underfeed Stokers: Power, Dec. 15, 1914, p. 838; Jan. 26, 1915, p. 132.

104. Sprinkling Stokers. — In this system of stoking the fuel in finely divided form is distributed by sprinkling uniformly over the entire area of the grate. With the proper adjustment of air supply and feed the volatile gases are distilled off continuously before the grate is covered by the new coal and without materially lowering the temperature of the incandescent fuel. Mechanically the operation involves considerable difficulty. Sprinkling stokers do not conform to Chicago requirements.

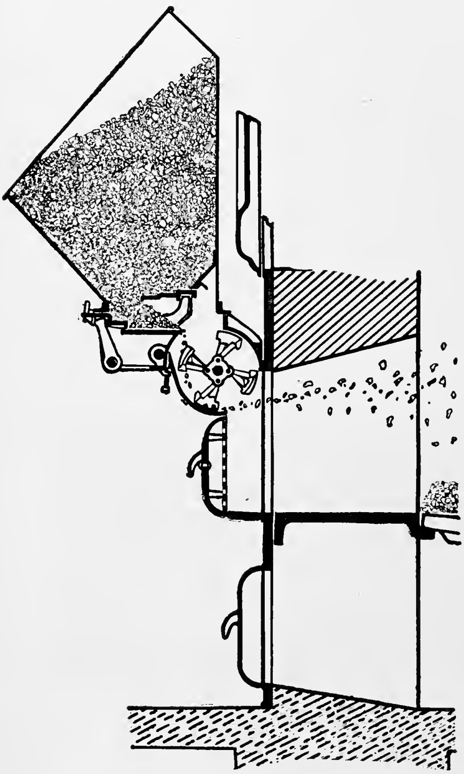


Fig. 104. Swift Sprinkling Stoker.

Fig. 104 gives the general details of the Swift stoker, illustrating a commercially successful stoker of this type. The apparatus is self-contained and is bolted to a frame casting in front of the setting, and takes the place of the fire door. It may be swung back from the fire-door opening in much the same manner as the ordinary fire door. Coal of nut size or smaller is fed into a small hopper, of about 300 pounds' capacity, from which it gravitates on to a berm plate and pusher plate. By means of the latter the fuel is fed to rapidly revolving spreaders, which crush it into small particles and throw it onto the grate. The fine or powdered coal is burned in suspension and the heavier coal falls to the grate. The spreaders are heavy pieces of cast steel, revolving about a common axis and shaped helically so as to throw the fuel in a direction at right angles to the face of the machine. There are several of these spreaders so arranged on the shaft that adjacent spreaders throw the fuel in different directions. This stoker is not self-cleansing, that is, the ashes must be removed by hand or by suitable shaking grates.

105. Smoke Determination. — Smoke measurements may be either quantitative or relative.

The most satisfactory method, at this writing, of determining the quantity of smoke passing through a chimney is that adopted by the Chicago Association of Commerce. A continuous sample of chimney gas is drawn from the stack by means of a special Pitot tube and exhauster, and the solid particles are entrapped in a filter. The tube is so arranged that the rate of flow through the apparatus is the same as that in the chimney. Since the area of the tube opening bears a fixed ratio to that of the chimney, the weight of carbon, cinders, soot and the like caught in the tube filter is a measure of the total weight emitted from the stack.

Quantitative measurements are of considerable value in estimating the amount of energy lost in the production of visible smoke, but are seldom attempted in regular practice.

There are several methods of determining smoke, relatively. The most common is that devised by Ringelmann, and is commercially known as the Ringelmann Smoke Chart. The chart, as published by

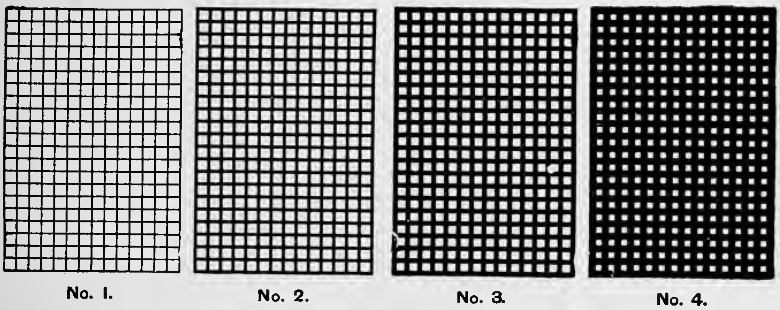


Fig. 105. Ringelmann Smoke Chart (Greatly Reduced).

the U. S. Geological Survey and used by the Smoke Department of the City of Chicago and other municipalities, consists essentially of a cardboard folder 12 by 26 inches over all. Four charts are printed on this folder, each chart consisting of 294 squares, 14 squares wide by 21 squares in length, the width of the lines and spacings varying as illustrated in Fig. 105. At a distance of 50 feet from the observer the lines become invisible and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer places the chart on a level with the eye (at the distance stated, and as nearly as possible in line with the chimney) and notes which card most nearly corresponds with the color of the smoke. Observations should be made at 15-second intervals and recorded as in Fig. 106. No smoke is recorded as No. 0, 100 per cent as No. 5, and the intermediate colors as indicated by the cards.

Experienced observers often record in half-chart numbers. Although these observations depend upon the personal element it is the opinion of the Chicago Smoke Department that only a little experience is necessary to effect consistent results with different observers.

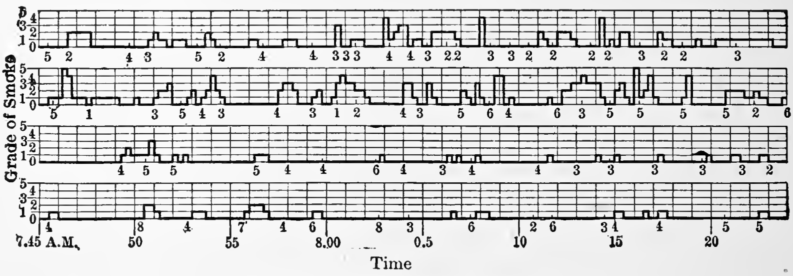


FIG. 106. Smoke Record Chart.

Prior to 1910 a chimney was held to be a smoke nuisance by the Chicago smoke inspection authorities when it emitted smoke of No. 3 density, according to the Ringelmann chart, for 7 minutes during one hour, as based on the original ordinance. With this standard the owners of a chimney which emitted but a very small total quantity of smoke might be liable to punishment, whereas, with a chimney which continuously emitted smoke of a density less than No. 3, the owners would be safe from legal prosecution, although the total quantity emitted might be many times as great.

The total smoke emitted is now taken into consideration. Observations are made on a given stack every 15 seconds throughout the entire day and the total "smoke units" are recorded, from which the average smoke density for the entire period is calculated.

A "smoke unit" is the equivalent of No. 1 smoke (Ringelmann scale) emitted for one minute. No. 1 smoke has a density of 20 per cent; No. 2, 40; No. 3, 60; No. 4, 80; and No. 5, 100 per cent. Thus, if a stack emits No. 3 smoke for 6 minutes, 18 smoke units are charged against it. If this smoke was emitted during one hour's observation, then

$$\frac{3 \times 6 \times 20}{60} = 6 \text{ per cent}$$

is the average density of smoke emitted during the period of observation.

If observations on a given stack show that the density averages more than 2 per cent, although the owner may not be legally liable, an appeal is made to his personal and civic pride by a representative of the smoke-inspection department. For example, if a certain hotel stack emits smoke of more than 2 per cent average density, the smoke department finds a plant record of similar design and equipment, preferably a hotel plant, which shows a record well below the 2 per cent mark. This plant is then pointed out to the owner or manager having the objectionable chimney and he is asked if he cannot do equally well when he has practically the same equipment, etc.

It has been found that this method of procedure often produces quicker and better results than a threat to go to law.

New Methods of Approaching the Smoke Problem: Osborne Monnett, Jour. Wes. Soc. Engrs., Nov. 4, 1912.

DIVISIONS OF MESH: RINGELMANN'S SMOKE CHART.

Numbers give Relative Smoke Density.	Thickness of Lines, mm.	Distance in the Clear between Lines, mm.
0	All white	All white
1	1	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5
5	All black	....

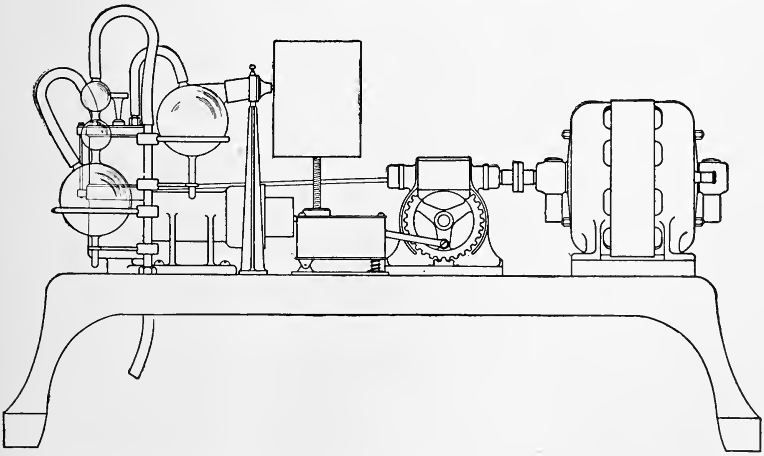


Fig. 107. Hammler-Eddy Smoke Recorder — Motor-driven Type.

The Hammler-Eddy smoke recorder, Fig. 107, is one of the most successful devices for automatically recording the density of the smoke

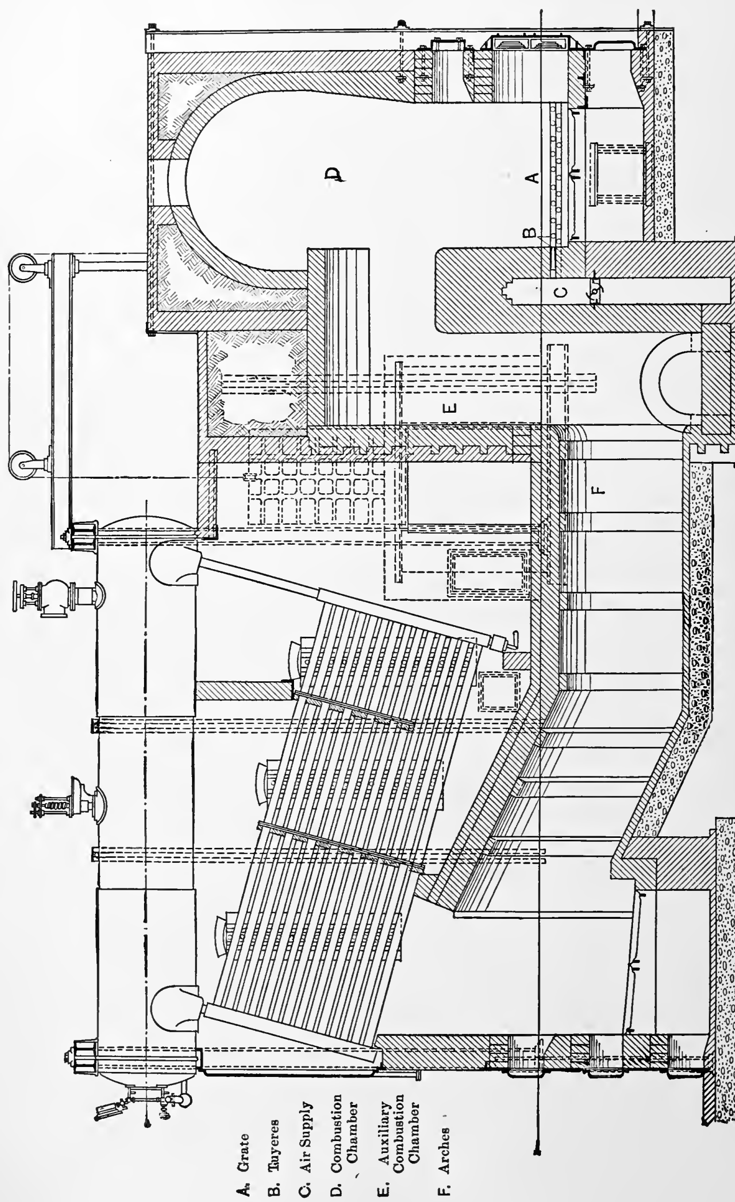


Fig. 108. Sectional Elevations of the Green Furnace for Burning Bagasse.

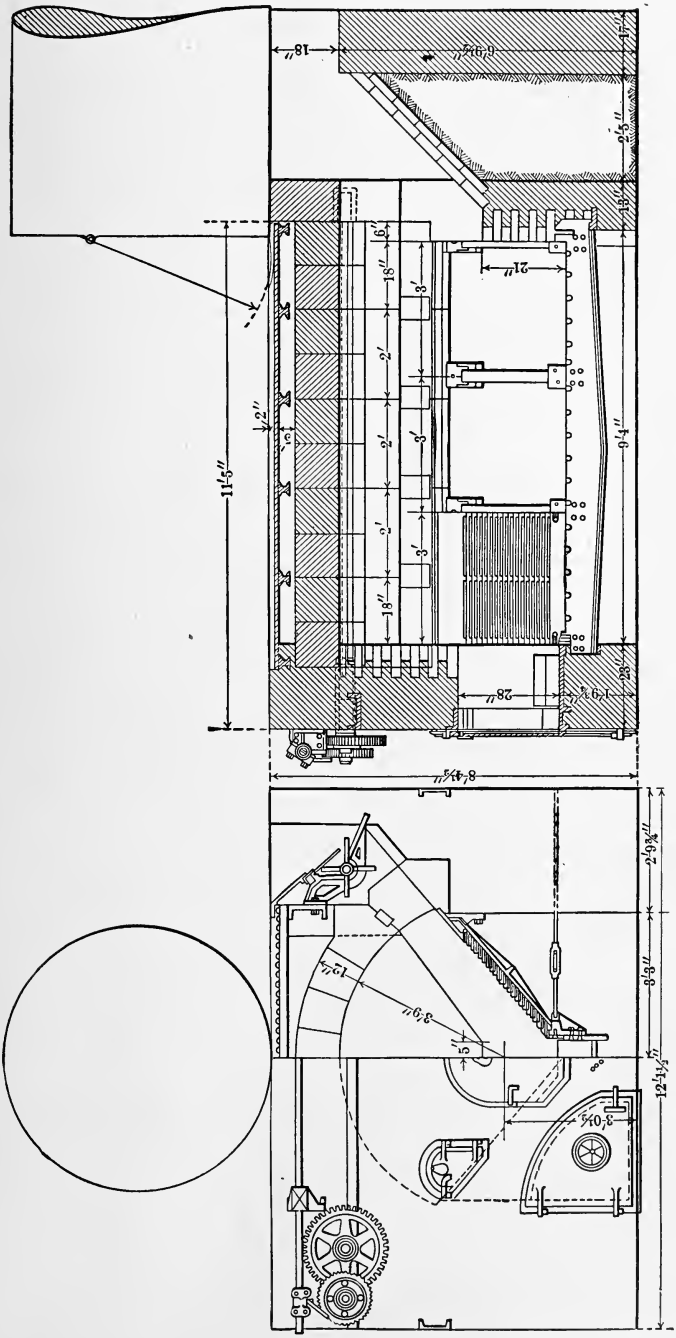


FIG. 109. End and Side Sectional Elevation of the Myers Tanbark Furnace.

independent of personal observations. This apparatus consists essentially of a small motor-driven vacuum pump, which draws a continuous sample of the products of combustion from the uptake, breeching or stack and discharges it against a paper-covered drum revolved by clockwork. The *density* of the smoke, the *time* at which visible smoke is being emitted and the *duration* of the smoke-production period are automatically recorded on the paper by the smoke itself. Before reaching the pumps the gases pass through a glass "emergency" condenser and a large portion of the vapor content is removed. The pump discharges the partially dried gases against a surface of sulphuric acid (which removes the last trace of moisture) and forces the smoke in the form of a small jet of dry powder onto the surface of the recording paper. The sampling tube leading from the flue to the pump is connected with a steam line and is "blown out" each time a card is changed. The instrument is very compact and portable and may be placed anywhere with respect to the chimney. A number of these appliances in Chicago power plants are giving excellent satisfaction. In a more recent design the pump is replaced by a steam siphon.

106. Cost of Stokers. — The following is the relative approximate cost of stokers suitable for a Babcock and Wilcox boiler of 350-horsepower rated capacity with 45 square feet of grate surface; height of chimney above grate, 175 feet; coal burned, Illinois screenings. The cost of installation included, exclusive of brickwork, is

1. Chain grate and appurtenances	$1500
2. Jones underfeed stoker	1400
3. Hawley down-draft furnace	1350
4. Burke smokeless furnace	1000
5. Roney stoker	1300
6. Murphy furnace and stoker	1350
7. Wilkinson stoker	1200

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CHAPTER V

SUPERHEATERS

107. Advantages of Superheating. — That superheated steam results in ultimate plant economy is evidenced by the fact that the largest and most economical plants in the world are equipped with superheaters. With very high pressures and temperatures, initial cost and upkeep may offset the thermal gain due to the use of superheated steam, but in general, a limited amount of superheat effects ultimate economy in nearly all cases. Practically all modern central turbo-generator stations and large isolated piston engine plants are designed for superheated steam. No general rules can be drawn as to the extent of the saving made because of the great number of variable factors entering into the problem. Each installation must be considered by itself and due consideration given to such items as the type and size of prime movers, character of service, nature and cost of fuel, piping, first cost, upkeep and the like. The logical procedure is to determine the saving in fuel regardless of other factors and then deduct the extra expense due to first cost and upkeep. The resulting net gain or loss will show whether or not the use of superheat is advisable.

Theoretically, all types of steam-driven prime movers show increased heat efficiency with superheated steam, but the gain is usually less than that actually realized in the commercial mechanism. Aside from the gain in the prime mover there is the possible added efficiency in the boiler plant. It is true that the heat required to superheat steam is furnished by the fuel and when a definite weight is superheated an added amount of fuel must be burned, but with a properly designed superheater integral with the boiler the over-all efficiency of boiler and superheater is usually somewhat higher than if saturated steam alone were generated, so that the added amount of fuel is less than the heat gained by the steam. In addition to the thermal gain in the prime mover and boiler there may be a reduction in heat losses in the piping system because smaller pipes may be used and because superheated steam gives up heat less rapidly than does wet steam. Furthermore, the increased economy of the prime mover may permit a reduction in the size of boilers, condensers and other auxiliary apparatus.

The principal advantages of superheated steam in connection with piston engine work are:

1. At high temperatures it behaves like a gas and is, therefore, in a far more stable condition than in the saturated form. Considerable heat may be abstracted without producing liquefaction, whereas the slightest absorption of heat from saturated steam results in condensation. If superheat is high enough to supply not only the heat absorbed by the cylinder walls but also the heat equivalent of the work done during expansion, then the steam will be dry and saturated at release. This is the condition of maximum efficiency in a single cylinder. (Ripper, "Steam Engine Theory," p. 155.) Greater superheat than this may result in a loss of energy unless the steam is exhausted into another cylinder. To obtain dry steam at release the steam at cut-off must be superheated from 100 to 300 deg. fahr. above saturation temperature, depending upon the initial condition of the steam and the number of expansions, a higher degree of superheat being required for earlier cut-off. A superheat of from 250 to 350 deg. fahr. at admission is necessary to insure dry steam at release in the average single-cylinder engine cutting off at one-fourth stroke, boiler pressure 100 pounds gauge. In most cases superheat is only carried so far as to reduce initial condensation, the steam becoming saturated at cut-off, thus permitting efficient lubrication. There will be a reduction of approximately 1 per cent in cylinder condensation for every 7.5 to 10 degrees of superheat. In compound and triple-expansion engines the steam is ordinarily superheated between each stage as well as before admission to the high-pressure cylinder.

2. A moderate amount of superheat produces a large increase in volume, the pressure remaining constant, and diminishes the weight of steam per stroke for a given amount of work. For example, the volume of one pound of saturated steam at 165 pounds pressure (absolute) is 2.75 cubic feet, and its temperature is 366 deg. fahr. The total heat of one pound of this steam above the freezing point is 1195 B.t.u. By adding 108 B.t.u. in the form of superheat its temperature will be increased to 565.8 deg. fahr. (superheated 200 deg. fahr.) and its volume to 3.68 cubic feet (specific heat taken as 0.54). Thus an increase of 9 per cent in the heat effects an increase of 34 per cent in the volume, which means a corresponding reduction in the weight of steam admitted to the engine per stroke. These figures are purely theoretical, as no allowances have been made for condensation of the saturated steam or for reduction in temperature of the superheated steam.

3. Superheated steam has a much lower thermal conductivity than saturated steam, and, therefore, less heat is absorbed per unit of time by the cylinder walls.

The water rate of the steam turbine is decreased by superheating but to a less extent than the piston engine. Theoretically the improvement in steam economy is the same for both types of prime movers, pressure and temperature ranges being the same in each case, but in actual practice the gain is more pronounced with the piston engine. In general, the less economical the steam motor the more is the gain effected by superheating. Aside from the gain in heat efficiency the use of superheated steam benefits the turbine by reducing erosion of the blades and by lowering skin friction and windage. The fact that nearly all modern steam turbine plants are operated with superheated steam is evidence that superheating results in ultimate plant economy.

108. Economy of Superheat. — Many comparative tests of engines and turbines using saturated and superheated steam under varying conditions of pressure and temperature have been made during the past few years, showing in all cases decreased steam consumption due to superheat. In the majority of moderately superheated steam installations the ultimate gain was a substantial one, but in a few cases involving the use of very high temperatures and pressures, the extra investment and cost of maintenance neutralized the reduction in steam consumption, resulting in an actual loss when measured in dollars and cents per unit output. With high degree of superheat (over 250 deg. fahr.) apparatus of a special nature is necessary and it is questionable whether the additional first cost, care and liability to operating difficulties, upkeep and maintenance will not offset any fuel saving accomplished.

As far as steam consumption per horsepower-hour is concerned, superheating usually increases the economy of the piston engine from 5 to 15 per cent and in some instances as much as 40, the latter figure referring to the more wasteful types. A fair estimate of the average reduction in steam consumption per horsepower-hour with moderate superheating, that is, from 100 to 125 deg. fahr., based on continuous operation of existing plants, is:

	Per Cent.
1. Slow running, full stroke, or throttling engines, including direct-acting pumps	40
2. Simple engines, non-condensing, with medium piston speed, including compound direct-acting pumps	20
3. Compound condensing Corliss engines	10
4. Triple-expansion engines	6

European builders guarantee steam consumption with highly superheated steam (total temperatures 750 to 850 deg. fahr.) as follows:

	Pounds per I.hp-hour
Single-cylinder condensing engines (uniflow)	8.5
Single-cylinder non-condensing engines (uniflow)	12.0
Compound condensing engines (locomobile)	8.0
Compound non-condensing engines (locomobile)	10.5

An exceptionally low steam consumption is credited to a locomobile compound using steam superheated to 806 deg. fahr. at an initial pressure of 220 pounds absolute. When exhausting against an absolute back pressure of 1.32 pounds the steam consumption was 6.95 pounds per i.hp-hour. (Zeit. des Ver. deut. Ingr., Mar. 18, 1911, p. 415.)

In high-pressure steam turbines the water rate is improved approximately one per cent for every 8 to 12 deg. fahr. superheat; the higher rate holding for about 50 degrees superheat and the lower for about 200 degrees. It is difficult to estimate the actual gain in heat economy due to superheating in very large turbines, since they are not designed for saturated steam and tests with the latter do not offer a true comparison. In a general way the average reduction in steam consumption for these large units is about 1 per cent for every 10 deg. fahr. increase in superheat. One of the best recorded performances is that of a 20,000-kilowatt turbo-generator installed in the New River station of the Buffalo General Electric Co.; with initial absolute pressure of 265 lb. per sq. in., 275 deg. fahr. superheat and absolute back pressure of 1 inch of mercury, the steam consumption was 10.25 lb. per kilowatt-hour.

In comparing the performances of engine and turbines using saturated steam it is advisable to base all results on the heat consumed per unit output rather than on the steam consumption, since the latter is apt to give a false idea of the relative economies. The real measure of economy is the cost of producing power, taking into consideration all charges, fixed and operating, and the next best is the coal consumption per unit output, but as a means of comparing the motors only, the heat consumption per unit output is very satisfactory. (See paragraph 162.)

See paragraph 182 for the influence of superheat on the economy of reciprocating engines and paragraph 221 for the influence on steam turbines.

109. Limit of Superheat. — In this country steam temperatures exceeding 600 deg. fahr. are seldom employed, while in Europe few if any plants are installed without superheaters, and 600 degrees is a common temperature with a maximum of about 850. There is no particular mechanical difficulty in designing power plant apparatus to withstand temperatures as high as 850 deg. fahr., and for industrial purposes steam temperatures of 1000 deg. fahr. are not uncommon, but first cost and maintenance usually offset any thermal gain accomplished except perhaps where fuel is very high and labor is cheap. In this country where fuel is comparatively cheap but material and labor are high, a moderate amount of superheat appears to effect the best economy.

Experience has shown that with engines of ordinary design, slide-valves and Corliss, the temperature at the throttle should not exceed 500 deg. fahr. This corresponds to a superheat of 160 degrees with steam at 100 pounds gauge pressure, and 130 degrees at 150 pounds. This degree of superheat insures practically dry steam at cut-off in the better grade of engines. Just how far superheating can be carried with a given engine of ordinary construction can be determined by experiment only, but a temperature of 500 degrees is probably an outside figure and 450 degrees a good average. Higher temperatures are apt to interfere with lubrication and sometimes cause warping of the valves. With temperatures below 450 degrees no difficulties are ordinarily met with.

With highly superheated steam involving temperatures of 600 deg. fahr. or more, the poppet-valve type of engine (Figs. 196, 203) is ordinarily employed, though balanced piston and specially designed Corliss valves are not uncommon. The poppet valve is not distorted by heat and requires no lubrication. In Europe these engines have been brought to a high state of efficiency, but have not been generally adopted in this country. The steam end of the composite gas-steam engines at the Ford Motor Company's plant, Detroit, are of Corliss valve design and though the steam at admission has a temperature of 700 deg. fahr., no difficulty is experienced with lubrication.

Owing to the absence of rubbing parts in contact with the steam, and because the casing is not subjected alternately to high and low temperatures, steam turbines may be designed to operate successfully with temperatures up to 800 deg. fahr., though temperatures above 600 deg. are exceptional. The majority of turbine installations in this country, including the very latest, are designed for temperatures under 650 degrees.

Properties of Superheated Steam. — See Chapter XXII.

How to Use Superheated Steam: Eng. Mag., May, 1916, p. 208; June, 1916, p. 413.

110. Types of Superheaters. — Superheaters are manufactured by practically all boiler builders, the characteristics of the boiler being embodied to a large extent in the design of the superheater. The superheater may be independently fired or placed in the boiler setting. In the latter arrangement the superheater may be located in the furnace, as in Fig. 53, at the end of the heating surface as in Fig. 116, or at some intermediate point, as in Figs. 55 and 110. Since the absorption of heat depends chiefly upon the average temperature difference between the gases and the steam and the extent of superheating surface, the required degree of superheat may be obtained from a small extent of heating surface in the furnace, a large amount in the rear of the heating surface or a proportionate amount in intermediate locations. In a general sense the sum of the boiler heating surface and superheating surface per boiler horsepower is practically the same for any degree of superheat. The cost of a superheated steam boiler is approximately equal to that of a saturated steam boiler since the superheated plant has less steam to generate. The requirements of a successful superheater are:

1. Security of operation, or minimum danger of overheating.
2. Economical use of heat applied.
3. Provision for free expansion.
4. Disposition so that it may be cut out without interfering with the operation of the plant.
5. Provision for keeping tubes free from soot and scale.

Superheaters may be *separately fired* or *indirectly fired.* The advantages of the separately fired superheater are:

1. The degree of superheat may be varied independently of the performance of the boiler.
2. It may be placed at any desired point.
3. Repairs are readily made without shutting down the boiler.

Some of the disadvantages are:

1. It requires separate attention.
2. Saturated steam only can be furnished to the prime movers in case of a breakdown to the superheater.
3. Extra piping is required.
4. Extra space is required.

The indirectly fired superheater arranged in the boiler setting has the advantage of:

1. Lower first cost.
2. Higher operating efficiency.
3. Minimum attention.
4. Minimum space requirements.

As ordinarily installed the indirectly fired superheater is subject to the fluctuating temperatures of the furnace so that forcing the boiler has a similar effect on the superheater. In some cases the superheater adjusts itself automatically to the load requirements maintaining a constant degree of superheat at all loads, but in most cases the degree of superheat increases with the load, see Fig. 124. Standard central station practice in this country favors the superheater contained within the boiler setting.

Figs. 110 and 111 show the application of superheating coils to a

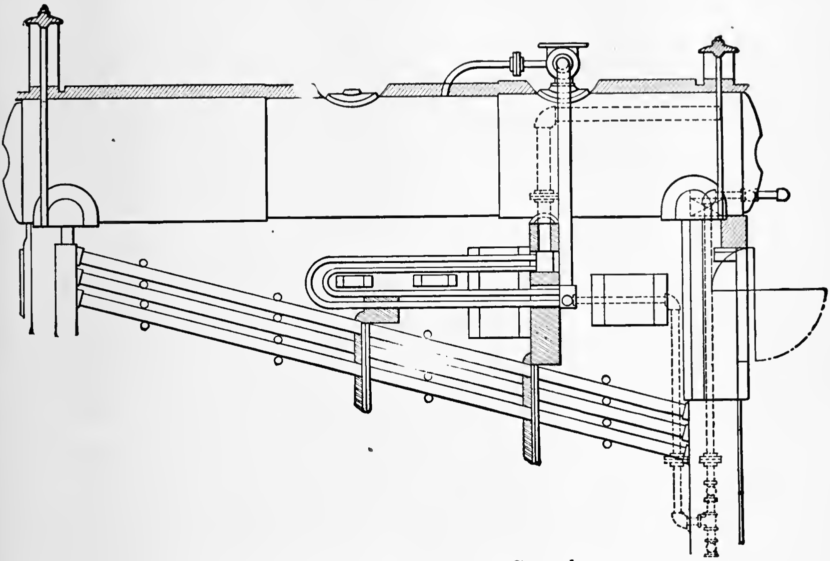


FIG. 110. Babcock and Wilcox Superheater.

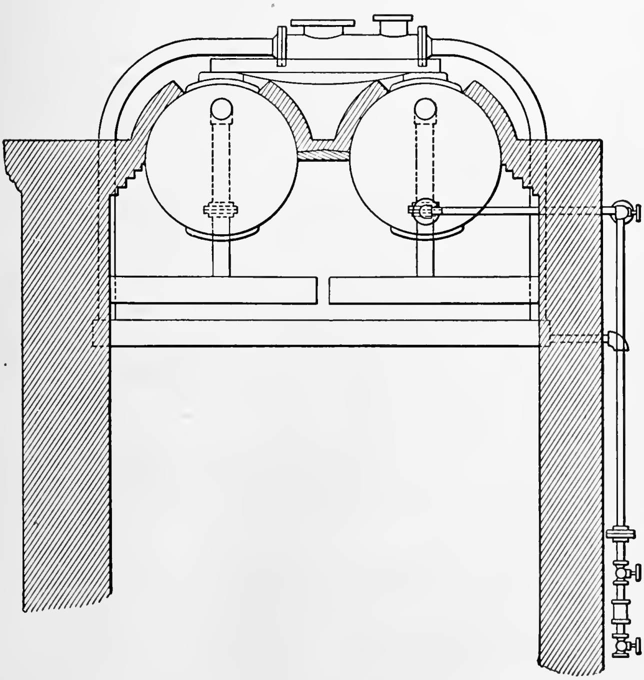


FIG. 111. Babcock and Wilcox Superheater.

Babcock and Wilcox boiler illustrating the usual location of the indirectly fired type. The superheater consists of two transverse square wrought steel manifolds into which two sets of 2-inch cold drawn seamless steel tubes bent to a "U" shape are expanded. The tubes ordinarily are arranged in groups of four. Saturated steam flows from the dry pipe located within the drums to the upper manifold. The latter is divided into as many sections as there are drums so as to avoid expansion strain. From the upper manifold the steam passes through the "U" shaped tubes to the lower one (which is continuous) and thence to a cast-steel "superheater center" fitting supported over the drum. The "superheater center" fitting is provided with a superheated steam outlet and an extra opening for the reception of the superheater safety valve. This safety valve is furnished as a part of the regular equipment and is set two pounds lower than the safety valves of the boiler. This is essential so as to provide a flow of steam through the superheater and to prevent any overheating of the latter in case the load should be suddenly thrown off the boiler. A small pipe connects the center fitting with the saturated steam space in the drum and is for the purpose of equalizing the pressure when the discharge from the superheater is closed. While a flooding device is not necessary its use is recommended by the Babcock & Wilcox Company. This consists essentially of a small pipe connecting the lower manifold with the water space of the boiler and by means of which the superheater may be flooded. Any steam formed in the superheater tubes is returned to the boiler drum through the collecting pipe which, when the superheater is at work, conveys saturated steam into the upper manifold. When steam pressure has been attained the superheater is thrown into action by draining the water away from the manifolds and opening the superheater stop valve. With the proportion of superheating surface to boiler surface ordinarily adapted the steam is superheated from 100 to 150 deg. fahr.

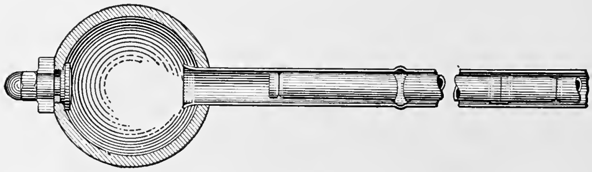


FIG. 112. Section Through Superheater Header and Tubes showing Method of Holding Core in Place: Babcock & Wilcox Superheater as Applied to Stirling Boiler.

When the boiler construction permits of only one inlet and one outlet connection to the superheater the Babcock and Wilcox superheater is modified by using one set of "U" tubes fitted with cores. Such a modified type is used in connection with the Stirling boiler. The cores are made of No. 13 B.W.G. tubes plugged at one end and inserted in the straight portion of the 2-inch superheater tubes, thereby causing the steam to flow through the annular space. Fig. 112 shows a cross section through an element of the superheater header and tubes, illustrating the method of holding the core in place.

Fig. 113 shows the application of a Foster superheater to a Babcock and Wilcox boiler. This device consists of cast-iron headers joined by a bank of straight parallel seamless drawn-steel tubes, each tube being encased in a series of annular flanges placed close to each other and

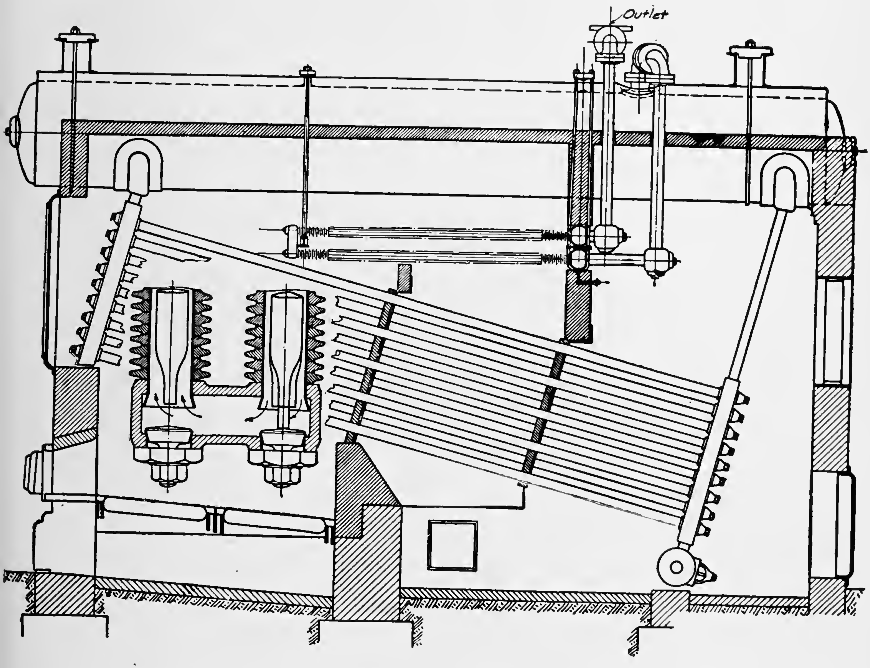


Fig. 113. Foster Superheater in Babcock and Wilcox Boiler.

forming an external cast-iron covering of large surface. The protection afforded by this external covering is ample to prevent damage from overheating during the process of steam raising, and flooding devices are unnecessary. The tubes are double, the inner tube serving to form a thin annular space through which the steam passes as indicated. Caps are provided at the end of each element for inspection and cleaning purposes. Foster superheaters are more costly than plain-tube superheaters, but are longer lived and offer a much larger heating surface in proportion to the space occupied.

The "Schwoerer" superheater, which is somewhat similar in external

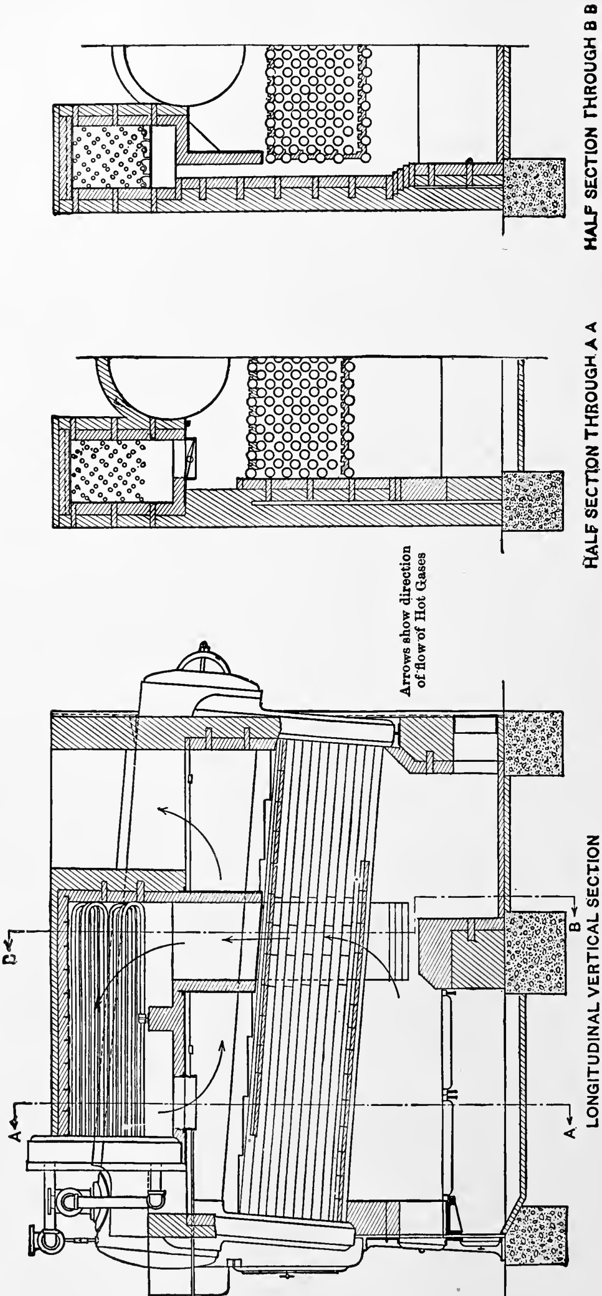


Fig. 114. Heine Superheater.

appearance to the Foster, differs from it considerably in detail, the heating surface being made up of suitable lengths of cast-iron pipe ribbed outside circumferentially and inside longitudinally. The ends of the pipes are flanged and connected by cast-iron U-bends. The intention is to provide ample heating surface internally and externally, with a compact apparatus.

Fig. 114 shows the application of a Heine superheater to a Heine boiler, illustrating the installation of a superheater within the boiler setting but entirely separated from the main gas passages. The superheater consists essentially of a number of $1\frac{1}{2}$-inch seamless steel tubes, bent to U-shape and expanded into a header box of the same type of construction as the standard Heine boiler water leg. The interior of this box is divided into three compartments by light sheet-iron diaphragms, so as to deflect the current of steam through the tubes. The superheater chamber is located above the steam drum as indicated. The gases of combustion are led to the superheater chamber through a small flue built in the side walls of the setting. A damper placed at the outlet of the flue controls the flow of gases and regulates the degree of superheat. No provision is necessary for flooding the superheating coils since the gases may be entirely diverted from the heating surface. Soot accumulations are readily removed by introducing a soot blower through the hollow stay bolts.

The Schmidt independently-fired superheater, Fig. 115, consists of two nests of coils, *A* and *D*, of equal size and dimensions, connected to cast-iron headers *O* and *I*. Saturated steam enters the first nest of coils through *C* and passes into header *O*. From *O* the steam, which is now dried, and partly superheated, flows through the cast-iron pipe *E* to header *I*, and thence through the second nest of coils into header adjoining *O*, and through pipe *R* to the engine. In chamber *D* the steam and gases flow on the counter-current and in chamber *A* on the concurrent principle. This combination permits of a low flue temperature and high steam temperature without subjecting the tubes to an excess of heat as would be the case if the steam left the coils *A* at header *I*, where the furnace gases are the hottest. A steam temperature of 750 deg. fahr. and a flue temperature of 450 deg. fahr. are easily maintained with this apparatus. A mercury pyrometer *T* is fitted where the superheated steam enters the discharge pipe *R*. A thermometer cup *L* permits of checking the pyrometer by means of a nitrogen-filled thermometer. Each coil can be taken out separately and a new one put in without removing the others or dismantling the plant. Water produced by condensation while the superheater is inoperative collects in the bottom header *N* and escapes through a drain cock. If the steam

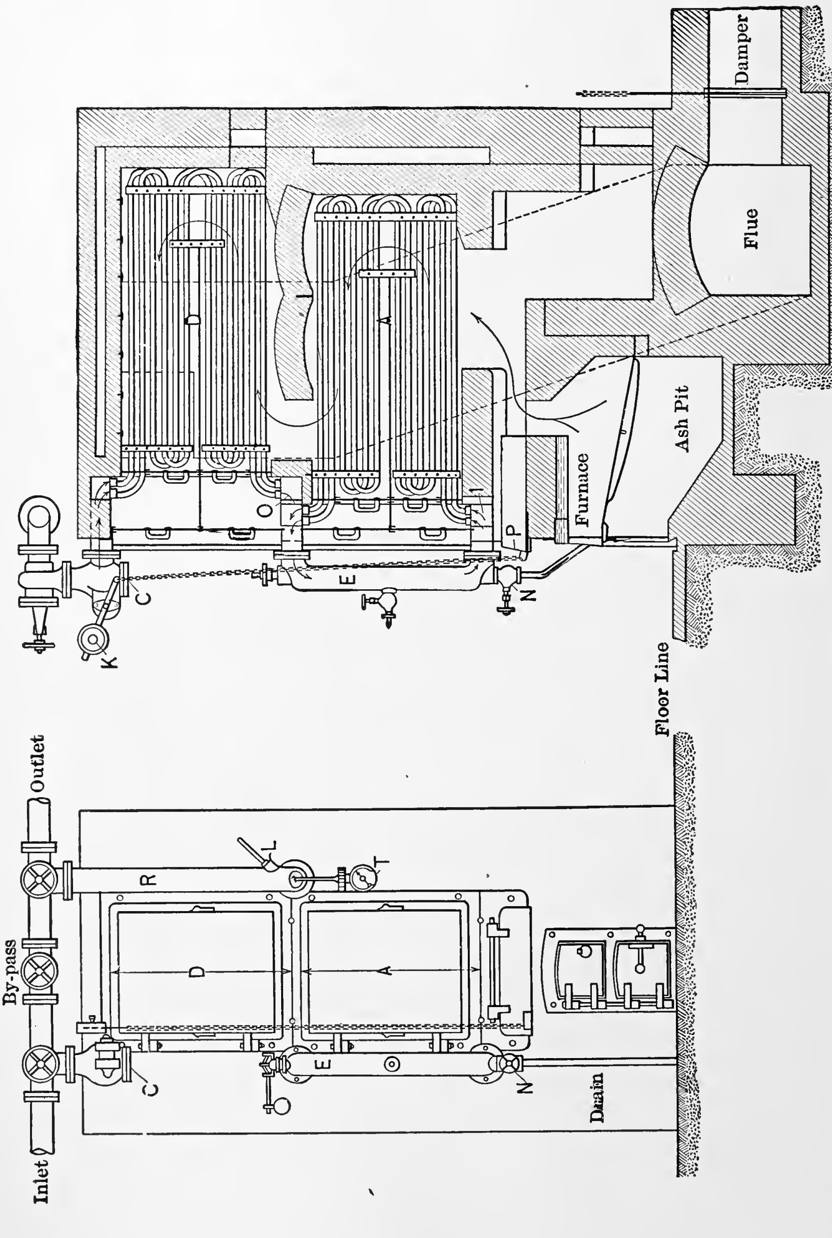
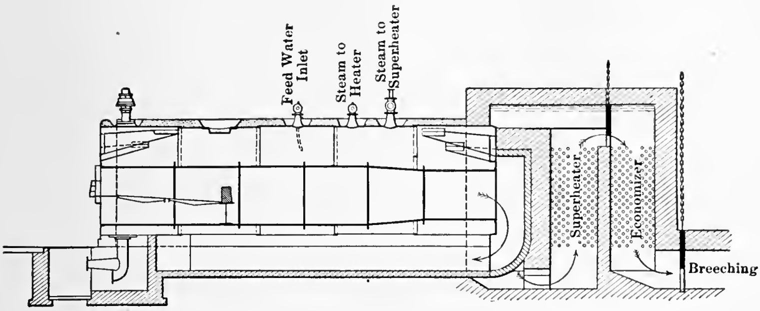


Fig. 115. Schmidt Independently-fired Superheater.



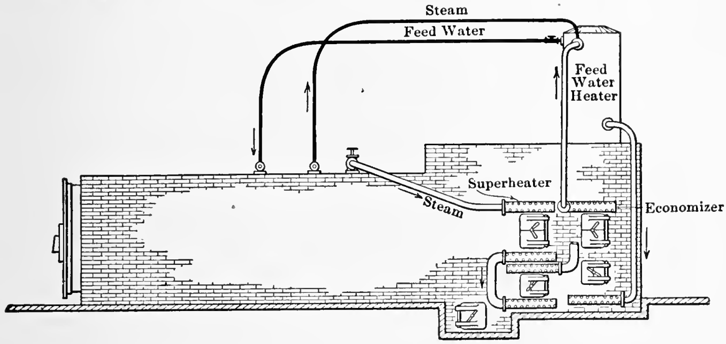


Fig. 116. Schmidt System of Combined Superheater, Feed-water Heater and Economizer.

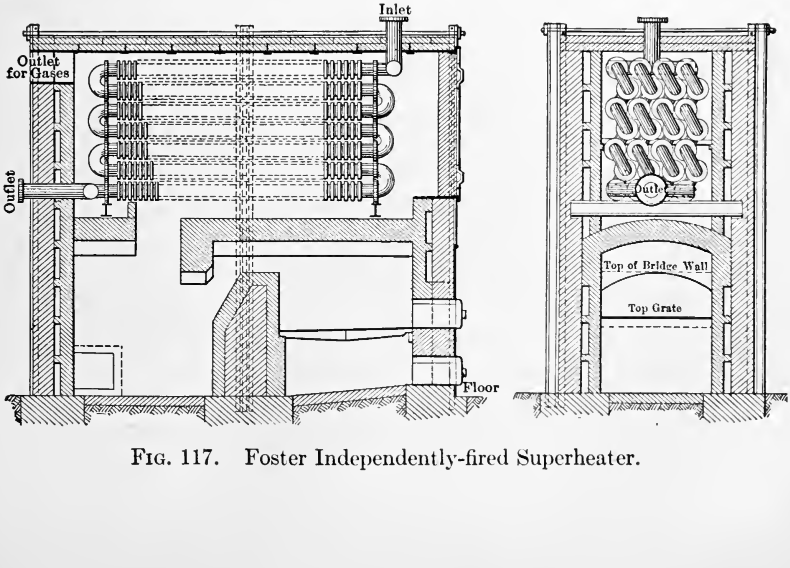


Fig. 117. Foster Independently-fired Superheater.

supply should be suddenly shut off, the air door P is opened automatically by weight K. As soon as steam begins to flow it raises the weight through the opening of valve C and the door closes. The Schmidt superheater when arranged in the flue has practically the same construction as the independently fired.

Modern Superheater and its Performance: Ry. Age Gaz., June 30, 1916.

111. Materials Used in Construction of Superheaters. — Most superheaters are constructed either of wrought steel, cast iron, or cast steel, the latter material having the advantage of not being damaged by any temperature to which it is likely to be subjected, which does away with the necessity of damper mechanisms and simplifies the installation. Cast-metal superheaters are usually ribbed after the fashion of an air-cooled gas engine, and are, therefore, very heavy and thick walled, necessitating a higher temperature for the same useful effect than in the case of the wrought-iron construction, but have the advantage of minimizing fluctuation of steam temperature which would otherwise be caused by a wide variation in temperature of furnace. One of the most successful cast-metal heaters is of European design and is constructed of a special alloy known as "Schwoerer" iron. Table 42 gives the yearly cost of repairs to piping and necessary brickwork for a number of installations equipped with cast-metal superheaters of the "Schwoerer" type.

Wrought steel offers the advantage of lightness, ease of construction, and low first cost, but cannot be exposed to very high temperatures without injury, and consequently provision must be made for diverting the direction of the heated gases or for flooding the coils while the boiler is being warmed before steam is generated.

The effect of temperature on superheater materials is shown in Fig. 118. It will be seen that the tensile strength drops off very rapidly for temperatures beyond 600 deg. fahr. Because of this rapid decrease in tensile strength of materials with the increase in temperature, steam is seldom superheated to temperatures above 850 deg. fahr.

For further information pertaining to the effect of temperature on various metals, consult "The Effect of High Temperatures on the Physical Properties of Some Metals and Alloys"; The Valve World, Jan., 1913, published by the Crane Co., Chicago.

Ordinary cast-iron valves and fittings have shown permanent increase in dimensions under high superheat and in numerous instances have failed altogether, but sufficient data are not available to prove conclusively the unreliability of cast iron if the iron mixture is properly compounded and the necessary provision is made for expansion and contraction. Authorities are of the opinion that the failure of cast-iron

TABLE 42.

AVERAGE YEARLY EXPENSES FOR REPAIRS FOR CAST-IRON SUPERHEATERS ("SCHWOÈRER" TYPE) AS OBTAINED FROM SIX PLACES OF INSTALLATION.

(Otto Berner, *Power*, August, 1904.)

Number.	Length of Time of Installation. Years.	Average Daily Use. Hours.	Place of Installation of Superheater with Reference to Boiler.	Average Temp. of Gases immediately in Front of the Superheater Surface. Degrees F.	Average Steam Pressure. Pounds per Square Inch (Absolute).	Average Temp. of Steam. Degrees F.	No. of Elements for One Superheater.	Length of One Element in Feet.	Average Yearly Cost of Repairs. 1 Superheater.	Average Yearly Cost of Repairs. For 1 Foot of Element.
1	8	11	Directly heated	....	99.56	518°	12	10.5	$71.40	$0.61
2	5	12	Directly behind the fire bridge	....	106.672	....	4	9.843	$14.09	$0.37
3	10	11	a. Directly behind the fire bridge .... b. Behind the first flue .	.... 716°–842°	78.232 113.79	.... 442.4°	2 6	8.202 9.843	11.90 0	$0.08 0
4	11	12	Behind the first flue	....	85.34	392°–572°	6	9.843	$8.33	$0.15
5	6	24	Behind the first flue (fire-tube boiler)	....	99.56 & 156.46	482°	8 to 10	6.562	$5.95	$0.08
6	7	11	Directly behind the fire bridge	....	170.68	500°–572°	....	....	0	0
Average Results	7.7	13.5	....	....	113.79	493°	6.9	5.25	$15.95	$0.28

fittings is due more to fluctuations in temperature than to the actual high temperature itself and cite numerous cases where ordinary cast-iron fittings under uniform temperature conditions are giving satisfaction with highly superheated steam. Notwithstanding the claims

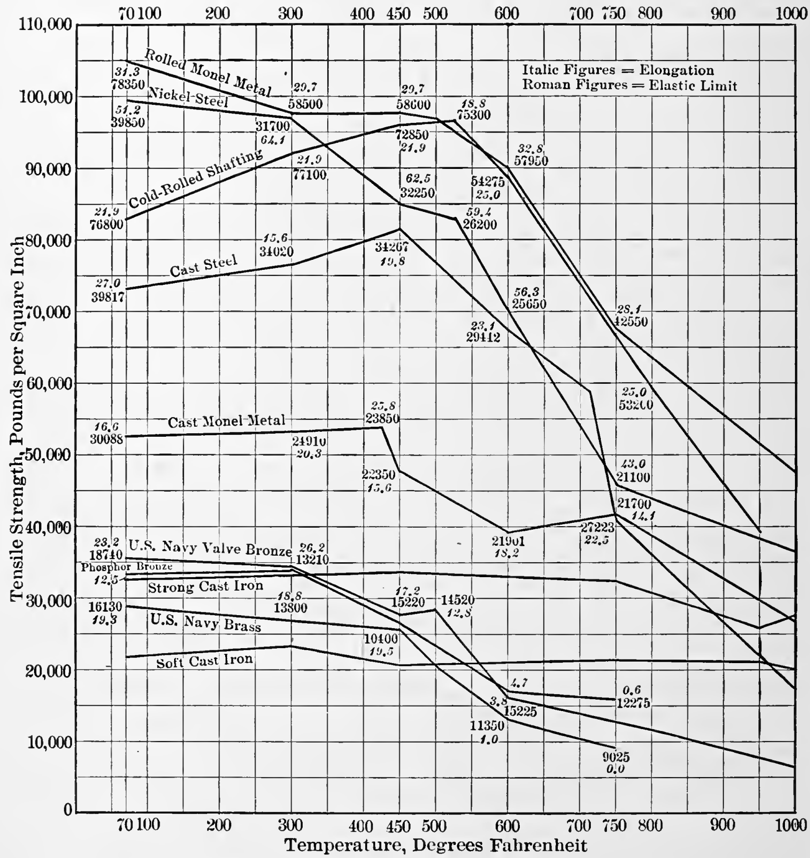


FIG. 118. Effect of Temperature on Strength of Materials.

that cast iron properly compounded is a perfectly reliable metal for fittings, engineers are inclined to use cast or forged steel, at least in this country. See "Effect of Superheated Steam on Cast Iron and Steel," Trans. A.S.M.E., Vol. 31, 1909, p. 989.

112. Extent of Superheating Surface. — The required extent of superheating surface for any proposed installation depends upon: (1) the degree of superheat to be maintained; (2) the velocity of the steam and gases through the superheater; (3) the character of the superheater; (4) the weight of steam to be superheated; (5) the moisture in the wet steam; (6) the temperature of the gases entering and leaving the superheater; (7) the conductivity of the material, and (8) cleanliness of the tubes.

Since the heat absorbed by the steam in the superheater is equal to that given up by the products of combustion, neglecting radiation, this relationship may be expressed

$$SUd = Wc\,(t_1 - t_2), \tag{53}$$

in which

S = square feet of superheating surface per boiler horsepower,
U = mean coefficient of heat transmission, B.t.u. per hour per degree difference in temperature,
d = mean temperature difference between the steam and heated gases, deg. fahr.,
W = weight of gases passing through the superheater per boiler horsepower-hour,
c = mean specific heat of the gases,
t_1 = temperature of the gases entering superheater, deg. fahr.,
t_2 = temperature of the gases leaving superheater, deg. fahr.

Transposing equation (53),

$$S = \frac{Wc\,(t_1 - t_2)}{Ud}. \tag{54}$$

The heat transfer from the products of combustion to the steam may also be expressed

$$SUd = wc'\,(t_s - t), \tag{55}$$

in which

w = weight of steam passing through the superheater, pounds per boiler horsepower-hour,
c' = mean specific heat of the superheated steam,
t_s = temperature of the superheated steam, deg. fahr.,
t = temperature of the saturated steam, deg. fahr.,
S, U, and d as in equation (53).

For wrought-iron or mild steel tubes U varies as follows:

U = 1 to 3 for superheaters located at the end of the heating surface
= 3 to 5 for superheaters located between the first and second pass of water tube boilers
= 8 to 12 for superheaters located immediately above the furnace in stationary boilers, in the smoke box of locomotive boilers, and in separately fired furnaces.

General practice allows $\frac{1}{4}$ to $\frac{1}{3}$ square foot of superheating surface per boiler horsepower for mild steel superheaters located in the furnace; from 2 to 2.5 square feet of surface at the end of the first pass, and from

3 to 4 square feet at the end of the heating surface for superheats of from 100 to 150 deg. fahr., boiler pressure 150 pounds absolute.

The Foster Superheater Company allows 6 B.t.u. per lineal foot per degree difference in temperature for their "two-inch" element where the average temperature of the gases is about twice the mean temperature of the steam.

For all engineering purposes d may be determined with sufficient accuracy from the relationship

$$d = \frac{t_1 + t_2}{2} - \frac{t_s + t}{2}.*$$

Notations as in equations (53) and (55).

An empirical formula for determining the extent of superheating surface in connection with indirect superheaters which appears to conform with practice is derived by substituting

$$U = 3, \quad d = t' - \frac{t_s + t}{2}, \quad w = 30, \quad c' = 0.5,$$

in equation (55) [J. E. Bell, Trans. A.S.M.E., 29–267]. Thus:

$$S \times 3\left(t' - \frac{t_s + t}{2}\right) = 30 \times 0.5 \times (t_s - t),$$

from which

$$S = \frac{10\,(t_s - t)}{2\,t' - t_s - t}; \tag{56}$$

t' (the mean temperature of the product of combustion where the superheater is located) may be approximated from equation

$$\frac{1}{(t' - t)^{0.16}} = 0.172\,H + 0.294, \tag{57}$$

in which

H = the per cent of boiler-heating surface between the point at which the temperature is t and the furnace,

t as in (56).

Equation (57) is based upon the assumption that the heat transferred from the gases to the water is directly proportional to the difference in temperature; that the furnace temperature is 2500 deg. fahr.; flue temperature 500 deg. fahr.; steam pressure 175 pounds per square inch gauge; one boiler horsepower is equivalent to 10 square feet of water-heating surface; and that there is no heat absorbed by direct radiation.

Example 15. What extent of heating surface is necessary to superheat saturated steam at 175 pounds gauge pressure, 200 deg. fahr., if the

* See also paragraph 286.

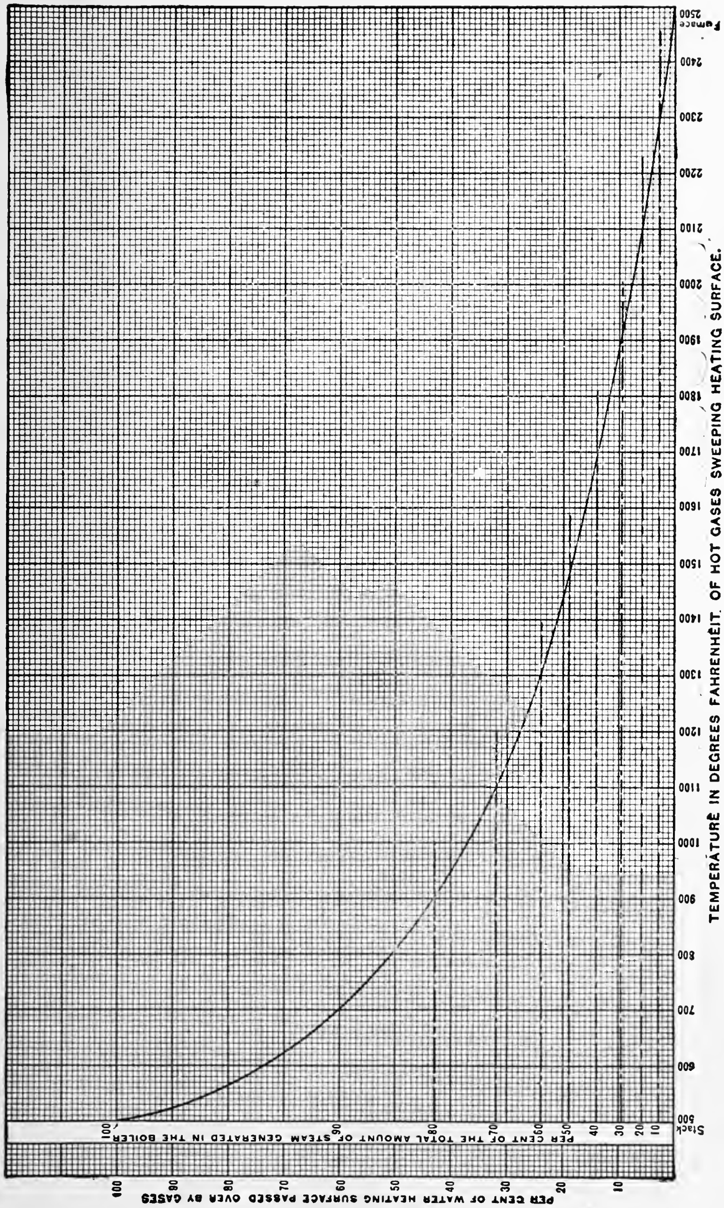


FIG. 119. Curve Showing Relation between Gas Temperature and Extent of Heating Surface Passed over.

superheater is placed in the boiler setting where the gases have already traversed 40 per cent of the water-heating surface?

Substitute $H = 0.4$ and $t = 378$ in equation (57),

$$\frac{1}{(t' - 378)^{0.16}} = 0.172 \times 0.4 + 0.294,$$

from which

$$t' = 950.$$

Substitute $t' = 950$, $t_s = 578$, and $t = 378$ in equation (56),

$$S = \frac{10\,(578 - 378)}{2 \times 950 - 578 - 378}$$
$$= 2.12 \text{ square feet.}$$

The curve in Fig. 119 was plotted from equation (57) and gives a ready means of determining t' and of observing the law governing heat absorption by the boiler between furnace and breeching. The abscissas represent the temperatures of the hot gases at different points in their path between furnace and breeching. The ordinates represent (1) the per cent of boiler-heating surface passed over by the hot gases, and (2) the per cent of the total heat generated which is absorbed by this heating surface.

In the use of equation (57) the probability of error is greatest when considering a point near the furnace, since large quantities of heat are transmitted to the tubes by radiation from the fuel bed which are not taken account of. For most practical purposes the assumption is sufficiently accurate.

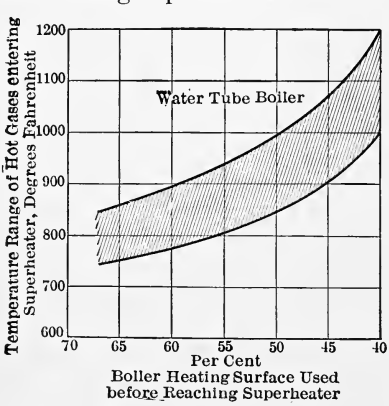


FIG. 120. Temperature Range of Gases in Superheater.

Fig. 120 gives the probable temperature range of gases entering superheater after passing over a given per cent of boiler-heating surface and Fig. 121 shows the relation between superheating surfaces and boiler heating surface. (See Power, Nov. 7, 1911, p. 696.)

It will be found that the boiler-heating surface per boiler horsepower will be decreased in almost the same proportion that the superheating surface is increased, so that the sum of the boiler-heating surface and superheating surface per boiler horsepower will be very nearly the same for any degree of superheat.

113. Performance of Superheaters. — Published tests of both directly and indirectly fired superheaters cover such a wide range of conditions

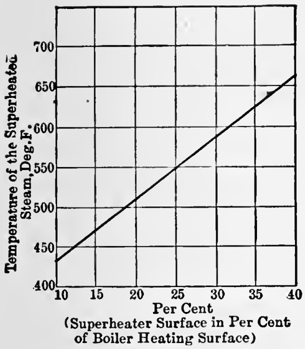


FIG. 121. Relation between Superheat and Boiler Heating Surface.

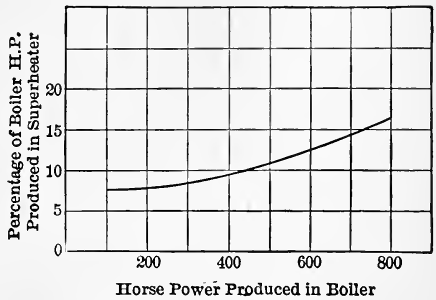


FIG. 122. Ratio of Horsepower Produced in the Superheater to that Developed in the Boiler.

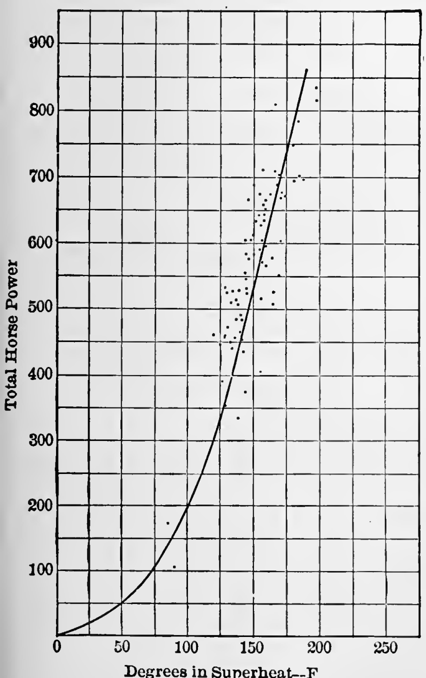


FIG. 123. Relation of Degree of Superheat to Total Horsepower Developed.

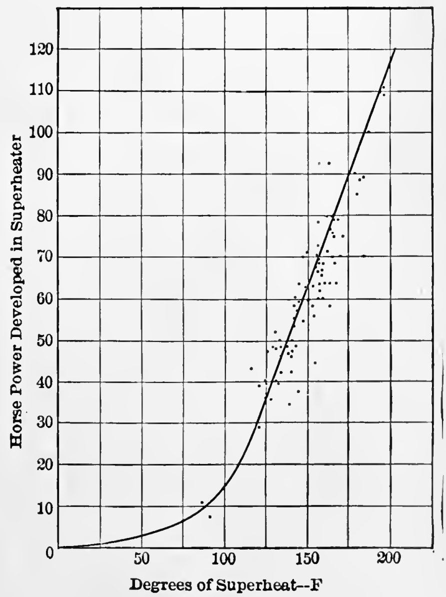


FIG. 124. Relation of Degree of Superheat to Horsepower of Superheater.

of installation and operation that general conclusions cannot be drawn, but it may be of interest to note briefly the performances in a few specific cases.

The curves in Figs. 122, 123, and 124 are plotted from tests of a Babcock and Wilcox boiler, with 5000 square feet of water-heating surface, equipped with superheating coils of 1000 square feet area, as illustrated in Fig. 93. The furnace with ordinary short ignition arch was provided with chain grate of 75 square feet area.

Fig. 122 shows the relation between degrees of superheating and total horsepower of boiler and superheater.

Fig. 123 shows the relation between horsepower produced in the boiler and the percentage of boiler horsepower produced in the superheater.

Fig. 124 shows the relation between the degree of superheat obtained and the horsepower developed in the superheater.

Tables 42 to 45 are taken from the report of Otto Berner ("Zeit. d. Ver. Deut. Eng." and reprinted in Power, August, 1904).

Table 42 compares the heat efficiency of a steam plant equipped with indirectly and with separately fired superheaters, the former showing a much higher efficiency.

Table 43 compares different boilers with and without flue superheaters, showing the effect upon the temperature of the flue gases. The gain in heat efficiency of the entire plant due to the use of the superheater is decisive in each case.

Table 45 shows the gain in heat efficiency due to the use of superheaters in a number of plants equipped with fire-tube boilers.

Table 46 gives the results of tests on one of the return tubular boilers at the Spring Creek Pumping Station of the Brooklyn Waterworks (Feb. 9, 1904) with and without a superheater. The superheater, of the Foster type, was installed between the rear wall of the setting and the tube sheet.

Although the results in Tables 42 to 46 represent practice of ten years ago, they agree substantially with current practice (1916).

TABLE 43.

DIFFERENCE IN HEAT EFFICIENCY OF SUPERHEATERS INSTALLED IN FLUE AND SEPARATELY FIRED SUPERHEATER.
(Otto Berner, *Power*, August, 1904.)

Style of Superheater	Separately Heated Superheater. (System Uhler.)		Boiler-Flue Superheater. (Patent Schwoerer.)	
Number of test	1	2	3	4
Heating surface of superheater ... Square feet	1,636.17	1,636.17	2,744.90	1,829.93
Number of boilers used in test	3	2	3	2
Boiler-heating surface ... Square feet	4,843.94	3,229.29	4,843.94	3,229.29
Grate area ... Square feet	104.28	69.52	104.28	69.52
Feed water per square foot of boiler-heating surface. Pounds per hour	1.96	2.75	2.04	2.8
Coal consumed per square foot of grate area ... Pounds per hour	9.31	14.05	9.96	14.48
Boiler pressure, absolute ... Pounds per square inch	177.48	175.64	179.91	176.35
Feed-water temperature ... Degrees F.	64.76	62.6	50.00	50.00
Steam temperature on leaving superheater ... Degrees F.	483.8	508.46	512.6	550.4
Heat generated per pound of coal: Boiler ... B.T.U.	11,553.39	11,187.46	....	....
Heat generated per pound of coal: Superheater ... B.T.U.	4,876.81	5,544.76	....	....
Heat generated per pound of coal: Entire plant ... B.T.U.	10,880.53	10,605.30	12,392.33	12,229.88
Heat efficiency when using separately fired superheater less than when using boiler only ... Per cent	5.70	5.33	....	....
Heat efficiency when using superheater installed in flues greater than when using boiler only ... Per cent	....	....	6.87	8.75
Heat efficiency when using flue superheater greater than when using separately fired superheater ... Per cent	....	....	**13.88**	**15.72**

TABLE 44.

DECREASE IN TEMPERATURE OF GASES OF COMBUSTION DUE TO SUPERHEATER INSTALLED IN FLUE.

(Otto Berner, *Power*, August, 1904).

Test	1		2		3		4		5		6	
Style of boiler	Water-Tube Boilers.											
Boiler-heating surface . . . Square feet	3,584.51		2,884.83		2,529.61		1,955.86		3,089.35		936.49	
Superheater surface . . . Square feet	....		344.45		645.85		Schwoerer Patent		1,054.9		182.99	
Grate area . . . Square feet	81.8		51.02		55.97		45.21		64.58		26.47	
Running of boiler	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.	Without Superheating.	With Superheating.
Feed-water consumption per square foot of heating surface . . . Pounds per hour	2.46	2.23	2.68	2.66	2.91	2.69	2.31	2.10	2.47	2.03	3.70	3.50
Coal consumption per square foot of grate area . . . Pounds per hour	14.23	13.55	19.57	19.74	27.84	26.00	15.7	14.02	14.54	11.79	26.32	18.51
Boiler pressure (absolute)	130.8	131.51	149.6	148.38	182.68	181.75	122.99	121.73	149.33	150.75	149.33	151.31
Feed-water temperature . . . Degrees F.	107.6	104.72	43.97	45.68	89.9	89.0	53.24	55.4	101.3	109.04	32	67.1
Temperature of gases of combustion . . Degrees F.	554	462.2	500	465.8	654.98	597.2	492.8	442.4	590	471.2	793.4	550.4
Steam temp. on leaving superheater . . Degrees F.	......	424.94	......	486.5	......	562.6	......	489.7	......	536	......	486.5
Efficiency of boiler plant . . . Per cent	......	......	......	......	55.9	59.1	55.63	60.18	71.9	78.6	49.3	62.5
Increase in heat efficiency of plant due to use of superheated steam . . . Per cent	......	2.3	......	3.3	......	5.4	......	7.6	......	8.5	......	21.2
Decrease in grate area requirements due to use of superheated steam . . . Per cent	......	4.8	......	0.8	......	6.6	......	10.7	......	18.9	......	23.7
Decrease in heating surface requirements due to use of superheated steam . . . Per cent	......	9.4	......	0.9	......	9.3	......	8.8	......	17.4	......	5.5

TABLE 45.

INCREASE IN HEAT EFFICIENCY OF THE BOILER DUE TO SUPERHEATER.

(Otto Berner, *Power*, August, 1904.)

	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater.	Boiler with Superheater.	Boiler without Superheater,	Boiler with Superheater.
Test	1	2	3	4	5	6	7	8	9	10
Boiler pressure (absolute) .... Lb. per sq. in.	77.21	77.21	71.25	71.25	72.53	72.53	72.24	72.24	86.18	86.18
Feed-water temperature, Deg. F	145.4	150.8	145.4	150.8	161.6	154.4	149.0	142.6	161.6	156.2
Steam temperature.....Deg. F.	....	361.4	....	363.2	....	372.2	....	366.8	....	372.2
Evaporation per square foot of heating surface. Lb. per hour	8.56	9.12	7.3	8.1	9.01	9.95	7.23	8.56	7.7	9.09
Coal consumption per square foot of grate area ......Lb. per hour	31.78	30.46	29.08	29.28	35.07	35.87	30.1	30.1	30.76	30.76
Increase in heat efficiency due to use of superheater..........Per cent	....	12.4	....	11.8	....	11.6	....	18.1	....	18.1

TABLE 46.

(*Engineer, U. S.*, May 1, 1904.)

	With Superheater.	Without Superheater
Time of start	12 noon, Feb. 8	11 A.M., Feb. 11
Time of finish	12 noon, Feb. 9	11 A.M., Feb. 12
Hours run	24	24
Average steam pressure	79.3 lb.	79.4 lb.
Average water pressure, triple expansion, head in feet	0.99	1.05
Average water pressure, compound, head in feet	7.10	7.10
Average vacuum of suction for triple and compound, inches of mercury	22.90	23.21
Total head on triple, feet of water	29.05	29.46
Total head on compound, feet of water	33.04	33.39
Total double strokes, triple	30,557	34,114
Total double strokes, compound	35,395	32,158
Gallons pumped from piston displacement, total, triple	2,854,023	3,186,247
Gallons pumped from piston displacement, total, compound	2,930,706	2,662,682
Gallons pumped from piston displacement, total, triple combined	5,784,720	5,848,930
Gallons, total, pumped as measured by weir	4,492,680	4,549,480
Per cent slip	22.3	22.2
Foot pounds, weir	1,163,815,819	1,184,983,596
Total coal consumed	5,015 lb.	6,410 lb.
Per cent refuse	23.7	18.7
Total refuse	1,188	1,203
Total feed water	38,399	50,960
Duty per 100 pounds coal	23,206,696	18,486,483
Duty per 1,000 pounds steam	30,308,498	23,253,213

Per cent increase of work per 100 pounds coal	25.5
Per cent increase of work per 1000 pounds steam	30.2
Per cent saving in coal per foot pound work	20.2
Per cent saving in feed water per foot pound work	23.2
Average temperature steam leaving superheater	527.4 deg. fahr.
Average temperature steam entering superheater	320.1 deg. fahr.
Average degree superheat	207.3 deg. fahr.

PROBLEMS.

1. Required the sq. ft. of superheating surface necessary to superheat 10,000 lb. of saturated steam per hour at 200 lb. abs., to 250 deg. fahr. if the superheater is placed in the boiler setting where the gases have already traversed 35 per cent of the water-heating surface.

2. Required the mean temperature of the products of combustion passing through a superheater of 3815 sq. ft. of heating surface if 66,000 lb. of steam are superheated from saturation at 265 lb. abs. to 250 deg. fahr.; lb. gas per lb. of steam, 2.0; mean coefficient of heat transfer, 5.

3. If the furnace gases enter a superheater at a temperature of 1200 deg. fahr. and leave at 900 deg. fahr., required the weight of steam superheated from saturation at 265 lb. abs. to 250 deg. fahr.; lb. gas per lb. of steam 2.0. Neglect all losses.

CHAPTER VI

COAL AND ASH-HANDLING APPARATUS

114. General. — The cost of coal and its delivery into the furnace are usually the largest items in the operating charges; hence large central stations are located, when practicable, adjacent to a railway line or water front, to minimize the cost of handling coal and ashes. Isolated stations in the business districts of large cities are usually unfavorably situated, so that the cost of handling coal and ashes is a large percentage of the total fuel cost. In large stations the amount of fuel and ash handled frequently warrants the expense of elaborate conveyor systems which would not be justified in smaller plants. In whatever way coal is supplied provision should be made for storing a quantity sufficient to operate the plant for some time in case the supply is interrupted, thereby guarding against an enforced shut-down.

If adjacent to a railway line, a side track must be provided for switching the cars. As the furnishing of bottom-dumping cars cannot be depended upon, provision should be made for unloading by hand or by grab bucket. If coal is delivered by water, clam-shell drop buckets are ordinarily used for unloading the barges. If the power house is located at some distance from the railroad or water the coal is generally hauled by teams or motor trucks in two- to five-ton loads.

115. Coal Storage. — In small stations the storage bins or coal bunkers may usually be located within the building, but in larger plants the quantity of coal consumed daily is frequently such that an immense space would be required to furnish storage capacity for even a short period of time. For example, the requirements of the Northwest Station of the Commonwealth Edison Company are approximately 3000 tons of Illinois coal per day of 24 hours. One day's supply would occupy a space 100 feet square and 12 feet high. Seventy-five railway cars per day would be required to supply this amount of coal and in addition about ten cars of ashes would have to be removed. The futility of storing the coal in cars is evidenced by the fact that about two and one-half miles of track would be necessary to carry only a four days' supply. In this particular plant there is yard space for storing 300,000 tons in two piles, or sufficient to run the plant for three months.

Exposed coal piles are objectionable, because of freezing in winter, the crust sometimes freezing so hard as to necessitate the use of dynamite to break it; moreover, a slow depreciation in heat value takes place, especially with bituminous coal. This depreciation is more rapid in warm weather and in the tropics. Stored coal is oftentimes subject to spontaneous combustion, particularly when there is a large content of iron pyrites. Storage under water minimizes spontaneous combustion and depreciation in heat value. (Consult references below.)

Coal bunkers or hoppers are ordinarily placed on the same level with the boiler-room floor or above the boiler setting. The former is the cheaper as far as the first cost is concerned, but necessitates additional handling of the fuel before it can be fed to the stokers. In the overhead system the coal gravitates to the stoker through down spouts. Overhead bunkers are usually found where real estate is costly. They are generally constructed of steel plates lined with concrete or of reënforced concrete. The bottoms slope at an angle of 35 to 45 degrees and empty into the coal chutes or down spouts. Fig. 130 shows the general appearance of a steel plate overhead bunker and Fig. 136 that of the suspended type. In some bunkers the floors are made with very slight slopes, but it is not advisable to use a slope less than the angle of repose of the coal, as it may be necessary to shovel the coal over the spouts. Convenience in framing makes the 45-degree slope the more desirable. Separate bunkers for each boiler are preferred to continuous bunkers, since fire in the coal is more readily prevented from spreading. In the new power house of Swift & Co., Chicago, Ill., the bunkers are of circular cross section instead of rectangular, as is the usual practice. The capacity of the cylindrical hopper is considerably less than that of a rectangular hopper of the same width, but is much cheaper to construct.

Ash bins are invariably lined with concrete or brickwork, since the corrosive action of the ashes would soon destroy the bare iron, and are usually located as in Fig. 130 so that they may be discharged by gravity. The angle of repose of most ashes is approximately 40 degrees, but the 45-degree angle is preferred on account of convenience in construction.

Coal Storage Under Water: Elec. Wld., Oct. 7, 1911, p. 885; Eng. News, Dec. 24, 1908; El. Ry. Jour., June 24, 1916, p. 1191.

Calorific Value of Weathered Coals: Bulletin No. 17, Univ. of Ill., Aug. 26, 1907; Eng. News, Jan. 11, 1912, p. 64.

Spontaneous Combustion of Coal: Jl. Ind. and Chem. Eng., Mar., 1911.

Suspended Coal Bins: Power, Apr. 23, 1912, p. 602.

Concrete Coal Cylinders: Eng. News, Mar. 2, 1916, p. 420.

116. Coal Handling Methods. — The best method of delivering coal to the furnace and of removing refuse from the ash pit is the one which will effect the desired result at the lowest ultimate cost. That this problem does not offer a simple solution is evidenced by the almost endless combinations found in practice for the same operating conditions. The principal factors which influence the choice of system are size and location of plant and cost of fuel and labor. In public service plants continuity of operation may be of even greater importance than reduction of cost and extra investment may be considered advisable to offset the unreliable labor element. Of the various methods found in current practice the following are the more common:

1. Hand shoveling.
2. Wheelbarrow or hand car and shovel.
3. Continuous conveyors:
   Spiral or screw,
   Scraper or flights,
   Apron and buckets,
   Overlapping pivoted buckets,
   Endless belt.
4. Hoist and hand car.
5. Hoist and automatic cable car.
6. Hoist and trolley: telpherage.
7. Clam shell buckets.
8. Vacuum system.
9. Combinations of the above.

117. Hand Shoveling. — Where possible, the coal is dumped direct from the cars or wagons into bins located in front of the boilers. In such instances one man may handle the coal and ashes and attend to the water level of 200 horsepower of boilers equipped with common hand-fired furnaces. With hand-shaking and dumping grates 300 horsepower may be fired by one man and from 800 to 1000 horsepower with automatic stokers. This refers, of course, to average good coal not too high in ash nor productive of much clinker. Sometimes the coal cannot be stored in front of the boilers but must be hauled by wheelbarrow, cart, or rail car. For distances over 100 feet and quantities over 20 tons per day the cost of handling the coal in this way may justify the installation of an automatic conveyor system. Hand-fired furnaces and manual handling of coal and ashes are usually associated with small plants of 500 horsepower and under, but a number of large stations are operated in this way with apparent economy. A notable example is the steam power plant of the Wood Worsted Mill, Lawrence, Mass., in which 40 return tubular boilers are fired by hand. A tipcart with a capacity of one ton brings the coal a distance of 100 to 200 feet to the firing floor, and firemen shovel it on to the grate. Four men are stationed at the coal pile. One man drives two carts (one of which is being filled while the other is gone with its load), sixteen firemen attend to the furnaces, and two men dispose of the ashes.

Most large plants, however, are equipped with conveying machinery, not so much because of the possible reduction in cost of operation, taking into consideration all charges fixed and operating, as because of the large and often unreliable labor staff which it dispenses with. Hand shoveling is sometimes necessary even with modern unloading devices on account of the freezing of coal in the cars. This is particularly true of washed coals, and it is not unusual to have an entire car load solidly frozen. In this case it has to be picked and shoveled by hand, or the unloading tracks must be equipped with steam pipes and outfits for thawing purposes. A good man is capable of shoveling 40 to 50 tons of coal in eight hours when unloading a car, provided it is only necessary to shovel the coal overboard. For cost of handling material by wheelbarrow and hand shoveling see end of paragraph 123.

118. Continuous Conveyors and Elevators. — The most popular method of automatically handling coal and ashes in the modern power plant is by means of continuous conveyors and elevators. They may be divided into two general classes:

1. Those which push or pull the material, but in which the weight of the load is not borne by the moving parts.
2. Those which actually carry the load.

A few of the more important types will be treated briefly.

Screw or Spiral Conveyors. — These consist of a stamped or rolled sheet steel spiral secured by lugs to a hollow shaft (usually a standard or extra heavy pipe) revolving in a trough which it fits approximately. Standard sizes range from 3 to 18 inches in diameter and in sections from 8 to 12 feet long.

TABLE 47.

SPEEDS AND CAPACITIES OF SCREW CONVEYORS.

(Fine Coal and Ashes.)

Diam. screw, in.	6	7	8	9	10	12	14	16	18
Max. r.p.m.	115	110	105	100	95	90	85	80	75
Capacity per hr., fine coal, tons	....	7	14	16	21	36	48	80	110
Ashes, cu. ft.	125	175	350	425	550	950	1200	2000	2700

On account of the torsional strain on the shaft the maximum length seldom exceeds 100 feet. Single sections may be used as feeders on inclines up to 15 degrees. Low first cost, compactness and adaptability to space requirements are the advantages of this type but these may be offset by high power consumption and excessive wear. The following equation gives a means of approximating the power requirements for horizontal runs:

$$\text{Horsepower} = C\frac{WL}{33{,}000} \tag{58}$$

in which

C = 0.7 for coal and 1.0 for ashes,
W = capacity in lb. per minute,
L = length in feet.

Fig. 125 shows an application of a screw conveyor for handling coal as installed in a modern isolated station.

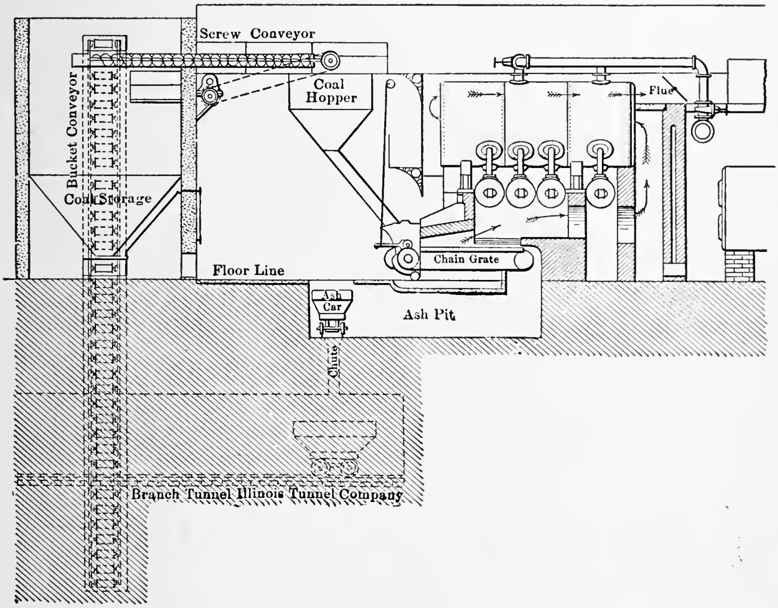


FIG. 125. Screw Conveyor as Installed in the Power Plant of a Tall Office Building.

Scraper or Flight Conveyor. This consists of a trough of any desired cross section and a single or double strand of chain carrying flights or scrapers of approximately the same shape as the trough. The flights scrape the material along the trough discharging at the end of trough gate controlled openings in the bottom of the conduit. Three types of flight conveyors are in common use; plain scraper, suspended flights and roller flight. In the *plain scraper* the flights are suspended from the chain and drag along the bottom of the trough. In the *suspended flight* conveyor the flights are attached to cross bars having wearing-shoes at each end and do not touch the trough at any point. The roller flight differs from the suspended type only in the substitution of rollers for the wearing-shoes. A typical installation of scraper and roller flight conveyors is illustrated in Fig. 126. The coal conveyor is a single strand roller flight, 80 feet in length between centers, driven by a 5-hp. electric motor. It has a capacity of 15 tons of buckwheat coal per hour. The ash conveyor is a single-strand drag-chain with 87 ft. centers on the horizontal run and 6 ft. between vertical centers. The

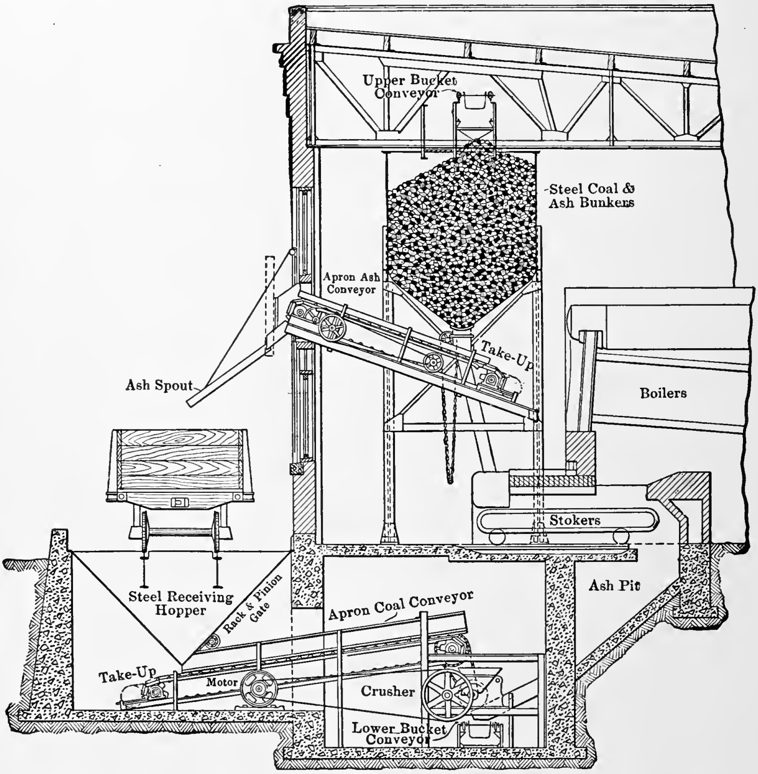


FIG. 126. Scraper and Drag Conveyor as Installed in a Power House of the Otis Elevator Company.

chain operates in an extra heavy cast-iron trough set in a cement trench and is operated by a 5-hp. motor. Flight conveyors are low priced and offer an economical and efficient means of handling coal and ashes in small plants.

Apron Conveyors are commonly used for conveying coal from track hopper to the main conveyor and elevator. The most elementary form consists of flat steel plates attached between two chains and forming a continuous platform or apron. Since the load is carried and not

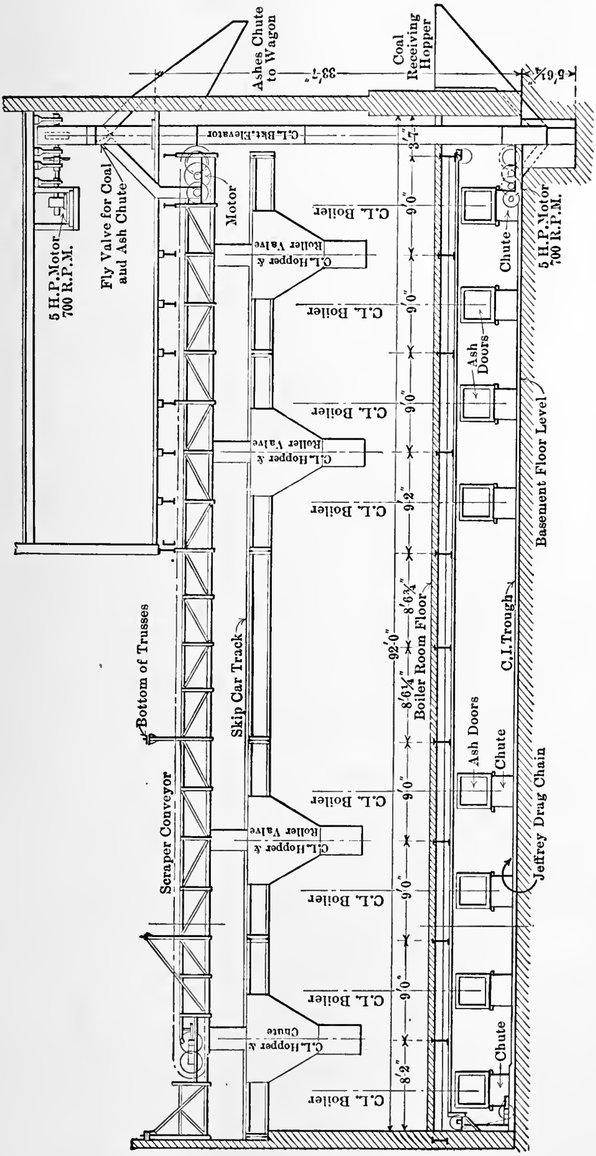


FIG. 127. Apron Conveyors as Installed in the Power Plant of the C. M. & St. P. R. R., Minneapolis, Minn.

dragged less power is required than with the scraper type and the maintenance is lower. These carriers are not suitable for elevating material except at an inclination not exceeding 30 degrees. End discharge only is possible. Fig. 127 shows a typical apron conveyor installation.

Pan Conveyors and Open Top Conveyors are similar to the apron carriers except that pans or buckets take the place of the flat or corrugated apron plates. These conveyors are used where pans deeper than those of an apron conveyor are required, as on inclines too flat for elevators

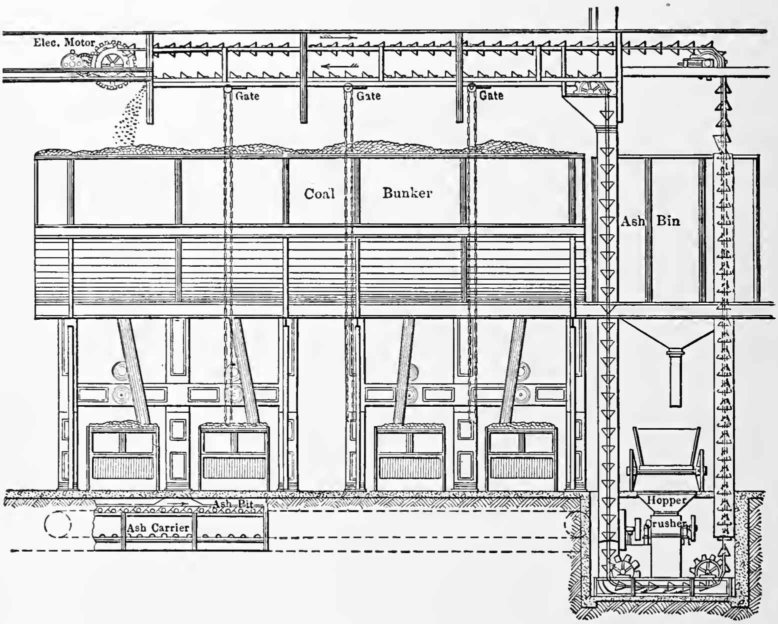


Fig. 128. Typical Installation of V-Bucket Conveyor for Handling Coal and Cast-iron Pan Conveyor for Handling Ashes.

and too steep for efficient operation of flight or apron conveyors. Conveyors of this type are usually run at speeds of 30 ft. to 50 ft. per minute and when equipped with self-oiling rollers of 6-inch to 8-inch diameter demand but little power for their operation above theoretical load requirements. Fig. 128 shows an installation of a cast-iron pan conveyor for handling ashes.

The power required to operate flight, apron and open top conveyors may be closely approximated by the following empirical equation.*

$$\text{Hp.} = \frac{AWLS}{1000} + \frac{BLT}{1000} + x \tag{59}$$

* C. K. Baldwin, The Robins Conveying Belt Co.

in which

Hp. = the horsepower required at the conveyor drive shaft,
A, B = constants as in Table 48,
W = weight of conveyor per ft. of run, lb.,
L = distance between centers of head and tail sprockets, ft.,
S = speed of conveyor, ft. per min.,
T = capacity of conveyor, tons (2000 lb.) per hour.
x = 1 for conveyors up to 100 ft. centers and 2 for longer conveyors.

If the conveyor is composed of portions on different inclines compute the power for each section separately and add 10 per cent for each change in direction.

The *V-Bucket Conveyor* consists of a series of V-shaped buckets rigidly fastened to the conveyor chain. The buckets act essentially as a drag conveyor on horizontal runs, each bucket pushing its half-spilled load ahead of it through a suitable trough. On vertical runs they act as elevators. A typical V-bucket conveyor for handling coal and a pan conveyor for handling ashes are illustrated in Fig. 128. The power requirements may be approximated from the following empirical equations:

$$\text{Hp.} = \frac{AWL'S}{1000} + \frac{BL_1T}{1000} + \frac{TH}{1000} + \tfrac{1}{2}x', \tag{60}$$

in which

L' = horizontal length of conveyor, ft.,
L_1 = total horizontal length traversed by the loaded bucket, ft.,
H = total vertical traverse, ft.,
x' = number of 90-degree turns in the conveyor.

Other notations as in equation (59).

TABLE 48.

VALUES OF CONSTANTS IN CHAIN CONVEYOR POWER FORMULAS.

Angle of Conveyor with Horizontal Deg.	*A.*				*B.* Scraper, Apron and Open Top.			*B.* V-Bucket and Pivoted Bucket.		
	Sliding Block.	3½-in. Roller, 3¼-in. Pin.	6-in. Roller, 1⅛-in. Pin.	6-in. Roller, 1½-in. Pin.	Anthracite Coal.	Bituminous Coal.	Ashes.	3½-in. Roller, ¾-in. Pin.	6-in. Roller, 1⅜-in. Pin.	6-in. Roller, 1½-in. Pin.
0	0.030	0.0043	0.0046	0.0050	0.33	0.60	0.54	0.071	0.076	0.083
6	0.030	0.0043	0.0046	0.0050	0.43	0.69	0.63	0.17	0.18	0.19
12	0.030	0.0042	0.0045	0.0049	0.54	0.79	0.73	0.28	0.28	0.29
18	0.029	0.0041	0.0044	0.0048	0.63	0.88	0.82	0.38	0.38	0.39
24	0.028	0.0039	0.0042	0.0046	0.72	0.95	0.90	0.48	0.48	0.49
30	0.026	0.0037	0.0040	0.0043	0.79	1.02	0.97	0.57	0.57	0.58
36	0.025	0.0035	0.0037	0.0040	0.86	1.08	1.03	0.65	0.66	0.66
42	0.023	0.0032	0.0034	0.0037	0.92	1.12	1.07	0.73	0.73	0.74
48	0.020	0.0029	0.0031	0.0033	0.97	1.15	1.11	0.80	0.80	0.81

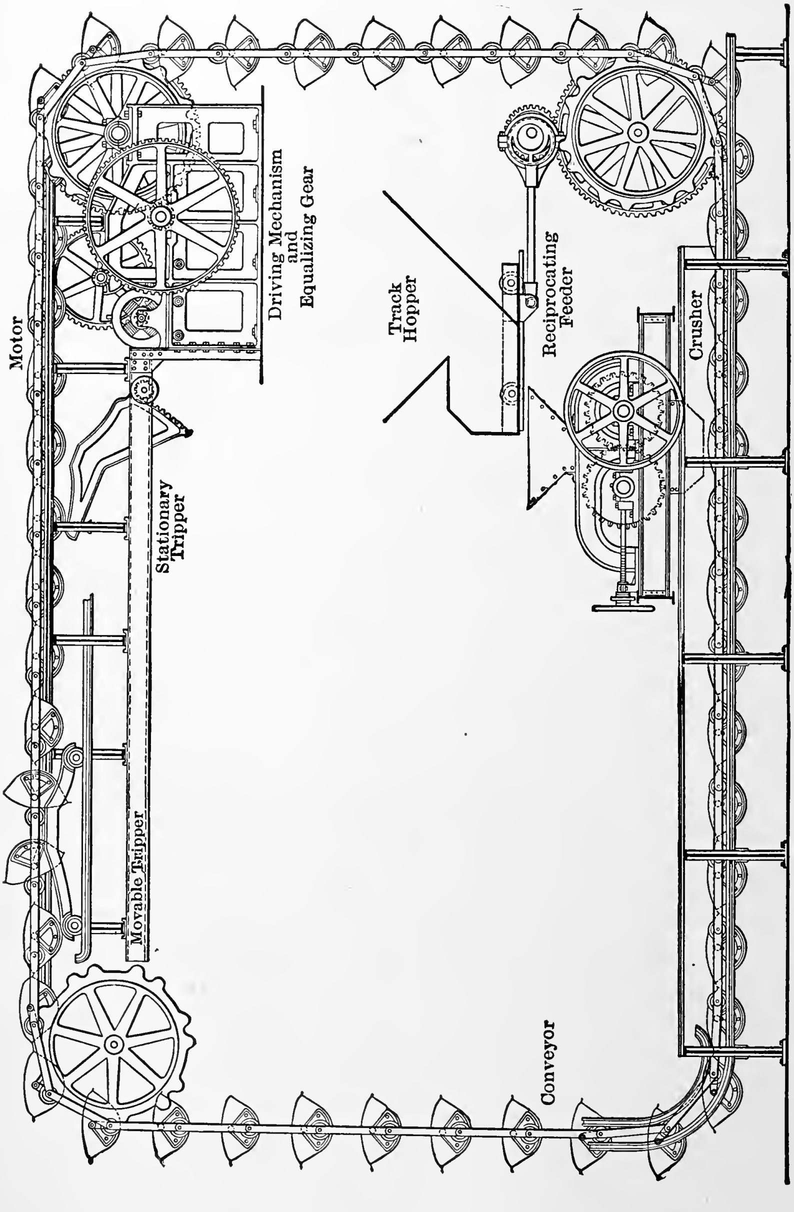


FIG. 129. Peck Carrier and Appurtenances.

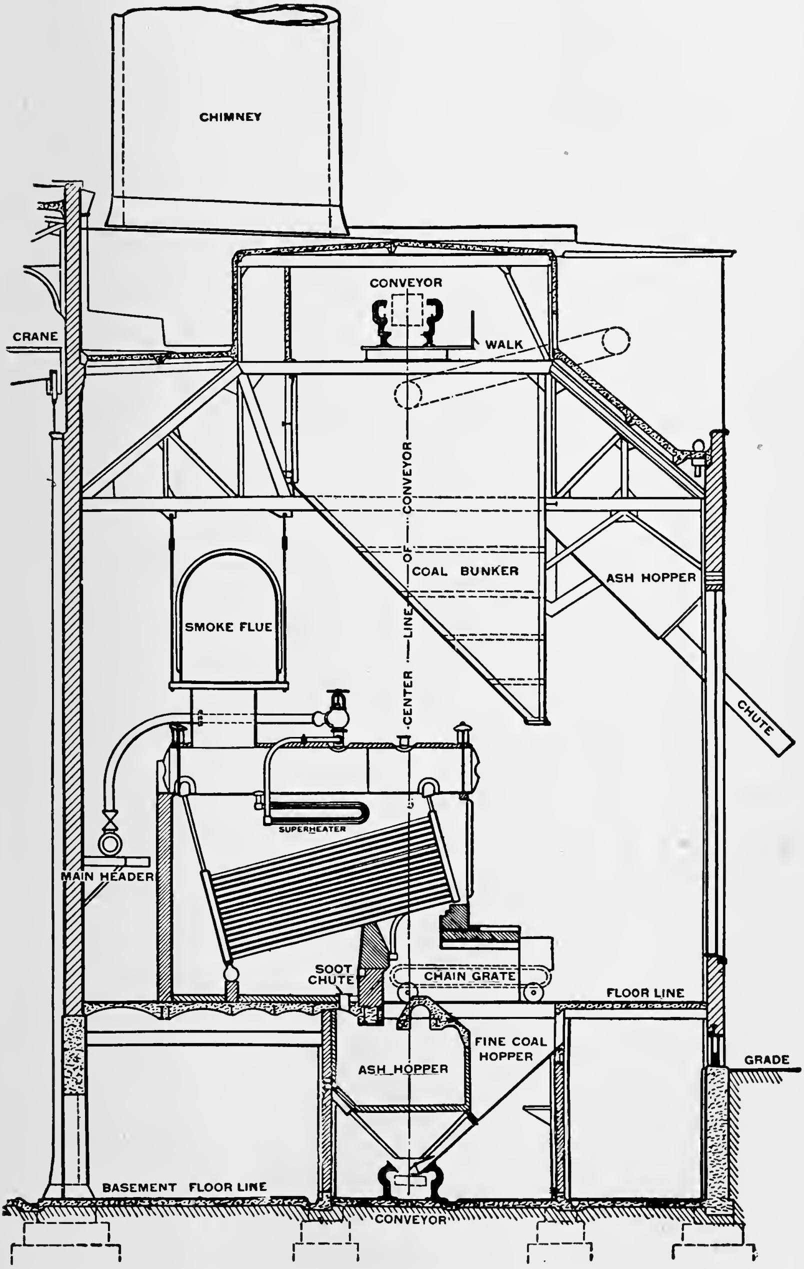


FIG. 130. Coal and Ash-handling System in the Power House of the South Side Elevated Railway Company, Chicago.

The *Pivoted Overlapping Bucket Conveyor* is perhaps the most popular type of continuous conveyor for large power plant service. It consists essentially of a continuous series of buckets pivotally suspended between two endless chains. The buckets at all times maintain their carrying position by gravity whether the chain is horizontal, vertical or inclined. By means of this system no transfer of material is necessary and discharge may be made at any desired point. Fig. 129 gives a diagrammatic arrangement of the Peck Carrier illustrating the principles of a complete coal and ash-handling system and Fig. 130 illustrates its application to a typical boiler plant.

Coal is discharged from the railway cars into a track hopper and from there delivered by a "feeding apron" into a crusher which reduces it

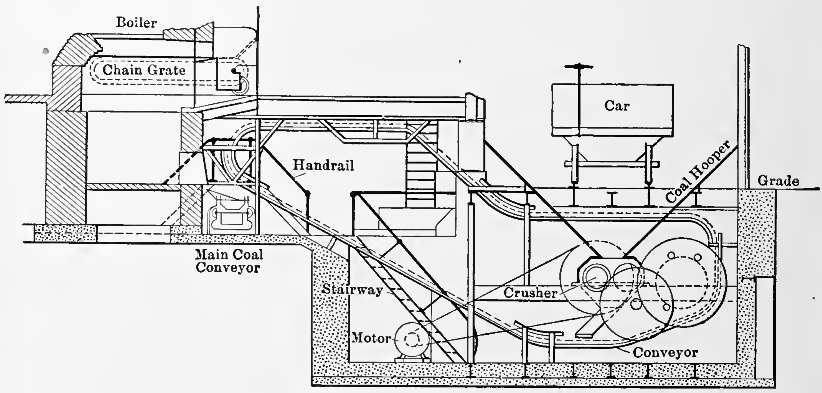


FIG. 131. Crusher and Cross Conveyor at the Power Plant of the South Side Elevated Station, Chicago.

to such a size as can be conveniently handled by the stokers. It is then discharged into a short apron or pan conveyor, which carries it to the main system of buckets, and it is elevated to the proper level and discharged into the overhead bunkers. The discharge is effected by special tripping devices which engage the buckets and turn them over. The ashes are dumped from the ash pit through a series of chutes into the lower run of buckets, by which they are elevated and discharged into the ash hopper alongside the coal bunkers. From the ash hopper the ashes discharge by gravity directly into the railway cars below. The system is operated by means of two motors, one driving the crusher and the other the main bucket system. The buckets are made of malleable iron.

In Fig. 129 the coal is fed to the crusher by the "reciprocating feeder," which is usually placed directly under the track hopper. The feeder consists of a heavy steel plate mounted on rollers and having a reciprocating movement effected by a crank mechanism from the carrier. The amount of coal delivered depends upon the distance the plate moves, and this can be varied by changing the throw of the eccentric.

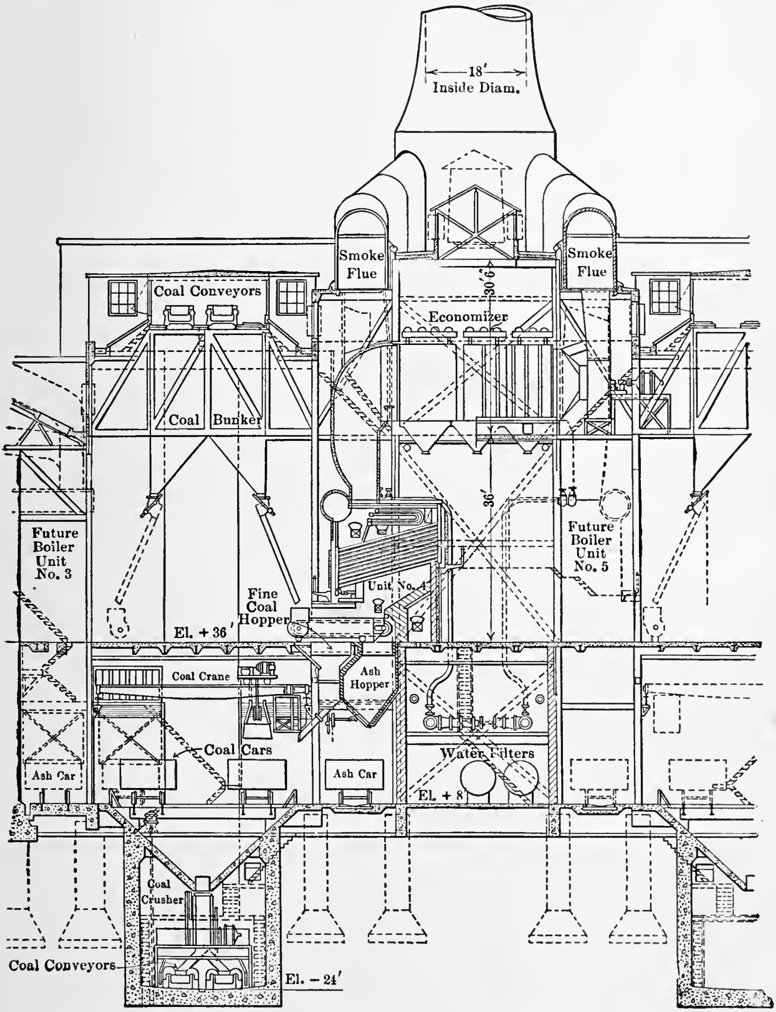


FIG. 132. Coal and Ash-handling System of the Commonwealth Edison Company, "Northwest Station."

The number of strokes corresponds to the number of buckets. Any size coal can be readily handled. When the distance from track hopper to carrier is so great that the reciprocating feeder is not practicable, a continuous or "belt" feeder is used to supply the crusher with fuel. The "equalizing gear" is designed to impart a pulsating motion to the driving sprocket wheel which will counteract the natural pulsation to which long pitch chains are subject, producing violent increase of the normal strain at frequent intervals. This is accomplished by driving the spur wheel with an eccentric pinion, causing the pitch line to describe a series of undulations corresponding to the number of sprockets on the chain wheel. Figs. 130 and 131 show the general arrangement of crusher and "cross conveyor" in the old portion of the South Side Elevated Power House, Chicago.

A coal and ash system similar to the one illustrated in Fig 129 for a plant consisting of eight 350-horsepower boilers will cost in the neighborhood of $8000, completely installed. This does not include the cost of coal and ash bunkers.

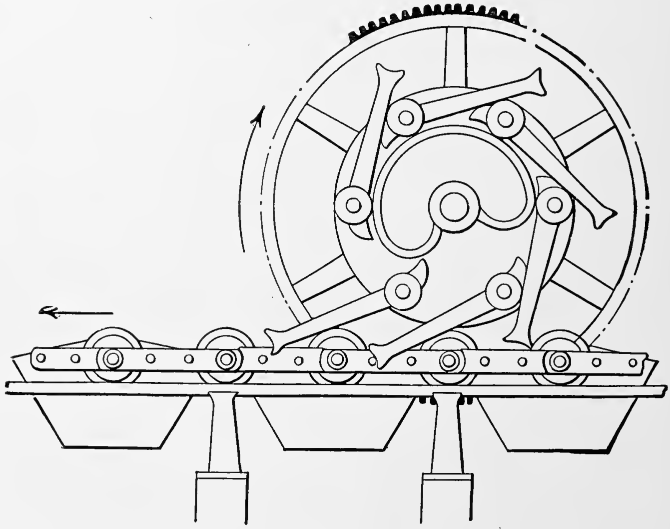


Fig. 133. Driving Mechanism of Hunt Conveyor.

The *Hunt Conveyor*, Fig. 133, while usually called a "bucket" conveyor, is in fact a series of cars connected by a chain, each having a body hung on pivots and kept in an upright position by gravity. The chain is driven by pawls instead of by sprocket wheels. The "buckets" are upright in all positions of the chain, consequently the chain can be driven in any direction. The change of direction of the chain is accomplished by guiding the carriers over curved tracks. The chain moves slowly, and the capacity is governed by the size of the buckets. The ordinary size buckets carry two cubic feet of coal and move at a rate of fifteen buckets a minute, carrying about 40 tons per hour. Two methods of filling the buckets are employed, the "measuring" and the "spout filler." In the former each bucket is separately filled with a predetermined amount by a suitable "measuring feeder." In the latter the material is spouted in a continuous stream, necessitating the use of overlapping buckets to prevent spilling of the material. Fig. 134 shows an application of the Hunt system to the power plant of the Rhode Island Suburban Railway, Providence, R. I.

The power required to operate carrier conveyors of the pivoted bucket type may be approximated from formula (60), using the proper value for B as given in Table 48.

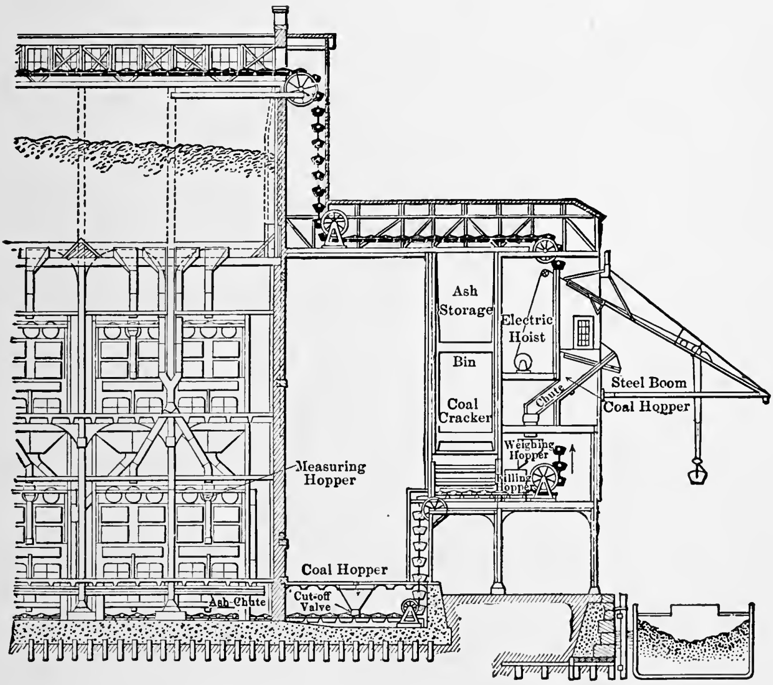


Fig. 134. Coal and Ash-handling System, Rhode Island Power House.

Belt Conveyors have a distinctive advantage over most other types of carriers in that they may be driven from any point in their length. The driving machinery is extremely simple; power is applied to one or more pulleys over which the conveyor belt passes. The maximum width of conveyors is limited only by the fiber stress in the belt. Conveyors 1000 feet from center to center handling 500 tons per hour have been successfully operated. Inclinations are limited by the angle of repose of the material. In power plant service they seldom exceed 20 degrees.

The *Robins Belt Conveyor*, Fig. 135, consists essentially of a thick belt of the required width driven by suitable pulleys and carried upon idlers so arranged that the belt becomes trough-shaped in cross section. For heavy duty five pulleys are employed instead of three as illustrated in order that the line of contact may more nearly approach the arc of a circle. The belt is constructed of woven cotton duck covered with a special rubber compound on the carrying side. The rubber is thicker at the middle than at the edges, since the wear is greatest in a line along the center, but the thickness of the belt is uniform throughout its entire width. The edges are reënforced with extra piles of duck to increase the tensile strength. The idlers are carried by iron or wooden framework, and are spaced from 3 to 6 feet between centers on the troughing side, according to the width of belt and the weight of the load. On the return side these distances range from 8 to 12 feet. High-speed rotary brushes with interchangeable steel bristles prevent wet, sticky material from clinging to the belt. Automatic tripping devices placed at the proper points cause the material to be discharged where it is needed. The trippers consist essentially of two pulleys, one above and slightly in advance of the other, the belt running over the upper and under the lower one, the course of the belt resembling the letter S. The material is discharged into chutes on the first downward turn of the belt. The trippers may be movable or fixed, single or in series. Movable trippers are used when it is desired to discharge the load evenly along the entire length, as, for instance, in a continuous row of bins, while fixed trippers are employed where the load is to be discharged at certain and somewhat separated points. The movable trippers are made in two forms, "hand-driven" and "automatic." In the former they are moved from point to point by means of a hand crank. The "automatic" tripper is propelled by the conveying belt through the medium of gearing. It reverses its direction automatically at either end of the run and travels back and forth continuously distributing its load. It can be stopped, reversed, or made stationary at will. Notable installations of this system are at the Hudson and Manhattan Railway Company's power house, Jersey City; L Street Station, Edison Illuminating Company of Boston; South Boston Power Station of the Boston Elevated Company and the Essex Power Station of the Public Service Electric Co., N. J.

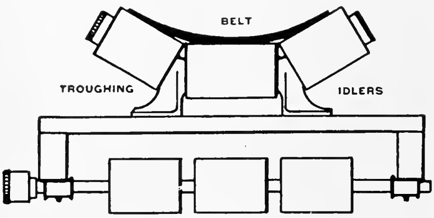


FIG. 135. Guide Pulleys, Robins Belt Conveyor.

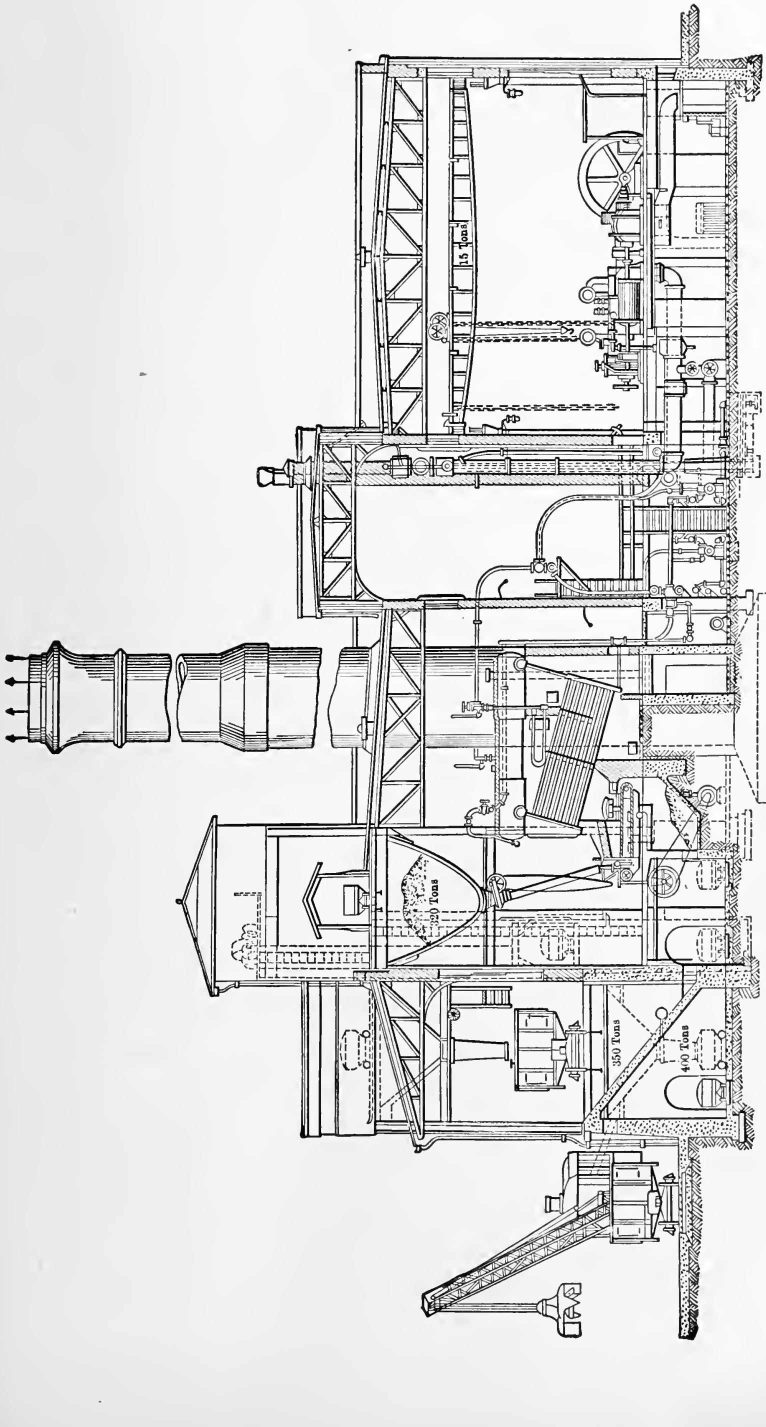


Fig. 136. Coal and Ash-handling System of the Rock Island R. R. Power Plant at Springfield, Mo.

The power required to drive belt conveyors may be approximated from the following empirical equation. (C. K. Baldwin, Trans. A.S.M.E., Vol. 30, p. 187.)

For level conveyors:

$$\text{Hp.} = \frac{CTL}{1000}. \tag{61}$$

For inclined conveyors:

$$\text{Hp.} = \frac{CTL}{1000} + \frac{TH}{1000}. \tag{62}$$

C = constant as given in Table 49,
T = load in tons (2000 lb.) per hr.,
L = length of conveyor between centers, ft.,
H = vertical lift of material.

For each movable or fixed tripper add the horsepower given in Table 49. For friction of conveyor ends and driver add the following:

Length of conveyor	25	50	75	100	200	500
Per cent added power	80	50	30	20	10	4

TABLE 49.

POWER REQUIREMENTS FOR BELT CONVEYORS.

(Coal and Ashes.)

Width of belt	12	16	20	24	28	32	36
C	0.234	0.220	0.205	0.195	0.175	0.163	0.157
Hp. required for each movable or fixed tripper	0.50	0.75	1.25	1.5	2.25	2.75	3.25

Belt-conveyor Operating Data: Power, Oct. 3, 1916, p. 490. *Economics of Conveyor Equipments:* Eng. Mag., Nov., 1916, p. 231.

119. Elevating Tower, Hand-car Distribution. — Fig. 137 illustrates the coal and ash-handling system as originally installed at the Aurora and Elgin Interurban Railroad power house, Batavia, Ill. Coal is delivered to the plant by railroad cars which dump directly into coal hoppers located inside a steel structure running the entire length of the building and spanned by two railroad tracks. There are 18 hoppers constructed of 17-inch brick walls fitted with steel-plate bottoms. Subdividing the storage space in this manner makes it possible to carry different grades of coal, prevents the spreading of fire, and affords a simple construction for the support of the railroad tracks. The basement of the boiler room extends underneath the hoppers, and two lines of narrow-gauge tracks are embedded in the concrete floor. Turntables at the center facilitate the switching of cars to the elevators which rise through the boiler room close to the chimney. The cars, of one ton capacity each, are of special construction, with roller-bearing axles and a combined ratchet lift and friction dump. The filled cars are pushed from underneath the hoppers to two elevators which lift them to the line of tracks supported overhead across the boiler fronts.

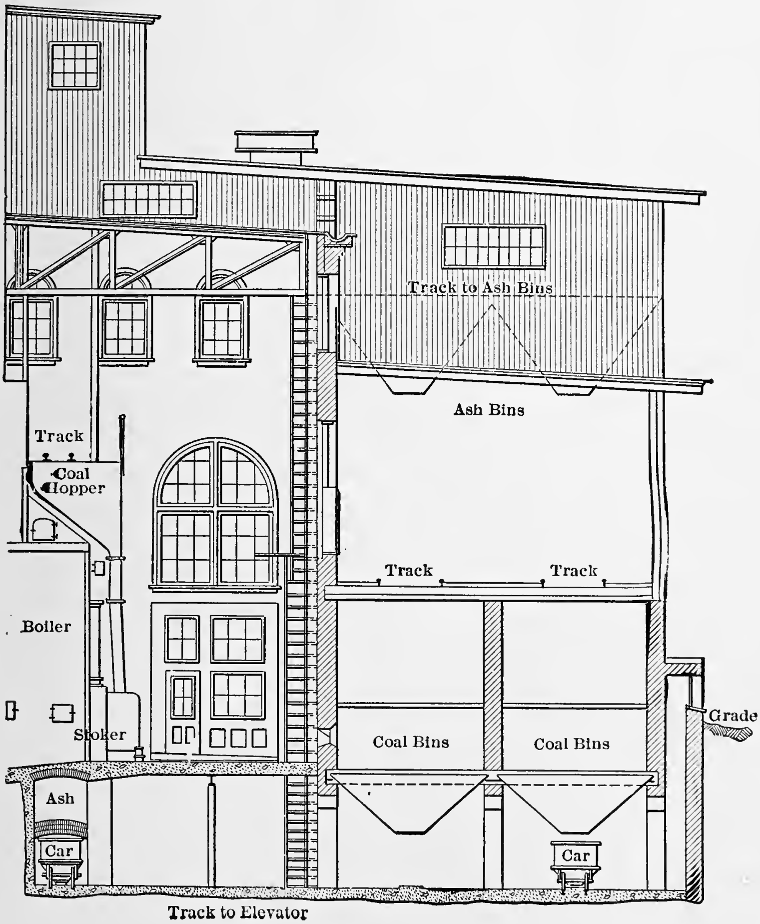


Fig. 137. Typical Coal and Ash-handling System Involving the Use of Elevating Tower and Hand-car Distribution.

They are then pushed to the hoppers suspended above the boiler setting and the coal is dumped. These hoppers have a capacity of six tons each. From the hoppers the coal is fed to the stoker by an ordinary down spout. The ashes fall from the stokers into an ash pit, from which they may be discharged into ash cars. The ash cars are elevated to a set of tracks running at right angles to the main tracks, and are transferred to ash bins located directly over the coal bins. Coal and ashes are weighed in the small cars. There are ten boilers in this plant and four men are required to handle the coal and ashes. The entire coal and ash-handling system cost about $10,000, and the cost of handling the coal and ashes, exclusive of fixed charges, is approximately 4 cents per ton. This does not include wages of firemen or water tenders. For a description of recent changes made in this plant see Prac. Engr., U. S., Nov. 1, 1916, p. 907.

120. Elevating Tower, Cable-car Distribution. — The coal and ash-handling system of the Delray Station of the Detroit Edison Company is a typical example of a large station equipped with elevating tower and cable-car distributers instead of the usual bucket conveyor. The system consists essentially of a lofty steel tower in which are housed at various levels a track receiving hopper, crushing rolls and feeders, weighing hopper, hoisting apparatus, etc., and a small cable railway for delivery to the bunkers. The railroad coal cars enter the tower on an elevated trestle 18 feet above grade, below which is a track receiving hopper. A two-ton "tub hoist" is filled with coal from the bottom of the receiving hopper and elevated to a 20-ton bin at the top, 120 feet above ground level. This bin has a grille bottom at one side and under the outlet a heavy duty coal crusher, thus allowing the fine coal to screen through directly while all the larger lumps are automatically delivered to the crusher. From the two bins the small cable cars are filled for dumping into the desired bunkers over the boiler rooms. The cars are arranged for automatic dumping by means of adjustable trips which may be located at any point. The entire system has a capacity of from 125 to 150 tons of coal per hour and is motor-driven. The ash-handling system consists of brick-lined concrete hoppers underneath each pair of stokers which discharge their contents by gravity into the small cars operated on the track system in the boiler-house basement.

When handling 600 tons per day of 24 hours the cost of operation is approximately 20 cents per ton from coal car to ash car. This includes wages of firemen and water tenders.

121. Hoist and Trolley; Telpherage. — The telpher is a form of electric hoist which lifts and transfers the load on overhead tracks from one point to another. Fig. 138 illustrates a very simple and economical method of handling coal and ashes as installed by the Jeffrey Manufacturing Company at the power plant of the Scioto Traction Company embodying the telpher systems. If the coal car is of the dump type the contents are discharged directly into the coal pit from which the coal is removed by grab bucket and transferred

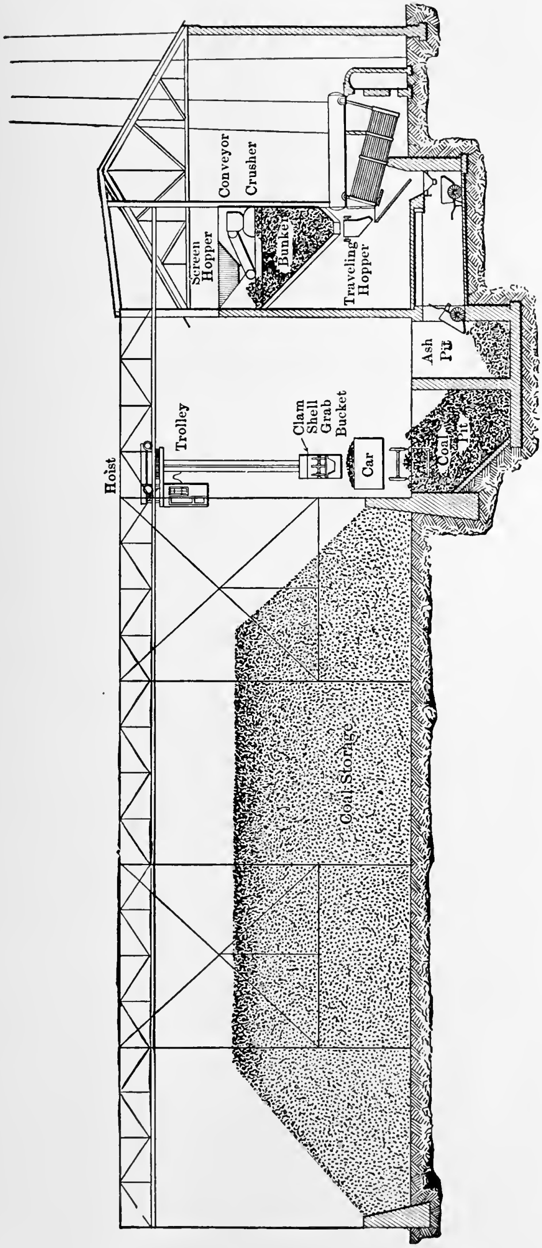


FIG. 138. Coal and Ash-handling System at the Power House of the Scioto Valley Traction Co.

either to the overhead bunker or to the storage pile. If the coal car is of the gondola type the coal is removed directly from the car by the grab bucket. The bucket is hoisted and carried on the trolley into the building over the screen hoppers where it discharges its contents; the finer particles fall directly into the bunker and the larger lumps are automatically delivered to the crusher. The grab bucket will take about 98 per cent of the coal in the car, leaving only 2 per cent to be handled by hand. Coal is fed to the stokers by means of a traveling electric hopper which receives its supply from the overhead bunkers. The present capacity of the plant is 50 tons per hour taken from the car or pit to stock pile.

122. Vacuum Conveyors. — This type of conveyor is finding favor with many engineers for handling dry ashes because of its simplicity in design and ease of application. It has also been used in a few cases for handling small nut coal and screenings. The system consists essentially of a pipe line through which air is flowing at a high velocity. The material to be conveyed is fed into the pipe through suitable openings and the momentum of the column of air carries it to the point of discharge. Velocity is imparted to the air either by a mechanical exhauster or by steam jets discharging in the direction of flow.

Fig. 139 gives a diagrammatic arrangement of a vacuum ash-conveying system as installed in the power plant of the Armour Glue Works, Chicago, Illinois. One end of special cast-iron header *F* leads to the ash pits of the various boilers by means of branch tubes, and the other end is connected with the closed storage tank. Each branch pipe is fitted with simple circular openings directly underneath each ash-pit door for admitting ashes. These openings are kept covered except when in operation. Exhauster *E* creates a partial vacuum in chamber *D* and draws in air at a high velocity from the opening in the ends of the branch pipes. Ashes raked into the pipes through the openings are caught by the rapidly moving column of air and forced into the storage tank. Air is withdrawn from the top of the separator chamber through pipe *G* and discharged to the stack or to waste. A spray is introduced into pipe *F* to reduce dust. In this particular installation the system is applied to a boiler plant of thirteen boilers, aggregating 4800 horsepower, and cost, completely installed, $5600. The ash bin has a capacity of 60,000 pounds of wet ashes and is constructed of five-sixteenths-inch sheet iron. The exhauster (a 30-foot Root blower) has a capacity of about 8000 cubic feet per minute at 265 r.p.m., and is driven by a 75-horsepower motor. Under normal conditions of operation the motor requires 50 horsepower when de-

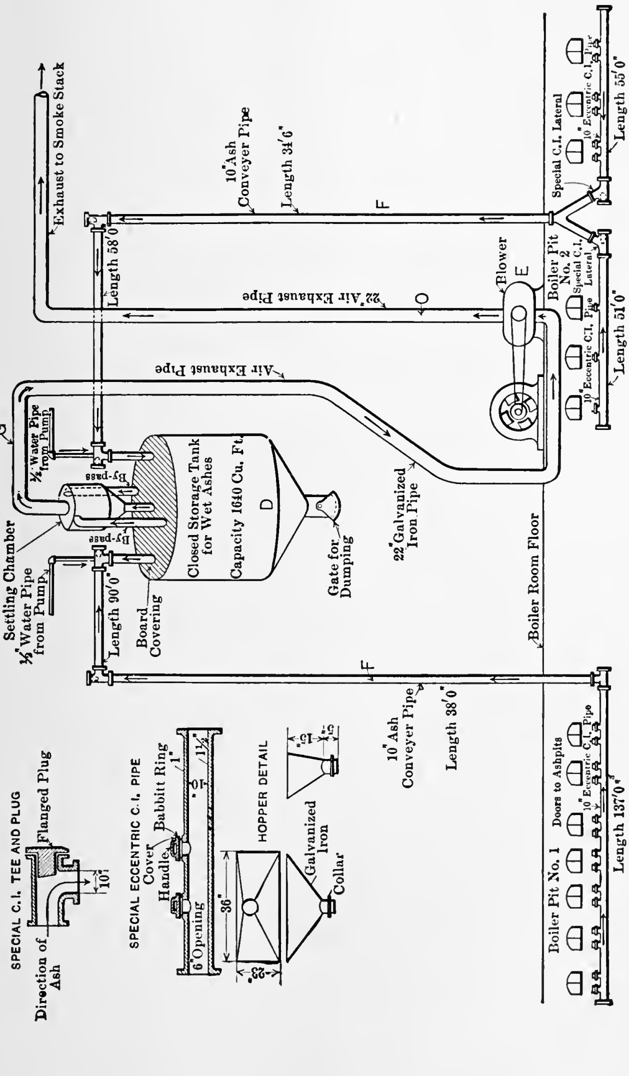


FIG. 139. Diagrammatic Arrangement of Vacuum Ash-handling System at the Armour Glue Works, Chicago, Ill.

livering 250 pounds of ash per minute, and the vacuum on the suction side of the exhauster is 3.3 inches of mercury. The pipe from the ash bins to the separating chamber is 10 inches in diameter and is constructed of number 16 and number 20 galvanized iron. The ashes are raked by hand from the ash pits to the suction openings of the branch pipes, and are handled dry, the dust being taken along with the ashes. Short elbows are soon worn out by the abrasive action of the ashes, and tees are used instead, since the accumulation in the "dead" end receives the impact and takes up the wear. Long radius bends may be used in place of the tees. The cost of power for handling the ashes in this installation is approximately 7 cents per ton.

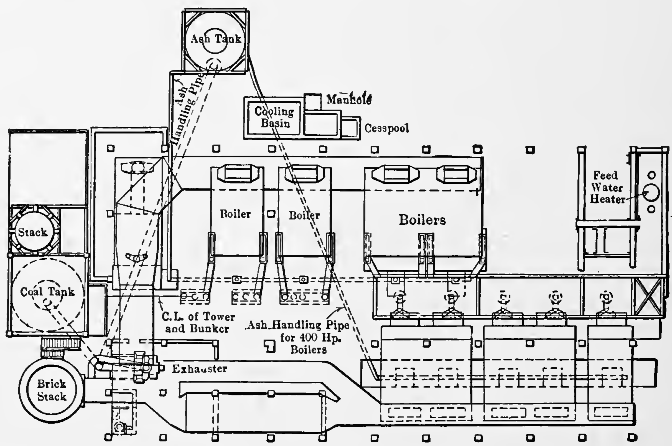


Fig. 140. Vacuum System for Handling Coal and Ashes at the Plant of the Pierce-Arrow Motor Car Co.

This type of vacuum system is used for handling coal at the power house of the Pierce-Arrow Motor Car Company, Buffalo, New York, and is giving satisfactory results. The steam jet system, however, is used for handling the ashes. It differs from the mechanical exhauster type in that steam jets are used for creating the vacuum. The jets are inserted in the pipe line between the last boiler and the point of delivery and discharge in the direction of flow. In the Pierce-Arrow plant the labor cost of handling the coal and ashes is 20.6 cents per ton on a basis of 26,000 tons per year. The entire equipment cost $34,000.

In the steam jet ash system, two conveying pipe sizes are in common use; one with 6-inch inside diameter for capacities up to three tons per hour and one with 8-inch inside diameter pipe for capacities from three to eight tons per hour. Larger sizes have not proved practicable because of the excessive amount of steam required to effect the desired result. For horizontal runs under 100 feet in length one jet placed in the elbow of the riser is sufficient to move the material, but for longer runs additional jets in the horizontal pipe are necessary. The jets as ordinarily installed require from 175 to 275 lb. of steam per ton of ashes per hour depending upon initial pressure and quality of the steam and size of pipe.

Vacuum Ash-Removal System: Power, April 7, 1914, p. 473; Jan. 13, 1914, p. 41.

123. Cost of Handling Coal and Ashes. — In large stations where a number of men are employed to handle coal and ashes only it is a simple matter to divide the cost of handling into the various stages, thus:

1. Cost of unloading cars or barges.
2. Cost of conveying coal to bunkers.
3. Cost of feeding coal to furnace.
4. Cost of removing ashes.

These costs are usually expressed in cents or dollars per ton of coal burned, or in terms of cents or dollars per horsepower-hour or kilowatt-hour of main prime-mover output. Item number 3 is oftentimes included under "boiler-room attendance" and items 1, 2, and 4 under "coal and ash handling." Not infrequently all four items are included under "attendance." So much depends upon the character of stokers and furnace, size of boilers, and the like, that general figures on the cost of handling the coal and ashes are of little value unless accompanied by a description of the equipment. For the sake of general comparison the most satisfactory method of expressing the cost is in dollars per ton of coal from coal car to ash car. This includes wages of coal and ash passers, repair men, and boiler tenders. In small stations the coal and ash handling is done by the boiler tenders, in which case it is impracticable to separate the items mentioned above, and the cost is ordinarily included under attendance. An average figure for handling coal by barrow and shovel is not far from 1.6 cents per ton per yard up to the distance of five yards, then about 0.1 cent per ton per yard for each additional yard. With automatic conveyors the operating cost, not including wages of firemen and water tenders, varies with the size of plant and the type of conveyor, and ranges anywhere from a fraction of a cent per ton to four or five cents per ton. The larger the plant and the greater the amount of coal handled the lower will be the cost per ton. In comparing the relative costs of manual and automatic handling, fixed charges of at least 15 per cent of the first cost of the mechanical equipment should be charged against the latter in addition to the cost of operation. In large central stations equipped with stokers and conveyors and consuming 200 tons or more of coal in twenty-four hours, the cost of handling the coal from coal car to ash car, including wages of firemen and water tenders but exclusive of fixed charges, will range between 18 cents and 25 cents a ton.

124. Coal Hoppers. — Fig. 141 shows a front and side elevation of a typical set of stationary weighing hoppers as applied to the boilers of the Quincy Point power plant of the Old Colony Street Railway

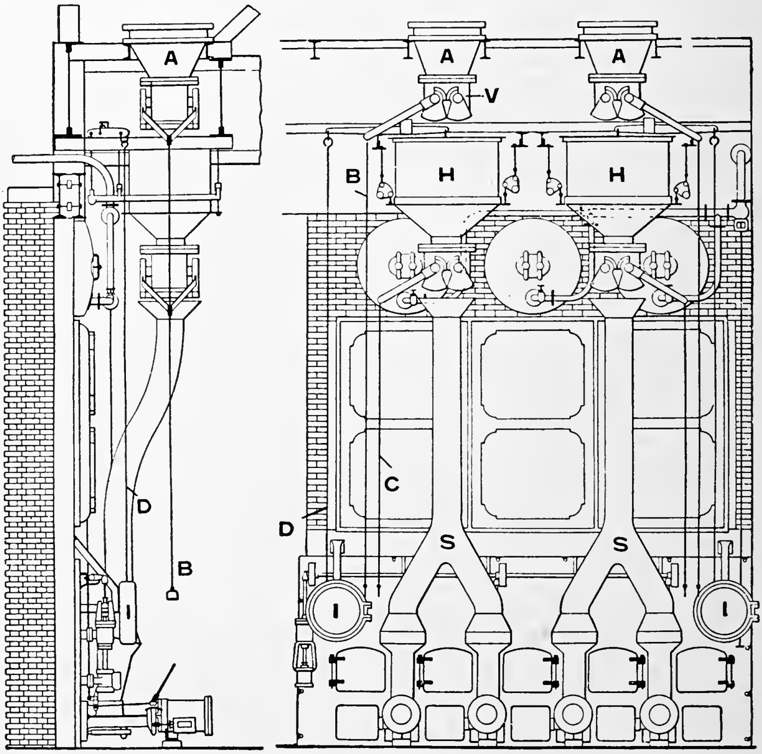


Fig. 141. Stationary Coal Weighing Hoppers.

Company, Quincy Point, Mass. Each battery of boilers is provided with an independent set of hoppers. The bottoms of the overhead coal bunkers lead into the small hoppers *A*, *A*. The operation of any single weighing hopper is as follows: Coal is fed from the overhead bunkers to weighing hopper *H* by means of valve *V*. The weight of coal in the weighing hopper is transmitted by a system of levers and knife edges to the inclosed scale beam *I* and noted in the usual way. The weighed charge of coal is then admitted to the down spout *S* by means of valves similar to those at *V*.

Although separate weighing hoppers for each battery, as illustrated in Fig. 141, offer many advantages, they are quite costly and it is not unusual to install one or more large weighing hoppers mounted on overhead traveling carriages so that one may supply a number of boilers (Fig. 142). At the Armour Glue Works, Chicago, the coal supply is stored in one large overhead bunker of 1000 tons' capacity. A five-ton motor-driven traveling hopper receives its supply from this central

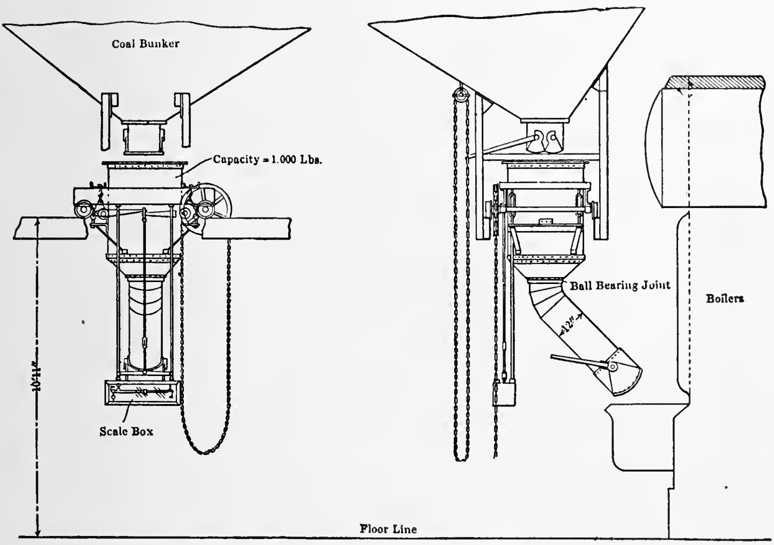


FIG. 142. Traveling Coal Hoppers.

bunker and delivers it to the various boilers. One man operates the traveling hopper, tends to the coal valves, and supplies all boilers with coal.

Weighing hoppers are sometimes made automatic; that is, the opening and closing of valves, feeding of coal, and recording of weight are automatically performed by the weight of the coal itself. The scale is set for discharges of a certain weight and continues to discharge this amount automatically. In the few plants which are equipped with automatic weighing hoppers the capacity of the hopper is approximately 100 pounds per discharge. These hoppers are necessarily more complicated and more costly than the ordinary weighing hoppers, and it is a question whether the advantages offset the extra first cost and maintenance charges. A small automatic hopper of 100 pounds discharge capacity costs approximately $400 as against $250 for the ordinary weighing device. For a description of a coal meter see paragraph 396.

125. Coal Valves. — Figs. 145 to 147 illustrate the principles of a few well-known coal valves. They may be conveniently grouped into two classes according to the location of the coal pocket: (1) those

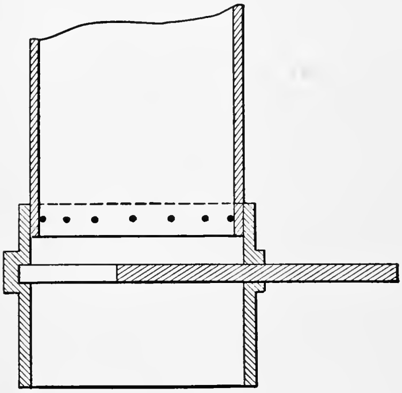


FIG. 143. Common Slide Coal Valve.

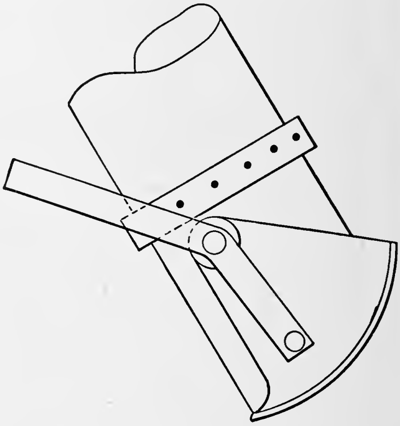


FIG. 144. Simplex Coal Valve.

drawing the coal from overhead bunkers and (2) those drawing from the side of a bin. In the first class come the simple *slide* valve and the *simplex* and *duplex rotating* valve. In the latter are the *flap* valve and the *rotating* valve. They are made in various sizes and designs, but those illustrated are examples of the most common types. The simple slide valve, Fig. 143, is applicable only to small size coal and to small spouts, since coarse or lump coal may get in the way and prevent proper closing. The simplex valve, Fig. 144, consists of a rotating jaw actuated by a lever. There are no rubbing surfaces, and the jaws cut through the material without jamming. The duplex valve, Fig. 145, consists of two rotating jaws connected to a common actuating lever. The jaws move simultaneously, so that even a partially open valve delivers the coal centrally. When closing the valve the flow is gradually stopped by the decreasing width of the opening and there is but little resistance to the movement of the jaws. The largest valve can easily be operated by hand.

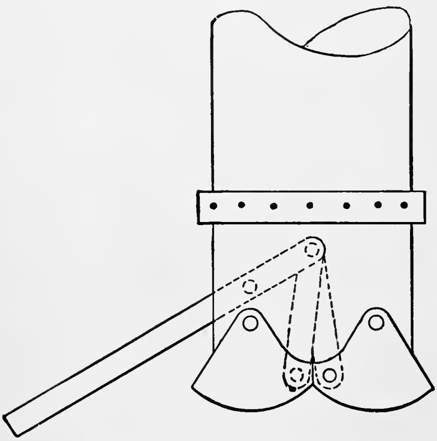


FIG. 145. Duplex Coal Valve.

The flap valve, Fig. 146, is the simplest form for drawing coal from a side bin. It consists merely of an iron flap hinged to the bottom of the chute. The valve is lowered to let the coal run over its top and is raised to stop the flow. It cannot be clogged or get jammed in closing. The flap is raised and lowered by a simple lever. For very large bins, where the valves are to be opened and closed frequently, the "Seaton" valve, Fig. 147, is usually preferred. This valve consists of two jaws *EE'*, and *TT'* pivoted to suitable framework at *O* and actuated by lever *A*.

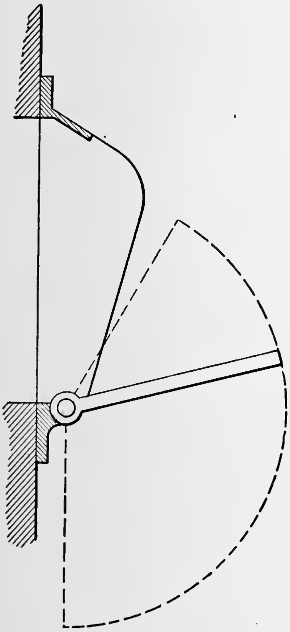


FIG. 146. Common "Flap" Coal Valve.

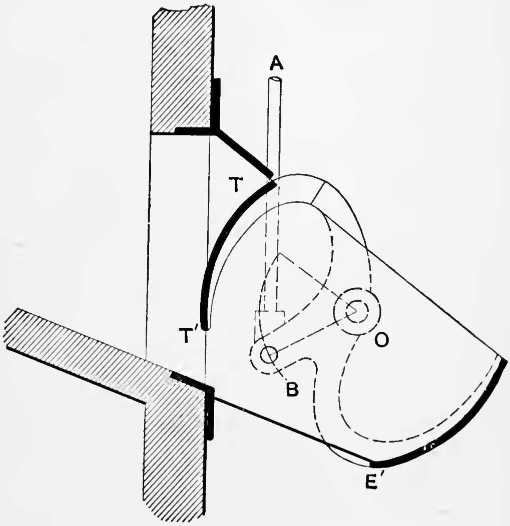


FIG. 147. "Seaton" Coal Valve.

The valve is shown fully closed. Raising lever *A* causes the cut-off blade *EE'* to rotate about *O* and permits the coal to flow through the space between the edge of the jaw *E* and the end of the chute. The rate of flow is regulated by the width of this opening. The cut-off blade does not reach a stop, hence there is no possibility of a lump of coal getting in the way and preventing the prompt closing of the valve.

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Coal and Ash Handling System at Pacific Mills, South Lawrence, Mass.: Prac. Engr., Oct. 1, 1913, p. 973.

Coal and Ash Handling System at Pierce-Arrow Plant: Power, Jan. 13, 1914, p. 41.

Coal and Ash Handling System at Victor Plant: Prac. Engr., June 15, 1916.

Mechanical Handling of Coal and Ashes in the Power Plant: Eng. Mag., Sept., 1915, p. 872; Oct., 1915, p. 65.

PROBLEMS.

1. If power costs 1.5 cents per kw-hr. approximate the cost of moving 200 tons of coal per hour a horizontal distance of 50 ft. by means of a screw conveyor.

2. Determine the power required to drive a scraper conveyor carrying 250 tons of bituminous coal per hour, sliding blocks to be used. The weight of the chain and flights with sliding blocks is 26 lb. per lineal ft., the capacity of the conveyor is 150 tons per hour. The distance between centers of head and last sprockets is 160 ft. and the angle of conveyor with the horizontal is 30 degrees.

3. Determine the power required to drive a pivoted bucket carrier having a capacity of 60 tons of coal per hour; rollers 6 in. in diameter with $1\frac{3}{8}$ in. pins; weight per ft. of empty carrier, 80 lb.; horizontal length of conveyor, 400 ft.; vertical lift, 60 ft.; 4 right angle turns; horizontal length traversed by loaded buckets, 300 ft.; speed of conveyor, 50 ft. per min.

4. Determine the power required to elevate 200 tons of coal per hour by means of a 24-inch belt. Speed of belt, 200 ft. per min.; vertical lift, 30 ft.; length of conveyor between centers, 300 ft. The system contains 3 fixed and 2 movable trippers.

CHAPTER VII

CHIMNEYS *

126. General. — In order to cause the necessary weight of air to flow through the fuel bed and force the products of combustion through the gas passages of the boiler and setting, a pressure difference between ash pit and uptake is necessary. This pressure difference is designated as *draft* whether the actual pressure is above or below atmosphere. Draft may be produced mechanically by means of fans, blowers and steam jets, or thermally by means of chimneys. Stacks or chimneys offer the simplest means of conducting the products of combustion to waste and since the latter must be discharged at a sufficient elevation to prevent their being a public nuisance the height of stack necessary to effect this result is often sufficient to create the required draft. Even if considerable height must be added to the stack over and above that required to discharge the gases at a given elevation the extra cost may be considerably less than incident to mechanical draft operation. For this reason the majority of steam power plants depend upon the chimney for draft. In large plants equipped with mechanical stokers or where fuel is burned at a high rate or where economizers are used for abstracting heat from the flue gases mechanical draft is commonly employed; but even in these cases if forced draft is used some chimney effect may be desirable. In view of the enormous amount of heat developed in forced draft, stoker-fired furnaces and the great weight of gas passing over the boiler heating surfaces it is now generally accepted that some means must be provided to remove these gases from the furnace promptly in order to protect the furnace brickwork, by preventing a "soaking up" action of the heat. The chimney provides such a suction draft throughout all parts of setting. (See Paragraph 153.)

When in operation, a chimney is filled with a column of gases with higher average temperature than that of the surrounding air. As a result the density of the gases within the stack is less than that of the outer air, and the pressure at the bottom of the column is less inside the stack than it is outside.

* In this text the terms "chimney" and "stack" are used synonymously. Builders usually apply the term "chimney" to the masonry and concrete structures and "stack" to the steel structures.

TABLE 51.

DENSITY AND SPECIFIC VOLUME OF AIR AND CHIMNEY GASES AT VARIOUS TEMPERATURES.

Air.				Chimney Gases.					
t	s	v	d	t	d	t	d	t	d
0	11.581	.935	.086353	200	.06334	430	.04695	660	.03730
5	11.706	.945	.085424	210	.06239	440	.04643	670	.03697
10	11.832	.955	.084513	220	.06147	450	.04592	680	.03665
15	11.931	.965	.083623	230	.06058	460	.04542	690	.03633
20	12.085	.976	.082750	240	.05971	470	.04493	700	.03602
25	12.211	.986	.081895	250	.05887	480	.04445	710	.03571
30	12.337	.996	.081058	260	.05805	490	.04398	720	.03540
32	12.387	1.000	.080728	270	.05726	500	.04353	730	.03511
35	12.463	1.006	.080238	280	.05648	510	.04308	740	.03481
40	12.589	1.016	.079434	290	.05573	520	.04264	750	.03453
45	12.715	1.026	.078646	300	.05499	530	.04221	760	.03424
50	12.841	1.037	.077874	310	.05428	540	.04178	770	.03396
55	12.967	1.047	.077117	320	.05358	550	.04137	780	.03369
60	13.093	1.057	.076374	330	.05290	560	.04096	790	.03342
62	13.144	1.061	.076081	340	.05224	570	.04056	800	.03316
65	13.220	1.067	.075645	350	.05159	580	.04017	900	.03072
70	13.346	1.077	.074930	360	.05096	590	.03979	1000	.02861
75	13.472	1.087	.074229	370	.05035	600	.03942	1100	.02678
80	13.598	1.098	.073541	380	.04975	610	.03905	1200	.02516
85	13.724	1.108	.072865	390	.04916	620	.03869	1300	.02373
90	13.851	1.118	.072201	400	.04859	630	.03833	1400	.02245
95	13.976	1.128	.071550	410	.04803	640	.03798	1500	.02131
100	14.102	1.138	.070910	420	.04749	650	.03764	1800	.01848
110	14.354	1.159	.069665	......	....	......	....	2000	.01698

d = density, pounds per cubic foot.
t = temperature, deg. fahr.
s = specific volume, cubic feet per pound.
v = comparative volume, volume at 32 deg. fahr. = 1.
Density of chimney gas taken 0.085 pound per cubic foot at 32 deg. fahr. and 29.92 inches of mercury. (Rankine, " Steam Engine," gives the density at 32 deg. fahr. as varying from 0.084 to 0.087.)

127. Chimney Draft. — The theoretical maximum static draft of a chimney is the difference in weight of the column of heated gas inside the stack and of a column of outside air of the same height, thus, if

D = maximum theoretical static draft, in. of water,
H = effective height of the chimney, ft.,
d_a = density of the outside air, lb. per cu. ft.,
d_c = density of the inside gas, lb. per cu. ft.,
0.192 = factor for converting pressure in lb. per sq. ft. to in. of water,

$$D = 0.192\,H(d_a - d_c). \tag{63}$$

Neglecting the influence of the relative humidity of the air

$$d_a = 0.0807\,\frac{P_a}{P}\cdot\frac{T}{T_a}, \tag{64}$$

in which

P_a = observed atmospheric pressure, lb. per sq. in.,
P = standard atmospheric pressure, lb. per sq. in.,
T = absolute temperature at the freezing point, deg. fahr.,
T_a = absolute temperature of the outside air, deg. fahr.

The density of chimney gas varies with the nature of the fuel and the air excess used in burning the fuel. An average value is 0.085 lb. per cu. ft. at 32 deg. fahr. and pressure P.

Therefore,

$$d_c = 0.085 \frac{P_c}{P} \cdot \frac{T}{T_c}, \tag{65}$$

in which

T_c = absolute temperature of the chimney gas, deg. fahr.

Other notations as in equation (64).

Substituting these values of d_a and d_c in equation (63),

$$D = 0.192\, H \frac{P_a}{P}\left(\frac{0.0807\, T}{T_a} - \frac{0.085\, T}{T_c}\right). \tag{66}$$

TABLE 52.

THEORETICAL DRAFT PRESSURE IN INCHES OF WATER. CHIMNEY 100 FEET HIGH.[1]

Temp. in the Chimney.	Temperature of the External Air — Barometer, 14.7 Pounds per Square Inch.[2]										
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
200	.453	.419	.384	.353	.321	.292	.263	.234	.209	.182	.157
220	.488	.453	.419	.388	.355	.326	.298	.269	.244	.217	.192
240	.520	.488	.451	.421	.388	.359	.330	.301	.276	.250	.225
260	.555	.528	.484	.453	.420	.392	.363	.334	.309	.282	.257
280	.584	.549	.515	.482	.451	.422	.394	.365	.340	.313	.288
300	.611	.576	.541	.511	.478	.449	.420	.392	.367	.340	.315
320	.637	.603	.568	.538	.505	.476	.447	.419	.394	.367	.342
340	.662	.638	.593	.563	.530	.501	.472	.443	.419	.392	.367
360	.687	.653	.618	.588	.555	.526	.497	.468	.444	.417	.392
380	.710	.676	.641	.611	.578	.549	.520	.492	.467	.440	.415
400	.732	.697	.662	.632	.598	.570	.541	.513	.488	.461	.436
420	.753	.718	.684	.653	.620	.591	.563	.534	.509	.482	.457
440	.774	.739	.705	.674	.641	.612	.584	.555	.530	.503	.478
460	.793	.758	.724	.694	.660	.632	.603	.574	.549	.522	.497
480	.810	.776	.741	.710	.678	.649	.620	.591	.566	.540	.515
500	.829	.791	.760	.730	.697	.669	.639	.610	.586	.559	.534
550	.863	.828	.795	.762	.731	.700	.671	.644	.618	.593	.585
600	.908	.873	.839	.807	.776	.746	.717	.690	.663	.638	.613

1. For any other height multiply the tabular figure by $\frac{H}{100}$, where H is the height in feet.

2. For any other pressure multiply the tabular figure by $\frac{P}{14.7}$, where P is the barometric pressure in pounds per square inch.

Assuming $P_a = P = 14.7$ and $T = 492$, equation (66) reduces to

$$D = H\left(\frac{7.64}{T_a} - \frac{7.95}{T_c}\right). \qquad (67)$$

By assuming the same density for the chimney gas and outside air, and $P = 14.7$, equation (66) may be written

$$D = 0.52\, P_c \left(\frac{1}{T_a} - \frac{1}{T_c}\right) H. \qquad (68)$$

Equation (66) gives the true maximum theoretical static draft provided the various factors entering into the formula are accurately known. In practice considerable variation exists in the composition of the gases and the temperatures are not uniform throughout the stack nor is the pressure the same at all points, hence the so-called theoretical value is correct only for an arbitrarily fixed set of conditions. Furthermore, with a quiet atmosphere the theoretical draft may be largely increased owing to the column of heated gases above the mouth of the chimney. Strong air currents passing over the mouth of the stack may also increase or decrease the draft. The actual maximum static draft can be realized only when there is no flow as when the ash pit doors are closed and there is no perceptible transfer of heat or leakage of air through the chimney walls, boiler setting and flue or breeching.

Example 16. Required the maximum theoretical draft obtained from a chimney 150 feet high, atmospheric pressure 14.5 pounds per square inch, temperature of outside air 60 deg. fahr., mean temperature of the chimney gases 550 deg. fahr.

Here $P_a = 14.5$, $T_a = 460 + 60 = 520$, $T_c = 460 + 550 = 1010$, $T = 460 + 32 = 492$.

Substituting these values in equation (66)

$$D = 0.192\,\frac{14.5}{14.7}\left(\frac{0.0807 \times 492}{520} - \frac{0.085 \times 492}{1010}\right) 150$$
$$= 0.994, \text{ or practically one inch of water.}$$

In this problem the mean temperature of the chimney gases is given. In practice it must be approximated from the flue gas temperature. Sufficient data are not available for predetermining the cooling action of the chimney walls and breeching except for a few special cases.* In view of the great variation in chimneys as to design, size, material, temperature difference and rate of driving, all assumptions are largely a matter of guess-work and equations based on a few isolated cases are equally untrustworthy. A common rule is to allow a drop of 80 degrees per 100 feet of unlined steel stacks and 40 degrees for brick or

* Peabody and Miller, Steam Boilers, p. 199.

lined steel chimneys. These values are too high for tall chimneys of large diameter and too low for small short stacks. Another rule is to allow 5 to 10 per cent of the theoretical maximum static draft as the pressure drop due to the cooling action. Both rules are purely arbitrary and may give results far from the truth.

For the influence of rate of driving on stack temperatures see Table 36 and Fig. 148.

With economizers stack temperatures are reduced to 250–350 deg. fahr. Because of the increased height of stack necessary to neutralize

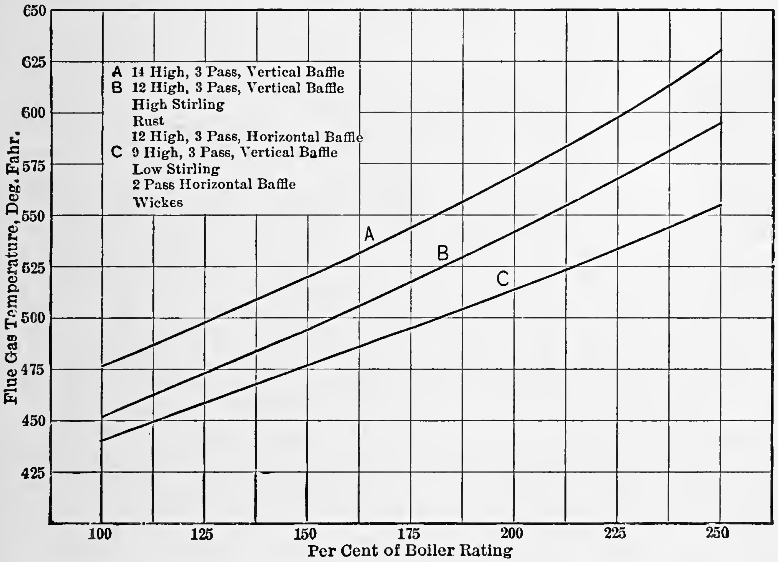


Fig. 148. Relation between Flue Gas Temperature and Increase in Boiler Rating. Natural Draft.

the reduction in stack temperatures economizer installations are commonly made with forced or induced draft.

As soon as a flow is established the static draft will decrease since part of this potential energy is required to impart velocity to the gases and overcome the resistance of the chimney walls. Furthermore, the breeching, boiler damper, baffles and tubes, and the bed and grate all retard the passage of the gases and the draft from the chimney is required to overcome these resistances. If an economizer is used this adds a further pressure drop. (See paragraph 285.) Neglecting leakage and minor influences, the various pressure losses may be expressed:

$$fD = D_g + D_b + D_v + D_d + F_f + D_c + D_r, \tag{69}$$

in which f is an empirical coefficient depending largely on the rate of cooling gases within the chimney, D is the maximum theoretical static draft, D_g the pressure drop through the fuel and grate necessary to effect the desired rate of combustion, D_b the drop through the boiler, D_v the draft required to impart velocity up to the damper, and D_d, D_f, D_c, D_r, the respective draft losses through the damper, flue, chimney, and right angle turns into the breeching. Transposing equation (69) we have

$$D_g + D_b + D_v + D_d = fD - D_c - D_f - D_r. \qquad (70)$$

$D_g + D_b + D_v + D_d$ is the *draft required at the stack side of the damper.* $fD - D_c$ is the *effective draft of the chimney and* $fD - D_c - D_f - D_r$ is the *available draft at the stack side of the damper.*

All these losses increase approximately with the square of the velocity of flow and may be expressed mathematically, but owing to extreme diversity in operating conditions many of the factors entering into the analysis can only be approximated, with the ultimate result that the calculated values are more or less arbitrary. Considering the losses in the order given in equation (69):

D, the total or maximum static draft, may be calculated from equation (66). The limitations of this formula have been previously shown.

D_g, the draft required to effect a given rate of combustion, depends upon the kind and condition of fuel, the thickness of fire, type of grate and efficiency of combustion and can only be found accurately by experiment. For every kind of fuel and rate of combustion there is a certain draft with which the best general results are obtained.

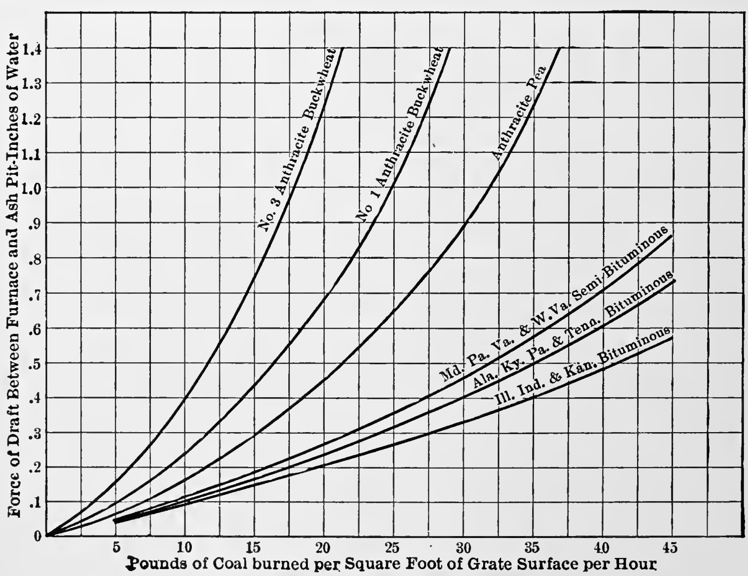


FIG. 149. Draft Required at Different Combustion Rates for Various Fuels.

The curves in Fig. (149)* give the furnace draft necessary to burn various kinds of coal at the indicated rate of combustion for average operating conditions. These curves allow a safe margin for economically burning coals of the kinds noted. Specific figures for various types of stokers may be obtained from the manufacturers.

TABLE 53.

AVERAGE PRESSURE DROP THROUGH BOILER AND SETTING.

(Boilers Operating at 100 to 150 Per Cent Rating.)

Boiler.	K.
Atlas, horizontal 2 pass, standard setting	25
Atlas, vertical 3 pass, standard setting	40
Babcock and Wilcox, Sewall baffle	35
Babcock and Wilcox, vertical 3 pass, standard setting	45
Cahall, standard baffling and setting	35
Continental, Dutch oven	35
Edge Moor, vertical 4 pass, underground breeching	36
Erie City, vertical boiler, standard baffling	40
Hawkes, horizontal baffles, standard setting	40
Keeler, vertical 3 pass, tube spacing 6×6 to 6×7	50
Keeler, vertical 3 pass, tube spacing $5\frac{1}{4} \times 7\frac{3}{4}$ to $5\frac{7}{8} \times 6\frac{3}{4}$	45
Return tubular, old style setting	44
Return tubular, double arch bridge wall setting	53
Return tubular, McGinnis arches, front and back	40
Stirling, standard setting and baffles	35
Stirling, 5 pass	19
Stirling, standard baffles, underground breeching	14
Wickes, standard setting and baffles	42
Worthington, standard setting and baffles	45

K = Percentage of the effective draft at the stack side of damper available in the combustion chamber. This factor applies only to hand-fired furnaces burning 20 to 30 pounds of Illinois coal per square foot of grate surface per hour and to mechanical stokers of the natural draft type burning 20 to 40 pounds per hour. Effective draft $= \dfrac{\text{Draft over fire} \times 100}{K}$.

D_b, the loss of draft through the boiler and setting, varies within wide limits depending upon the type and size of boiler, arrangement of tubes and baffles, design of setting, type of grate, and rate of driving, and ranges from less than 0.1 inch to 1.0 inch and over. The data given in Table 53 † are based upon the investigations of O. Monnett, former Chief Smoke Inspector of the City of Chicago, and may be used as a guide in predetermining the extent of these losses for different types of boilers and settings. The figures in the table apply to hand-fired grates having an air space of 45 to 55 per cent and rates of combustion ranging from 20 to 30 pounds of Illinois coal per square foot of grate surface. They also apply to mechanical stokers of the natu-

* "Steam"; Babcock & Wilcox Co., p. 246.

† Power, June 2, 1914, p. 768.

ral draft type, burning 20 to 40 pounds of coal per square foot of grate surface with the capacities in either case ranging from rating to 50 per cent overload. The relative pressure drop increases with the load but there appears to be no close relationship between those two

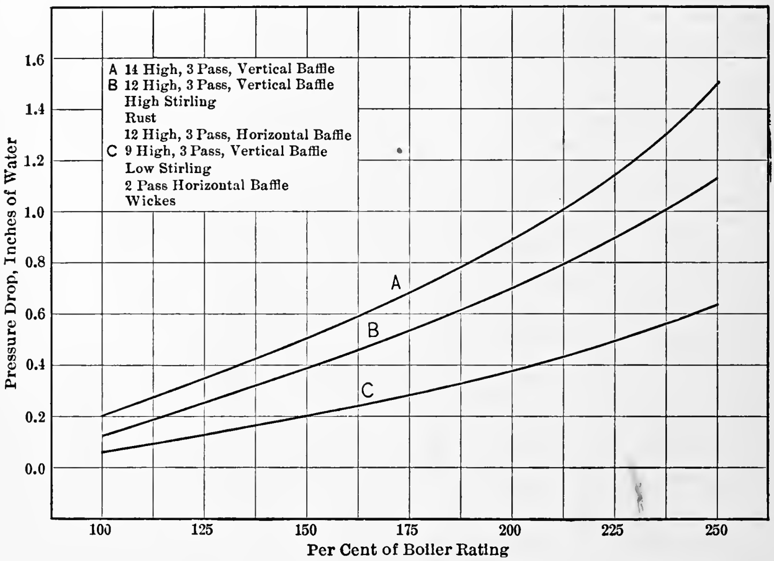


FIG. 150. Pressure Drop through Boilers — Furnace to Stack Damper — Natural Draft.

factors for different boiler equipments. Specific figures may be obtained from boiler manufacturers. The curves in Fig. 151 are plotted from a series of tests conducted by A. P. Kratz, on a Babcock & Wilcox boiler located in the new power plant of the University of Illinois. (A Study of Boiler Losses. A. P. Kratz, Bul. 32, University of Illinois, April 12, 1915.) The futility of assuming an "average value for general practice" is evidenced from the extreme range even in this particular installation.

D_v, the draft required to accelerate the gases, varies in accordance with the law

$$h = \frac{V_1^2 - V_2^2}{2\,g}, \tag{71}$$

in which

h = head in feet of gas producing the velocity,
V_2 = initial velocity, ft. per sec.,
V_1 = final velocity, ft. per sec.,
g = acceleration of gravity = 32.2 (approx.).

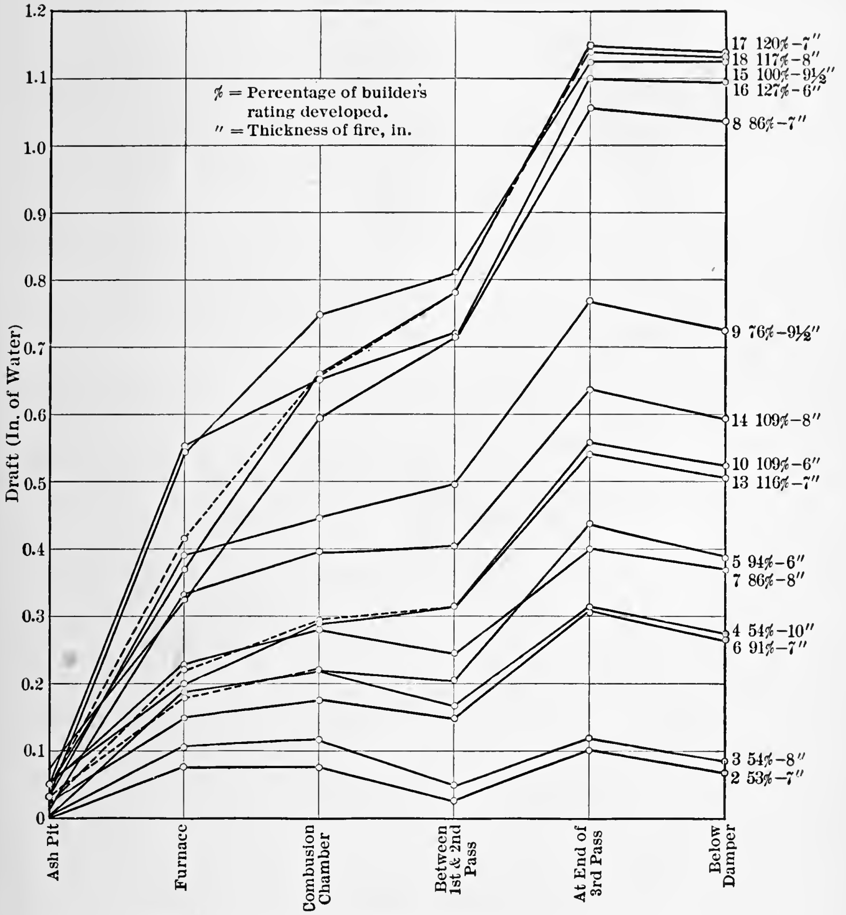


FIG. 151. Pressure Drop through Boiler Setting. 508 Horsepower B. & W. Boiler, 3-Pass Vertical with Tile-roof Furnace. — Chain Grate.

Assuming a gas density of 0.085 lb. per cubic foot at 32 deg. fahr. and 14.7 lb. per sq. in. pressure, and reducing head in feet of gas to pressure in inches of water, equation (71) reduces to

$$D_v = 0.124 \frac{P_a}{P}\left(\frac{V_1^2 - V_2^2}{T_c}\right), \tag{72}$$

in which

P_a = observed barometric pressure, lb. per sq. in.,
P = one standard atmosphere = 14.7 lb. per sq. in.,
T_c = absolute temperature of the chimney gases, deg. fahr.

The draft required tc accelerate at sea level from zero velocity to velocities 10, 20, 30, and 40 feet per second at a temperature of 550 deg. fahr. is 0.012 in., 0.048 in., 0.108 in., and 0.192 in., respectively. Except at high velocities the draft is small and may be neglected in the problem of chimney design.

D_d, the loss of draft through the damper, is varied arbitrarily to meet the load requirements. The minimum value of D_d corresponding to "wide open damper" is usually included in the boiler loss D_b. For the influence of damper area on the draft in fire-tube boilers, see "Draft in Fire Tube Boilers," S. H. Viall, Power, April 11, 1916, p. 509. See also, "Dampers for Water-Tube Boilers," Osborn Monnett, Power, May 26, 1914, p. 729.

The commonly accepted rules for determining the friction loss D_c through the chimney are all based on Chezy's formula

$$h = f\,\frac{V_2}{2\,g}\cdot\frac{l}{m}, \tag{73}$$

in which

f = coefficient of friction,
l = length of the conduit, ft.,
m = mean hydraulic radius, ft.,
V = mean velocity of the gases, ft. per sec.
Other notations as in equation (71).

Because of the great variation in the assumed value of the coefficient of friction, the rules referred to give widely discordant results for the same set of conditions. Satisfactory results have been obtained by assuming $f = 0.012$ for unlined steel stacks and $f = 0.016$ for brick and brick-lined steel stacks. Until further experiments prove to the contrary these values may be accepted as being as accurate as any used in this connection.

For circular chimneys, and square chimneys, D ft. square, Chezy's formula may be reduced to the convenient form

$$D_c = K\,\frac{V^2H}{DT_c}, \tag{74}$$

in which

D_c = friction loss in inches of water,
K = coefficient including the coefficient of friction and the various reduction factors,
= 0.006 for unlined steel stack, 0.008 for brick or lined steel stack,
V = velocity of the gases, ft. per sec.,
H = height of stack above the breeching, ft.,
D = diameter of the stack, ft.,
T_c = absolute temperature of the chimney gases.

Considering the weight instead of the velocity of flow, equation (74) reduces to the form

$$D_c = k\,\frac{W^2HT_c}{d^5}, \tag{75}$$

in which

k is a coefficient including the coefficient of friction and the various reduction constants,
W = weight of gas flowing, lb. per sec.,
d = diameter of the stack, inches.

For the assumed values of f (0.012 and 0.016):

k = 1.6 for unlined steel stacks, 1.9 for brick or brick-lined stacks.

C. R. Weymouth, Trans. A.S.M.E., Vol. 34, 1912, gives k a value of 2.3 for lined and unlined stacks.

According to Kingsley's experiments * the loss of draft due to skin friction, displacement of the atmosphere by the issuing stream and change of direction of the gases upon entering the stack conforms approximately to the following equation

$$D_c = 0.00036\ V^2. \tag{76}$$

Notations as in equation (74).

Equations (74) and (75) may be used for determining D_f, the loss in the flue or breeching, by substituting the length of the flue for H and the diameter for d. A common allowance for the friction drop in flues, round or square, is 0.1 inch of water per 100 feet of straight conduit.

D_r, the draft resistance due to right-angle turns, is ordinarily taken as 0.05 inch of water per turn. Another rule is to assume this resistance to be equivalent to a length of flue twelve diameters in length.

An examination of equations (73) to (76) will show that the friction draft loss of the chimney cannot be calculated directly unless the height and diameter and weight or velocity of flow are known. Since these are the quantities to be determined it is evident that the problem lends itself only to be a "cut and trial analysis," provided the equations are to be satisfied. If the various pressure drops influencing the height of the stack could be calculated or estimated with any degree of accuracy there would be some reason for exact analysis, but the arbitrary values assigned in practice vary so widely that such analyses are ordinarily without purpose. Furthermore, the friction loss through the chimney is only a comparatively small percentage of the total loss (except for high velocities), hence a careful calculation of the chim-

* Engineering Record, Dec. 21, 1907, p. 679.

ney friction, and guess-work in estimating the other losses is highly inconsistent. Scattering tests made on a number of tall chimneys in successful operation show that the *effective* pressure at 100 to 150 per cent rating is not far from 80 per cent of the theoretical maximum static pressure. Assuming this to hold true for chimneys in general, the problem of determining the height becomes a comparatively simple one. In view of the uncertainty of the coefficient of friction, results based upon this assumption are perhaps fully as reliable as those calculated from the various formulas.

Example 17. Determine the height of a stack suitable for burning 30 pounds of Illinois bituminous coal per sq. ft. of grate surface per hour for a hand-fired, return tubular boiler with double-arch bridge wall furnace, when the temperature of the outside air is 60 deg. fahr., the mean temperature of the flue gases 550 deg. fahr., and the flue is 100 feet long with two right-angle bends.

This loss will be approximately as follows:

Loss through fuel and grate (from curves in Fig. 149)	0.33
Loss in boiler (from Table 53), $\frac{0.33}{0.53}$	0.62
Loss in flue, 100 ft. at 0.10 per 100	0.10
Loss in turns, 2×0.05	0.10
Total required draft at the breeching entrance to the stack	1.15

On the assumption that the effective or required draft is 80 per cent of the theoretical maximum static draft,

$$D = \frac{1.15}{0.8} = 1.43.$$

And from equation (67)

$$1.43 = H\left(\frac{7.64}{520} - \frac{7.95}{1010}\right),$$

from which

$$H = 210 \text{ feet, height above damper.}$$

The vertical passes in any boiler act as chimneys and are capable of furnishing a draft pressure in much the same manner as the chimney proper. The greater the length of the vertical passes the greater will be the "chimney action." The pressure difference due to the chimney action may decrease or increase the draft of the stack, depending upon the direction of flow of the gases. If the flow is upward the vertical pass acts as an additional height of stack, if downward, it tends to retard the flow. Thus, in the Wickes boiler, Fig. 56, the vertical path of the gases through the boiler itself causes considerable chimney action. At low rating the pressure at C may be atmospheric or even slightly above, although the draft in the combustion chamber B may be 0.10 inch of water below that of the atmosphere. This means that the boiler itself furnishes sufficient chimney action to operate the boiler at this load. Similarly the draft at D may be higher than at C due to the negative chimney action and resistance combined. The difference in temperature of the gases due to the cooling action of the heating surface must of course be considered in calculating the chimney action. In practically all boilers the chimney action of the vertical passes influences the pressure drop throughout the setting and the effect is more marked when the rate of flow is low. See "Draft in Furnaces and Flues," E. G. Bailey, Power, Nov. 9, 1915, p. 638.

A well-designed central chimney serving several boilers and subject to considerable load variation should have comparatively low stack and breeching friction in order to insure "draft regulation." While a certain draft margin is necessary it should be the aim to provide a chimney with the least possible excess draft over the necessary maximum. For very high stacks, such as are required in tall office buildings, the diameter is made very small so that a considerable portion of the pressure drop will occur in the stack and breeching, otherwise the draft will be excessive even with throttled damper. In designing stacks for this purpose the assumed draft loss in the stack and breeching should be made to conform with the law expressed in equation (75).

Boiler Draft: Power, Mar. 20, 1917, p. 374; April 11, 1916, p. 509; Nov. 9, 1915, p. 638; Aug. 10, 1915, p. 196; May 18, 1915, p. 675; Jan. 12, 1915, p. 39; July 7, 1914, p. 7; June 9, 1914, p. 806.

The Significance of Drafts in Steam Boiler Practice: Bulletin 21; U. S. Bureau of Mines, 1911.

Proportioning Chimneys on a Gas Basis: A. L. Menzin, Jour. A.S.M.E., Jan. 1916, p. 31.

Dimensions of Boiler Chimneys for Crude Oil: C. R. Weymouth, Trans. A.S.M.E., Vol. 34, 1912.

Calculating the Dimensions of Chimneys and Stacks: G. A. Orrok, Power, Aug. 22, 1916, p. 274; Sept. 12, 1916, p. 384.

128. Chimney Area. — A study of equation (75) will show that any required effective draft may be obtained from various combinations of heights and diameters. Evidently there must be a certain height and diameter which will produce the cheapest structure. In practice this particular combination cannot be predetermined with any degree of accuracy because of the uncertainty of the various factors entering into the problem of calculating the height and diameters. For an assumed set of conditions the logical procedure is to calculate a trial height for the required maximum rate of combustion, and then to proportion the area according to equation (75) so that the maximum weight of gases generated may be discharged at a rate corresponding to the assumed friction loss through the stack. By cut and trial a number of combinations of heights and diameters may be calculated in this manner which will give the required effective draft. The costs of the various structures may then be estimated and a selection made.

In general practice this degree of refinement is seldom attempted and the usual procedure is to calculate a height compatible with the assumed pressure losses (subject, of course, to community laws) and proportion the area by rules which are more or less empirical. Thus, if the area is to be proportioned on a gas basis the maximum volume of the gases to be discharged is computed and an arbitrary velocity is assumed.

If specific data are not available for computing the volume of the gases the area may be calculated by one of the various empirical equations outlined in Table 56.

Example 18: Proportion a brick-lined stack for water-tube boilers (vertical three-pass standard baffling) rated at 6000 horsepower, equipped with chain grates and burning Illinois coal; boilers rated at 10 square feet of heating surface per horsepower; ratio of heating surface to grate surface, 50 to 1; flue 100 feet long with two right-angle bends; stack to be able to carry 50 per cent overload; atmospheric temperature 60 deg. fahr.; sea level; temperature of flue gases at overload 540 deg. fahr.; calorific value of the coal 11,200 B.t.u. per pound.

A modern plant of this type and size should be able to maintain a combined boiler, furnace and grate efficiency of 75 per cent at 150 per cent rating. To be on the safe side assume it to be 70 per cent, then

Maximum boiler horsepower $= 6000 \times 1.5 = 9000.$

Heat equivalent of 1 boiler horsepower-hr. $= 34.5 \times 970 = 33{,}479$ B.t.u.

$$\text{Coal per boiler hp-hr.} = \frac{33{,}479}{11{,}200 \times 0.70} = 4.3 \text{ lb. approx.}$$

$$\text{Total grate surface} = \frac{6000 \times 10}{50} = 1200 \text{ sq. ft.}$$

Total coal burned per hour $= 4.3 \times 9000 = 38{,}700$ lb.

$$\text{Maximum rate of combustion} = \frac{38{,}700}{1200} = 32.3 \text{ lb. per sq. ft. grate surface per hour.}$$

Assumed pressure losses at maximum rating:

	Inches of Water.
Loss through fuel and grate (from curves in Fig. 149)	0.34
Loss in boiler (furnace to stack side of damper)	0.55
Loss in flue 100 ft. at 0.1 in. per 100	0.10
Loss in turns, 2 × 0.05	0.10
Total loss or required effective pressure measured at flue entrance of stack	1.09

$$\text{Theoretical draft} = \frac{1.09}{0.8} = 1.36 \text{ in.}$$

Height of stack above damper, equation (67),

$$1.36 = \left(\frac{7.64}{520} - \frac{7.95}{1000}\right) H.$$
$$H = 202 \text{ ft.}$$

For 70 per cent combined efficiency the air excess with chain grate and Illinois coal may range from 50 to 75 per cent. To take care of possible reduction in efficiency, leakage and other adverse influences assume a total air excess of 100 per cent.

Theoretical air per 10,000 B.t.u. = 7.5 lb. (See Table 13.)

$$\text{Theoretical air per lb. of coal} = 7.5 \frac{11{,}200}{10{,}000} = 8.4 \text{ lb.}$$

Actual air per lb. of coal = 8.4 × 2 = 16.8 lb.

Probable weight of flue gas per lb. of coal = 17.5 lb.

(If the ultimate analysis of the coal is known the weight of the products of combustion may be calculated as shown in paragraph (22). If the per cent of CO_2 is assured this quantity may be calculated or it may be taken directly from Table 54.)

$$\text{Weight of flue gas} = \frac{17.5 \times 38{,}700}{3600} = 188 \text{ pounds per sec.}$$

$$\text{Total volume of flue gas } \frac{188}{0.0418} = 4500 \text{ cu. ft. per sec.}$$

(The density of the flue gas varies considerably with the nature of the fuel and the air excess.)

TABLE 54.

WEIGHT OF GASES FOR DIFFERENT PERCENTAGE OF CO_2 WHEN CO = O.

Per cent CO_2 in the dry gases by volume	18.7	18.0	17.0	16.0	15.0	14.0	13.0	12.0
Excess air in per cent of the theoretical minimum	0	4.0	10.0	17.0	24.0	33.0	43.0	54.0
Weight of gases per 10,000 B.t.u.	7.8	8.1	8.6	9.1	9.6	10.3	11.0	11.9
Per cent of CO_2 in the dry gases by volume	....	11.0	10.0	9.0	8.0	7.0	6.0	5.0
Excess air in per cent of the theoretical minimum	....	68.0	85.0	105.0	130.0	162.0	206.0	267.0
Weight of gases per 10,000 B.t.u. in the coal	....	12.9	14.2	15.7	17.6	20.0	23.3	27.8

TABLE 55.

AVERAGE VELOCITY OF CHIMNEY GASES.

Volume of chimney gases discharged, cu. ft. per sec.	10	100	500	2500	5000	8000	12,000
Average velocity at maximum load, ft. per sec.	10	15	20	25	30	35	40

These values are based upon data compiled from 200 modern chimney installations of various heights and diameters. There appeared to be no definite relationship between volume and velocity and the values in the table represent gross averages only.

Assume 30 ft. per sec. as the average velocity of the gases. (See Table 55.)

$$\text{Area} = \frac{4500}{30} = 150 \text{ sq. ft.}$$

Corresponding diameter = 13.8 ft. or 165 inches.

It will be noted that numerous assumptions have been made in the foregoing analysis, consequently the reliability of the results depends entirely upon the accuracy of these assumptions. Because of the possible variation in practice of these assumed values, and because in many situations they cannot be approximated with any degree of accuracy, many engineers prefer to proportion the area on such empirical equations as (5) and (12), Table 56.

Thus, Kent's rule, equation (5), gives

$$\text{Effective area} = \frac{0.3 \times 9000}{\sqrt{202}} \times 0.86 = 163 \text{ sq. ft.}$$

Corresponding actual diameter = 177 inches.

Kent's equation is based on a coal consumption of 5 lb. per boiler horsepower-hour, therefore $4.3 \div 5.0 = 0.86$ is the correction factor for the given conditions, hence the effective area as calculated from Kent's equation should be multiplied by 0.86.

According to equation (12), Table 56,

$$\begin{aligned} D &= 4.92 \text{ hp.}^{0.4} \\ &= 4.92 \times 6000^{0.4} \\ &= 160 \text{ inches.} \end{aligned}$$

For small hand-fired plants it is sufficiently accurate to adopt the following proportions:

Internal area of the chimney, one-fifth to one-sixth of the connected grate area for bituminous coal and one-seventh of the grate area for anthracite.

The following heights have been found to give good results in plants of moderate size:

	Feet
With free-burning bituminous coal	90
With anthracite, medium and large sizes	120
With slow burning bituminous	140
With anthracite pea	150
With anthracite buckwheat	175
With anthracite slack	200

For plants of 800 horsepower or more the height of stack for coal burning should never be less than 150 feet, regardless of the kind of coal used. Natural draft greater than 1.5. in. of water is seldom necessary and higher intensities can be obtained much better by forced or induced draft. This limits the height of chimney to about 225 to 250 ft.

In proportioning the area of the stack on a gas basis the data in Tables

54 and 55 may be used as a guide. By plotting the data compiled from a number of modern chimneys the relation between velocity and area appeared to be approximately as follows:

$$V = (0.2 + 0.005\,D)\,V', \tag{77}$$

in which

V = average actual maximum velocity of the chimney gases, ft. per sec.,

D = diameter of the chimney, ft.,

V' = theoretical velocity, ft. per sec., assuming that the total theoretical draft is available for producing velocity.

129. Empirical Chimney Equations. — The various empirical formulas outlined in Table 56 are occasionally used in proportioning chimneys. They give good results within the limits of the assumptions upon which they are based, but otherwise may lead to absurd results, their applicability depending largely upon the available data covering the various losses with the particular kind, quality, and condition of coal, and conditions of operation. Occasionally practical and local considerations fix the height of the stack irrespective of theoretical deductions.

Referring to Table 56, equations (1), (2), (6), (7), and (9) are based upon a fuel consumption of 13 to 15 pounds of anthracite and 22 to 26 pounds of bituminous coal per square foot of grate area per hour. In equations (3), (4), and (9), the diameter is dependent solely upon the quantity of coal burned per hour and the height is determined mainly by the rate of combustion per square foot of grate. The results accord well with practice. With western coals equation (3) gives results rather too large and the constant should be 120 instead of 180. Equation (5) is perhaps the most used and has met with much approval. It is based on the assumptions that:

1. The draft of the chimney varies as the square root of the height.
2. The retardation of the ascending gases by friction may be considered due to a diminution of the area of the chimney or to a lining of the chimney by a layer of gas which has no velocity and the thickness of which is assumed to be 2 inches. Thus, for square chimneys,

$$E = D^2 - \frac{8\,D}{12} = A - \frac{2}{3}\sqrt{A}, \tag{78}$$

and for round chimneys,

$$E = \frac{\pi}{4}\left(D^2 - \frac{8\,D}{12}\right) = A - 0.591\,\sqrt{A}. \tag{79}$$

For simplifying calculations the coefficient of $\sqrt{A}$ may be taken as 0.6 for both square and round chimneys, and the equation becomes

$$E = A - 0.6\,\sqrt{A}. \tag{80}$$

3. The horsepower capacity varies as the effective area E.

4. A chimney should be proportioned so as to be capable of giving sufficient draft to permit the boiler to develop much more than its rated power in case of emergencies or to permit the combustion of 5 pounds of fuel per rated horsepower per hour.

5. Since the power of the chimney varies directly as the effective area E and as the square root of the height H, the equation for horsepower for a given size of chimney will take the form

$$\text{Hp.} = CE\sqrt{H}, \tag{81}$$

in which C is a constant, found by Mr. Kent to be 3.33, obtained by plotting the results from numerous examples in practice.

The equation then assumes the form

$$\text{Hp.} = 3.33\,E\sqrt{H}, \tag{82}$$

or

$$\text{Hp.} = 3.33\,(A - 0.6\sqrt{A})\sqrt{H}, \tag{83}$$

from which

$$H = \left(\frac{0.3\ \text{Hp.}}{E}\right)^2. \tag{84}$$

Table 57 has been computed from equation 5, Table 56.

130. Stacks for Oil Fuel. — In designing stacks for oil fuel or gas firing the procedure is the same as for coal burning, that is, the height is made sufficiently great to maintain the required draft in the furnace at maximum overload and the area is proportioned to take care of the maximum volume of gases generated. Excessive draft greatly influences the economy of oil-fired furnaces, whereas with coal firing there is rarely danger of too much draft. Consequently greater care must be exercised in estimating the various draft losses through the boiler and breeching. With oil fuel there is practically no loss of draft through the fuel bed and grate and the pressure loss through the boiler will be less because of the smaller volume of gases discharged per boiler horsepower hour. Furthermore, the action of the burner itself acts to a certain degree as a forced draft. Therefore, both the height and area of the stack for a given capacity of boiler will be less for oil-firing than for coal-firing. Table 58 calculated by C. R. Weymouth (Trans. A.S. M.E., Vol. 34, 1912) after an exhaustive study of data pertaining to the subject may be used as a guide in proportioning stacks for oil fuel.

131. Classification of Chimneys. — Chimneys may be grouped into three classes according to the material of construction:

1. Steel.
2. Reinforced concrete.
3. Masonry.

TABLE 56.—CHIMNEY EQUATIONS.

Index	Author.	References.	Formulas. Horse Power.	Formulas. Height, Feet.	Area, Square Feet.
1	Adams.	Adams, "Handbook for Mechanical Engineers," p. 155.	....	$H=\left(\frac{F}{14\,A}\right)^2.$	$A=\frac{F}{14\sqrt{H}}.$
2	Christie.	Christie, "Chimney Design," p. 22.	$HP=3.24\,A\sqrt{H}.$	$H=\left(\frac{F}{KA}\right)^2.$	$A=\frac{F}{K\sqrt{H}}.$
3	Gale..........	Trans. A.S.M.E., Vol. XI, p. 463.	....	$H=\frac{180}{t}\left(\frac{F}{G}\right)^2.$	$A=0.07\,F^{\frac{4}{5}}.$
4	"Ingenieurs Taschenbuch"	....	....	$H=0.216\left(\frac{F}{G}\right)^2+6\,D.$	....
5	Kent..........	Trans. A.S.M.E., Vol. XII, p. 81.	$HP=3.33\,E\sqrt{H}.$	$H=\frac{\overline{0.3\,HP}^2}{E^2}.$	$E=\frac{0.3\,HP}{\sqrt{H}}.$
6	Lange........	Eng. Record, July 20, 1901, p. 52.	....	$H=15\,D+32.5.$	$A=0.00049\,BF.$
7	Molesworth...	Molesworth, "Pocket Book"...	$HP=1.28\,A\sqrt{H}.$	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{F}{12\sqrt{H}}.$
8	Nagle........	Power, November, 1902, p. 29...	$HP=2\,D^2\sqrt{H}.$	$H=\frac{\overline{HP}^2}{4\,D^4}.$	$D=\left(\frac{HP}{2\sqrt{H}}\right)^{\frac{1}{2}}.$
9	Nystrum.	Nystrum, "Mechanics," ed. 1882, p. 423.	$HP=1.45\,A\sqrt{H}.$	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{HP}{1.45\sqrt{H}}.$
10	Rankine......	Christie, "Chimney Design," p. 9.	....	$H=\frac{13\,\frac{V}{2\,g}\left(\frac{w}{n\delta}\right)^2\left(\frac{T_1}{T_0}\right)^2}{0.96\,\frac{T_1}{T_2}-1-\frac{0.06\,D}{2\,gA}\left(V^2\,\frac{wT_1}{n\delta T_0}\right)^2}.$	....
11	Smith........	Smith, "Boiler Practice," p. 423.	....	$H=\left(\frac{F}{12\,A}\right)^2.$	$A=\frac{0.0825\,F}{\sqrt{H}}.$
12	Stirling.......	1905 issue of "Stirling," Stirling Boiler Company.	....	$H=\frac{d'}{0.8\,M}.$	$D'=C\,(HP)^{0.4}.$

H. Height above grate; ft.
A. Inside area at top; sq. ft.
E. Effective area $=A-0.6\sqrt{A}$.
D. Inside diameter at top; ft.
HP. Boiler horse power.
F. Lbs. coal burned per hour.

G. Grate surface; sq. ft.
d'. Net available draft; in. H_2O.

$$K=\begin{cases}\frac{F}{1.89\,G} & \text{for bituminous coal.}\\ \frac{F}{G} & \text{for anthracite.}\end{cases}$$

$$C=\begin{cases}4.68 \text{ for unlined stack.}\\ 4.92 \text{ for lined stack.}\end{cases}$$

t. Temperature flue gas; deg. *F*.
M. Theoretical draft per foot of chimney height.
B. Weight of air per lb. of coal.
V. Cu. ft. air per lb. of coal, 32° *F*.

w. Lb. fuel burned per second.
n. Ratio grate to chimney area.
δ. Weight of 1 cu. ft. chimney gas, 32° *F*.
T_2. Temperature external air; deg. *F*., abs.
T_1. Temperature flue gas; deg. *F*., abs.
T_0. = 461.
D' = Diameter in inches.

TABLE 57.

SIZE OF CHIMNEYS FOR STEAM BOILERS.

Kent's Formula.

Diam. Inches.	Area (a) Sq. Ft.	Effective Area $E=A-0.6\sqrt{A}$ Sq. Ft.	Height of Chimney.														Equivalent Sq. Chimney Side of Sq. $\sqrt{E}+4$ Inches.
			50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	225 ft.	250 ft.	300 ft.	
			Commercial Horse Power of Boiler.*														
18	1.77	.97	23	25	27	29	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	16
21	2.41	1.47	35	38	41	44	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	19
24	3.14	2.08	49	54	58	62	66	.....	.....	.....	.....	.....	.....	.....	.....	.....	22
27	3.98	2.78	65	72	78	83	88	.....	.....	.....	.....	.....	.....	.....	.....	.....	24
30	4.91	3.58	84	92	100	107	113	119	.....	.....	.....	.....	.....	.....	.....	.....	27
33	5.94	4.48	.....	115	125	133	141	149	156	.....	.....	.....	.....	.....	.....	.....	30
36	7.07	5.47	.....	141	152	163	173	182	191	204	.....	.....	.....	.....	.....	.....	32
39	8.30	6.57	.....	.....	183	196	208	219	229	245	.....	.....	.....	.....	.....	.....	35
42	9.62	7.76	.....	.....	216	231	245	258	271	289	316	.....	.....	.....	.....	.....	38
48	12.57	10.44	.....	.....	.....	311	330	348	365	389	426	.....	.....	.....	.....	.....	43
54	15.90	13.51	.....	.....	.....	.....	427	449	472	503	551	595	.....	.....	.....	.....	48
60	19.64	16.98	.....	.....	.....	.....	536	565	593	632	692	748	.....	.....	.....	.....	54
66	23.76	20.83	.....	.....	.....	.....	.....	694	728	776	849	918	981	.....	.....	.....	59
72	28.27	25.08	.....	.....	.....	.....	.....	835	876	934	1,023	1,105	1,181	1,253	.....	.....	64
78	33.18	29.73	.....	.....	.....	.....	.....	.....	1,038	1,107	1,212	1,310	1,400	1,485	1,565	.....	70
84	38.48	34.76	.....	.....	.....	.....	.....	.....	1,214	1,294	1,418	1,531	1,637	1,736	1,830	2,005	75
90	44.18	40.19	.....	.....	.....	.....	.....	.....	.....	1,496	1,639	1,770	1,893	2,008	2,116	2,318	80
96	50.27	46.01	.....	.....	.....	.....	.....	.....	.....	1,712	1,876	2,027	2,167	2,298	2,423	2,654	86
102	56.75	52.23	.....	.....	.....	.....	.....	.....	.....	1,944	2,130	2,300	2,459	2,609	2,750	3,012	91
108	63.62	58.83	.....	.....	.....	.....	.....	.....	.....	2,090	2,399	2,592	2,771	2,939	3,098	3,393	96
114	70.88	65.83	.....	.....	.....	.....	.....	.....	.....	.....	2,685	2,900	3,100	3,288	3,466	3,797	101
120	78.54	73.22	.....	.....	.....	.....	.....	.....	.....	.....	2,986	3,226	3,448	3,657	3,855	4,223	107
132	95.03	89.18	.....	.....	.....	.....	.....	.....	.....	.....	3,637	3,929	4,200	4,455	4,696	5,144	117
144	113.10	106.72	.....	.....	.....	.....	.....	.....	.....	.....	4,352	4,701	5,026	5,331	5,618	6,155	128

* Based on a consumption of 5 pounds of fuel per boiler horse power. For any other rate multiply the tabular figure by the ratio of 5 to the maximum expected coal consumption per horse power per hour.

Steel chimneys have many advantages and are finding much favor in large power plants, especially where economy of space warrants the erection of the stack over the boiler, in which case the structural work of the boiler setting answers for both boiler and chimney. Among the advantages over the masonry construction are: (1) ease and rapidity of construction; (2) less weight for a given internal diameter and height; (3) less surface exposed to the wind; (4) lower cost; (5) smaller space required; (6) slightly higher efficiency if properly calked, for there can be no infiltration of cold air as is likely through the cracks in masonry. The chief disadvantage is the cost of keeping the stack well painted to prevent rust and the corrosive action of the sulphur in the coal.

TABLE 58.

STACK SIZES FOR OIL FUEL.

Stack Diameter, Inches.	Height in Feet Above Boiler-room Floor.					
	80	90	100	120	140	160
33	161	206	233	270	306	315
36	208	253	295	331	363	387
39	251	303	343	399	488	467
42	295	359	403	474	521	557
48	399	486	551	645	713	760
54	519	634	720	847	933	1000
60	657	800	913	1073	1193	1280
66	813	993	1133	1333	1480	1593
72	980	1206	1373	1620	1807	1940
84	1373	1587	1933	2293	2560	2767
96	1833	2260	2587	3087	3453	3740
108	2367	2920	3347	4000	4483	4867
120	3060	3660	4207	5040	5660	6160

Figures represent nominal rated horsepower; sizes as given are good for 50 per cent overloads. Based on centrally located stacks, short direct flues and ordinary operating efficiencies.

Steel chimneys may be:

1. Guyed.
2. Self-sustained.

132. Guyed Chimneys. — Guyed sheet-iron or steel chimneys or stacks held in position by guy wires are employed in small sizes on account of their relative cheapness. They seldom exceed 72 inches in diameter and 100 feet in height. A heavy foundation is unnecessary for the smaller sizes and the stack may be supported by the boiler breeching. The small short stacks are ordinarily riveted in the shop, ready for erection, larger sizes being shipped in sections and riveted at the place of installation. In addition to a liberal allowance for corrosion the material is made heavy enough to support its own weight and to prevent buckling under initial tension of the guy wires and the stress due to wind action. The thickness of shell is ordinarily based on arbitrary rules of practice and no attempt is made to calculate this value by stress analysis. Table 59 gives the thickness of material as advocated by a number of manufacturers.

TABLE 59.

APPROXIMATE WEIGHT AND COST OF GUYED SHEET-STEEL CHIMNEYS.

Height, Feet.	Diameter, Inches.	Thickness of Shell, B.W.G.	Approximate Weight per Foot, Pounds.
40	18	16	13
45	20	16	14
45	22	14, 16	20, 15
50	24	14, 16	22, 16
50	26	14	23.5
55	28	14	25
60	30	12, 14	34, 27
65	32	12, 14	36, 28
70	34	10, 12	48, 39
75	36	10, 12	51, 41

Approximate cost per pound, 4 cents to 10 cents, including cost of sections riveted and punched, ready for assembling, the higher figure referring to the smaller stacks.

Guy wires are furnished in one to three sets of three to six strands each and are attached to angle or tee iron bands at suitable points in the height of the stack. The lower ends of the guys are ordinarily anchored at angles of 50 or 60 degrees with the vertical. A rational analysis of the proper size of guy wires for a specified maximum wind pressure is impracticable because of the number of unknown variables entering into the problem, such as initial tension and stretch of the wires and flexure of the shaft. A common rule is to assume the entire overturning load to be resisted by one strand in each set of guys; thus, if there are two sets of guys the entire load is assumed to fall on two wires. An additional stress of one-half the overturning load is allowed for initial tension. A lattice bracing is frequently used between stacks when a number of stacks are placed in a continuous row.

133. Self-sustaining Steel Chimneys. — Steel chimneys over 52 inches in diameter are usually self-supporting. They may be built with or without a brick lining, but the lining is preferred, since it prevents radiation and protects the inside from the corrosive action of the flue

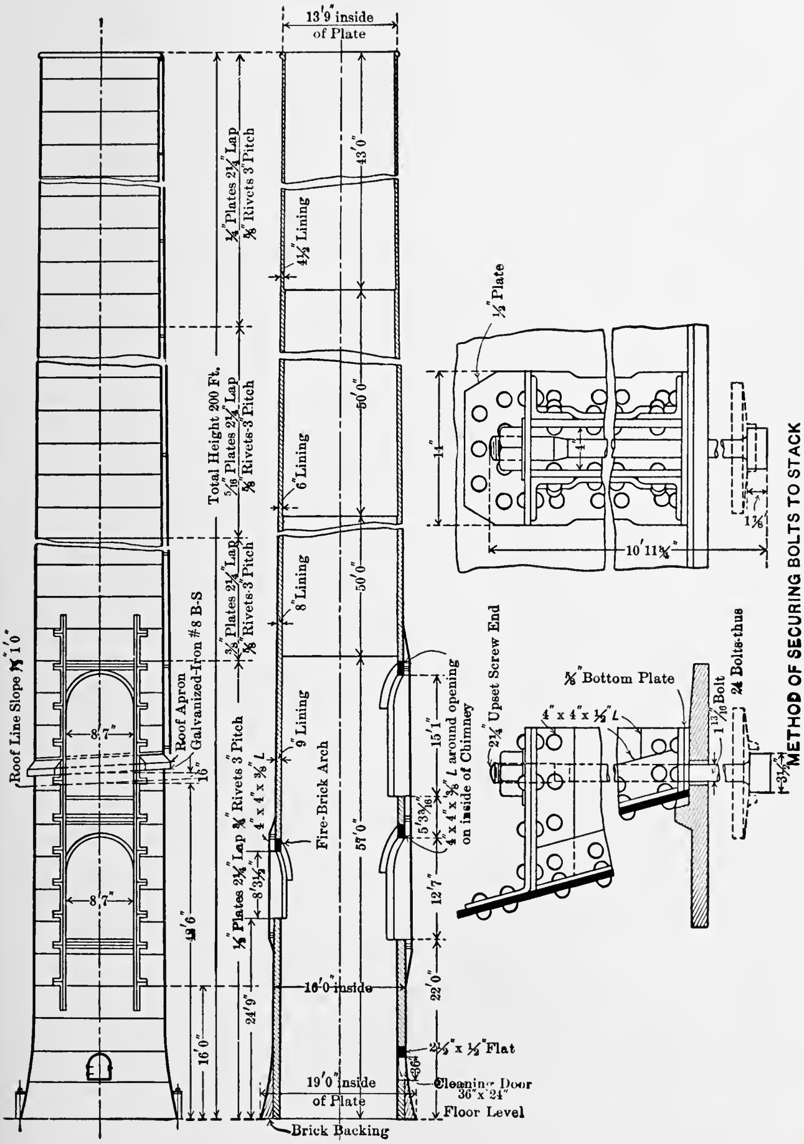


FIG. 152. Steel Chimney of the South Side Elevated Railway Power House, Chicago.

gases. Since the lining plays no part in the strength of the chimney, it is made only thick enough to support its own weight, and usually of a low-grade fire brick or carefully burned common brick or both. In average practice the fire brick extends 20 or 30 feet above the breeching, the remainder of the lining being of common brick. In chimneys up to 80 inches internal diameter, the upper course is $4\frac{1}{2}$ inches thick and increases $4\frac{1}{2}$ inches in thickness for each 30 to 40 feet to the bottom. In larger chimneys about 8 inches is the minimum thickness. The lining is generally set in contact with the shell and thoroughly grouted, otherwise depreciation will be very great.

In several recent designs vertical stiffeners are riveted to the shell which support horizontal rings or shelves on which the lining is built. The vertical stiffeners are spaced about 5 feet apart and the horizontal rings about 20 feet apart. By this method any section of the lining may be replaced without disturbing the rest. The lining is ordinarily of uniform thickness throughout the length of the shaft and seldom exceeds 4 inches in thickness.

Self-sustaining stacks may be straight or tapered, and are generally made with a flared or bell-shaped base whose diameter and length are $1\frac{1}{2}$ to 2 times the internal diameter of the stack. The base is riveted to a heavy cast-iron plate bolted to a concrete foundation of sufficient mass to insure stability. In the modern large station the stack is frequently carried on a steel structure over the boilers, thereby reducing ground space requirements. Such a design is illustrated in Fig. 130.

Fig. 152 gives the details of one of the steel chimneys at the power house of the South Side Elevated Railroad, Chicago, Illinois.

134. Wind Pressure. — Sufficient data are not available to show conclusively the relation between wind velocity and the resulting effective pressure on surfaces of different shapes. Practically all authentic tests have been conducted on small flat surfaces and there is evidence to believe that the unit pressure exerted on large surfaces is somewhat less than that obtained from the former. Experiments conducted by different authorities show that the pressure per square foot of flat surface bears the following relationship to the wind pressure:

$$P = KV^2,$$

in which

K = coefficient determined by experiment,
P = wind pressure, lb. per sq. ft.,
V = wind velocity, miles per hour.

The value of K as determined by the different investigators varies from 0.0029 to 0.005. The most authentic tests give an average value $K = 0.0032$. This corresponds to a pressure of 32 lb. per sq. ft. of flat surface for a wind velocity of 100 miles per hour. Practically all chimneys are proportioned on a maximum wind velocity of 100 miles per hour, but the unit pressure corresponding to this velocity is generally assumed to be 50 lb. per sq. ft. of flat surface. Considering the unit pressure on a flat surface as 1, according to Rankine, the effective pressure for the same projected area is 0.75 for the hexagonal, 0.6 for octagonal, and 0.5 for round columns. Henry Adams, Industrial Engineering, 1912, p. 197, states that these figures are not in accordance with modern experiments and that the factors should be 0.785 for round and 0.82 for octagonal shafts. Current practice allows 25 to 30 lb. per sq. ft. of projected area as the maximum unit pressure on round shafts. That 25 lb. per sq. ft. allows sufficient margin for safety is evidenced by the fact that chimneys proportioned on this basis are successfully withstanding the most violent gales.

135. Thickness of Plates for Self-sustaining Steel Stacks. — If there is no wind blowing the only stress to be considered in the shell at any section is that due to the weight of the material itself, thus:

$$S_1 = W \div \frac{\pi}{4}(d_1^2 - d_2^2), \tag{85}$$

in which

S_1 = stress (compression) due to the weight of the material, lb. per in. If the shaft is in perfect alignment this stress is uniformly distributed over the entire cross section under consideration.

W = weight of the shaft above the section under consideration, lb. If the lining is independent of the steel structure then the weight of the latter only is to be considered, but if the lining is supported by ledges secured to the shaft then the weight of the lining must be added to that of the steel.

d_1 = external diameter of the tube, in.,

d_2 = internal diameter of the tube, in.

When the wind is blowing there is an additional stress due to bending. This is a tension on the windward side and a compression on the leeward side, thus,

$$S_2 = Ph \div \frac{I}{e}, \tag{86}$$

in which

S_2 = stress in the outer fiber due to wind pressure, lb. per sq. in.,

P = the total wind pressure, lb.,

h = distance from the section under consideration to the center of wind pressure, in. For a cylindrical shaft, $h = \frac{1}{2}$ height of shaft above section.

$\frac{I}{e}$ = sectional modulus $= \frac{\pi}{32}\left(\frac{d_1^4 - d_2^4}{d_1}\right)$.

The net stress, S, is therefore

$$S = S_1 \pm S_2 = \frac{W}{\frac{\pi}{4}(d_1^2 - d_2^2)} \pm \frac{Ph}{\frac{\pi}{32}\left(\frac{d_1^4 - d_2^4}{d_1}\right)}. \tag{87}$$

Equation (87) may be written

$$S = \frac{[W(d_1^2 + d_2^2) \div 8] \pm Ph}{\frac{\pi}{32}\left(\frac{d_1^4 - d_2^4}{d_1}\right)}. \tag{88}$$

$(d_1^2 + d_2^2) \div 8$ is commonly called the radius of the statical moment (see paragraph 143). Designating this quantity by q, equation (88) reduces the convenient form

$$S = (Wq \pm Ph) \div \frac{I}{e}. \tag{89}$$

Because of the liberal factor allowed for the safe working stress and because a tube of large diameter with thin walls will probably fail by flattening or buckling on the leeward side and not by tension of the windward side, the influence of the weight of the material is ordinarily neglected and the shaft is treated as a cantilever subject to wind pressure only. Wq therefore is neglected and equation (88) becomes

$$S = Ph \div \frac{I}{e}. \tag{90}$$

Since the thickness of the wall is a small fraction of the diameter the section modulus $\frac{I}{e}$ becomes, approximately,

$$\frac{I}{e} = 0.7854\, d_1^2 t,$$

in which

t = thickness of the shell in inches.

Substituting this value in equation (90)

$$S = \frac{Ph}{0.7854\, d_2 t}. \tag{91}$$

A number of steel stack builders simplify equation (91) still further by making the constant 0.8, thus

$$S = \frac{Ph}{0.8\, d^2 t}. \tag{92}$$

Considering the stress, S', per lineal inch instead of that per sq. in. equation (92) becomes

$$S' = \frac{Ph}{0.8\, d_1^2}. \tag{93}$$

Example 19: Determine the thickness of plate at a section 150 feet from the top of a cylindrical steel stack 12 feet in diameter and 200 feet high. Horizontal seams to be single riveted.

The total wind pressure on the section is

$$P = 150 \times 12 \times 25^* = 45{,}000 \text{ lb.}$$

The moment arm is

$$h = \frac{150}{2} \times 12 = 900 \text{ inches.}$$

$S = 8000$ lb. per sq. in. (A common allowance for safe stress is 8000 lb. per sq. in. for single riveted and 10,000 for double riveted joints.)

Substituting these values in equation (93)

$$8000 = \frac{45{,}000 \times 900}{0.8 \times 144\,t},$$

from which

$$t = 0.305.$$

The nearest commercial size lies between $\frac{9}{32}$ and $\frac{5}{16}$.

TABLE 60.

STEEL STACKS. — SIZES OF RITER CONLEY COMPANY, PITTSBURG.

Diameter of Flue.		Total Height.	Total Weight.	How Made.
Ft.	In.	Ft.	Lb.	
5	6	165	67,000	40 ft. of $\frac{3}{16}$ in., 45 ft. of $\frac{1}{4}$ in., 50 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
7	0	160	79,000	30 ft. of $\frac{3}{16}$ in., 50 ft. of $\frac{1}{4}$ in., 50 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
8	6	150	94,000	60 ft. of $\frac{1}{4}$ in., 60 ft. of $\frac{5}{16}$ in., 30 ft. of $\frac{3}{8}$ in.
10	0	200	150,000	90 ft. of $\frac{1}{4}$ in., 60 ft. of $\frac{5}{16}$ in., 50 ft. of $\frac{3}{8}$ in.
12	0	200	175,000	35 ft. of $\frac{1}{4}$ in., 35 ft. of $\frac{9}{32}$ in., 35 ft. of $\frac{5}{16}$ in., 35 ft. of $\frac{11}{32}$ in., 35 ft. of $\frac{3}{8}$ in., 25 ft. of $\frac{13}{32}$ in.
11	6	225	232,000	40 ft. of $\frac{1}{4}$ in., 40 ft. of $\frac{9}{32}$ in., 40 ft. of $\frac{5}{16}$ in., 40 ft. of $\frac{11}{32}$ in., 40 ft. of $\frac{3}{8}$ in., 25 ft. of $\frac{7}{16}$ in.
12	0	255	256,000	75 ft. of $\frac{1}{4}$ in., 65 ft. of $\frac{5}{16}$ in., 55 ft. of $\frac{3}{8}$ in., 35 ft. of $\frac{7}{16}$ in., 25 ft. of $\frac{1}{2}$ in.

136. Riveting. — The diameter of rivets should always be greater than the thickness of the plate but never less than one-half inch. The pitch should be approximately $2\frac{1}{2}$ times the diameter of the rivet, and always less than 16 times the thickness of the plate. Single-riveted joints are ordinarily used on all sections except the base, where the joint should be double riveted with rivets staggered, although in very large stacks all horizontal seams are double riveted to give greater stiffness to the shaft.

* See Paragraph 133.

137. Stability of Steel Stacks. — For stability the resisting moment $W_t q'$ must be greater than the Ph_1 overturning moment (see paragraph 143), that is

$$W_t q' > Ph_1,^* \tag{94}$$

in which

W_t = total weight of the structure, including that of the foundation, and the earth filling over the base, lb.,
q' = radius of the statical moment of the foundation base, ft.,
h_1 = distance from the center of wind pressure to the base, ft.

For a square base the minimum value of q_1, see equation (106), paragraph 143, is

$$q' = \frac{L}{6},$$

and the condition for stability is

$$W_t \frac{L}{6} > Ph_1. \tag{95}$$

Expressed graphically: Lay off GP, Fig. 153, equal to the total wind pressure in direction and amount and acting at the center of pressure of the shaft; lay off GW equal to the weight of the stack and foundation; find the resultant GR and produce it to intersect the base line as at R'; if R' falls within the inner third of the base the stack is stable, provided, of course, that the chimney is properly designed and constructed. Therefore the heavier the combined weight of the chimney and its foundation the more stable the structure.

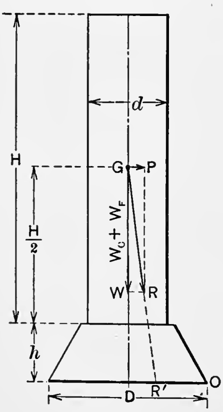


FIG. 153.

L in Fig. 153 varies from one-tenth to one-fifteenth H, depending upon the character of the subsoil. For the ordinary concrete foundation, Christie ("Chimney Design and Theory," p. 57) gives as an average value for L,

$$L = \frac{H^2 d}{26{,}000} + 10. \tag{96}$$

138. Foundation Bolts for Steel Stacks. — There is no generally accepted rule for proportioning foundation bolts for steel stacks. The various rules differ principally in the assumed location of the center of moments or neutral axis of the bolts when stressed by the overturning moment. In lieu of proof to the contrary and considering the number of unknown factors entering into the problem the neutral axis may be taken as passing through and tangent to the bolt circle,

* Axis of the shaft assumed to be vertical.

and the fiber stresses in the bolts may be assumed to be proportional to their distances from the axis. Thus

$$Ph - Wq = SaL, \tag{97}$$

in which

Ph = wind moment at the base ring, in-lb.,
Wq = statical moment, in-lb.,
S = maximum fiber stress in the bolts, lb. per sq. in. (To allow for initial stress due to tightening up, a low fiber stress of 12,000 lb. per sq. in. is commonly assumed.)
a = area of each bolt at the root of the thread, sq. in. (All bolts assumed to be of the same diameter.)
L = equivalent mean length of the bolt resisting moment, in.

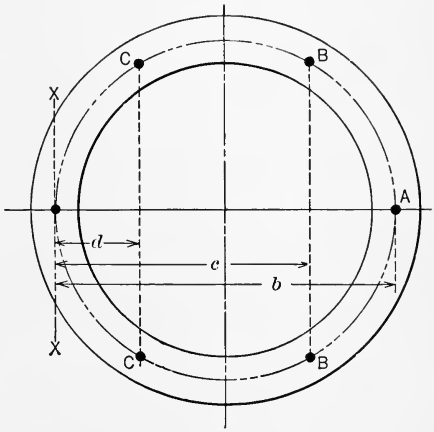


Fig. 154.

Referring to Fig. 154,

$$SaL = S_1b + 2\,S_2c + 2\,S_3d, \tag{98}$$

in which

S_1, S_2, S_3 = stresses in bolts, A, B–B, and C–C, respectively, lb.,
b, c, d = respective moment arms relative to neutral axis XX, in.

Since the stress in each bolt is assumed to be directly proportional to its distance from the neutral axis, $S_2 = S_1 \frac{c}{b}$ and $S_3 = S_1 \frac{d}{b}$. Substituting these values in equation (98) and noting that $S_1 = Sa$, equation (98) reduces to

$$L = \frac{1}{b}\,(b^2 + 2\,c^2 + 2\,d^2). \tag{99}$$

The value of L becomes

Number of bolts	6	8	10	12	16	24	36
$L = b \times$	2.25	3.00	3.88	4.58	6.00	8.90	12.40

Example 20: Calculate the size of bolts necessary for a steel stack with conditions as follows: Overturning moment 2,750,000 in-lb., bolt circle diameter 82 in., 6 bolts, allowable stress 12,000 lb. per sq. in.

Here $Ph - Wq = 2{,}750{,}000$; $S = 12{,}000$; $L = 2.25 \times 82 = 184.5$. Substituting these values in equation (97)

$$2{,}750{,}000 = 12{,}000 \times a \times 184.5;$$
$$a = 1.24 \text{ sq. in.}$$

Nearest commercial size corresponding to this area, $1\frac{1}{2}$ in. diam.

Foundation Bolts for Steel Chimneys: D. A. Hess, Power, Oct. 5, 1915.

Design of Steel Stacks: Eng. & Contr., Nov. 22, 1916, p. 440; Oct. 25, 1916, p. 369.

Design and Construction of a 400-ft. Steel Stack: Eng. & Contr., Aug. 25, 1915, p. 140.

Reasons for Corrosion of Steel Smokestacks and Ways to Prevent it: Elec. Wld., Nov. 6, 1915, p. 1033.

139. Brick Chimneys. — By far the greater number of power-plant chimneys are of brick construction and usually of circular section, though octagonal, hexagonal, and square sections are not uncommon. The round chimney requires the least weight for stability, and the others in the order mentioned.

Brick chimneys may be divided into two general classes:

1. Single shell, Fig. 158, and
2. Double shell, Fig. 156.

The double shell is the more common and consists of an outer shaft of brickwork and an inner core or lining extending part way or throughout the entire length of the shaft.

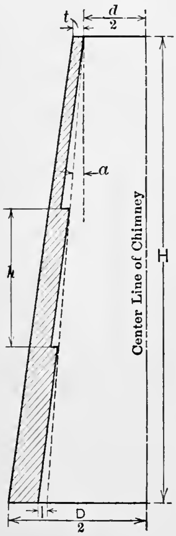


Fig. 155.

The single shell is the general construction where carefully burned and selected brick not easily affected by the heat are used. As the inner core or lining is independent of the outer shell and has no part in the strength of the chimney, the rules for determining the thickness of the walls are practically the same for both single and double shell.

140. Thickness of Walls. — The thickness of the wall should be such as to require minimum weight of material for the proper degree of stability, due consideration being paid to the practical requirements of construction. The thickness does not vary uniformly, but decreases from bottom to top by a series of steps or courses as in Fig. 155. In general, the thickness at any section should be such that the resultant stress of wind and weight of shaft will not put the masonry in tension on the windward side or in excessive compression on the leeward side.

For circular chimneys using common red brick for the outer shell the following approximate method gives results in conformity with average practice:

$$t = 4 + 0.05\,d + 0.0005\,H, \tag{100}$$

where

t = thickness in inches of the upper course, neglecting ornamentation, and should, of course, be made equal to the nearest dimension of the brick in use. Ordinary red bricks measure $8\frac{1}{4} \times 4 \times 2$.
d = clear inside diameter at the top, inches,
H = height of stack, inches.

Beginning at the top with this thickness, add one-half brick, or 4 inches, for each 25 or 30 feet from the top downwards, using a batter of 1 in 30 to 1 in 36.

The minimum value of t for stacks built with inside scaffolding should be 7 inches for radial brick and $8\frac{1}{4}$ inches for common brick, as a thinner wall will not support the scaffold. Radial brick for chimneys are made in several sizes, so that the thickness of the walls when they are used increases by about 2 inches at the offsets.

For specially molded radial brick or for circular shells reinforced as in Fig. 156 the length of the different courses may be much less than stated above. The external form of the top is a matter of appearance, and may be designed to suit the taste, but should be protected by a cast-iron or tile cap and provided with lightning rods. Ladders for reaching the top of the chimney are generally located inside the brick stacks and outside the steel structures.

Professor Lang's rule (Eng. Rec., July 20, 1901, p. 53) for determining the length of the different courses is (Fig. 155):

$$h = C\left(20\,t + 60\,i + 0.1056\,G + 2.5\,\frac{d}{2} + 656\tan\alpha - 0.007\,H \right.$$
$$\left. - 0.453\,p - 18.7\right), \tag{101}$$

in which

h = length of the course under consideration,
C = constant = 1 for a circular, 0.97 for an octagonal, and 0.83 for a square, chimney,
i = increase in thickness for each succeeding section in feet,
G = weight per cubic foot of brickwork,
p = wind pressure, pounds per square foot,
α = angle of the internal batter.

All other notations as indicated in Fig. 155.

For chimneys over 100 feet in height he recommends that 100 be

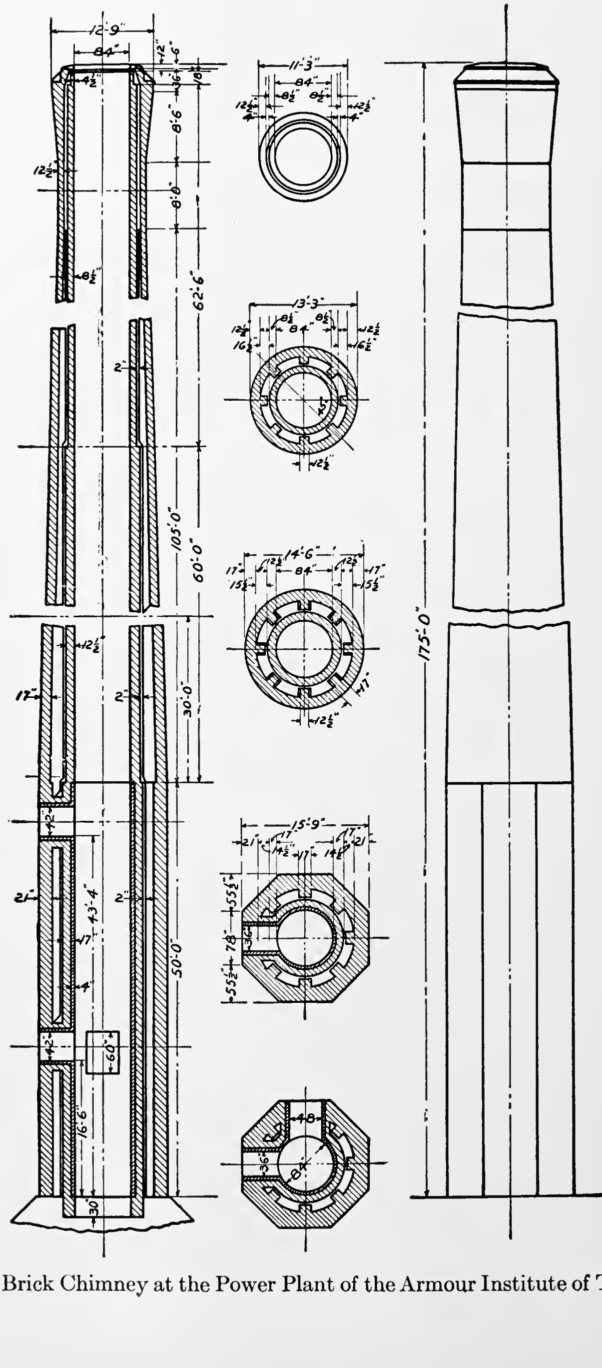


FIG. 156. Brick Chimney at the Power Plant of the Armour Institute of Technology.

used instead of the actual height, since the critical point will be in one of the lower sections and not at the base.

If a value of h is obtained which is not contained an even number of times in H, it may be slightly increased or decreased so as to effect this result.

To determine the stresses at any section the shaft is treated as a cantilever uniformly loaded with a maximum wind pressure of 25 pounds per square foot. If the tension on the windward side subtracted from the compression leaves a positive remainder, the chimney will be stable; if the remainder is negative, the masonry will be in tension, which it withstands but feebly. The sum of the compressive stresses on the leeward side due to wind pressure and weight must be less than the crushing strength of the masonry. The practice, however, of assuming a fixed value for allowable pressure irrespective of the height of the stack gives dimensions that are too low for small stacks and too high for large stacks. According to Professor Lang, compressive stress on the leeward side in pounds per square inch with single chimneys should not exceed

$$p = 71 + 0.65\,L, \tag{102}$$

where

p = pressure in pounds per square inch,
L = distance in feet from top of chimney to the section in question.

With double shell $$p = 85 + 0.65\,L. \tag{103}$$

The tension on the windward side should not exceed,

$$\text{for single shell:}\quad p = (18.5 + 0.056\,L), \tag{104}$$

$$\text{for double shell:}\quad p = (21.3 + 0.056\,L). \tag{105}$$

Example 21. Determine the maximum stress in the outer fiber of the brickwork at the base of section 8 of the chimney illustrated in Fig. 158 when the wind is blowing 100 miles an hour. Assume the weight of the brickwork 120 pounds per cubic foot.

A wind velocity of 100 miles per hour is estimated to exert a pressure 25 pounds per square foot of projected area on a cylindrical surface. (See paragraph 133.) The height of the chimney to section 8 is 131.4 feet. The projected area as computed from the figure is 1800 square feet. Hence p, the total wind pressure, is 1800 × 25 = 45,000 pounds. The volume of brickwork above section 9 may be calculated, and is 6150 cubic feet, hence the weight W = 6150 × 120 = 738,000 pounds.

The area of the joint at this section is 75.3 square feet, therefore the pressure due to the weight of the superimposed brickwork is 738,000 divided by 75.3 = 9800 pounds per square foot. To find the stress due to the wind pressure, substitute the proper values in equation (86):

$$Ph = S\,\frac{I}{e} = 0.0983\left(\frac{d_1^4 - d^4}{d_1}\right)S.$$

Here

P = 45,000 as computed above,
h = 55 feet (found by laying out the section and locating the center of gravity),
d_1 = 16.2,
d = 12.9,

whence

$$45{,}000 \times 55 = 0.0983 \frac{16.2^4 - 12.9^4}{16.2} S,$$

from which S = 9907 pounds per square foot.

The net stress on any part of the section is the resultant of that due to the weight of the stack and that caused by the wind, the net stress on the windward side being

$$9907 - 9800 = 107 \text{ pounds per square foot,}$$

which is evidently a tensile stress and should never exceed the value given by formula (104):

$$\begin{aligned} p &= (18.5 + 0.056\,L) \\ &= (18.5 + 0.056 \times 131.4) \\ &= 25.8 \text{ pounds per square inch} \\ &= 3715 \text{ pounds per square foot.} \end{aligned}$$

The net compressive stress on the leeward side is 9800 + 9907 = 19,707 pounds per square foot, which should not exceed that given by formula (102):

$$\begin{aligned} p &= 71 + 0.65\,L \\ &= 71 + 0.65 \times 131.4 \\ &= 156.4 \text{ pounds per square inch} \\ &= 22{,}521 \text{ pounds per square foot.} \end{aligned}$$

141. Core and Lining. — The core or lining of a brick chimney is commonly carried to the top of the shaft, though it sometimes extends only part of the distance. The inside diameter is generally uniform, the offsets being made on the outside. The core and outer shell should be independent to prevent injury due to expansion of the core. The rules for the thickness of lining in steel chimneys apply also to brick chimneys. The batters for the inner and outer shells should be such as to allow at least 2 inches clearance between the two shafts at the top, and the top should be protected by an iron ring or by a projecting ledge from the outer shell.

142. Materials for Brick Chimneys. — Brick for the external shaft should be hard burned, of high specific gravity, and laid with lime mortar strengthened with cement. Lime mortar itself is more resistant to heat, but hardens slowly and may cause distortion in newly erected tacks, and hence should be used only when a long time is taken in building. Mortar of cement and sand alone is not to be recommended, since it does not resist heat well and is attacked by carbon dioxide, particularly in the presence of moisture. A mortar consisting of 1 part by volume of cement, 2 of lime, and 6 of sand may be used for the upper brickwork, 1, $2\frac{1}{2}$, and 8 respectively for the lower part, and 1, 1, and 4 respectively for the cap. The harder the brick the more cement is necessary, as lime does not cling so well to hard, smooth surfaces. The inner core may be constructed of second-class fire brick, since the temperature seldom exceeds 600 deg. fahr. Lime mortar is invariably used for the core.

143. Stability of Brick Chimneys. — When there is no wind blowing and the chimney is built symmetrically about a vertical axis the pressure due to weight is uniformly distributed over the bearing surfaces, and the center of pressure lies in the line XX, Fig. 157. But when the wind blows the pressure exerted tends to tilt the shaft as a whole column in the direction of the current, and the resultant pressure at the windward side of the base decreases, until, with a sufficiently high velocity of wind, it may become zero, in which case the center of pressure moves a distance q towards the leeward side of the base. As soon as the pressure at A becomes zero the joint begins to open (assuming no adhesion between chimney and base) and the shaft is evidently in the condition of least stability. The distance q through which the center of pressure has moved is called the *radius of the statical moment.* For any column it may be shown that

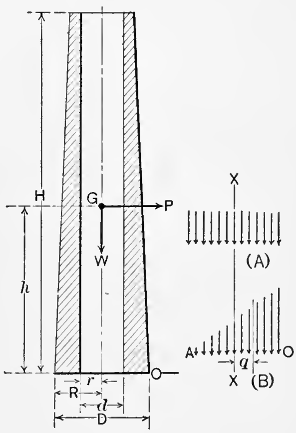


FIG. 157.

$$q = \frac{I}{Ae} \text{ (Rankine, ``Applied Mechanics,'' p. 229)}, \tag{106}$$

in which

I = moment of inertia of the section,
A = area of the section,
e = distance from the center of the shaft to the outer edge of the joint.

Thus for a solid circular section, $q = \dfrac{D}{8}$.

For a solid square section, $q = \dfrac{L}{6}$.

For an annular circular ring, $$q = \frac{D^2 + d^2}{8\,D}.$$

For a hollow square, $$q = \frac{L^2 + l^2}{6\,L}.$$

The relationship between weight of shaft and wind pressure for the condition of least stability is

$$Ph = Wq, \tag{107}$$

in which

P = total wind pressure, pounds,

h = distance in feet from the base line of the section under consideration to center of gravity of that section,

W = weight of shaft in pounds above the assumed base line,

q = radius of the statical moment.

The condition of least stability for round chimneys requires, therefore, that

$$Ph = W\,\frac{D^2 + d^2}{8\,D}. \tag{108}$$

For many purposes it is sufficiently accurate to assume $D = d$, and equation (77) becomes

$$Ph = W\frac{D}{4} \text{ for round chimneys,} \tag{109}$$

$$Ph = W\frac{L}{3} \text{ for square chimneys.} \tag{110}$$

Another rule gives for the condition of least stability:

$$W\left(\tfrac{1}{2}\,R + \tfrac{1}{4}\,r\right) = Ph. \quad \text{(Eng. Rec., July 27, 1901, p. 82.)} \tag{111}$$

Notations as in Fig. 108, all dimensions in feet.

This permits of a lighter chimney than equation (108), and the maximum wind pressure may be assumed to put the joint on the windward side in tension or even to permit a slight opening of same.

A rule of thumb for stability is to make the diameter of the base one-tenth of the height for a round chimney; for any other shape to make the diameter of the inscribed circle of the base one-tenth of the height.

The factor of stability is the quotient obtained by dividing the value of q from formula (107) by that from (106). If less than unity, the chimney is in tension at the outer fiber on the windward side, and must be redesigned unless the tension is less than that allowed by equation (104). Calculations for stability should be made for various sections.

Example 22. Analyze the chimney illustrated in Fig. 158 for stability at, say, section 8, the following data referring to the portion above the base line of this section.

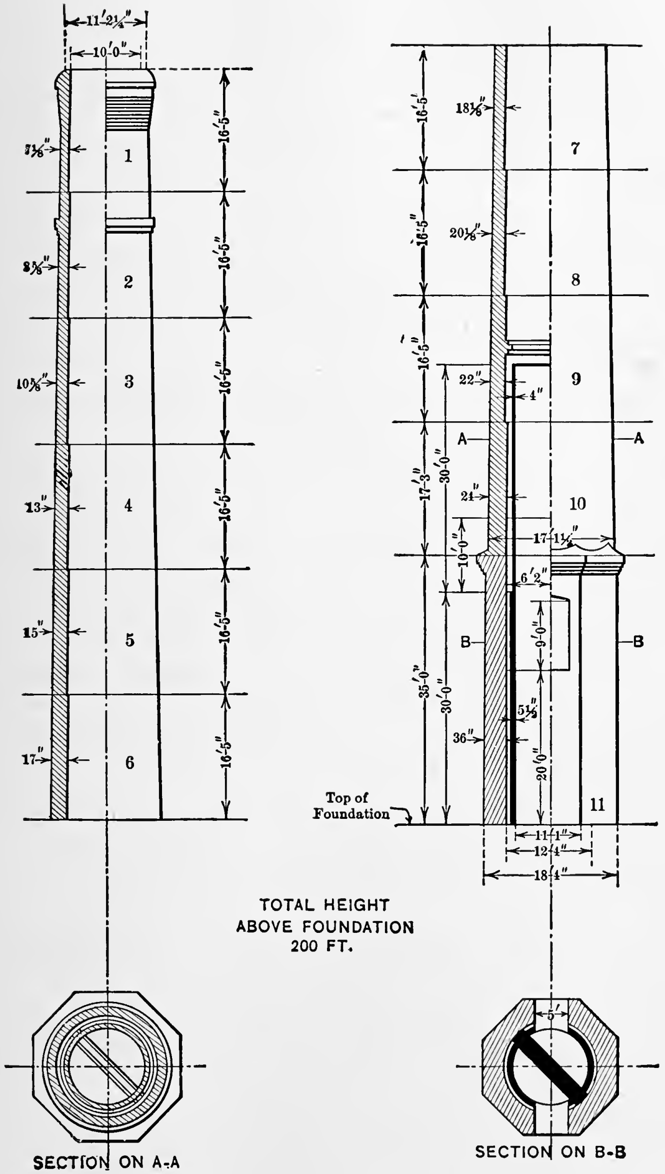


FIG. 158. Custodis Radial Brick Chimney.

From the drawing:
Projected area of the stack, 1800 square feet.
Volume of brickwork, 6150 cubic feet.
Outside diameter of base, 16.2 feet.
Inside diameter of base, 12.9 feet.
Center of pressure to base line, 55 feet.
Total height above base line, 131.4 feet.
Maximum total wind pressure:

$$P = 1800 \times 25 = 45{,}000 \text{ pounds.}$$

Weight of shaft:

$$W = 6150 \times 120 = 738{,}000 \text{ pounds.}$$

For stability, according to equation (55),

$$Ph < W \frac{D^2 + d^2}{8\,D}.$$

Substituting the proper values:

$$Ph = 45{,}000 \times 55 = 2{,}475{,}000 \text{ foot-pounds.}$$

$$W \frac{D^2 + d^2}{8\,D} = 738{,}000 \left(\frac{16.2^2 + 12.9^2}{8 \times 16.2} \right) = 2{,}441{,}000.$$

While Ph is slightly greater than $W \frac{D^2 + d^2}{8\,D}$, for practical purposes the shaft at this section would be called stable under maximum allowable wind pressure.

For stability, according to equation (111),

$$Ph < W\left(\tfrac{1}{2} R + \tfrac{1}{4} r\right),$$
$$Ph = 2{,}475{,}000, \text{ as determined above,}$$

$$W \left(\frac{1}{2} R + \frac{1}{4} r \right) = 738{,}000 \left(\frac{8.1}{2} + \frac{6.45}{4} \right)$$
$$= 4{,}177{,}000.$$

Ph is therefore considerably less than $W\left(\tfrac{1}{2} R + \tfrac{1}{4} r\right)$, and the condition imposed in equation (111) is more than fulfilled.

The Design of Tall Chimneys: Henry Adams, Industrial Engineering, March, 1912, p. 198. *Design of a Brick Chimney:* Eng. News, May 9, 1912, p. 866.

144. Custodis Radial Brick Chimney. — Fig. 158 gives the details of a 200 × 10-foot radial brick chimney constructed of special molded radial brick, formed to suit the circular and radial lines of each section, thus permitting them to be laid with thin, even mortar joints. The blocks are much larger than common brick and the number of joints is proportionately reduced. They are molded with vertical perforations, as shown in Fig. 159, which permits thorough burning, thereby increasing the density and strength and at the same time reducing the weight of the block. In laying, the mortar is worked into the perforations about one-half inch. The first 60 feet above the base are octagonal in section, with 36-inch walls, and the balance of circular section, with walls tapering gradually from 22 inches to $7\frac{1}{8}$ inches in thickness. A radial brick lining extends 60 feet from the base as indicated. The chimney was designed to furnish draft for a 3500-horsepower boiler plant and cost, erected, $8,800. The entire weight of the chimney exclusive of foundation is 870 tons.

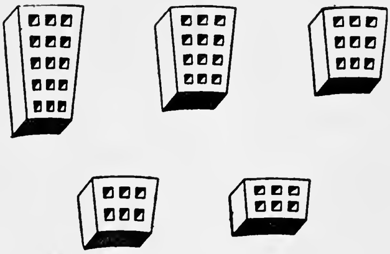


Fig. 159. Custodis Radial Brick.

Radial brick chimneys without the inner lining are likely to be unduly affected by temperature changes.

The largest chimney of this type is located at Great Falls, Mont., and is used for leading off the gases from the smelter plant of the Boston and Montana Consolidated Copper and Silver Mining Company. The height above the top of the foundation is 506 feet, and the internal diameter at the top 50 feet. The chimney and foundation cost approximately $200,000.

Custodis Chimney Details: Eng. Rec., Oct. 1, 1904, p. 385; Power, May, 1900, p. 12.

145. Wiederholt Chimney. — This type of chimney consists essentially of a combination of the masonry and reinforced concrete structures. The inner and outer surfaces of the shaft are formed by hard burned fire clay tile of special design as illustrated in Fig. 160. When placed in position these tile form a permanent mold into which the reinforcing bars and concrete may be introduced. Both vertical and horizontal reinforcing bars are incorporated in the structure in much the same manner as in the Weber type. Because of the tile lining much higher temperatures may be safely carried than with concrete type and the color may be readily made to match that of the power house or adjoining buildings.

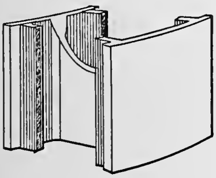


Fig. 160. Tile for Wiederholt Chimney.

146. Steel-Concrete Chimneys. — The use of concrete reinforced with iron or steel for the construction of chimneys is rapidly increasing. The advantages claimed for this class of stack are:

1. Light weight of the whole structure, being but one-third as great as an equivalent common brick chimney. The space occupied is much

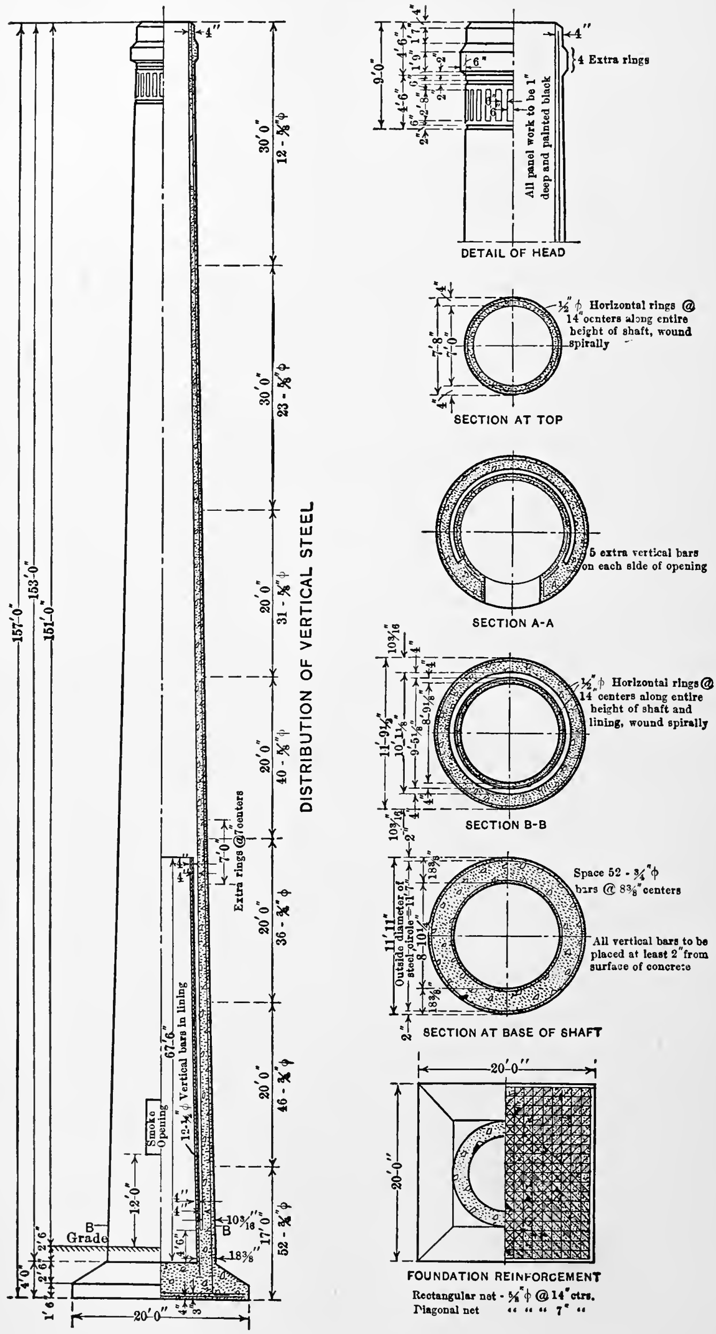


Fig. 161. Weber "Coniform" Reinforced Concrete Chimney.

less than with either brick or steel stack, on account of the thinness of walls at the base and the absence of any flare or bell.

2. Total absence of joints, the entire structure including foundation being a monolith.

3. Great resisting power against tension and compression.

4. Rapidity of construction. May be erected at an average rate of six feet per day.

5. Adaptability of the material to any form.

This type of chimney being comparatively new, little data concerning depreciation are available, but some which have been in use ten years show little or no deterioration.

Fig. 161 gives the details of a Weber "coniform" steel-concrete chimney as erected at Grafton, Mass., for the Grafton State Hospital. The entire structure, foundation, shaft and lining is monolithic, 157 feet in total height, seven feet internal diameter and weighs only 344 tons. It occupies but 108 square feet of ground space at grade level. The weight of the shaft and lining is 249 tons.

The shaft is of the double shell type with inner core extending 65 feet above the grade. The core is but 4 inches in thickness and the shaft varies from $10\frac{3}{16}$ inches at the junction of the core and shaft to 4 inches at the top. The core reinforcement consists of twelve vertical $\frac{1}{2}$-inch twisted steel bars and similar horizontal bars wound spirally at 14-inch centers. The vertical reinforcement in the outer shell varies from fifty-two $\frac{3}{4}$-inch twisted bars at the grade to twelve $\frac{5}{8}$-inch bars at the top. The horizontal reinforcement consists of $\frac{1}{2}$-inch twisted steel rings spaced at 14-inch centers along the entire height of shaft and wound spirally. The steel bars vary from 16 to 30 feet in length and where they meet lengthwise are lapped not less than 24 inches. The use of different lengths of steel prevents the laps from concentrating in any given section.

The tallest chimney in the world is of this type and is located in Japan. It is 567 feet high and 26 feet 3 inches in diameter at the top.

The determination of the amount of steel reinforcement does not permit of simple mathematical calculation because of the number of variables entering into the problem and graphical charts plotted from semi-rational formulas offer a simple solution. The curves in Fig. 162 are reproduced from "Principles of Reinforced Concrete," p. 408, Turneaure and Maurer, and are used extensively in this connection. The use of the chart is best illustrated by a specific example.

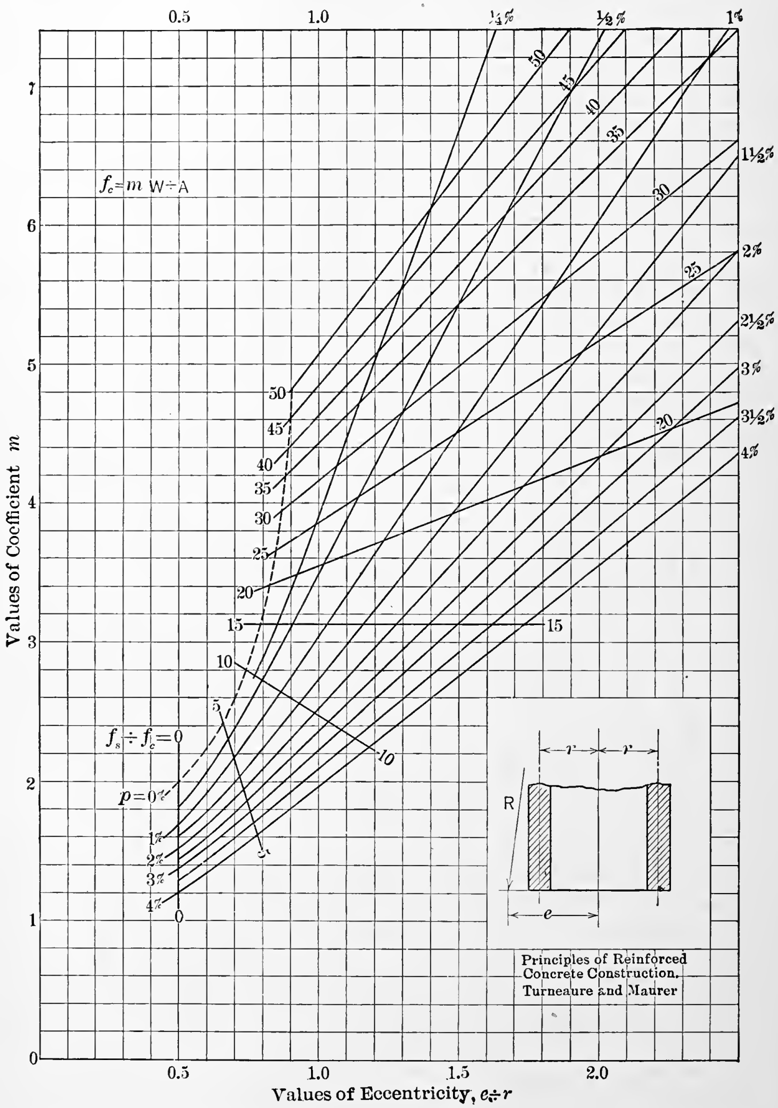


FIG. 162. Wind Stresses in Steel-Concrete Chimneys. (Turneaure and Maurer.)

Strain Sheet.

Example 23. Determine the amount of reinforcement required for the chimney illustrated in Fig. 161 at section BB.

From the drawing we find:

$D = 11$ ft. $9\frac{1}{2}$ in. $\qquad r =$ radius of the steel circle $= 5.79$ ft.
$d = 10$ ft. $1\frac{1}{8}$ in. $\qquad h = 153$ ft.

The following values may be obtained by simple arithmetic computations, but the actual calculation will be omitted for the sake of brevity.

W, weight of shaft above section BB, 409,000 lb.
A, area of shaft above section BB, 4320 sq. in.
M, wind moment above section BB, 2,600,000 ft-lb.

$$e, \text{ eccentricity } = \frac{M}{W} = 6.36 \text{ ft.}$$

$$\frac{e}{r} = 1.1.$$

Assume a maximum compression in the concrete of $f_c = 360$ lb. per sq. in. (In practice this assumed value varies from 350 lb. per sq. in. for chimneys under 150 ft. in height to 500 lb. per sq. in. for chimneys 350 ft. high.)

$$m, \text{ a coefficient } = \frac{f_c A}{W} = 3.8.$$

From the curves in Fig. 162 the intersection of $m = 3.8$ and $\frac{e}{r} = 1.1$ gives p (per cent of steel required) as 0.53.

$$\text{But } p = \frac{\text{area steel}}{\text{area section}}.$$

Whence area of steel $= 0.0053 \times 4320 = 23$ sq. in. corresponding to 52, $\frac{3}{4}$-inch steel bars.

Other sections at 20 ft. intervals have been analyzed in a similar manner and the results inserted in Fig. 161.

In the earlier types of steel concrete chimneys designed and built by the Weber Company the amount of steel reinforcement was calculated from formula (89), but all recent structures are proportioned on the Turneaure and Maurer chart. The resultant stress R as calculated from equation (89) necessitates the use of more reinforcement than that derived from the chart.

Evasé Stacks. See paragraph 153.

Design, Construction, and Cost of a 137-ft. Reinforced Concrete Chimney: Eng. & Contr., Aug. 11, 1915, p. 111.

147. Breeching. — The area of the flue or breeching leading from the boilers to the chimney is generally made equal to or a little larger than the internal area of the chimney at the top, 10 per cent greater being an average figure. A common rule is to allow 1 to $4\frac{1}{2}$ or 5 as the ratio of breeching area to grate area for ordinary service, and 1 to $3\frac{1}{2}$ for large boilers operating continually at 150 to 200 per cent rating. The flue may be carried over the boilers or back of the setting or even under the fire-room floor, but in any case should be as short as possible and free from abrupt turns. Underground breechings cause excessive pressure drop and are difficult to clean. Short right-angled turns reduce the draft approximately 0.05 inch for each turn, and a convenient rule is to allow 0.1 inch loss for each 100 feet of flue if of circular cross section and constructed of steel, and double this amount for brick flues of square section. Each additional boiler connected to the breeching

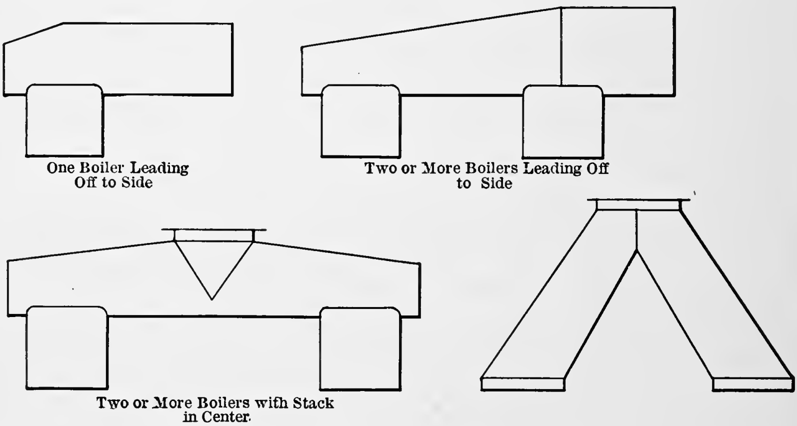


FIG. 163. Types of Breeching Connections.

will cause a pressure drop due to friction or interference of the gases as they enter the breeching, or to leakage through the dampers when the boiler is out of service. A common rule is to allow a pressure drop of 0.05 in. of water for each boiler connected to the breeching. The cross section of the flue need not be the same throughout its entire length, but may be tapered and proportioned to the number of boilers. Where two flues enter the stack on opposite sides, a diaphragm is inserted as indicated in Fig. 158. Flues should be covered on the outside with heat-insulating material, because a lining on the inside is difficult to repair and deterioration might readily escape detection. A damper ratio of 1 to 4 expressed in terms of grate surface has given good satisfaction.

148. Chimney Foundations. — On account of the concentration of weight on a small area the foundation of a chimney should be carefully designed. In most cities the building laws limit the maximum loads allowed for various soils and materials, and although they vary considerably the average is approximately as follows:

Material.	Safe Load, Lb. per Sq. Ft.
Hard-burned brick masonry, cement mortar, 1 to 2	20,000–30,000
Hard-burned brick masonry, cement mortar, 1 to 4	18,000–24,000
Hard-burned brick masonry, lime mortar	10,000–16,000
Concrete, 1 to 8	8,000–10,000

Kind of Soil.	Safe Load, Tons per Sq. Ft.
Quicksands and marshy soils	0.5
Soft wet clay	1.0
Clay and sand 15 feet or more in thickness	1.5
Pure clay 15 feet or more in thickness	2.0
Pure dry sand 15 feet or more in thickness	2.0
Firm dry loam or clay	3.0–4.0
Gravel well packed and confined	6.0–8.0
Rock broken but well compacted	10.0–15.0
Solid bed rock	Up to ⅕ of its ultimate crushing strength.
	Tons per Pile.
Piles in made ground	2.0
Piles driven to rock or hardpan	25.0

Chimney foundations as a rule are constructed of concrete except where the low sustaining nature of the soil necessitates the use of piles or a grillage of timber or steel. For masonry chimneys the foundation is designed to give the necessary support to the shaft without particular reference to its mass or distribution, as the shape of the foundation has virtually no effect on its stability as a column. In steel and reinforced concrete chimneys the shape and weight of the foundation are a function of the desired factor of stability, since the shaft is securely anchored to the foundation and the two form practically one mass. The foundation should be designed to fulfill the conditions for shear and flexure in addition to the requirements for stability.

Practically all chimney foundations are square in plan and the maximum pressure on the supporting surface may be calculated from the following equation: *

$$P = \frac{4\,W}{3\left(1 - \frac{2\,e}{b}\right)b^2}, \tag{112}$$

in which

P = maximum pressure due to wind and weight, lb. per sq. in.,
W = total weight of the chimney and foundation, lb.,
e = eccentric $\frac{M}{W}$ = wind moment divided by the weight,
b = width of the foundation.

* Principles of Reinforced Concrete Construction, Turneaure and Maurer, p. 423.

For the chimney illustrated in Fig. 158

$$P = \frac{4 \times 738{,}000}{3\left(1 - \frac{2 \times 3.35}{20}\right)20^2}$$

$$= 3900 \text{ lb. per sq. ft.}$$

The pressure due to weight only is $\frac{738{,}000}{400} = 1845$ lb. per sq. ft.

Table 61 gives the least diameter and depth of foundation for steel chimneys of various diameters and heights.

TABLE 61.

SIZES OF FOUNDATION FOR STEEL CHIMNEYS.

Diameter, Feet.	Height, Feet.	Least Diameter of Foundation.	Least Depth of Foundation.
3	100	15′ 9″	6′ 0″
4	100	16′ 4″	6′ 0″
4	125	18′ 5″	7′ 0″
5	150	20′ 4″	9′ 0″
5	200	23′ 8″	10′ 0″
6	150	21′ 10″	8′ 0″
6	200	25′ 0″	10′ 0″
7	150	22′ 7″	9′ 0″
7	250	29′ 8″	12′ 0″
9	150	23′ 8″	10′ 0″
9	275	33′ 6″	12′ 0″
11	250	24′ 8″	10′ 0″
11	350	36′ 0″	14′ 0″

149. Chimney Efficiencies. — The chimney as a *mover of air* has a very low thermodynamic efficiency. Compared with that of a fan its performance is very poor, and mechanical-draft concerns sometimes use this as an argument.

Example 24. A chimney 200 feet high and 10 feet in diameter furnishes draft for a battery of boilers rated at 3500 horsepower. Average outside temperature 60 deg. fahr.; temperature of flue gases 500 deg. fahr.; calorific value of the fuel 14,000 B.t.u. per pound. Compare the thermal efficiency of the chimney as a mover of air with that of a forced-draft apparatus of equivalent capacity.

From Table 52 we find that a chimney 200 feet high, with temperatures as stated above, will furnish a theoretical draft of 1.27 inches, equivalent to a pressure of 6.6 pounds per square foot. Neglecting friction, the height H of a column of external air which would produce this pressure is

$$H = \left(\frac{d_1 - d}{d_1}\right)h, \tag{113}$$

in which

h = height of the chimney in feet,
d = density of the hot gases in the stack,
d_1 = density of the outside air.

Substitute in equation (113)

$$d_1 = 0.0763, \quad d = 0.0435, \quad \text{and} \quad h = 200.$$

$$H = \left(\frac{0.0763 - 0.0435}{0.0763}\right)200$$
$$= 85.9 \text{ feet.}$$

The theoretical velocity of the air entering the base of the chimney under this head is

$$v = \sqrt{2\,gH}$$
$$= \sqrt{2 \times 32.2 \times 85.9}$$
$$= 74.5 \text{ feet per second.}$$

The weight of the gas escaping per second

$$= 74.5 \times \text{area of the stack} \times 0.0763$$
$$= 446 \text{ pounds.}$$

The displacement of this volume of gas is the result of heating it from 60 to 500 deg. fahr. Taking the specific heat of the gas as 0.24, the heat necessary to displace 446 pounds per second is

$$\text{Heat required} = 446 \times 0.24 \times (500 - 60)$$
$$= 47{,}000 \text{ B.t.u. per second.}$$

The work actually performed is that of overcoming a total resistance of $6.6 \times 78.5 = 518$ pounds (78.5 = internal area of the chimney) through a space of 74.5 feet; i.e.,

$$\text{Work done} = 74.5 \times 518 = 38{,}591 \text{ foot-pounds per second}$$
$$= 49.7 \text{ B.t.u. per second.}$$

$$\text{Efficiency} = \frac{49.7}{47{,}000} = 0.00107, \text{ or about } \tfrac{1}{10} \text{ of 1 per cent.}$$

If a fan be substituted for the chimney and we allow say 8 per cent for the efficiency of engine and boiler, 40 per cent for the fan, and 25 per cent for friction, the combined efficiency will be

$$0.08 \times 0.40 \times 0.75 = 0.024, \text{ or 2.4 per cent.}$$

The fan then will be $\frac{0.024}{0.00107} = 22.4$ times more efficient than the chimney *as a mover of air.*

150. Cost of Chimneys. — Christie ("Chimney Design and Theory") gives the following costs of chimneys 150 feet high and 8 feet internal diameter:

Common red brick	approximate	cost	\$8,500.00
Radial brick	do.	do.	6,800.00
Steel, self-supporting, full lined	do.	do.	8,300.00
Steel, self-supporting, half lined	do.	do.	8,800.00
Steel, self-supporting, unlined	do.	do.	5,820.00
Steel, guyed	do.	do.	4,000.00

The following approximate costs of various sizes of a well-known radial brick chimney give an idea of the variation in cost due to increase in diameter and height:

TABLE 62.

Size of Chimney.		Cost.	Size of Chimney.		Cost.
Height.	Diameter.		Height.	Diameter.	
Feet.	Feet.		Feet.	Feet.	
75	4	$1,350.00	175	8	$7,050.00
75	6	1,950.00	175	10	7,925.00
75	8	2,650.00	175	12	8,950.00
75	10	3,725.00	175	14	9,725.00
125	6	3,500.00	200	8	9,250.00
125	8	4,250.00	200	10	10,500.00
125	10	4,675.00	200	12	11,100.00
125	12	5,125.00	200	14	12,500.00
150	8	6,150.00	250	10	16,500.00
150	10	7,125.00	250	12	18,250.00
150	12	7,750.00	250	14	21,500.00
150	14	8,275.00	250	16	24,250.00

PROBLEMS.

1. Determine the maximum theoretical draft obtainable from a chimney 200 ft. high; altitude 2250 ft. (barometer 27.5 in.); temperature outside air 80 deg. fahr.; temperature of the flue gas 500 deg. fahr.

2. Calculate the height of stack suitable for burning 20 lb. of anthracite buckwheat per sq. ft. of grate surface per hr. for a hand-fired return tubular boiler, standard setting, when the temperature of the outside air is 70 deg. fahr. and that of the flue gas is 450 deg. fahr. Assume a pressure loss in the boiler of 0.45 in.

3. Determine the height and diameter of stack for a battery of Wickes vertical water-tube boilers rated at 4000 horsepower, equipped with chain grates and burning Illinois screenings; boiler rated at 10 sq. ft. of heating surface per hp.; ratio of heating surface to grate surface 65 to 1; flue 50 ft. long; stack to be able to carry 100 per cent overload; atmosphere temperature 60 deg. fahr., average barometric pressure 29 in.; temperature of flue gas at overload 650 deg. fahr.; calorific value of the coal 11,000 B.t.u. per lb. Assume pressure drop through boiler from the curves in Fig. 150.

4. Determine the thickness of plates at various sections for a self-supporting steel stack of the height and diameter as calculated in Problem 3.

5. Determine the size of foundation for the chimney in Problem 4.

6. Design a brick chimney suitable for the data in Problem 3.

Analyze the various sections for strength and stability.

CHAPTER VIII

MECHANICAL DRAFT

151. General. — The intensity of natural draft in a chimney depends mainly upon the height of the stack and the temperature of the chimney gases, and the chimney should be designed to meet the maximum requirements, permitting the damper to be partly shut at times. There is usually no practicable means of increasing natural draft *per se* after the maximum has been reached. Again, chimney draft is peculiarly susceptible to atmospheric influence and may be seriously impaired by adverse winds and air currents. Notwithstanding these apparent limitations, by far the greater number of steam power plants depend upon chimneys for draft because of the disposition of the waste gases. In many cases artificial draft has a great advantage and under certain conditions is indispensable; it is very flexible and readily adjusted to effect various rates of combustion, irrespective of climatic influences, and permits any degree of overload without undue expenditure of energy.

Artificial draft may be broadly classified under three heads:

1. The vacuum or induced draft.
2. The plenum or forced draft, and
3. The "balanced" draft method.

In the induced draft system a partial vacuum is produced above the fire by suitable apparatus, and the effect is substantially that of natural draft.

In the forced-draft system pressure is produced in the ash pit, the air being forced through the fuel bed.

The so-called "balanced draft" system is a combination of forced draft and induced or chimney draft. The pressure created by forced draft is made sufficient to overcome the resistance of the fuel bed while the chimney or induced draft is depended upon for creating a suction throughout the furnace and setting. The adjustment is such that practically atmospheric or a slight suction pressure exists in the combustion chamber.

In all these systems the artificial draft is usually produced by either:

1. Steam jets, or
2. Centrifugal fans or exhausters.

152. Steam Jets. — Fig. 164 shows an application of a ring jet to the base of a stack. The apparatus is very simple, inexpensive in first cost, and easily applied. It consists essentially of a ring or a series of concentric rings of 1-inch pipe, perforated on the upper side

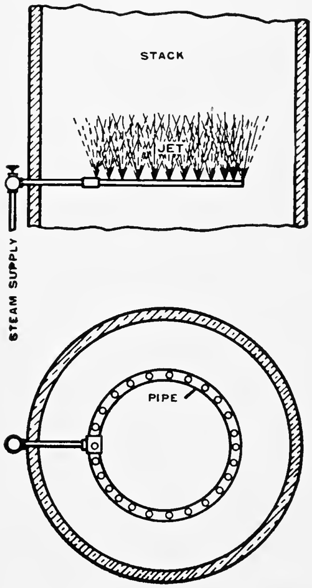


Fig. 164. Ring Steam Jet.

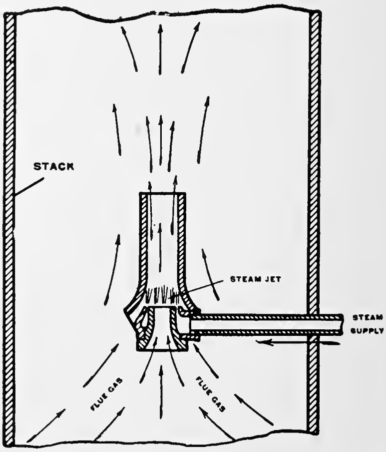


Fig. 165. Bloomsburg Jet.

with $\frac{1}{16}$- or $\frac{1}{8}$-inch holes, and placed in the base of the stack, so that the jets are discharged upward, thus creating a draft independent of the temperature of the flue gases. The steam connection to the jet is

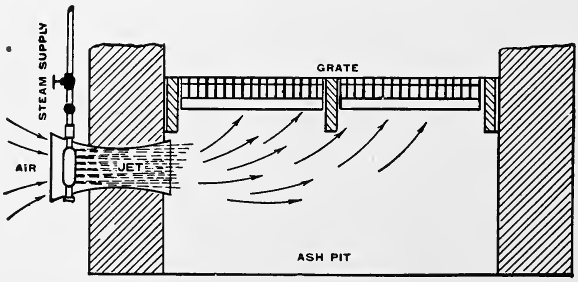


Fig. 166. McClaves Argand Blower.

generally made direct to the boiler and not to the steam main, though the jet is often produced by exhaust steam.

Fig. 165 illustrates a Bloomsburg jet, which involves to some extent the principle of the ejector.

The increase in draft produced by these devices as ordinarily installed is not great, although in locomotive practice where the entire exhaust is discharged up the stack an intense draft is obtained.

Fig. 166 shows the application of a "McClaves argand blower." The steam is discharged below the grate through a perforated hollow ring, as indicated, drawing the air through the funnel by inspiration. This creates a powerful draft by forming an air pressure in the ash pit, and is an especially useful system of forcing fires for boilers which need forcing for short periods only.

Steam jets, as ordinarily installed, are very uneconomical, since a large amount of steam is required to produce good results. Table 63, based on experiments at the New York Navy Yard, to determine the

TABLE 63.

RESULTS OF EXPERIMENTS UPON STEAM JETS AT NEW YORK NAVY YARD.*

Index of Jet.	Pounds of Water Evaporated per Hour.				
	A	B	C	D	E
In boiler making steam	463.8	580.0	361.25	528.5	545.00
In boiler supplying jets	97.5	120	30	63.2	76.25
Per cent of steam used by jet	21.2	20.7	8.3	12.0	19.0

* Annual Report of the Chief of the Bureau of Steam Engineering, U. S. Navy, 1890.

TABLE 64.

CONSUMPTION OF STEAM BLASTS COMPARED.†

Coal.	Name of Blower.	Per Cent of Air Openings in Grate.	Pounds of Dry Coal burned per Hour per Square Foot of Grate.	Per Cent of Total Steam Generated in the Boilers that is required to operate the Steam Blasts.
Rice	Young	11	25.8	11.1
Do	do	11	17.9	7.0
Do	Wilkinson	7	27.0	10.8
Buckwheat	Young	11	27.3	10.8
Do	do	11	16.7	4.6
Do	do	26	31.4	8.9
Do	McClave	11	16.4	6.7
Do	do	11	26.1	9.3
Do	Wilkinson	7	32.5	7.8
Do	do	7	45.4	10.2

† Trans. A.S.M.E., Vol. XVII. — See Whitham.

best form of steam jet for producing draft in launch boilers, shows steam consumptions of from 8.3 to 21.2 per cent of the total steam made. Table 64 gives the steam consumption of a number of types of steam jet blowers as determined by A. J. Whitman. The best performance is 4.6 per cent and the poorest 11.1 per cent of the total boiler steam generated. Steam jets below the grate are said to prevent clinkers from forming where fine anthracite coals are used, and thus to assist in keeping the fire free and open. They also assist in the economical combustion of certain low-grade fuels. See paragraph 93 for the influence of steam jets in effecting smokeless combustion.

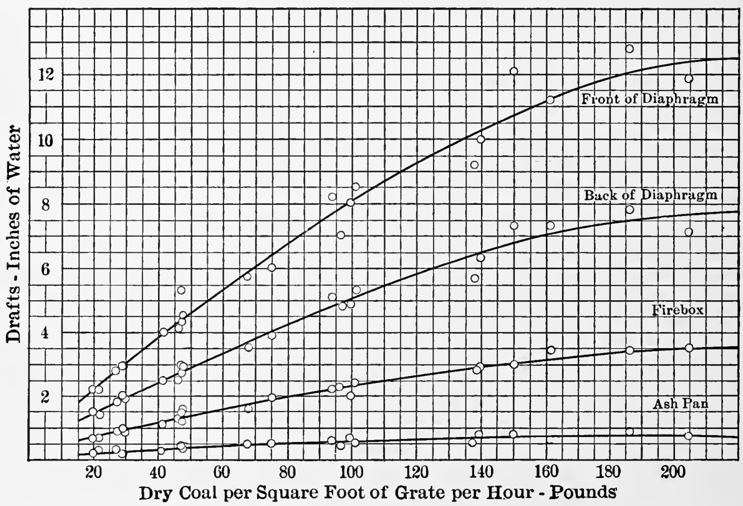


FIG. 167. The Relation between Draft and Rate of Combustion. Consolidated Locomotive.

The curves in Fig. 167 are of interest in showing the intensity of draft created by steam jets in the modern locomotive and the influence of the draft on the rate of combustion. These curves are taken from Bulletin 2, Univ. Ill., Sept. 13, 1915, p. 16.

In large modern central stations where boiler overloads of from 150 to 250 per cent above rating are desirable, steam jets and mechanical blowing and stoking appliances use but a nominal percentage of the steam generated. The results in Table 65, taken from the tests of the large Stirling boilers at the Delray Station of the Detroit Edison Company, show what may be expected from installations of this class (Jour. A.S.M.E., Nov., 1911).

153. Fan Draft. — Fig. 168 shows a typical installation of a centrifugal fan on the *forced-draft* or *plenum* principle, the fan creating a

TABLE 65.

STEAM CONSUMPTION OF DRAFT APPLIANCES AND STOKER ENGINES, 2365 H.P. STIRLING BOILER.

(Delray Station, Detroit Edison Co.)

RONEY STOKER.

No. of Test.	Per Cent of Rating.	Dry Coal per Sq. Ft. G. S. per Hr.	Steam Consumption, Per Cent of Total Generated.			Draft, Inches of Water.		
			Stoker Engines.	Steam Jets.	Total.	Below Dampers.	In Furnace.	Ash Pit.
5	94	14.81	0.19	1.56	1.75	0.16	0.24	0.10
4	152	25.97	0.15	1.43	1.58	0.55	0.22	0.02
18	195.7	33.60	0.13	1.19	1.32	1.11	0.33	0.05

TABLE 65 — *Concluded.*

TAYLOR STOKER.

No. of Test.	Per Cent of Rating.	Dry Coal per Sq. Ft. G. S. per Hr.	Steam Consumption of Stoker Engines and Turbine Blower.	Draft, Inches of Water.		
				At Blast in Tuyeres.	Suction Below Boiler Dampers.	Suction in Ash Pit.
10	92.9	16.43	2.63	0.67	0.20	0.15
9	162.8	29.23	2.87	1.73	0.53	0.06
11	211.0	38.75	3.41	2.53	0 84	0.02

All of the steam exhausted from the Taylor equipment may be returned to the feed-water heater, whereas only that exhausted from the engines in the Roney equipment may be used in this manner, hence the *net heat* used is approximately the same in both cases.

For application of steam jets to mechanical stokers see Chapter IV.

pressure in the ash pit and forcing air through the fuel. The most approved method is to pass the air through the bridge wall, thence toward the front of the grate, though it may enter through an underground duct or through the side of the setting. Forced draft is usually adopted in old plants where increased demands for power require that the boilers be forced far above their rating to save the heavy expense of new boilers, or in plants burning refuse, anthracite culm or screenings, which require an intense draft for efficient combustion. Forced draft is also well adapted for underfeed stokers of the retort type, hollow blast grates, and the closed fire-hole system. The air supply may be taken from an air chamber built around the breeching, thereby supplying the heated air to the fan and effecting a lower temperature in the breeching and a higher temperature in the furnace. The objection is sometimes raised against forced draft that the gases tend to pass outward through the fire door when the fire is cleaned or replenished, since the pressure in the furnace is greater than atmospheric. This objection may usually be overcome by suitable dampers in the blast pipe which are closed on opening the fire doors or by having sufficient stack action to create a partial vacuum in the combustion chamber. With a boiler plant of 1000 horsepower or more the cost of a forced-draft fan, engine, and stack will approximate from 20 to 30 per cent of the outlay for an equivalent brick chimney. The power consumption will depend upon the character and efficiency of the motor or engine and will range from 1 to 5 per cent of the total capacity.

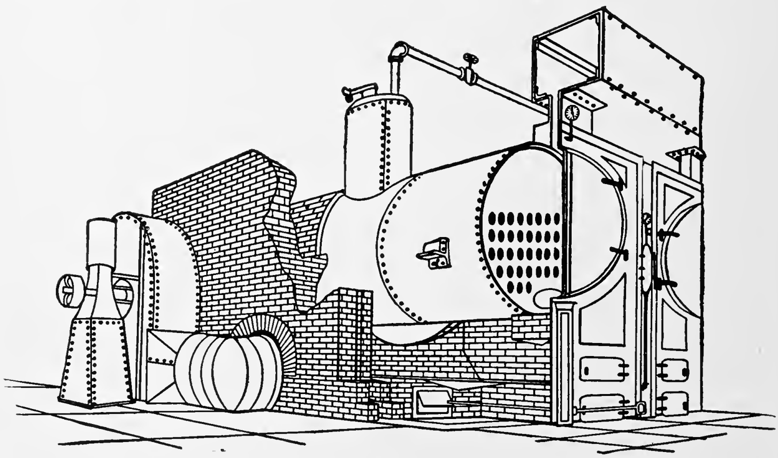


Fig. 168. Typical Forced-draft System.

Induced draft as illustrated in Fig. 168 is perhaps the most common substitute for natural draft and is extensively used in street railway and lighting plants which have high peak loads, being ordinarily installed in connection with fuel economizers. The suction side of the fan is connected with the uptake or breeching of the boiler or batteries of boilers and the products of combustion are usually exhausted through a stub stack. The illustration shows a typical installation in which two fans of the duplex type are placed above the boiler setting. The fan ducts are generally designed with a by-pass direct to the stack to be used in case of accident or when mechanical draft is not required.

Since the fan handles hot gases it must, under the ordinary conditions of practice, have a capacity approximately double that of a forced-draft fan delivering cold air, but the gases being of lower density the power required per cubic foot moved is less.

With forced draft from 200 to 300 cubic feet of air are required per pound of coal; with induced draft the fan must handle twice this volume if the gases are exhausted at 500 deg. fahr. or 300 to 450 cubic feet if exhausted at 300 deg. fahr., a temperature to be expected in connection with economizers.

The advantages of induced draft over forced draft are very pronounced. The pressure in the furnace is less than atmospheric, therefore it is not necessary to shut off the draft in cleaning fires of ash pit, and the fire burns more evenly over the entire grate area, since the

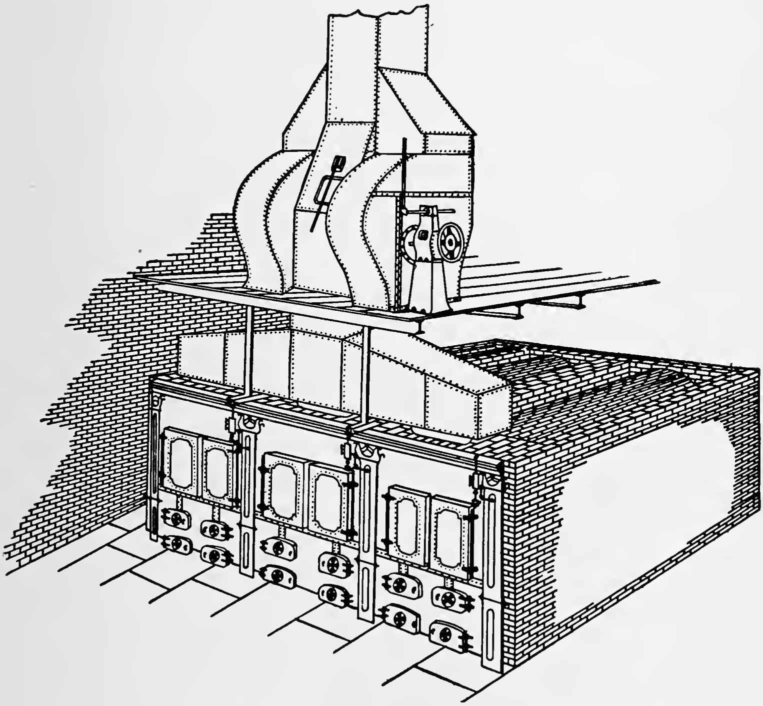


FIG. 169. Typical Induced-draft System.

draft pressures are ordinarily less than with forced draft. An induced-draft plant costs considerably more than forced draft on account of the larger fan required, but the operating expenses are but little greater. With a boiler plant of 1000 horsepower or more the cost of a single induced-draft fan, engine, stack, etc., will approximate from 40 to 50 per cent of the outlay required for a brick chimney of equivalent capacity, and the double-fan outfit will approximate from 50 to 60 per cent. The double-fan system is particularly adapted to plants which operate continuously and where even a temporary break-down is a serious inconvenience.

Turbo-undergrate draft blowers, installed in each setting, are finding favor with many engineers because of the low cost of installation. They consist essentially of small impulse steam turbines direct connected to specially designed propeller fans set in the side walls of the setting by means of wall thimbles. The fan discharges below the grate, and may be automatically controlled by damper regulation. The turbine exhaust may be discharged into the ash pit to prevent clinkers, or it may be used in the feed-water or other heating devices. They are more economical in heat consumption than the ordinary jet device.

In Europe induced draft created by a fan discharging into the base of an *evasé stack* is finding favor with many engineers. A few installations have been made in this country by the Schutte and Koerting Company, but data relative to their performance are not available. In this system a short stack (seldom exceeding 70 ft. in height) and resembling a Venturi tube is fitted with a small pressure blower near the base. The stack action is based on the injector principle and is sufficient to operate the boiler at low rating without the use of the fan. For higher ratings air is discharged into the stack just below the Venturi throat and the suction in the breeching is greatly increased. These stacks are usually applied to single boilers or batteries. The general dimensions of an evasé stack as installed in the power plant of the Ingersoll Rand Company, Phillipsburg, Pa., is shown in Fig. 170. The fan requires about 5 per cent of the rated boiler horsepower for operation and the static pressure of the blower is approximately eight times the draft requirements in the breeching.

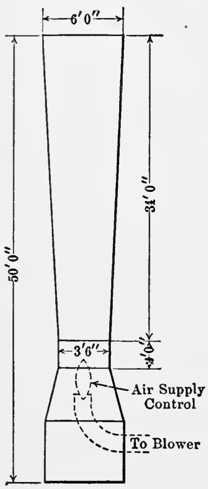


FIG. 170. Evasé Stack Capable of Furnishing Draft for the Combustion of 6000 lb. of Coal per hour with Maximum Suction at Breeching of 1.5 Inches of Water.

Mechanical Draft and the Evasé Stack: Eng. Mag., **July,** 1915, p. 525.

Tall chimneys are a necessity in most cities since legislation requires the gases to be discharged at a height above that of adjacent buildings. In such situations, with stokers of the forced-draft type, tall stacks or induced draft would at first thought appear to be a necessary evil. Experience, however, shows that suction draft is an important factor in effecting efficient combustion and in prolonging the life of the furnace brickwork. By mutually adjusting the pressure created by the forced-draft apparatus and the suction of the chimney or its equivalent, a so-called "*balanced-draft*" effect can be produced in the combustion chamber; that is, the pressure in the combustion chamber becomes practically atmospheric. The relative pressure drops are shown graphically in Fig. 171. This condition of positive pressure under the fire bed, zero or slightly suction pressure in the combustion chamber and a suction pressure throughout the rest of the setting (1) prevents discharge of the furnace gases into boiler room through leaky fire doors, inspection doors and cracked settings; (2) minimizes stratification and short circuiting of the air supply and combustible gases; (3) reduces the "soaking up" action of heat by the furnace brickwork; (4) assists reduction of air excess and (5) effects increase in overall boiler, furnace

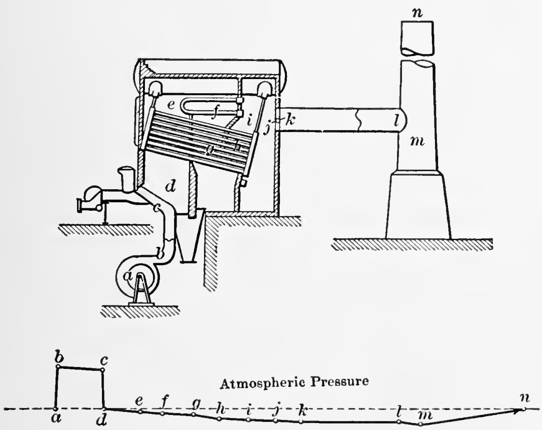


Fig. 171. Pressure Drops through Boiler — Combined Forced Draft and Chimney.

and grate efficiency. Many of our modern central stations are operating with practically balanced-draft conditions as will be seen from the data in Table 66. In these plants the stoker speed, fan speed and stack damper are automatically controlled so as to effect the desired result.

In the Essex Power Station of the Public Service Electric Company, New Jersey, which is representative of the very latest practice (1917) the chimneys are 250 feet high and are served with both forced- and induced-draft fans. The induced-draft fan gives a maximum suction in the uptake of two inches of water pressure and the forced-draft equipment is capable of maintaining a pressure of six inches water under the grates. After the gases have passed from the boiler this may be discharged directly into the stack or by closing proper dampers in the breeching can be made to pass through the economizer and then to the stack; by closing a second damper the gases will pass through the induced-draft fans before going to the stack. This makes it possible to operate the boilers under the most economical conditions at all times.

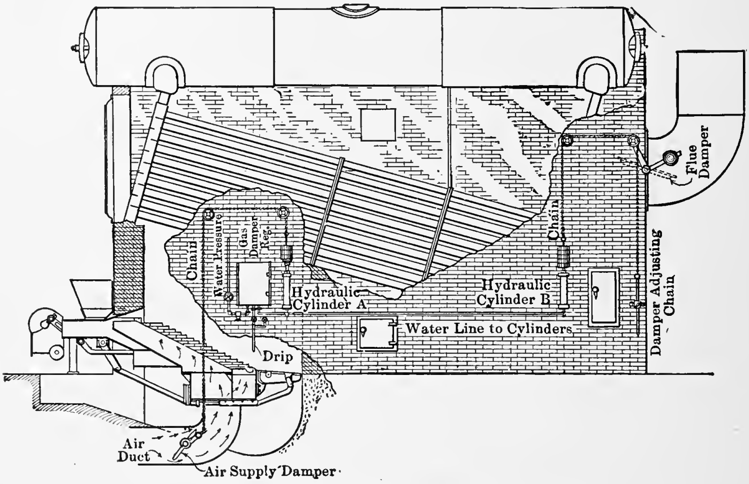


FIG. 172. McLean "Balanced-draft" System.

The term "balanced draft" as applied to furnace work originated with Embury McLean and refers strictly to his system of control, see Fig. 172, but the term is now applied to any system in which chimney and fan draft are controlled so that the pressure in the combustion chamber is approximately atmospheric.

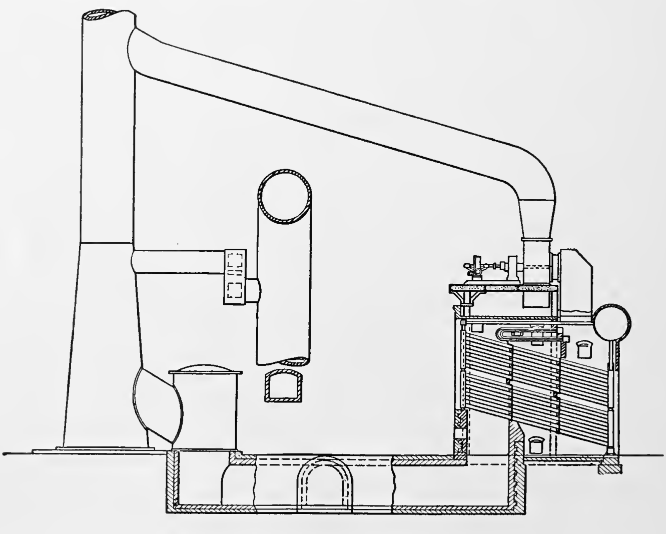


FIG. 173. Mechanical Draft as Applied to Waste-Heat Boiler for Open-Hearth Furnace.

TABLE 66.

DRAFT PRESSURES IN MODERN CENTRAL STATION AT OVERLOADS.

(Underfeed Stokers.)

Plant.	Type of Boiler.	Type of Stoker.	Boiler Rating, Per Cent of Rated Capacity.	Height of Stack Above Breeching.	Flue Temperature, Deg. Fahr.	Static Draft, Inches of Water.		
						Ash Pit.	Combustion Chamber.	Stack Side of Damper.
Delray No. 1	Stirling	Taylor	*175	242	550	+3.5	−0.03	−1.2
Delray No. 2	Stirling	Taylor	*175	196	600	+3.5	−0.03	−1.2
Delray No. 2	Stirling	Taylor	220	196	620	+4.2	−0.07	−1.3
Connors Creek	Stirling	Taylor	*175	240	580	+3.8	−0.10	−0.8
Boston Elevated	B. & W.	Taylor	240	165	515	+4.0	−0.03	−0.76
59th St. Interborough	B. & W.	Taylor	200	200	523	+3.3	−0.14	−0.52
74th St. Interborough	B. & W.	Riley	335	242	631	+5.8	−0.02	−0.74
74th St. Interborough	B. & W.	Westinghouse	292	242	609	+3.8	−0.15	−0.65

* Normal operating maximum.

Draft and Stoker Control at Waterside: Power, Nov. 7, 1914, p. 698.
Boiler Control Boards at Delray: Power, Sept. 28, 1915, p. 435.
The Essex Power Station: Power, Nov. 28, 1916, p. 739.
Performance of Boilers with Balanced Draft: Elec. Wld., Sept. 9, 1916, p. 522; Aug. 12, 1916, p. 321.

154. Types of Fans. — Centrifugal fans for mechanical draft may be divided into two general classes; those having rotors with a few straight or slightly curved blades of considerable length radially, Fig. 174, commonly designated as *steel-plate fans*, and those having rotors

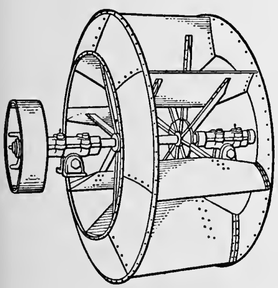


FIG. 174. Standard Steel-plate Fan Wheel.

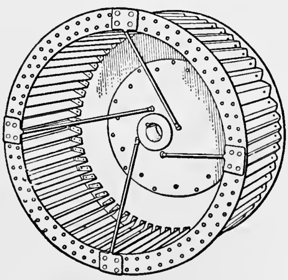


FIG. 175. "Sirocco" Wheel — Turbine Type Impeller.

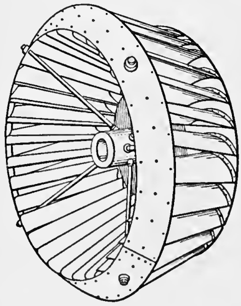


FIG. 176. Single Conoidal Fan Wheel.

with a number of short curved blades, Figs. 175 and 176, generally known as *multi-vane fans*. Both of these types are found in the modern power plant though the multi-vane construction is the more common. Tall, narrow steel-plate fans are frequently used for induced-draft work partly because the narrow wheel permits of shorter overhang on the fan bearing and partly because they may be operated at low speed and are suitable for direct connection to steam engines. Multi-vane fans require less space than steel-plate fans of equal capacity and efficiency and on account of higher speed requirements are more suitable for direct connection to electric motors or steam turbines. Each type has different characteristics, the nature of which controls the selection for a given set of operating conditions. The housings may be arranged for top or bottom horizontal discharge, up or down blast, or special, depending upon the arrangement of the draft system.

155. Performance of Fans. — On account of the great number of variables involved in the construction and operation of fans simple equations or formulas for proportioning the various elements are practically impossible. The design of a new fan is largely a matter of trial and error based on experiments. For this reason no attempt will be made to analyze the problem of design and only such elementary theory will be discussed as is necessary for a clear understanding of the principles of operation.

Pressure. If the delivery pipe of a fan is sealed against discharge there is but one pressure in the conduit, namely, *static pressure.* Referring to Fig. 177, *A* and *B* represent Pitot tubes inserted in the discharge or suction pipe of a centrifugal fan, *A* being bent to face the

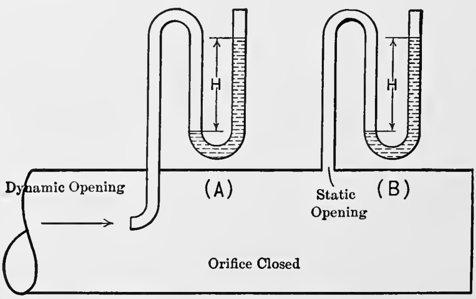


Fig. 177.

current while *B* is flush with the inside wall of the casing at right angles to it. *A* receives the full impulse of the stream, and the manometer indicates the total or *dynamic pressure*, while *B* registers the *static pressure* only. With the pipe sealed against discharge, resistance to flow is a maximum, there is no flow and the water depression in both manometers will be the same, that is, there is only static pressure in the conduit.

If the discharge orifice is opened to its maximum and there are no frictional resistances the static pressure indicated by manometer *B*,

Fig. 178, becomes zero while that in A stands at a height equivalent to the full impulse of the stream, that is, there is only velocity pressure in the conduit.

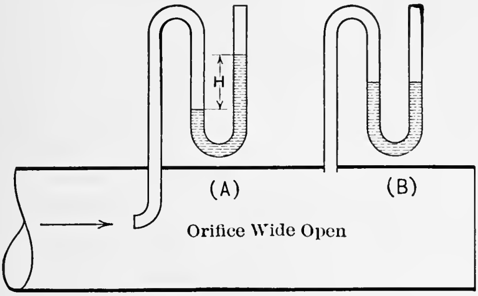


FIG. 178.

If the orifice is partly closed, as in Fig. 179, there will be a water depression in both manometers A and B, that is, there is both velocity and static pressure in the conduit. The difference between the depression in A and B is the pressure due to velocity. By connecting the two manometers as indicated in Fig. 179 (C), the *velocity pressure* is given directly.

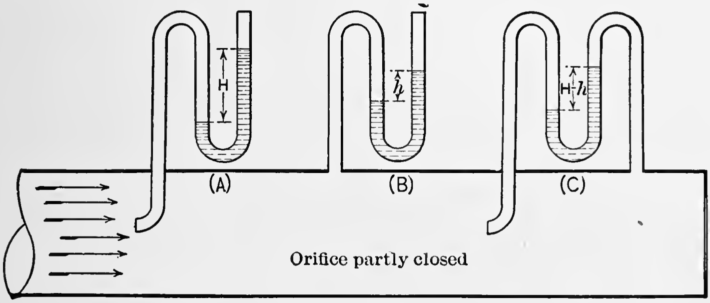


FIG. 179.

Pressure resulting from the impulse of a current of air flowing at a velocity corresponding to that of the tip of the blades is commonly designated as the *peripheral velocity pressure.*

The ratios between the various pressures are of great importance in fan engineering and manufacturers publish characteristic curves showing this relationship for various conditions of operation. These characteristics vary with the type of fan and the design of the blades and housing. A few examples are shown in Figs. 180–183.

The "ratio of opening," Fig. 180, refers to the actual percentage of opening compared with the maximum. The "ratio of effect" is the relative effect produced by restricting the discharge.

Suppose a steel-plate fan with an unrestricted inlet and outlet delivers 25,000 cu. ft. of air per minute against a dynamic head of 2.14 in. with a peripheral velocity requiring 4.5 horsepower. It appears from the curves in Fig. 180 that if the discharge outlet is restricted to 50 per cent of the full area, only 12,500 cu. ft. will be delivered. The dynamic pressure will be increased to 4.28 in., and the power required drops to 2.7 horsepower. If the outlet be still further reduced to 20 per cent of the full opening the capacity will drop to 5000 cu. ft., the pressure

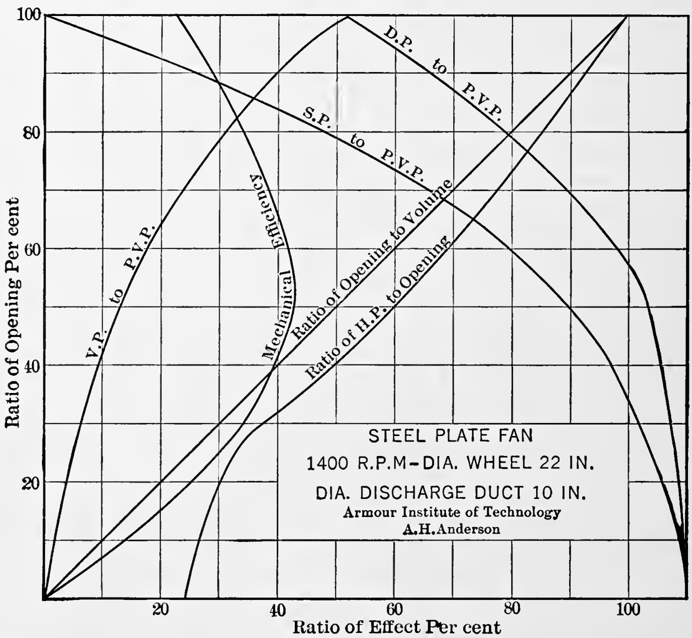


FIG. 180. Characteristic Curves of 22-in. Buffalo Steel-plate Blower (at 1400 R.P.M.).

will increase to 4.41 inches, and the power will be decreased to 1.35 horsepower. With a discharge area of 50 per cent, the mechanical efficiency is a maximum, and equal to about 43 per cent. With orifice closed the horsepower required to drive the fan is about 24 per cent of that required when discharging the maximum volume of air.

Velocity: In a centrifugal fan operating under constant orifice conditions and at known air density, the theoretical velocity and pressure developed bear a definite relation to the peripheral velocity of the fan. For ordinary fan work where air is at a low pressure the relationship between pressure and velocity is substantially

$$V = \sqrt{2\,gh}, \tag{114}$$

in which

V = velocity, ft. per sec.,
g = acceleration of gravity; 32.2 (approximately),
h = head of air causing flow, ft.

Equation (114) may be reduced to the convenient form

$$v = 1096.5 \sqrt{p \div \delta}, \tag{115}$$

in which

v = velocity, ft. per min.,
p = pressure drop producing velocity, in. of water,
δ = density of the air, lb. per cu. ft.

For standard conditions, dry air at 70 deg. fahr. and 29.92 barometer:

$$v = 4005 \sqrt{p}. \tag{116}$$

Where quietness of operation is necessary the velocity should be limited to 2000 ft. per min. but where this is not essential duct velocities as high as 4000 ft. per min. may be used. Since the friction losses of a piping system vary with the square of the velocity the usual compromise must be made between size and velocity, otherwise the pressure losses become excessive.

Capacity. For a given fan size, piping system and air density the capacity, Q, varies directly as the velocity and hence as the speed of the fan, thus,

$$Q = vA, \tag{117}$$

in which

Q = volume, cu. ft. per min.,
v = velocity, ft. per min.,
A = area of the conduit.

Since the velocity varies as the square root of the pressure drop

$$Q = KA \sqrt{p}, \tag{118}$$

in which

K = coefficient determined by experiment; other notations as in equation (115).

Horsepower. The horsepower required to operate a fan varies directly with the capacity and the total or dynamic pressure, thus:

$$\text{Hp.} = \frac{5.2\, Q \times P_d}{33{,}000 \times E} = 0.000157\, \frac{Q \times P_d}{E}, \tag{119}$$

in which

E = total efficiency of the blower,
P_d = dynamic pressure, in. of water.

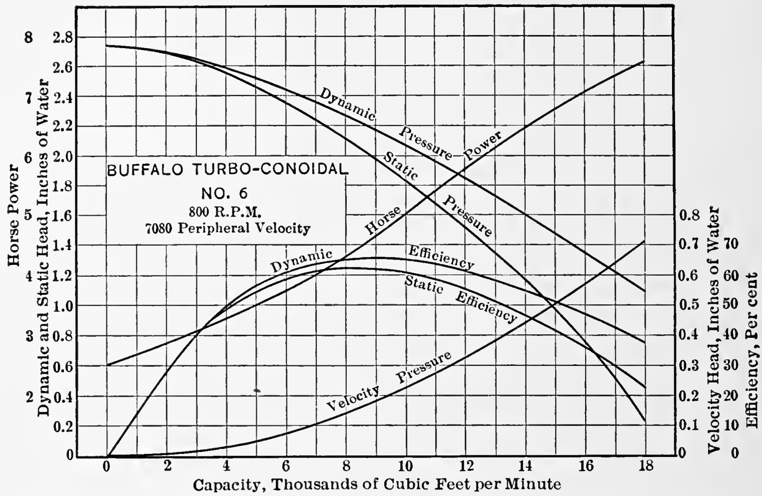


FIG. 181. Typical Characteristic Curves of Buffalo "Conoidal" Blower.

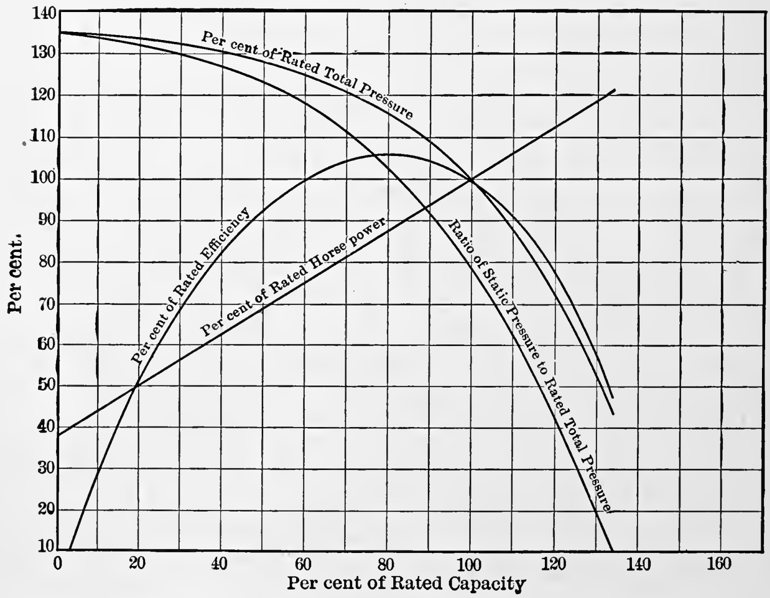


FIG. 182. Typical Characteristic Curves of Buffalo "Planoidal" Steel-plate Exhausters.

Combining equations (118) and (119) and reducing, remembering that for constant orifice conditions and at known air density the velocity pressure bears a definite relation to the peripheral velocity, we have

$$\text{Hp.} = Bp^{\frac{3}{2}}, \tag{120}$$

in which

B = coefficient involving all constants and reduction factors.

Equation (120) shows that the horsepower varies as the cube of the square root of the pressure.

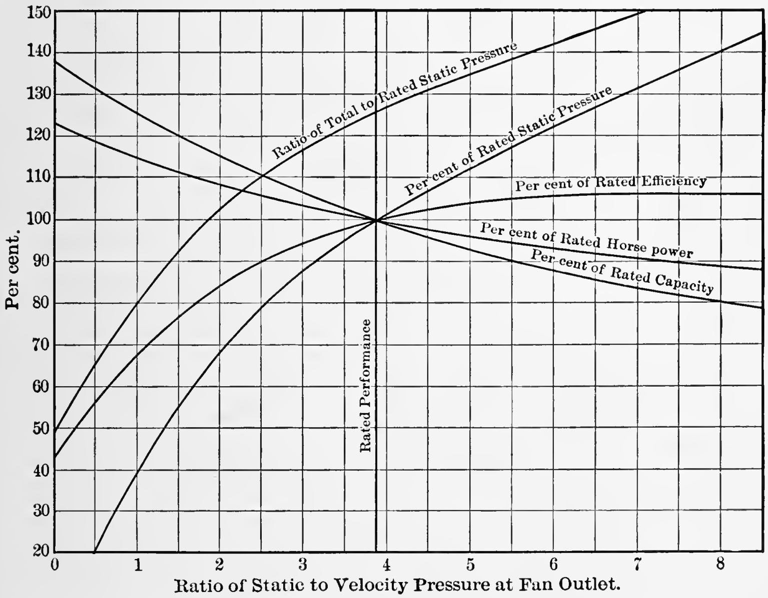


FIG. 183. Typical Characteristic Curves of Buffalo "Planoidal" Steel-plate Exhausters.

Since the capacity is directly proportionate to the peripheral velocity or fan speed, and the pressure developed varies directly as the square of the speed it follows that the horsepower varies as the cube of the speed, thus:

$$\text{Hp.} = MN^3, \tag{121}$$

in which

M = coefficient involving all constant and reduction factors,
N = speed of the fan, r.p.m.

The marked increase in power required for even a moderate increase in speed should be borne in mind in selecting a fan. It is, as a rule, more economical to err in selecting too large a fan than one which must be forced above its rated speed.

The capacity varies directly with the speed; therefore the horsepower also varies with the cube of the capacity.

Manometric Efficiency. This efficiency is the ratio of the dynamic head as actually observed to the maximum theoretical dynamic head, or

$$E_{\text{man.}} = \frac{h}{H}, \tag{122}$$

in which h is determined from the actual manometer reading and H is the theoretical maximum head.

Volumetric Efficiency. This is the ratio between the actual volume of air passing in a given time divided by the impeller displacement for the same period, or

$$E_{\text{vol.}} = \frac{4\,Q}{D^2NB}, \tag{123}$$

in which

Q = volume discharge, cubic feet per minute,
D = diameter of the impeller, feet,
B = width of the impeller, feet,
N = r.p.m.

Mechanical efficiency, or simply fan efficiency is the ratio of the total work done by the fan in moving the air to the horsepower input to the fan, or

$$E_{\text{mec.}} = \frac{Qh}{Hi \times 33{,}000}, \tag{124}$$

in which

Q = weight discharged, pounds per minute,
h = dynamic head, feet of air,
Hi = horsepower input.

Two efficiencies are sometimes given, (1) that based on the dynamic head as in equation (124) and that based on the static head. See Fig. 181.

An analysis of the performance of fans under various operating conditions is beyond the scope of this test and the reader is referred to the accompanying bibliography for an extended study.

Measurement of Air in Fan Work: C. H. Treat, Jour. A.S.M.E., Sept., 1912, p. 1341. *Some Experiences with the Pitot Tube on High and Low Air Velocities:* F. H. Kneeland, Jour. A.S.M.E., Nov., 1911, p. 1407. *Experiments with Ventilating Fans and Pipes:* Capt. D. W. Taylor, Soc. Naval Arch. and Marine Engrs., 1905, p. 35. *The Measurements of Gases:* Carl C. Thomas, Jour. Frank. Inst., Nov., 1911, p. 411. *Experiments with the Pitot Tube in Measuring the Velocities of Gases:* R. Burnham, Eng. News, Dec. 21, 1905, p. 660. *Pressure Fans vs. Exhaust Fan:* Bulletin Am. Inst. Min. Engrs., Feb., 1909. *A.S.M.E.* 1915 *Code for Testing Fans:* Trans. A.S.M.E., Vol. 37, 1915, p. 1342.

156. Selection of Fan. — In general, the multi-vane is more efficient than the steel-plate fan as ordinarily constructed, and requires less space than the latter for equal capacity and efficiency. Another important advantage lies in the fact that the higher speed of the multi-vane fan permits of direct connection to high-speed prime movers. The steel-plate blower, however, is not necessarily a low efficiency device since by special design it may be made to give higher efficiencies than obtained from the curved short blade construction. Where first cost is a consideration and where space limitation is of little consequence the steel-plate fan may be used to advantage. In small plants the power requirements for the mechanical draft system are low and the type of fan has but little effect on the overall cost of operation, but in large central stations the power requirements are considerable and the type and attending pressure characteristics greatly influence the ultimate economy.

Having selected the type of fan the first step is to determine the size best suited to the required conditions. The influencing factors are primarily the volume of air or gas to be delivered and the static pressure necessary to overcome the frictional resistances of the system. The air requirements and the pressure drop through the boiler equipment may be calculated as shown in paragraphs 23 and 127. The frictional resistance of the air ducts and dampers must be included in determining the maximum static pressure.

The next step is to select from capacity tables (furnished by fan builders) the nearest commercial size which will meet the volume and static pressure requirements. If the conditions are different from those published in the tables the performance under specified conditions may be approximated by calculation or taken directly from "characteristic curves" of the particular type under consideration.

Two sets of capacity tables are found in practice, the "rated capacity" (such as are reproduced in Tables 67 and 69) and the "variable capacity" (one element of which is given in Table 70). The former gives the capacity, speed, and horsepower of the different fans for various static or total pressures, when operating at what is approximately the highest efficiency. These rated capacity tables are self-explanatory and require no particular discussion. The variable capacity tables give the performance of each size of fan on either side of the condition for maximum efficiency and offer practically the same information as the characteristic curves. By means of these tables the performance of the fan may be readily obtained for practically all conditions of operation.

TABLE 67.

CAPACITIES OF FORCED-DRAFT FANS.

(Steel Plate Fans.)

For Forced Draft, Temperature of Air 60°.

Diameter of Fan.	Cubic Feet of Air Delivered to Furnace per Minute.	Pressure in Inches of Water. 0.5		0.75		1.00		1.25		1.50		2.00		2.50	
		R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.
2′ 6″	4,200	510	1.6	560	1.8	600	1.9	640	2.1	710	2.3	780	2.5	850	2.7
3′	5,800	430	2.2	460	2.4	490	2.6	530	2.8	590	3.1	640	3.4	710	3.8
3′ 6″	7,800	360	3.0	400	3.3	420	3.5	450	3.8	500	4.2	550	4.6	610	5.1
4′	10,000	320	3.9	350	4.2	370	4.4	400	4.9	440	5.4	480	5.9	530	6.5
4′ 6″	12,400	290	4.8	310	5.2	330	5.6	360	6.0	400	6.7	430	7.3	470	8.0
5′	15,200	250	5.9	270	6.4	290	6.8	310	7.4	350	8.2	380	8.9	420	9.8
5′ 6″	18,200	230	7.0	250	7.7	270	8.2	300	8.8	330	9.8	360	10.6	390	11.8
6′	21,400	210	8.3	230	9.1	250	9.6	260	10.4	290	11.5	320	12.5	350	13.9
7′	28,800	180	11.2	200	12.2	210	13.0	230	14.0	250	15.5	280	16.8	300	18.7
8′	37,200	160	14.4	170	15.7	190	16.7	200	18.1	220	20.1	240	21.8	270	22.5
9′	46,800	140	18.1	160	19.8	170	21.1	180	22.7	200	25.3	220	27.4	240	30.3
10′	57,400	130	22.2	140	24.3	150	25.8	160	27.9	180	3.1	200	33.6	210	37.2

Discharge velocity 2000 feet per minute.

TABLE 68.

CAPACITIES OF INDUCED-DRAFT FANS.

(Steel Plate Fans.)

For Induced Draft, Temp. of Flue Gases 500°.

Diameter of Fan.	Cubic Feet of Air at 60° Temp. Drawn into Furnace per Minute.	Pressure in Inches of Water. 0.5		0.75		1.00		1.25		1.50		2.00		2.50	
		R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.	R.P.M.	H.P.
2′ 6″	3,000	688	2.2	756	2.4	810	2.6	864	2.8	958	3.1	1053	3.4	1147	3.6
3′	4,200	580	3.0	621	3.2	661	3.5	715	3.8	796	4.2	864	4.6	958	5.1
3′ 6″	5,700	486	4.0	540	4.5	567	4.7	607	5.1	675	5.7	742	6.2	823	6.9
4′	7,300	432	5.3	472	5.7	500	6.1	540	6.6	594	7.3	648	8.0	715	8.8
4′ 6″	9,300	390	6.5	418	7.0	445	7.5	486	8.1	540	9.0	580	9.8	634	10.8
5′	11,100	337	8.0	364	8.6	391	9.2	418	10.0	472	11.1	513	12.0	567	13.2
5′ 6″	13,300	310	9.5	337	10.4	364	11.1	405	11.9	445	13.2	486	14.3	526	15.9
6′	15,600	283	11.2	310	12.3	337	13.0	351	14.0	391	15.5	432	16.9	472	18.7
7′	,000	243	15.1	270	16.5	283	17.5	310	18.9	337	20.9	378	22.6	405	25.2
8′	27,100	216	19.4	230	21.2	256	22.5	270	24.4	297	27.1	324	29.4	364	30.4
9′	34,200	189	24.4	216	26.7	230	28.5	243	30.6	270	34.1	297	37.0	324	40.9
10′	41,900	175	30.0	190	32.8	202	34.8	216	37.6	243	41.8	270	45.3	283	50.2

TABLE 69.

CAPACITIES OF FORCED-DRAFT FANS.*

(Sirocco Type.)

(Figures given Represent Dynamic Pressures in Ozs. per Sq. In. For Static Pressure Deduct 28.8 Per Cent. For Velocity Pressure Deduct 71.2 Per Cent.)

Diam. Wheel.		¼ Oz.	½ Oz.	¾ Oz.	1 Oz.	1¼ Oz.	1½ Oz.	1¾ Oz.	2 Oz.	2½ Oz.	3 Oz.
6	Cu. Ft.	155	220	270	310	350	380	410	440	490	540
	R.P.M.	1,145	1,615	1,980	2,200	2,560	2.800	3,025	3,230	3,616	3,960
	B.H.P.	0.0185	0.052	0.095	0.147	0.205	0.270	0.34	0.42	0.58	0.76
12	Cu. Ft.	625	880	1,080	1,250	1,400	1,530	1,650	1,770	1,970	2,170
	R.P.M.	572	808	990	1,145	1,280	1,400	1,512	1,615	1,808	1,980
	B.H.P.	0.074	0.208	0.381	0.588	0.82	1.08	1.36	1.66	2.32	3.05
18	Cu. Ft.	1,410	1,990	2,440	2,820	3,160	3,450	3,720	3,980	4,450	4,880
	R.P.M.	381	538	660	762	850	933	1,010	1,076	1,204	1,320
	B.H.P.	0.167	0.470	0.862	1.33	1.85	2.43	3.07	3.75	5.25	6.9
24	Cu. Ft.	2,500	3,540	4,340	5,000	5,600	6,120	6,620	7,080	7,900	8,680
	R.P.M.	286	404	495	572	640	700	756	807	904	990
	B.H.P.	0.296	0.832	1.53	2.35	3.28	4.32	5.44	6.64	9.3	12.2
30	Cu. Ft.	3,910	5,520	6,770	7,820	8,750	9,600	10,350	11,050	12,350	13,550
	R.P.M.	228	322	395	456	510	560	604	645	722	790
	B.H.P.	0.460	1.30	2.40	3.68	5.15	6.75	8.53	10.4	14.5	19.1
36	Cu. Ft.	5,650	7,950	9,750	11,300	12,640	13,800	14,900	15,900	17,800	19,500
	R.P.M.	190	269	330	381	425	466	504	538	602	660
	B.H.P.	0.665	1.87	3.44	5.30	7.40	9.72	12.25	15.0	20.9	27.5
48	Cu. Ft.	10,000	14,150	17,350	20,000	22,400	24,500	26,500	28,300	31,600	34,700
	R.P.M.	143	202	248	286	320	350	378	403	452	495
	B.H.P.	1.18	3.32	6.10	9.40	13.1	17.2	21.75	26.6	37.1	48.8
60	Cu. Ft.	15,650	22,100	27,100	31,300	35,000	38,400	41,400	44,200	49,400	54,200
	R.P.M.	114	161	198	228	255	280	302	322	361	396
	B.H.P.	1.84	5.20	9.58	14.7	20.6	27.0	34.1	41.6	58.2	76.5
72	Cu. Ft.	22,600	31,800	39,000	45,200	50,600	55,200	59,600	63,600	71,200	78,000
	R.P.M.	95	134	165	190	212	233	252	269	301	330
	B.H.P.	2.66	7.48	13.7	21.2	29.6	38.9	49.0	59.8	83.6	110
84	Cu. Ft.	30,800	43,400	53,200	61,600	68,700	75,200	81,200	86,800	97,100	106,400
	R.P.M.	81	115	142	163	182	200	216	231	258	283
	B.H.P.	3.61	10.2	18.7	28.9	40.4	53.0	66.8	81.7	114	150
90	Cu. Ft.	35,250	49,800	61,000	70,500	78,800	86,400	93,300	99,600	111,200	122,000
	R.P.M.	76	107	132	152	170	186	201	214	241	264
	B.H.P.	4.14	11.7	21.5	33.1	46.2	60.7	76.7	93.6	131	172

* A number of sizes have been omitted.

157. Chimney vs. Mechanical Draft. — The choice of chimney or mechanical draft depends largely upon local conditions. Where there are no limitations to the height of stack mechanical draft offers many advantages over chimney draft. With certain types of grates and for low-grade fuels and anthracite culm or dust, it is indispensable. Again, where a fair quality of fuel is obtainable the size of plant may determine the choice.

First Cost: In small plants of, say, 100 to 150 horsepower the cost of a guyed steel chimney, 75 feet in height or less, would cost practically nothing for operation, while the power required to operate a fan in so small a plant would amount to 5 per cent or more of the total steaming capacity.

A tall, self-supporting chimney for larger plants, however, is very costly as compared with a fan system of equal capacity. For example, a brick chimney 175 feet high and 10 feet in diameter, foundation and all, capable of furnishing the necessary draft for a 3000-horsepower plant, will cost about $10,000. A two-fan induced system of equivalent capacity will cost in the neighborhood of $5000, a one-fan system $3500, and a forced-draft system $2500. See Fig. 179. With interest at 5 per cent, depreciation 5 per cent, taxes 1 per cent, and insurance one-half per cent, the annual fixed charges will be $575, $402.50, $287.50 respectively, for the fan equipment.

TABLE 70.

TYPICAL VARIABLE CAPACITY CHART.

Performance of No. 6 Buffalo Turbo-Conoidal Fan.

Outlet Velocity, Ft. Per Min.	Capacity, Cu. Ft. Per Min.	Add for Total Pressure.	Static Pressure, Inches of Water.									
			½		1		1½		2		3	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	5,250	0.063	415	0.71	....	....	....	....	....	....	....	......
1200	6,300	0.090	443	0.94	563	1.60	663	2.29	748	3.13	....	......
1400	7,350	0.122	472	1.23	587	1.99	682	2.76	767	3.56	910	5.57
1600	8,400	0.160	503	1.56	613	2.43	705	3.31	785	4.19	930	6.09
1800	9.450	0.202	535	1.96	642	2.93	730	3.92	808	4.93	947	6.88
2000	10,500	0.250	562	2.41	670	3.51	757	4.61	833	5.71	967	7.91
2200	11,550	0.302	602	2.94	702	4.19	785	5.37	858	6.58	988	9.01
2400	12,600	0.360	637	3.53	733	4.93	813	6.23	885	7.54	1013	10.19
2600	13,650	0.422	670	4.21	765	5.76	843	7.18	915	8.61	1040	11.45
2800	14,700	0.489	708	5.01	798	6.69	873	8.24	945	9.76	1067	12.85
3000	15,750	0.560	745	5.87	832	7.71	907	9.42	973	11.09	1093	14.29
3200	16,790	0.638	....	....	....	....	970	12.06	1037	14.02	1152	17.71

This chart has been considerably condensed. In the original table static pressures and velocities are given for a wider range and at narrower increment.

Depreciation and Maintenance: The depreciation of a well-designed masonry or concrete stack is very low, and 2 per cent is a liberal factor. Maintenance is practically negligible, as it requires no attention whatever for years. A steel stack, however, must be kept well painted or corrosion will take place rapidly. The depreciation and maintenance charges on a mechanical-draft system will range from 4 per cent to 10 per cent of the original outlay.

Cost of Operation: Once erected, the comparative cost of operating a chimney is practically nothing; that is, of course, on the assumption that the chimney and fan exhaust equal volumes of gas per pound of fuel and at the same temperature. A fan system requires for its operation from one and one-half per cent to five per cent of the total steaming capacity of the plant, depending upon the type and character of the fan engine or motor, and the conditions of operation.

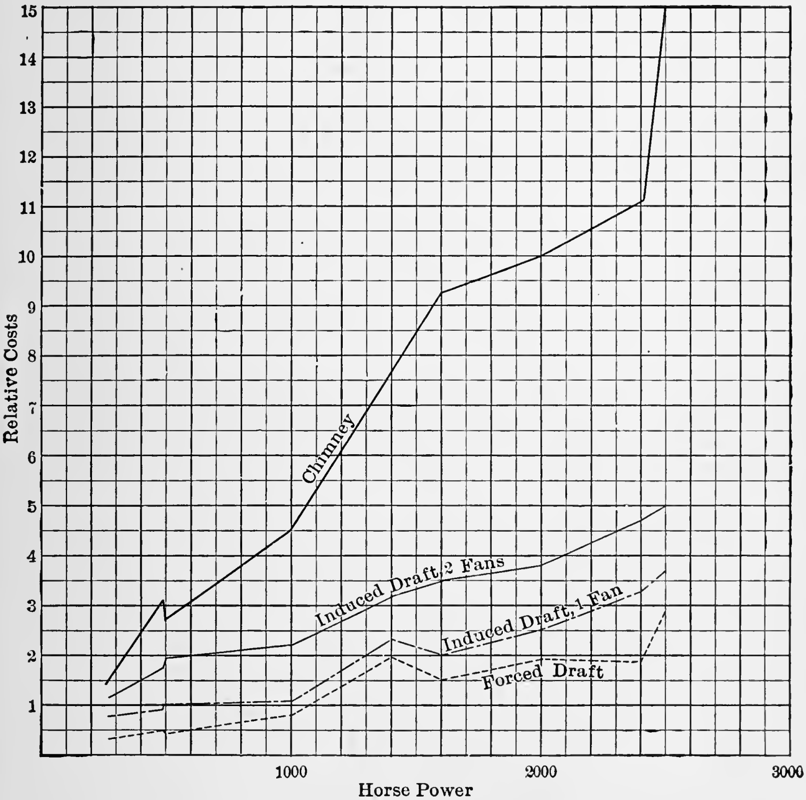


FIG. 184. Comparative Costs of Chimneys and Mechanical Draft (W. B. Snow).

Efficiency: With fan draft a very thick fire can be maintained on the grate, thus permitting a high rate of combustion, and minimum air per pound of fuel, both of which result in increased boiler efficiency. The influence of the rate of combustion on air supply in a specific case is illustrated in Fig. 185. For the same temperature of discharge each pound of air in excess of theoretical requirements results in a loss of about one per cent of the total heat in the fuel. With fan draft an average figure is 18 pounds of air per pound of bituminous coal against 24 pounds for the chimney, a saving of 5 per cent in favor of the fan.

Again, a fan permits of a low temperature of the flue gases without affecting the draft, while lowering the temperature in the chimney reduces the draft as shown in Table 36. From Table 14 we see that a reduction in flue gas temperature of 25 deg. fahr. will increase the

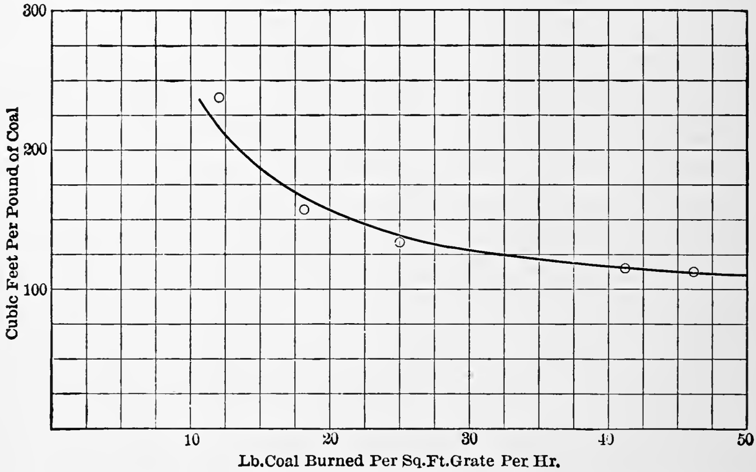


FIG. 185. Influence of Rate of Combustion on Air Supply — Forced Draft.

boiler efficiency about one per cent. With an economizer the flue gases may be reduced to 350 deg. fahr., with a net saving of about $500 - 350 = 150$, or 6 per cent of the total fuel. It is in this connection that the fan draft is peculiarly suitable. Of course, the chimney may be provided with an economizer, effecting the same reduction in temperature, but its height must be made sufficiently great to overcome the additional resistance of the economizer and the reduction in temperature of the chimney gases.

Flexibility: With a fan the draft may be readily regulated for sudden increased or decreased requirements, independent of the boiler performance. Damp and muggy days appreciably affect the draft of a chimney, as do adverse air currents and high winds.

Smoke: Smokeless combustion is more readily effected with artificial draft than with natural draft, as a thicker fire can be carried, and the correct proportion of air can be more readily adjusted.

Advantages of Forced Draft on Peak Loads: Elec. Wld., July 8, 1916, p. 68; Sept. 16, 1916, p. 583.

Notes on Fans: Power, June 15, 1915, p. 816.

PROBLEMS.

1. Dry air is flowing through a conduit, the velocity head (as indicated in Fig. 179) being one inch water. If one cu. ft. air weighs 0.074 lb., required the velocity in ft. per sec.

2. Let the cross-sectional area of the conduit in Prob. 1 be 2 sq. ft. and the static pressure 0.5 in water. Required the output horsepower of the fan.

3. It is required to supply 20,000 cu. ft. air per min. to a furnace under a static pressure of one inch water. The cross-sectional area of the conduit is 3.33 sq. ft. The mechanical efficiency of the fan is 40 per cent. One cu. ft. of air weighs 0.074 lb. Calculate the horsepower required to drive the fan.

4. Required the horsepower necessary to operate the fan in Problem 3 if the static pressure is increased in 2 in. water.

5. If the rated speed of fan in Problem 3 is 2000 r.p.m. required the horsepower if the speed is increased to 4000 r.p.m.

6. The demands on a fan running 2000 r.p.m. have increased and it is estimated the fan will deliver the required volume of air if speed is increased to 4000 r.p.m. Show why it will be much more economical to replace blower with one designed to deliver the required volume.

7. Required the capacity of an induced fan suitable for the conditions stated in Problem 3, Chapter VI.

8. Required the power necessary to operate the fan in Problem 7, if its mechanical efficiency is 60 per cent.

CHAPTER IX

RECIPROCATING STEAM ENGINES

158. Introductory. — The type of prime mover best suited for a given installation is the one which delivers the required power at the lowest cost, taking into consideration all charges, fixed and operating. These include not only the cost of fuel, labor, supplies and repairs, but all overhead charges such as interest on the investment, depreciation, maintenance and taxes. Space requirements and continuity of operation are often of vital importance, and may greatly influence the selection of type of prime mover and auxiliary apparatus. In many situations the gas engine and producer are productive of the highest commercial economy; in others the choice lies between the reciprocating steam engine or turbine, occasionally the hydroelectric plant offers the best returns, but each proposed installation is a problem in itself, and general rules are without purpose.

The reciprocating steam engine is the most widely distributed prime mover in the power world, and although its field of usefulness has been greatly encroached upon in recent years by the steam turbine and gas engine it is still an important heat engine and will probably continue to be a factor for years to come. In a general sense the piston engine is superior to the turbine for variable speed, slow rotative speeds and heavy starting torque, while the turbine has superseded the engine for large central station units and for auxiliaries requiring high rotative speed. The high-speed turbine in connection with efficient reduction gearing has many advantages over the piston engine for low-speed drives and is rapidly replacing the latter in this connection. From a purely thermal standpoint the Diesel type of internal combustion engine is superior to the steam engine and the turbine is more economical in space requirements, but taking into consideration all of the items affecting the production of power, the reciprocating engine may still prove to be the better investment in many situations, at least for sizes under 1000 horsepower.

Improvement in the heat efficiency of the piston engine within the past three years has been remarkable and single cylinder units are being operated with steam consumptions lower than that obtained from the older types of compound units. A few years ago the piston engine appeared to be doomed to the scrap heap but the unusual economies effected in the later designs has made it once more a formidable competitor of the steam turbine, at least for moderate power requirements and non-condensing service.

Present Status of Prime Movers: Pro. A.I.E.E., June, 1914, p. 953; Jan., 1915, p. 102.

159. The Ideal Engine. — In every heat engine the working fluid goes through a circuit or cycle of operation. Beginning at a particular condition it passes through a series of successive states of pressure, volume and temperature and returns to the initial condition. An ideally perfect engine which effects the highest possible conversion of heat into mechanical work for a given cycle is taken as a standard of comparison for the performance of the actual engine. There are several cycles which simulate more or less the action of steam in the actual engine, but the Rankine cycle meets the conditions of most engines and for that reason has been adopted as a standard. The various cycles are treated at length in Chapter XXIV and need not be considered here.

In order to realize the ideal Rankine cycle the walls of the cylinder and the piston must be non-conducting, expansion after cut-off must be adiabatic and carried down to the existing back pressure, the action of the valves must be instantaneous and steam passages must be sufficiently large to prevent wire drawing. None of these conditions is fulfilled by the actual engine. The various losses which prevent the actual engine from obtaining the efficiency of the ideal are outlined in paragraphs 169 to 177.

The heat supplied, heat consumption, efficiency and water rate of a perfect engine operating in the Rankine cycle are treated at length in Chapter XXIV and may be summed up as follows:

$$\text{Heat supplied} = H_i - q_n, \text{ B.t.u.,} \tag{126}$$

$$\text{Heat absorbed} = H_i - H_n, \text{ B.t.u.,} \tag{127}$$

$$\text{Efficiency, } E_r = \frac{H_i - H_n}{H_i - q_n}, \tag{128}$$

$$\text{Water rate, } W_r = \frac{2546}{H_i - H_n}, \text{ lb. per hp-hr.,} \tag{129}$$

in which

H_i = initial heat content of the steam,

H_n = final heat content after adiabatic expansion from initial condition to final condition n,

q_n = heat of the liquid corresponding to exhaust temperature.

The average engine seldom expands to the existing back pressure and though this is a fault chargeable against it, occasion may arise to

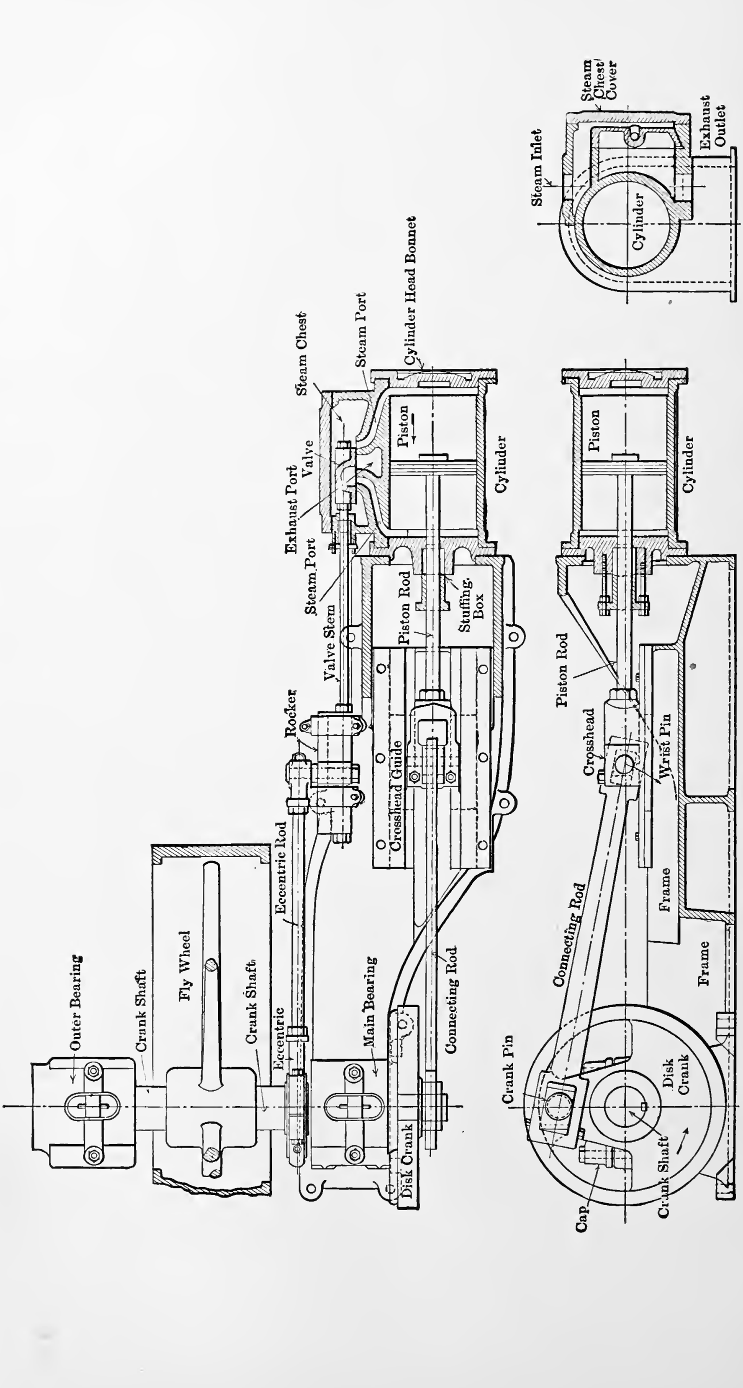


Fig. 186. Typical Piston Engine, Single Cylinder, Throttling Governor.

compare the actual cycle with the theoretical in which expansion is not complete. The various theoretical quantities for this condition of incomplete expansion (see paragraph 462) may be calculated as follows:

$$\text{Heat supplied} = H_i - q_n \text{ B.t.u. per lb.} \tag{130}$$

It will be noted that this is the same as for complete expansion.

$$\text{Heat absorbed} = H_i - H_c + \tfrac{1}{778}(P_c - P_2)\,v_c \text{ B.t.u. per lb.,} \tag{131}$$

$$\text{Efficiency, } E_r' = \frac{H_i - H_c + \frac{1}{778}(P_c - P_2)\,v_c}{H_i - q_n}, \tag{132}$$

$$\text{Water rate, } W_r' = \frac{2546}{H_i - H_c + \frac{1}{778}(P_c - P_2)\,v_c} \text{ lb. per hp-hr.,} \tag{133}$$

in which

H_c = heat content at release pressure P_c after adiabatic expansion,
P_c = release pressure, lb. per sq. ft.,
v_c = specific volume of the fluid under release conditions, cu. ft. per lb. (Other notations as for complete expansion.)

TABLE 71.

THEORETICAL EFFICIENCIES AND WATER RATES OF PERFECT ENGINES OPERATING IN THE CARNOT AND RANKINE CYCLES.

(Saturated Steam.)

Initial Pressure, Pounds Absolute.	Condensing.*				Non-condensing.			
	Efficiency.		Water Rate.		Efficiency.		Water Rate.	
	C	R	C	R	C	R	C	R
50	27.18	24.98	10.13	8.98	9.32	8.98	29.56	28.51
100	31.51	28.47	9.10	7.85	14.70	13.88	19.48	18.22
150	34.10	30.60	8.65	7.26	17.90	16.65	16.46	15.08
200	35.91	31.88	8.41	6.94	20.19	18.60	14.94	13.44
250	37.34	32.93	8.25	6.70	21.97	20.05	14.02	12.42
300	38.51	33.76	8.14	6.52	23.42	21.22	13.39	11.71
400	40.37	35.10	8.00	6.25	25.74	23.07	12.53	10.73
500	41.79	36.06	7.988	6.07	27.54	24.46	12.12	10.10
600	43.00	36.84	7.987	5.94	29.02	25.57	11.83	9.66

* Absolute back pressure 0.5 lb. per sq. in.

Direct-acting steam pumps and engines taking steam full stroke have the following theoretical possibilities (see paragraph 463):

$$\text{Heat supplied} = H_i - q_n \text{ B.t.u. per lb.,} \tag{134}$$

$$\text{Heat absorbed} = \tfrac{1}{778}(P_1 - P_2)\,v_1 \text{ B.t.u. per lb.,} \tag{135}$$

$$\text{Efficiency, } E_r'' = \frac{(P_1 - P_2)\,v_1}{778\,(H_i - q_n)}, \tag{136}$$

$$\text{Water rate, } W_r'' = \frac{2546 \times 778}{(P_1 - P_2)\,v_1}, \tag{137}$$

in which

v_1 = specific volume of the steam at pressure P_1, cu. ft. per lb.

(Other notations as above.)

160. Efficiency Standards. — The performance of the actual engine is variously stated as follows:

1. Steam consumption, pounds per hour or hp-hr.
2. Heat consumption, B.t.u. per hp-hr. or per hp. per minute.
3. Thermal efficiency, per cent.
4. Mechanical efficiency, per cent.
5. Rankine cycle ratio, per cent.
6. Cylinder efficiency, per cent.
7. Commercial efficiency.
8. Duty.

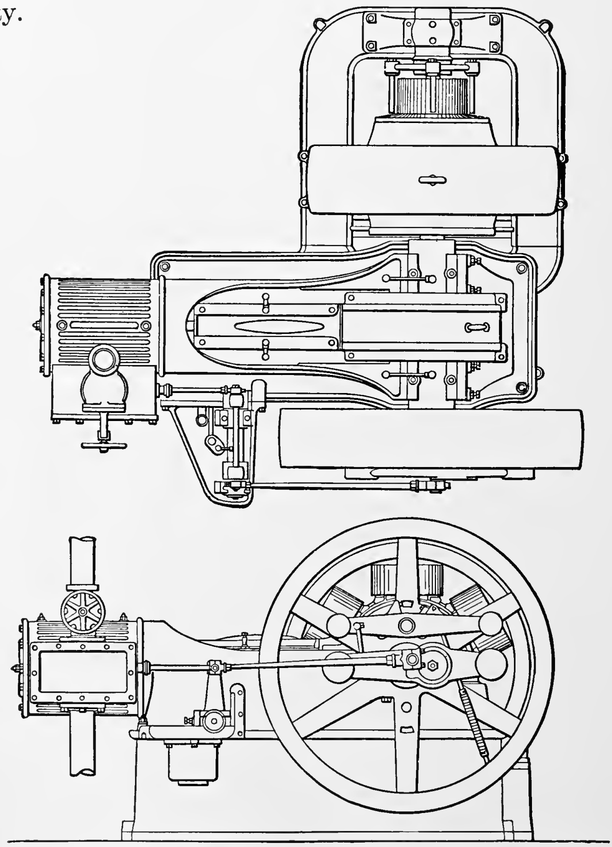


Fig. 187. Typical Piston Engine, Single Cylinder, Automatic Governor.

The indicator offers the simplest means of measuring the output of a piston engine, and for this reason the performance is usually stated in terms of *indicated horsepower.* The indicated horsepower is always greater than the net available power by an amount equivalent to the friction of the mechanism. The power actually developed, or brake horsepower, is not readily obtained except for small sizes, and it is customary to approximate this value by deducting the indicated horsepower when running idle from the indicated horsepower when running under the given load. This does not give the true effective power, but is sufficiently accurate for most commercial purposes. (See paragraph 174.) The output of steam turbines and piston engines driving electrical machinery is conveniently stated in electrical horsepower or kilowatts, since the electrical measurements are readily made. The electrical output as measured at the switchboard gives the net effective work, and automatically deducts the machine losses. Large turbines are usually tested at the factory by means of suitable water brakes, and the brake horsepower may be obtained from the makers.

161. Steam Consumption or Water Rate. — The most generally used measure of the performance of a steam engine is the steam consumption per hour or per unit of work output. Since the indicator offers the simplest means of measuring the output the performance is usually stated in terms of *indicated horsepower.** By plotting the total weight of steam passing through the engine as ordinates and the indicated horsepower as abscissas the performance of the engine at all loads may be seen at a glance. Just what form the total water rate curve will take depends largely on the type of governing and the form of valve gear. If the control is by throttling, the total water rate curve is substantially a straight line, and the relation is commonly called the *Willans line*, Fig. 214. When two points on this line, or one point and the slope, are given the line can be drawn at once. If the control is by "cutting off" the curve departs somewhat from a straight line, but in many cases the departure is insignificant, Fig. 222. Dividing the ordinates by corresponding abscissas gives the steam consumption per indicated horsepower-hour or unit water rate, Fig. 214.

For electrically driven machinery the economy is given as steam consumption per electrical horsepower-hour or per kilowatt-hour. A study of the unit water rate curve will show that the steam consumption decreases with the load up to a certain point and then increases.

* This must not be confused with the *steam accounted for by the indicator diagram*, or, as it is commonly called, the *indicated steam consumption.* The former refers to the actual weight of fluid flowing through the cylinder and the latter to the weight of steam calculated from the indicator card. (See paragraph 8, Appendix B.)

This point of minimum steam consumption corresponds ordinarily to the rated load. If the initial pressure, quality and back pressure were constant for all conditions of operation the water rate would be a true measure of heat efficiency, but since this is not the case the water rate under actual conditions is of little value in comparing performances. The water rate may be used as a means of comparison provided suitable corrections are made for pressure and quality, but this procedure is not common. The water rates for different types of engines are given further on in the chapter.

162. Heat Consumption. — Because of the extreme variation in steam conditions the performance of all engines and turbines is best expressed in terms of the heat consumption per unit output measured above the maximum theoretical temperature at which the condensation can be returned to the boiler. This temperature is called the *ideal feed-water temperature.* Thus the ideal feed-water temperature of an unjacketed non-condensing engine without receiver coils exhausting at standard atmospheric pressure is 212 deg. fahr., and that of a condensing engine exhausting against an absolute back pressure of two pounds is 126 deg. fahr. If the engine is fitted with jackets and reheating coils the heat of the liquid at jacket and coil pressure should be added to that of the exhaust in determining the ideal feed-water temperature. For example, if a condensing engine exhausts against an absolute back pressure of two pounds, and ten per cent of the total weight exhausted is condensed in the jackets under a pressure of 150 pounds absolute, the ideal feed-water temperature will be 159.5 deg. fahr. (Heat of the liquid at 150 pounds absolute = 330 B.t.u. per pound. Heat added by the jackets to the feed water = 330 × 0.1 = 33. Heat of the liquid at two pounds absolute = 94 B.t.u. 94 + 33 = 127 B.t.u., which corresponds to an actual temperature of 159.5 deg. fahr.)

Example 25. (1) A compound condensing engine develops one brake horsepower-hour on a steam consumption of 8.5 pounds, initial pressure 200 pounds absolute, superheat 250 deg. fahr., exhaust pressure 0.5 pound absolute, release pressure two pounds absolute. (2) The same engine when using wet steam develops one brake horsepower-hour on a steam consumption of 12 pounds per hour, initial pressure 150 pounds absolute, quality 98 per cent, exhaust pressure two pounds absolute, release pressure four pounds absolute.

Determine the comparative heat consumption of the two engines.

Superheated steam engine:

$$H_i = 1332 \text{ approx. (from steam tables)},$$
$$q_n = 48,$$

Heat supplied per br. hp-hr. = 8.5 (1332 − 48) = 10,914 B.t.u.,
Heat supplied per br. hp. per minute = 181.9 B.t.u.,

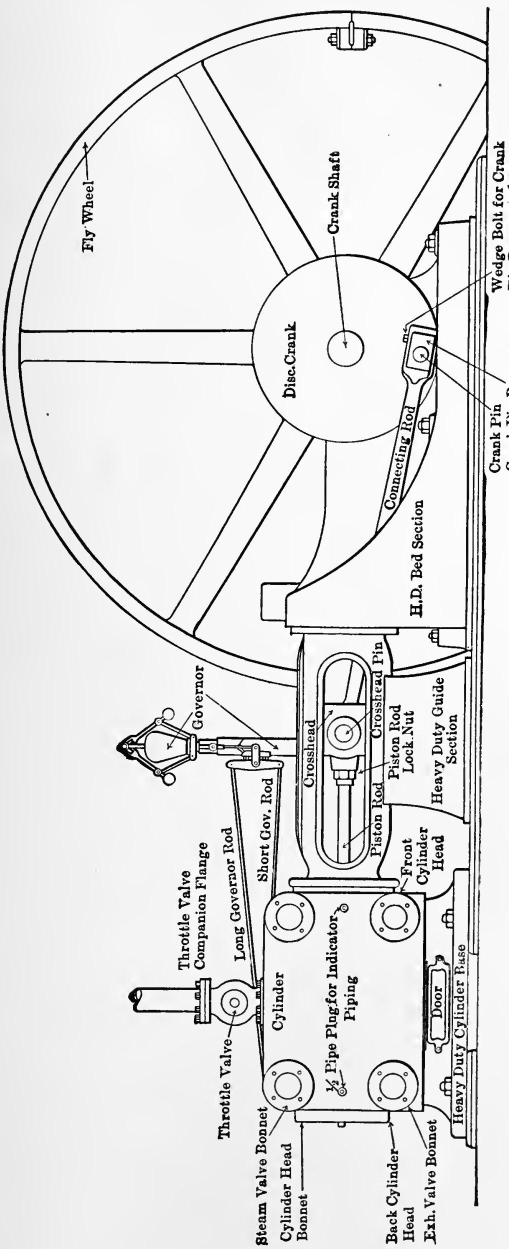


FIG. 188. Side Elevation of a Typical Corliss Engine, Single Cylinder, Belt Drive.

Saturated steam engine:

$$H_i = x_1 r_1 + q_1$$
$$= 0.98 \times 863.2 + 330.2 = 1176.1,$$

(This may be obtained directly from the Mollier diagram.)

$$q_n = 94,$$

Heat supplied per br.hp-hr. = 12 (1176.1 − 94) = 12,985 B.t.u.,
Heat supplied per br.hp. per minute = 216.4.
Economy of superheated steam

(1) in steam consumption, $100 \frac{12 - 8.5}{12} = 29.2$ per cent,

(2) in heat consumption, $100 \frac{216.4 - 181.9}{216.4} = 15.9$ per cent.

The heat consumption for different types of piston engines is given further on in the chapter.

163. Thermal Efficiency. — The *thermal efficiency* of a steam engine or turbine is the ratio of the heat converted into *useful* work to that *supplied,* measured above the heat of the liquid at exhaust steam temperature.* If the heat consumption is expressed in terms of i.hp-hr., the ratio becomes the *indicated thermal efficiency.* Since the heat equivalent of one horsepower, using the latest accepted values, is 42.44 B.t.u. per minute or 2546 B.t.u. per hour, this relationship may be expressed

$$E_t = \frac{2546}{\text{B.t.u. supplied per br. hp-hr.}}$$
$$= \frac{2546}{W(H_i - q_n)}, \tag{138}$$

in which

W = the weight of steam supplied, pounds per developed horsepower-hour.

H_i and q_n as in equation (129).

If measured in electrical units this relationship becomes

$$E_t' = \frac{3413}{W_1(H_i - q_n)}, \tag{139}$$

in which

W_1 = pounds per kilowatt-hour; other notations as in (129).

* The heat supplied is often measured above the *actual* feed-water temperature but the latter is not dependent upon the performance of the engine and hence is not satisfactory for purposes of comparison.

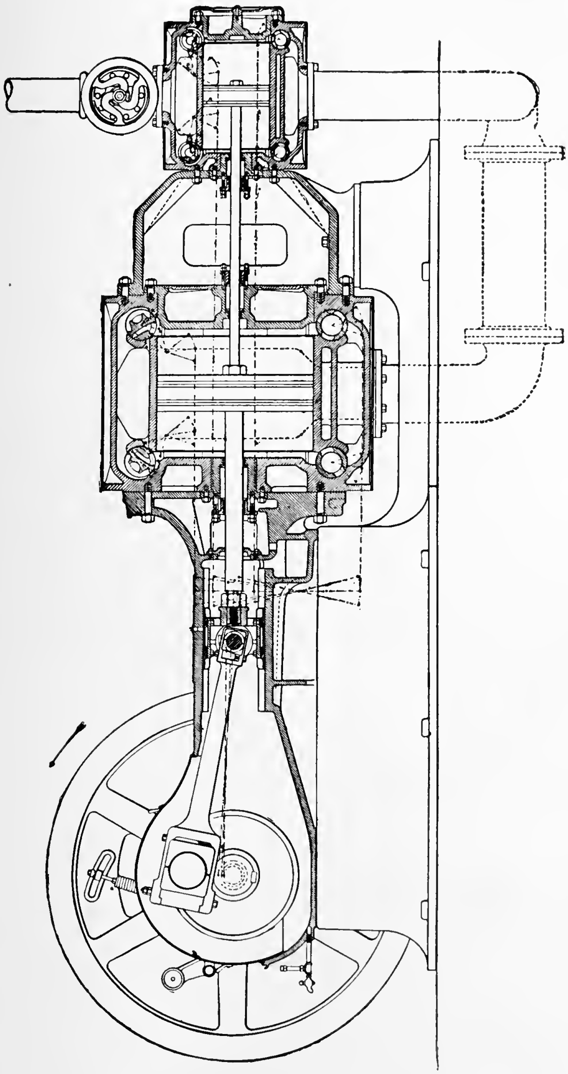


Fig. 189. Fleming-Harrisburg Four-valve High-speed Tandem Compound Engine.

Example 26. Determine the thermal efficiencies for the two engines using the data of the preceding example.

Superheated steam engine,

$$E_t = \frac{2546}{8.5\,(1332 - 48)} = 0.233 = 23.3 \text{ per cent.}$$

Saturated steam engine,

$$E_t = \frac{2546}{12\,(1176.1 - 94)} = 0.196 = 19.6 \text{ per cent.}$$

The thermal efficiency of the actual engine varies from 5 per cent for the poorer grades of non-condensing engines to 25 per cent for the best recorded performance to date. As far as thermal efficiency is concerned the piston engine still leads the turbine for sizes under 2000 horsepower.

164. Mechanical Efficiency. — The ratio of the developed or brake horsepower to the indicated power is the mechanical efficiency of the engine; the ratio of the electrical horsepower to the indicated power is the mechanical efficiency of the engine and generator combined; and the ratio of the pump horsepower to the indicated power of the engine is the mechanical efficiency of the engine and pump combined. The percentage of work lost in friction is therefore the difference between 100 per cent and the mechanical efficiency in per cent. (See also paragraph 174.)

TABLE 72.

MECHANICAL EFFICIENCIES OF ENGINES.

Kind of Engine.	Horse Power.	Efficiency at Full Load.
Simple:		
1. High-speed, non-condensing	150	95.5
2. High-speed, condensing	170	96
3. Low-speed, non-condensing	275	94
Compound:		
4. High-speed, non-condensing	150	94
5. High-speed, condensing	160	98
6. Low-speed, non-condensing	900	95
7. Low-speed, condensing	1000	95
8. Do	5500	95.2*
9. Do	7500	93.0*
Triple: (combined efficiency of engine and pump)		
10. Pumping engine	865	97.4
Quadruple: (combined efficiency of engine and pump)		
11. Pumping engine	712	93

* Combined efficiency of engine and generator.

The mechanical efficiency of piston engines at rated load varies from 85 per cent for the cheaper grades of engines to 95 per cent and even 98 per cent for the better types. The size of engine has practically no influence on the mechanical efficiency, though the smaller machines are apt to have a lower efficiency because of the cheaper construction. Generator efficiencies at full load vary from 86 per cent for the 15-kilowatt size to 94 per cent for units of 2000 kilowatts rated capacity. The generator efficiency of very large turbo-alternators, 25,000 kilowatt rated capacity or more, is in the neighborhood of 96 per cent. The overall or combined efficiency at rated load varies from 75 per cent for small units to 93 per cent for larger ones. A few examples of high engine efficiency are cited in Table 72. The efficiency at fractional loads for a specific case are illustrated in Fig. 190.

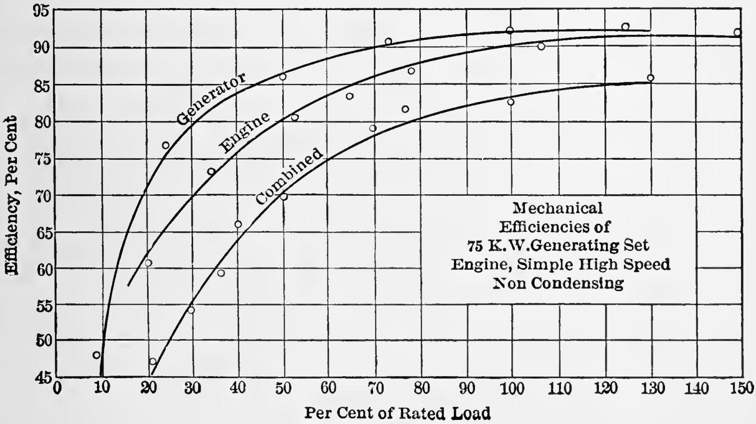


FIG. 190.

Lucke, *Engineering Thermodynamics*, p. 370, states that the mechanical efficiency of the piston engine is independent of the speed and that it may be expressed

$$E_m = 1 - K_1 - \frac{K_2}{\text{m.e.p.}}, \tag{140}$$

in which

K_1 = constant, varying from 0.02 to 0.05,

K_2 = constant, varying from 1.3 to 2.0,

m.e.p. = mean effective pressure, lb. per sq. in.

165. Rankine Cycle Ratio. — The degree of perfection of an engine or the extent to which the theoretical possibilities are realized is the ratio of the thermal efficiency of the actual engine to that of an ideally perfect engine working in the Rankine cycle with complete expansion.

This is called the *Rankine cycle ratio* or *potential efficiency*.* It is the accepted standard for comparing the performance of steam engines and steam turbines.

If E = Rankine cycle ratio †

E_t = thermal efficiency of the actual engine,

E_r = efficiency of the ideal engine work in the Rankine cycle with complete expansion.

Then
$$E = \frac{E_t}{E_r}. \tag{141}$$

From equation (138)
$$E_t = \frac{2546}{W(H_i - q_n)}.$$

And from equation (128)
$$E_r = \frac{H_i - H_n}{(H_i - q_n)}.$$

Whence
$$E = \frac{2546}{W(H_i - q_n)} \div \frac{H_i - H_n}{H_i - q_n} \tag{142}$$
$$= \frac{2546}{W(H_i - H_n)}. \tag{143}$$

Example 27. Determine the Rankine cycle ratio of the two engines specified in paragraph 162.

Superheated steam engine:
$$E = \frac{2546}{8.5\,(1332 - 908)} = 0.706 = 70.6 \text{ per cent.}$$

Saturated steam engine:
$$E = \frac{2546}{12\,(1176 - 898)} = 0.763 = 76.3 \text{ per cent.}$$

Tables 81 and 83 give the best recorded Rankine cycle ratics for current practice.

166. Cylinder Efficiency. — The piston engine seldom expands the steam down to the existing back pressure but releases from two to five pounds above this point in condensing engines and from 15 to 20 pounds above in non-condensing engines. The ideal cycle corresponding to this condition is the Rankine cycle with incomplete expansion. The ratio of the thermal efficiency of the actual engine to that of the ideal engine working in the incomplete cycle is a true measure of the degree

* The term "thermodynamic efficiency" or "efficiency" without qualification is ordinarily interpreted as the Rankine cycle ratio though some authorities apply the name "thermodynamic efficiency" to the "thermal efficiency" as defined in paragraph 163.

† This may be based on either indicated, brake or electrical horsepower.

of perfection of the engine under the given conditions. This rate is called *cylinder efficiency* and may be expressed as

$$E' = \frac{2546}{W\,[(H_1 - H_c) + (P_c - P_2)\,x_c u_c \div 778]}. \tag{144}$$

See equations (138) and (132).

Example 28. Determine the cylinder efficiency of the two engines specified in paragraph 162.

Superheated steam engine:

$$E' = \frac{2546}{8.5\dagger\,[1332 - 980 + \frac{144}{778}\,(2.0 - 0.5)\,0.866 \times 173.5]}$$
$$= 0.761 = 76 \text{ per cent.}$$

Saturated steam engine:

$$E' = \frac{2546}{12\,[1176 - 935 + \frac{144}{778}\,(4 - 2)\,0.808 \times 90.5]}$$
$$= 0.808 = 80.8 \text{ per cent.}$$

Summing up the various efficiencies for the two cases analyzed in paragraphs 162 to 166:

	Saturated Steam Engine.	Superheated Steam Engine
Pressure, pounds per square inch, absolute:		
Initial	150	200
Release	4	2
Condenser	2	0.5
Degree of superheat, deg. fahr.	0.98*	250
Steam consumption, pounds per developed horsepower-hour:		
Actual engine	12.00	8.50
Ideal engine, Rankine cycle, with incomplete expansion	9.69	6.46
Ideal engine, Rankine cycle, with complete expansion	9.16	6.00
Ideal engine, Carnot cycle	10.59	....
Thermal efficiency, per cent:		
Actual engine	19.6	23.3
Ideal engine, Rankine cycle, with incomplete expansion	24.3	30.7
Ideal engine, Rankine cycle, with complete expansion	25.8	33.3
Ideal engine, Carnot cycle	28.3	....
Heat consumption, B.t.u., per developed horsepower-minute, Actual engine	216.4	181.9
Ideal engine, Rankine cycle, with incomplete expansion	190.4	154.6
Ideal engine, Rankine cycle, with complete expansion	174.8	138.4
Ideal engine, Carnot cycle	152.5	....
Rankine cycle ratio, per cent	76.3	70.6
Cylinder efficiency, per cent	80.8	76.1

* Quality.

† If the steam consumption per i.hp-hour is used in this connection instead of the consumption per br.hp-hour this ratio becomes the *indicated* cylinder efficiency.

167. Commercial Efficiencies. — There is no accepted standard for rating the commercial efficiency of an engine or turbine. The various measures used in this connection, such as *pounds of standard coal per d.hp-hour, cents per horsepower per year* and the like include the economy of the boiler and auxiliaries and are not a true indication of the performance of the engine alone. From a commercial standpoint it is important to know the weight of coal required to develop a horsepower-hour, taking into consideration all of the losses of transmission and conversion, and a knowledge of the *overall efficiency* from switchboard to coal pile is of value in basing the cost of power, but these items are in reality measures of the plant economy and are of little value in comparing the performance of the prime mover. The various efficiencies under this heading are treated in Appendices B to G.

168. Heat Losses in the Steam Engine. — The principal losses which tend to lower the efficiency of the steam engine and which prevent it from realizing the performance of the ideal engine are due to:

(*a*) Cylinder condensation.
(*b*) Leakage.
(*c*) Clearance volume.
(*d*) Incomplete expansion.
(*e*) Wire drawing.
(*f*) Friction of the mechanism.
(*g*) Presence of moisture in the steam at admission.
(*h*) Radiation, convection and minor losses.

169. Cylinder Condensation. — The weight of steam apparently used per revolution, as determined from the indicator card, or the *indicated steam consumption** (see paragraph 8, Appendix B), is considerably less than that actually supplied. The difference or *missing quantity* is due chiefly to *cylinder condensation.* This is by far the greatest loss in the steam engine with the exception of that inherent in the ideal engine. When steam is admitted to the cylinder a considerable portion of the heat is given up to the comparatively cool skin surface of the cylinder walls. If the steam is saturated at admission this heat absorption causes condensation, or *initial condensation* as it is called; if superheated at admission the temperature is lowered to a corresponding point. After cut-off heat continues to be given up to the walls until the temperature of the steam falls below that of the skin surface, when the process is reversed and part of the heat is returned to the steam. With saturated steam the heat absorption causes *condensation during expansion,* and the heat rejected, *reëvaporation during expansion.*

* Also called the *steam accounted for by the diagram* or *diagram steam.*

With superheated steam an equivalent heat exchange takes place. Unless the cylinder is of a compound series the heat absorbed from the cylinder walls during exhaust does no useful work and is lost. Cylinder condensation, measured as the proportion of the mixture present, varies with the size of the engine, speed, length of cut-off, valve design, temperature range, location of ports and port passages, jacketing, lagging, and other variables. It ranges from 16 to 30 per cent, and is occasionally as high as 50 per cent of the total weight of steam admitted to the cylinder. Cylinder condensation and leakage cannot be conveniently separated and are ordinarily classified together. Fig. 191

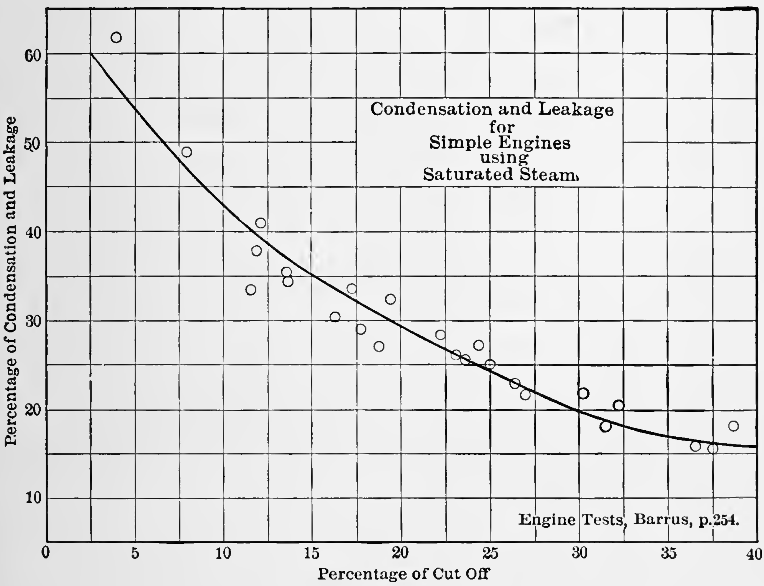


FIG. 191.

shows the relation between cylinder condensation and leakage losses for various percentages of cut-off for simple high-speed non-condensing engines.

Empirical formulas for calculating the extent of these losses, and which involve the various influencing factors, are unwieldy and only approximately accurate. One of the most satisfactory formulas of this class is that deducted by R. C. H. Heck, "The Steam Engine and Turbine," 1911, p. 175.

The various heat exchanges between working fluid and the cylinder walls, including cylinder condensation and leakage, are approximately determined by transferring the indicator diagram to the temperature entropy chart. (See paragraph 466.)

For use and application of the temperature-entropy diagram in engine tests consult Power, Dec., 1907, p. 834; Jan. 21, 1908, p. 96; Jan. 28, 1908, p. 145.

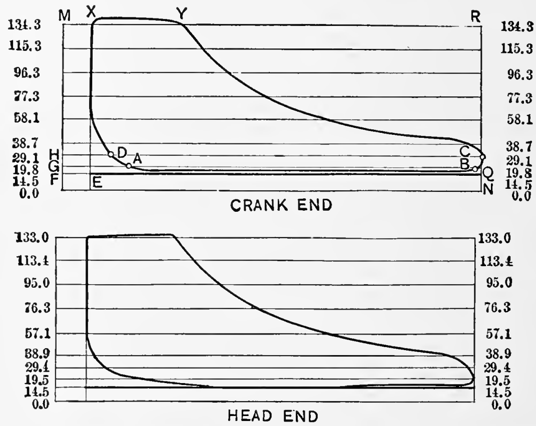


Fig. 192. Diagrams Taken from a 12-in. by 24-in. Corliss Engine.

A comparatively simple method for approximating cylinder condensation and leakage losses is given by J. Paul Clayton, Proc. A.S.M.E., April, 1912, and consists in transferring the indicator diagram to logarithmic cross-section paper. By means of the logarithmic diagram

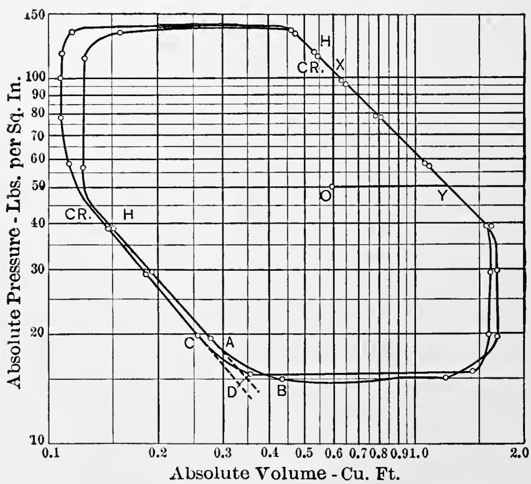


Fig. 193. Logarithmic Diagrams Plotted from Fig. 192.

Clayton found that (1) free from certain abnormal influences, expansion and compression take place in the cylinder substantially according to the law $PV^n = C$, (2) the value n bears a definite relation in any given cylinder to the proportion of the total weight of steam mixture which was present as steam at cut-off, (3) the relation of the value n to the value x_c (quality of steam at cut-off) for the same class of cylinder as regards jacketing is practically independent of engine speed and of cylinder size, and (4) by means of the experimentally determined relation of x_n and n the actual steam consumption may be obtained from the indicator card to well within 4 per cent of the true value. The curves in Fig. 193 were plotted on logarithmic cross-section paper from the pressure-volume diagrams, shown in Fig. 192, and illustrate

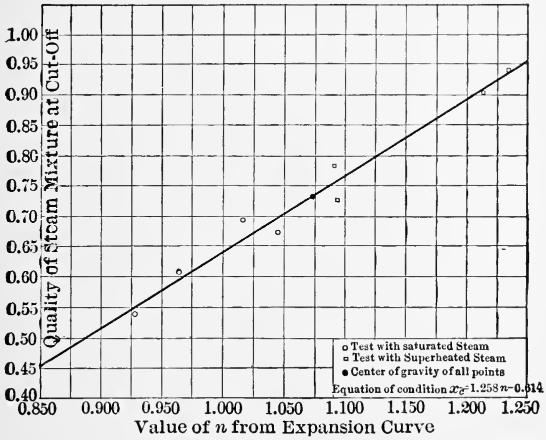


FIG. 194. Relation of Quality and the Value of n.

Mr. Clayton's method of analysis. The curves in Fig. 194 show the relation between quality x_c and exponent n or for a given set of conditions. See also paragraphs 470–3.

Cylinder Condensation: Power, Jan. 3, 1911, p. 25; Jan. 30, 1912, p. 145; Nov. 5, 1912, p. 664.

170. Leakage of Steam. — The loss due to leakage is a variable factor depending upon the design and condition of the engine, and is greater with saturated than with superheated steam. The usual method of measuring leakage past the valves and piston while the engine is at rest is likely to give erroneous results, as demonstrated by Callender and Nicolson (Peabody, "Thermodynamics," p. 351) in tests made on a high-speed automatic balanced-valve engine and on a quadruple expansion engine with plain unbalanced slide valves. With the engines at rest they found that the leakage past valves and piston was insignificant, but when in operation the leakage from the steam chest into the exhaust was considerable. It was thought that a large proportion of the leakage was probably in the form of water formed by condensation of steam on the seat uncovered by the valve.

According to the report of the Steam Engine Research Committee (Eng. Lond., March 24, 1905, p. 208), leakage through a plain slide valve is independent of the speed of the sliding surfaces, and directly proportional to the difference in pressure on the two sides; with well-fitted valves the leakage is never less than 4 per cent of the volume of steam entering the cylinders, and is often greater than 20 per cent.

The various leakage losses may be approximated by transferring the indicator diagram to logarithmic cross-section paper. Fig. 195 shows the application of the logarithmic diagram to a specific case and illustrates this method of determining leakage losses. See paragraph 465.

Leakage Past Piston Valves: Engr., Feb. 9, 1912.

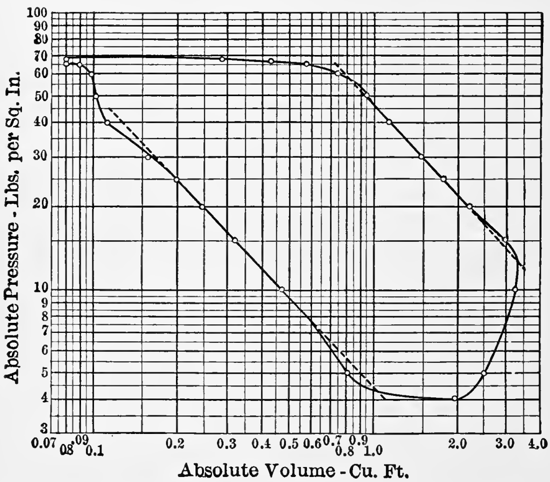


FIG. 195. Diagram from a 14-in. by 35-in. Corliss Engine, Showing Leakage at Beginning and End of Expansion and Compression.

171. Clearance Volume. — The portion of the cylinder volume not swept through by the piston but which is nevertheless filled with steam when admission occurs is called the clearance volume. It is the space between the end of the piston when on dead center and the inside of the valves covering the ports. It varies from about 1 per cent of the piston displacement in very large engines with short steam passages to 10 per cent or more in small high-speed engines.

The extent of surface in the clearance space greatly influences the amount of cylinder condensation since the piston is moving slowly near the end of the cylinder, and the time of exposure of the steam to these surfaces is comparatively long. The greater part of the cylinder condensation usually occurs in the clearance space, therefore the steam passages and clearance space should be designed so as to present a minimum amount of surface consistent with the proper cushioning volume for smooth operation. Theoretically if steam is compressed adiabatically to the initial pressure there is no loss due to clearance but in practice compression carried to initial pressure does not necessarily improve the economy. For a constant time element the shorter the cut-off the greater will be the ratio of the weight of cushion steam to that of the steam supplied and hence the greater the loss. In large, slow-moving engines the loss due to clearance may be greater than that in high-speed, short-stroke engines because of the longer time of exposure to the clearance surface.

The ratio of expansion is decreased by clearance; for example, an engine cutting off at one-fifth, neglecting clearance, has an apparent ratio of expansion of 5, but if the clearance volume is 10 per cent the actual ratio is only 3.66. One of the few recorded tests relative to the influence of clearance on the economy of a high-speed engine was conducted on a 14-in. by 15-in. Allfree engine. (Power, May, 1901.) With a clearance volume of 2.2 per cent, initial pressure 105 pounds gauge, and 172 r.p.m., the best performance was 23.7 pounds of dry steam per i.hp-hour. With the same steam pressure and speed but with clearance volume increased to 6 per cent by the use of a shorter piston, the best performance was 28.3 pounds per i.hp-hour. In both cases the compression was carried up to admission pressure.

Independent tests made by Prof. Boulvin and by A. H. Klemperer on single-cylinder Corliss engines gave a minimum water rate when the clearance volume was approximately one-half the compression volume. See end of paragraph 172.

Engine Clearance and Compression: Power, July 5, 1910, Dec. 27, 1910; Sibley Journal, Dec., 1910.

172. Loss Due to Incomplete Expansion and Compression. — Theoretically the loss due to incomplete expansion is considerable. For example, the theoretical steam consumption of a perfect engine (Rankine cycle) expanding from 120 pounds absolute to a condenser pressure of 2 pounds absolute is 9.6 pounds per horsepower-hour. If the expansion were carried to only 5 pounds absolute, the exhaust pressure remaining the same, the steam consumption would be increased to 11.8 pounds per horsepower-hour, a difference of 22 per cent for an increase in terminal pressure of only 3 pounds per square inch. The theoretical water rates for various terminal pressures are given below.

Terminal Pressure, Pounds per Square Inch Absolute.	Steam Consumption of Perfect Engine.	Terminal Pressure, Pounds per Square Inch Absolute.	Steam Consumption of Perfect Engine.
1	8.5	3	10.4
1.5	9.1	4	11.1
2	9.6	5	11.8
2.5	10	6	12.3

In actual engines expansion is seldom complete, since it would necessitate increased bulk and weight of engine, and the work done by the steam in the last stages would not compensate for the increased cost.

In single-cylinder engines maximum economy is effected when the terminal pressure is considerably above that of the exhaust, since the gain due to complete expansion is more than offset by the increased cylinder condensation. This is true to a certain extent in all engines irrespective of the number of cylinders. Tests by G. H. Barrus ("Engine Tests," 1900) to determine the terminal pressures effecting maximum economy for various types of engine gave results as follows:

	Terminal Pressure, Pounds Absolute.
Simple slide-valve engines, non-condensing	30 to 40
Simple slide-valve engines, condensing	25 to 30
Simple Corliss engines, non-condensing	20 to 25
Simple Corliss engines, condensing	15 to 18
Compound engines, non-condensing	18 to 22
Compound engines, condensing	3 to 5

In high-speed engines a certain amount of compression is desirable for its cushioning effect; outside this mechanical feature compression may or may not be of benefit to the engine, as will be seen from the results of tests stated below. Zeuner in his treatise on theoretical thermodynamics proves deductively that in an engine with a large clearance volume the loss due to clearance is completely eliminated if the compression is carried up to admission pressure, a conclusion which tests by Jacobus, Carpenter, and others fail to confirm. A series of tests by Professor Jacobus (Trans. A.S.M.E., 15–918) on a 10-in. by 11-in. high-speed automatic engine at Stevens Institute show decreasing economy with increase of compression, the initial pressure, cut-off, and release remaining constant. The results were as follows:

Proportion of initial pressure up to which the steam is compressed	$\frac{5}{8}$	$\frac{2}{3}$	Full
Steam, pounds per i.hp-hour	34.8	36.7	38

Tests by Carpenter (Trans. A.S.M.E., 16–957) on the high-pressure cylinders of the Corliss engine at Sibley College gave:

Compression, per cent	11.4	25	35.2
Brake horsepower	30	29	26
Steam, pounds per br.hp-hour	33	33.3	34

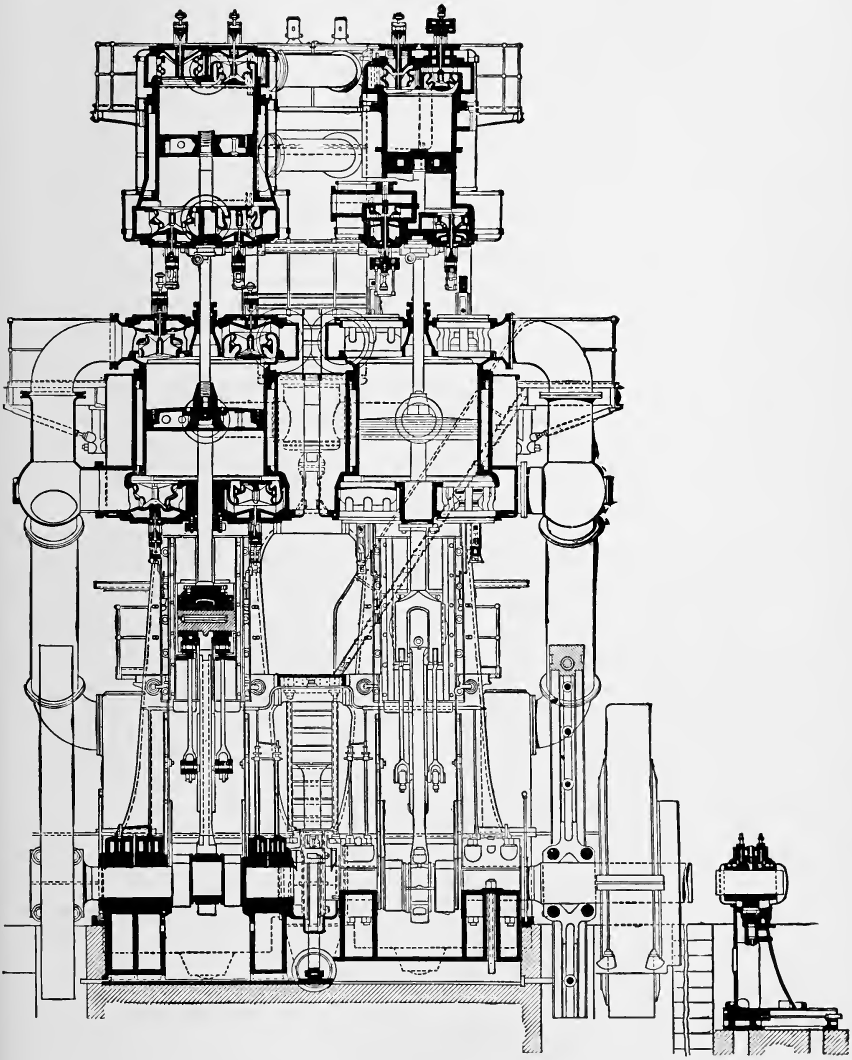


Fig. 196. 3000-hp. Sulzer Engine Designed for Highly Superheated Steam.

Tests made by A. H. Klemperer on a 7.1-in. by 17.7-in. Corliss engine, at Dresden, gave decreasing steam consumption for increase in compression up to a compression of about twice the clearance volume, beyond which the water rate increased with the increase in compression. (Zeit. d. Ver. deut. Ingr., Vol. I, 1905, p. 797.)

Tests made by Prof. Boulvin on a 9.8-in. by 19.7-in. Corliss engine at University of Ghent gave results agreeing with those of Klemperer. (Revue de Mécanique, 1907, Vol. XX, p. 109.)

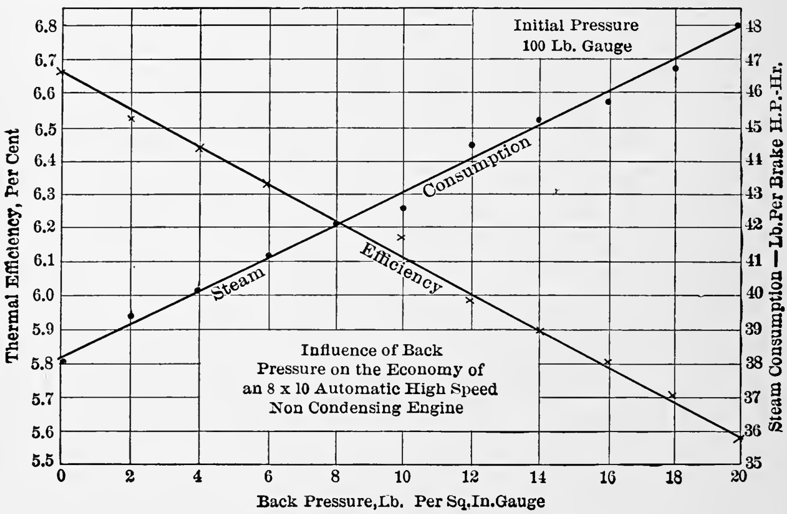


Fig. 197.

Fig. 197 shows the influence of increasing back pressure on the economy of an 8-in. by 10-in. automatic high-speed engine at the Armour Institute of Technology.

The Effect of Compression: Power, Oct. 27, 1914, p. 595.

173. Loss Due to Wire Drawing. — Wire drawing, or the drop in pressure due to the resistances of the ports and passages, has the effect of reducing the output and the economy of the engine to some extent, since the pressure within the cylinder is less than that at the throttle during admission and greater than discharge pressure at exhaust. The steam may be dried to a small extent during admission, but because of the drop in pressure the *heat availability* is reduced. The loss in available heat may be calculated as shown in paragraph 456. In single-valve engines the effects of wire drawing are decidedly marked and the true points of cut-off and release are sometimes difficult to locate on the indicator card. In engines of the Corliss, poppet, or gridiron-valve type the effects are hardly noticeable.

174. Loss Due to Friction of the Mechanism. — The difference between the indicated horsepower and that actually developed is the power consumed in overcoming friction, and varies from 4 to 20 per cent of the indicated power, depending upon the type and condition of the engine. Engine friction may be divided into (1) initial or no-load friction and (2) load friction. The stuffing-box and piston-ring friction is practically independent of the load, while that of the guides, bearings, and the like increases with the load. In Fig. 198, curve *A* gives the relation between the frictions for a four-slide-valve horizontal cross compound engine, and *B* that for a simple non-condensing Corliss.

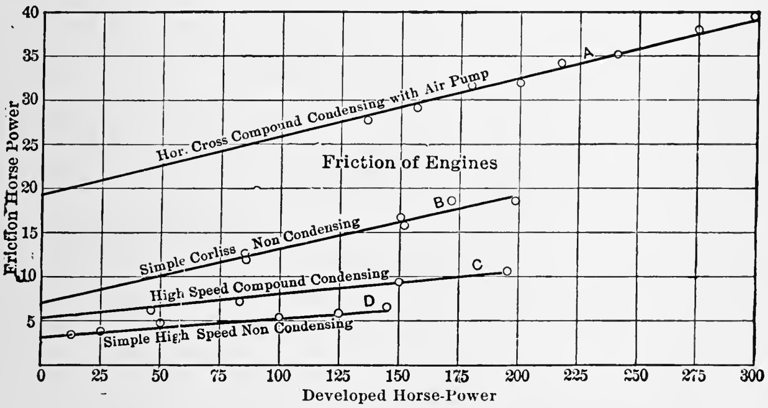


FIG. 198. Typical Curves of Steam Engine Friction.

(Peabody's "Thermodynamics," pp. 433 and 437.) Curve *C* is plotted from the tests of a Reeves vertical cross compound condensing engine (Engineering Record, July 1, 1905, p. 24), and *D* from the test of an Ames simple high-speed non-condensing engine. (Engineering Record, Vol. 27, p. 225.) A large number of recorded tests show less friction at full load than at no load, but this is probably due to error or to variations in lubrication. With first-class lubrication it is usually sufficiently accurate to assume the friction to be constant and equal to the initial friction at zero load. The distribution of the frictional losses in a number of engines is given in Table 73.

175. Moisture. — The presence of moisture in the steam pipe is due to condensation caused by radiation or to priming at the boiler. Unless removed by some separating device between boiler and engine the amount of moisture entering the cylinder may be from 1 to 5 per cent of the total weight of steam, and the work done per pound of fluid is correspondingly reduced. This loss should not be charged against the engine, however, and its performance should be reckoned on the dry steam basis. Experiments reported by Professor R. C. Carpenter (Trans. A.S.M.E., 15–438) in which water in varying quantities was introduced into the steam pipe, causing the quality of the steam to range from 99 per cent to 57 per cent, showed that the consumption of *dry steam* per i.hp-hour was practically constant, the water acting as an inert quantity. An efficient separator will remove practically all the entrained water.

TABLE 73.

DISTRIBUTION OF FRICTION IN SOME DIRECT-ACTING STEAM ENGINES.

(Thurston.)*

Parts of Engines where Friction is Measured.	Percentage of Total Engine Friction.				
	"Straight Line" Balanced Valve.	"Straight Line" Unbalanced Valve.	Traction Engine Locomotive Valve Gear.	Automatic Balanced Valve.	Condensing Engine Balanced Valve.
Main bearings	47.0	35.4	35.0	41.6	46.0
Piston and piston rod	32.9	25.0	21.0		
Crank pin	6.8	5.1	13.0	49.1	21.8
Crosshead and wrist pin	5.4	4.1			
Valve and valve rod	2.5	26.4	22.0	9.3	21.0
Eccentric strap	5.4	4.0			
Link and eccentric	....	....	9.0	....	....
Air pump	....	....	....	....	12.0
	100.0	100.0	100.0	100.0	100.0

* "Friction and Lost Work in Machinery," p. 13.

176. Radiation and Minor Losses. — The radiation and conduction of heat from the cylinder, piston rod and valve stem has the effect of increasing the cylinder condensation. In jacket engines this loss may be approximated by the quantity of steam condensed in the jacket when the engine is not running. In unjacketed engines the loss is practically undeterminable since the heat exchange between cylinder walls and the steam is exceedingly complex. The heat loss due to radiation, measured in terms of the total heat supplied, varies from 0.3 per cent in very large units with efficiently lagged cylinders and steam chests to approximately 2 per cent in small engines as ordinarily insulated.

177. Heat Lost in the Exhaust. — Most of the heat supplied to the engine is rejected to the exhaust; this varies from 70 per cent in the most economical type of prime mover to 95 per cent in the poorer types. If the exhaust steam is used for heating purposes the heat chargeable to power is the difference between the heat supplied and that utilized from the exhaust. In passing through a prime mover heat is abstracted from the steam by:

(1) Conversion of part of the heat into mechanical energy.
(2) Loss through radiation.

If w represents the water rate or steam consumption per indicated horsepower-hour or the equivalent, then $\frac{2546}{w}$ = B.t.u. utilized per hour from each pound of steam in producing one indicated horsepower. Considering H_1 as the initial heat content, B.t.u. per pound above 32 degs. fahr., and H_r as the loss due to radiation, the heat content H_2 per pound of exhaust will be

$$H_2 = H_1 - H_r - \frac{2546}{w}. \tag{145}$$

As previously stated the heat loss due to radiation in terms of the total heat supplied varies from 0.3 per cent in very large units with efficiently lagged cylinders and steam chests to approximately 2 per cent in small engines of 25 horsepower rated capacity. An average value of one per cent may be assumed for most practical purposes.

If the exhaust contains moisture as is usually the case, we have

$$H_2 = x_2 r_2 + q_2, \tag{146}$$

in which

x_2 = quality of the exhaust,
r_2 = latent heat corresponding to exhaust pressure,
q_2 = heat of the liquid at exhaust pressure.

Combining equations (145) and (146) and reducing

$$x_2 = \frac{H_1 - H_r - q_2 - \frac{2546}{w}}{r_2}. \tag{147}$$

If the exhaust is superheated

$$H_2 = r_2 + q_2 + C_m t_2', \tag{148}$$

in which

C_m = mean specific heat of the superheated steam at exhaust pressure,
t_2' = degree of superheat of the exhaust steam, deg. fahr.

Assuming that the moisture in the exhaust is rejected to waste, the heat available per pound of exhaust steam at the exhaust nozzle is $x_2 r_2 + q_2$ and the net heat chargeable to power is

$$w\,[H_1 - (x_2 r_2 + q_2)] \text{ B.t.u. per i.hp-hr.} \tag{149}$$

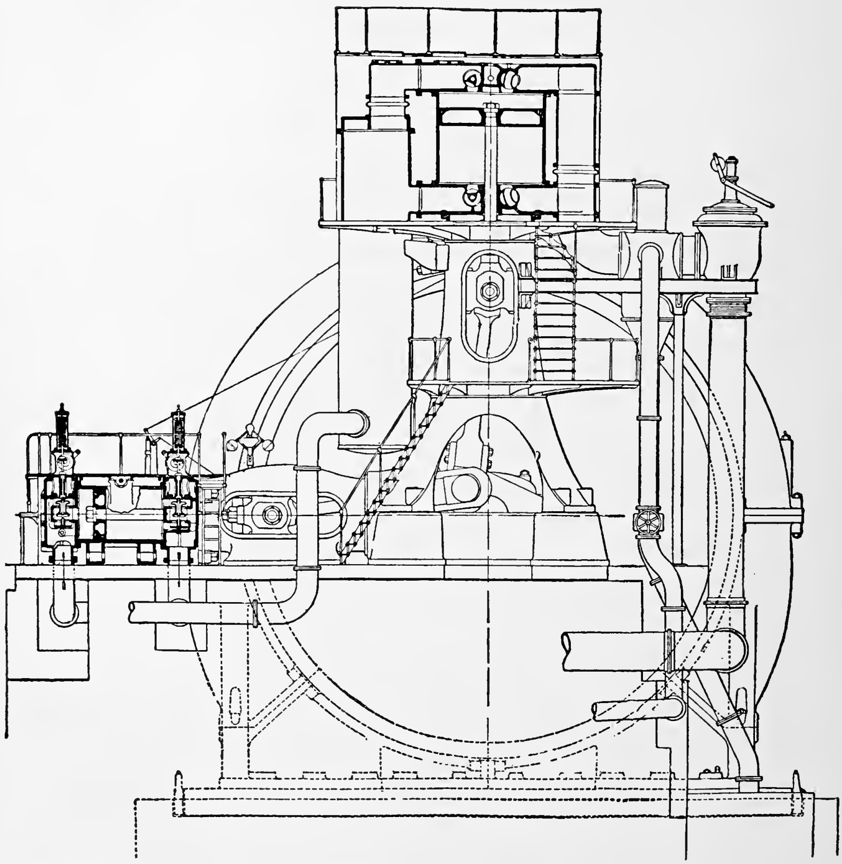


FIG. 199. 7500-kw. Vertical-horizontal Double-compound Engine as Installed at the 59th Street Station of the Interborough Rapid Transit Co. (Manhattan Type.)

All of this heat is not available for commercial heating purposes because of the condensation losses in the exhaust main. The extent of the latter depends upon the size and length of main, rate of flow and efficiency of the pipe covering. Representing this loss, per pound of steam, by H_x, the total heat chargeable to power is

$$w\,[H_1 - (x_2r_2 + q_2) + H_x] \text{ B.t.u. per i.hp-hr.,} \qquad (150)$$

and the equivalent water rate for power only is

$$\frac{w\,[H_1 - (x_2r_2 + q_2) + H_x]}{H} \text{ lb. per i.hp-hr.,} \qquad (151)$$

in which

H = net heat supplied to the engine, B.t.u. per pound.

Very little information is available relative to the quality of exhaust as determined by actual test but such as has been published is in accord with the results calculated from equation (147).

Moisture in Exhaust Steam.—Trans. Am. Soc. Heat & Vent. Engrs., Vol. 21, 1915, p. 85; Trans. A.S.M.E., Vol. 32, p. 331.

Example 29. A 23-inch by 16-inch simple engine, direct connected to a 200-kilowatt generator installed at the Armour Institute of Technology uses 35 pounds of steam per indicated horsepower-hour at full load, initial pressure 115 pounds absolute, back pressure 17 pounds absolute, initial quality 98 per cent.

Calculate the quality of the exhaust, assuming a radiation loss of one per cent.

From steam tables

$$\begin{aligned} H_1 &= xr + q \\ &= 0.98 \times 879.8 + 309 \\ &= 1171+, \\ r_2 &= 965.6, \\ q_2 &= 187.5. \end{aligned}$$

By assumption, $H_r = 0.01 \times 1171 = 11.7$.

Substituting these values in equation (147)

$$x_2 = \frac{1171 - 11.71 - 187.5 - \dfrac{2546}{35}}{965.6} = 0.933 \text{ or } 93.3 \text{ per cent.}$$

(Actual calorimeter tests gave a quality of 92.5 per cent, indicating a somewhat larger radiation loss than the assumed value of one per cent.)

Total heat chargeable to power (equation 150).

$$35\,[1171 - (901 + 187.5) + H_x] = 2918 + 35\,H_x \text{ B.t.u. per i.hp-hr.}$$

H_x varies within such wide limits that general assumptions are apt to lead to serious error. Where specific figures are not available it is customary to allow 2 per cent of the heat value of the exhaust as the extent of this loss. With this assumption we have as the heat chargeable to power

$$2918 + 35 \times 0.02\,(901 + 187.5) = 3680 \text{ B.t.u. per i.hp-hr.}$$

Assuming that the condensation from the heating system, including that exhausted from the engine, is returned to the boiler at a temperature of 192 deg. fahr., the net heat supplied per pound of steam is

$$\begin{aligned} H &= 1171 - (192 - 32) \\ &= 1011 \text{ B.t.u.} \end{aligned}$$

And the equivalent water rate for *power only* is

$$\frac{3680}{1011} = 3.63 \text{ lb. per i.hp-hr.}$$

The low fuel consumption for power when the exhaust steam is used for heating purposes is at once apparent.

178. Methods of Increasing Economy. — Various methods have been adopted for bettering the economy of piston engines; among them may be mentioned:

(*a*) Increasing boiler pressure.
(*b*) Increasing rotative speed.
(*c*) Decreasing back pressure by condensing.
(*d*) Superheating.
(*e*) Use of steam jackets.
(*f*) Reheating receivers.
(*g*) Compounding.
(*h*) Use of uniflow or straight-flow cylinders.
(*i*) Use of binary fluids.

179. Increasing Boiler Pressure. — A glance at Table 74 will show that increase in initial pressure, other conditions remaining the same, results in increased theoretical efficiency. This increase is so marked that engineers are considering the possibilities of employing pressures far above any now in use. There is no question but that working pressures as high as 600 pounds per sq. in. abs. will necessitate radical departure from the present type of boiler and may involve costs which are prohibitive, but the present tendency is toward the higher pressures. The design of engines for high pressures is not a difficult one since pressures as high as 800 pounds per sq. in. abs. are used successfully in Diesel engines. Several steam turbine plants are now under construction in which initial pressures of 350 pounds gauge and temperatures of 650 deg. fahr. are to be used, but until actual operating data are available no conclusions can be drawn as to the ultimate commercial economy effected by this practice. With the ordinary type of

TABLE 74.

THEORETICAL EFFICIENCY.

Rankine Cycle.

Initial Temperature Constant (600 Deg. Fahr.).

Initial Pressure, Lb. per Sq. In. Abs.	Superheat, Deg. Fahr.	Efficiency, Per Cent. Condensing Back Pressure ½ Lb. Per Sq. In. Abs.	Efficiency, Per Cent. Non-condensing Back Pressure 14.7 Lb. Per Sq. In.
1574	0	40.3	30.6
600	113.4	37.3	26.0
500	132.7	36.7	25.0
400	155.2	36.1	23.7
300	182.5	34.5	22.0
200	218.1	32.9	19.7
100	272.2	29.8	15.4

double-flow engine the heat economy increases with the pressure up to the point where increased condensation losses and leakage neutralize the theoretical gain. This point of maximum efficiency varies with the size and type of engine and the grade of workmanship.

Fig. 200 shows the results of tests made at the Armour Institute of Technology on an 8-in. by 10-in. automatic high-speed piston-valve engine. A marked gain will be noted up to a pressure of 115 lb. per

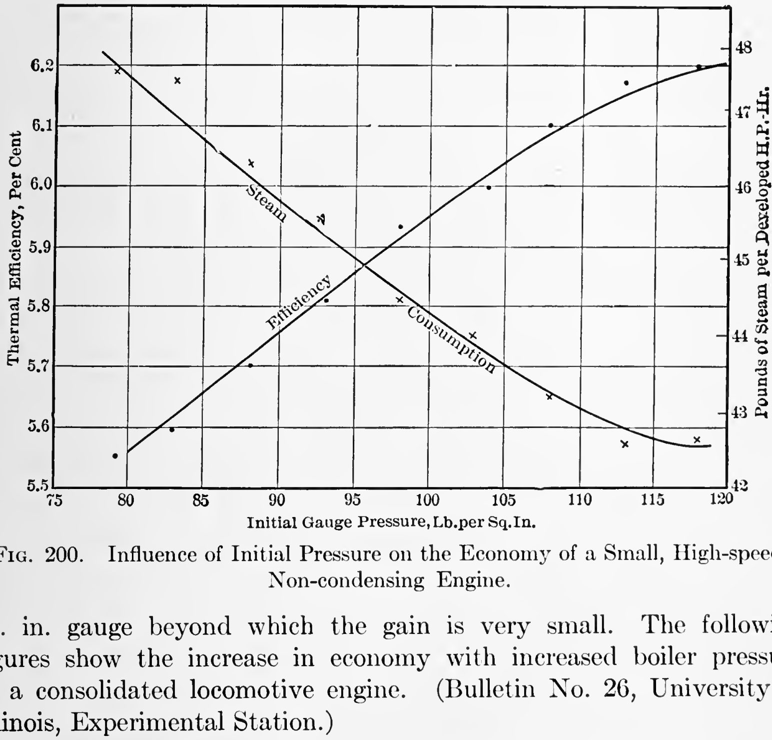


FIG. 200. Influence of Initial Pressure on the Economy of a Small, High-speed, Non-condensing Engine.

sq. in. gauge beyond which the gain is very small. The following figures show the increase in economy with increased boiler pressure in a consolidated locomotive engine. (Bulletin No. 26, University of Illinois, Experimental Station.)

Boiler pressure, lb. per sq. in...	120	140	160	180	200	220	240
Steam per i.hp-hr., lb..........	29.1	27.7	26.6	26.0	25.5	25.1	24.7

A small Willans engine, non-condensing, gave results as follows:

Initial pressure, lb. per sq. in...	36.3	51.0	74.0	85.0	97.0	110.0	122.0
Steam per i.hp-hr., lb..........	42.8	36.0	32.6	29.7	26.9	27.8	26.0

The uniflow engine offers possibilities for high pressures which are very promising but the art, at least in America, is still in an experimental stage. Tests on 100 hp. condensing engines by Lentz gave the following results:

Initial steam pressure, lb. per sq. in. abs..............	235	461
Initial temperature, deg. fahr........................	923	1018
Steam consumption, lb. per i.hp-hr...................	6.52	5.67
Heat consumption, B.t.u. per i.hp-minute	162	144

The range of pressures sanctioned by modern practice for different types of engines is as follows:

Type of Engine.	Range in Pressure (Gauge).	Average.
Simple slow-speed (standard type)	60–120	90
Simple high-speed (standard type)	70–125	100
Simple, uniflow (condensing)	125–225	175
Compound high-speed, non-condensing	100–180	150
Compound high-speed, condensing	100–180	150
Compound slow-speed, condensing	125–200	170
Triple expansion, condensing	140–250	200
Quadruple expansion, condensing	175–300	250

Higher Steam Pressures: R. Cramer, Trans. A.S.M.E., Vol. 37, 1915, p. 597.
High Pressure Steam for Superheating: Power, Dec. 28, 1915, p. 892.

180. Increasing Rotative Speed. — High rotative speed does not necessarily mean high piston speed. An 8-in. by 10-in. engine running at 300 r.p.m. has a piston speed of only 500 feet per minute, whereas a 36-in. by 72-in. Corliss running at 60 r.p.m. has a piston speed of 720 feet per minute. The classification "high speed" and "low speed" refers to rotative speed only, the former above and the latter below, say 150 r.p.m.

On account of the reduction of thermodynamic wastes, a high-speed engine should give theoretically a higher efficiency than the same engine at a lower speed, all other conditions being the same. The effect of speed upon economy is decidedly marked in engines and pumps taking steam full stroke. For example, tests of a 12-in. by $7\frac{1}{4}$-in. by 12-in. simplex direct-acting steam pump at Armour Institute of Technology showed a steam consumption of 300 pounds per i.hp-hour at 10 strokes per minute, and only 99 pounds at 100 strokes per minute. (See Figs. 381 and 382.)

Tests of engines using steam expansively, however, do not furnish conclusive evidence on this point, some showing a decided gain (Peabody, "Thermodynamics," p. 425), others little or no gain (Barrus, "Engine Tests," p. 260). For example, a small Willans engine showed an increase in economy of 20 per cent in increasing the rotative speed from 111 to 408 r.p.m. (Peabody, "Thermodynamics," p. 402), whereas the compound locomotive at the Louisiana Purchase Exposition showed a loss in economy for the higher speeds (Publication by the Pennsylvania Railroad Company). On the other hand, a comparison of the performances of high- and low-speed Corliss engines shows little difference in economy, and a general comparison between high- and low-speed engines furnishes little information, since nearly all high-speed engines are of a different class from the low-speed ones. High-speed engines are comparatively small in size, require larger clearance volume, and are usually fitted with a single valve. Rotative speed is limited by design, material, workmanship, and cost of subsequent maintenance. Speeds of 400 r.p.m. and more are not unusual with single-acting engines, whereas 300 r.p.m. is about the limit for double-acting machines with strokes over 12 inches in length. A comparison of tests of high-speed and low-speed engines in this country, irrespective of design and construction, shows the former to be less economical than the latter in most cases. In Europe high-speed engines are developed to a high degree of efficiency, and their performances are comparable with the best grade of low-speed engines.

High-speed engines as a class have the advantage of being more compact for a given power, are simple in construction and relatively low in first cost; on the other hand, they are subject to comparatively rapid depreciation, excessive vibration, and are less economical in steam consumption.

181. Decreasing Back Pressure by Condensing. — The effect of the condenser upon the power and economy of engines is indicated in Table 75. The curves in Figs. 201 and 202 were plotted from tests made by Professor R. L. Weighton on a 7, $10\frac{1}{2}$, $15\frac{1}{2}$ by 18-in. triple-expansion engine at Durham College of Science, Newcastle-on-Tyne. The straight line shows how the mean effective pressure would vary with the degree of vacuum if the power increased directly with the reduction in back pressure. The curved line shows the actual m.e.p., which increases almost along the theoretical line up to a 10-inch vacuum, from which point on the increase is less marked. At 26 inches the actual m.e.p. reaches an apparent maximum. These figures are not applicable to all engines but give a good idea of the limitation of the vacuum with the average type of reciprocating engine with restricted exhaust port openings. With specially designed ports and passages of large cross-sectional area the piston engine shows increase in steam economy up to the highest vacuum carried in the condenser. (See Power, Jan. 16, 1912, p. 72.)

The gain in steam consumption due to the condenser does not indicate a corresponding gain in heat consumption. For example, Engine No. 2, Table 75, shows an apparent gain in steam consumption, due to condensing, of 12.5 per cent, the temperature of the feed water returned to the boiler being 120 deg. fahr. With a suitable heater the exhaust of the non-condensing engine would be capable of heating the feed water to 210 deg. fahr. The non-condensing engine should therefore be credited with 210 − 120 or 90 heat units per pound of steam

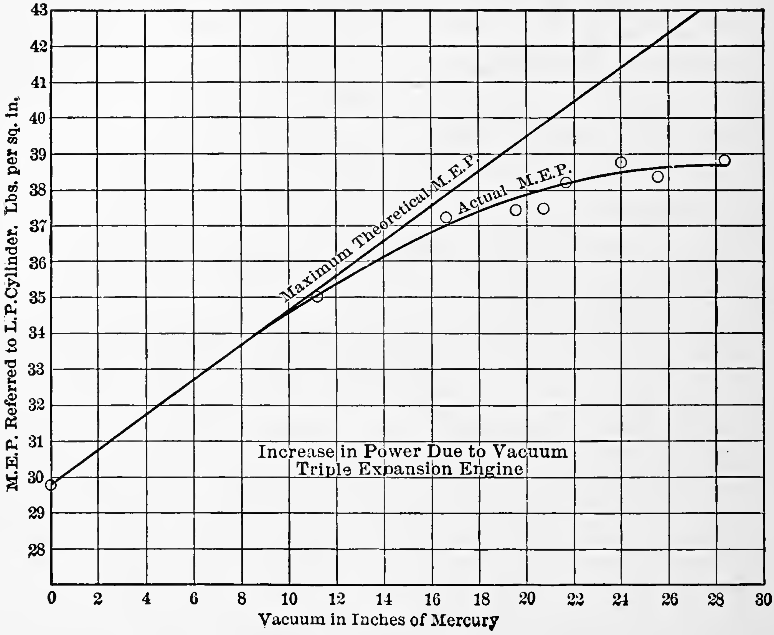


FIG. 201.

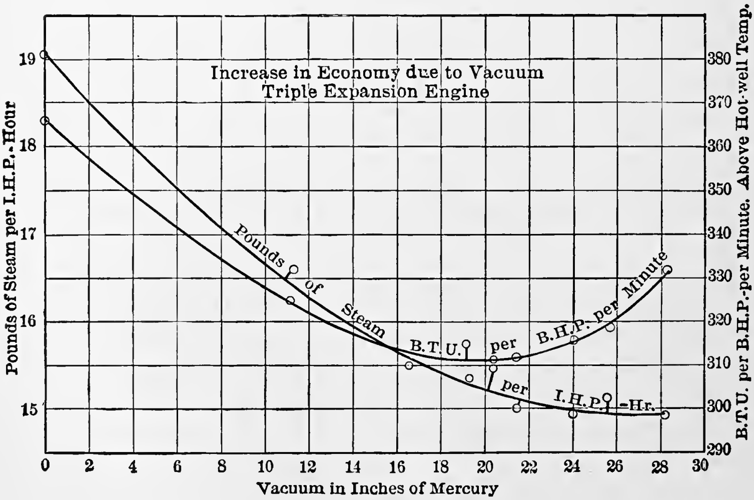


FIG. 202.

used, or, in round numbers, 9 per cent. The difference between 12.5 per cent and 9 per cent, or 3.5 per cent, represents the net gain in favor of condensing, provided the power necessary to create the vacuum is ignored. Actually, the steam consumption of the condenser pumps might be equal to or greater than 3.5 per cent of the steam generated and the net gain becomes zero or even negative. Referring to Fig. 202, plotted from tests of the 7, $10\frac{1}{2}$, $15\frac{1}{2}$ by 18-in. triple-expansion engine mentioned above, the curves show the feed-water consumption per i.hp-hour and the heat units consumed per brake horsepower per minute measured above the hot-well temperature. The engine efficiency, based upon the water consumption, increases as the vacuum increases, reaching a maximum between 26 and 28 inches, whereas the heat-unit curve gives the maximum between 20 and 21 inches. Between 22 and 28 inches the heat-unit curve shows a rapid falling off in economy. Tests of the 5500-horsepower engine at the New York Edison Company's Waterside Station showed that increasing the vacuum from 25.3 to 27.3 inches decreased the water rate only 0.06 pound per i.hp-hour. (Power, July, 1904, p. 424.) The results are illustrated in Fig. 220. In most cases, and particularly with large compound engines, the net gain due to condensing is considerable, but the feed-water temperatures and power consumed by the auxiliaries should be taken into account.

TABLE 75.

EXAMPLES OF THE EFFECT OF CONDENSING ON THE ECONOMY OF SMALL RECIPROCATING ENGINES.

	Non-Condensing.			Condensing.				Increase Due to Condensing.	
Reference Number.	Initial Gauge Pressure.	Horse Power Developed.	Steam Consumption, Pounds per H.-P. Hour.	Initial Gauge Pressure.	Back Pressure, Pounds per Square Inch Abs.	Horse Power Developed.	Steam Consumption, Pounds per H.-P. Hour.	In Power, per Cent.	In Economy, per Cent.
1	147	54.7	19.2	149	1.6	83.4	14.8	52.5	25
2	148	540	19.3	147	4	......	16.9	*	12.5
3	126	83	23.8	130	7.4	116	19.1	39.8	19.7
4	67.6	209	28.9	67	4.5	213	22	1.9	23.5
5	103.8	177.5	22.1	103.8	1.2	155	16.5	*	25.1
6	114	160	31	114	......	168	27	2	12.9
7	96	120	23.9	96	4	145	19.4	20.8	18.8
8	118	267	23.24	119	4.2	276.9	16	3.7	31
9	75.9	310	25.6	79	6.4	336	20.5	8.7	19.9
10	62.5	451	30.1	63.6	7.8	444	23	*	23.6
11	186.7	40.4	18.7	184.6	1.6	29.8	12.7	*	32

* Cut-off changed for best economy.

182. Superheating. — The theoretical gain due to the use of superheat is comparatively small as will be seen from Table 76. Considering the additional expense of equipment and maintenance of superheating apparatus the ultimate gain would appear to be a negative quantity. Practically, however, the heat economy of the piston engine is greatly increased by superheating. This apparent anomaly is due to the fact that the theoretical engine is assumed to operate in a non-conducting cycle and no condensation takes place except in doing work, whereas, in the actual mechanism the cylinder is far from being non-conducting and considerable initial condensation takes place. The reduction of cylinder condensation due to the use of superheated steam is the principal reason for the marked gain in economy of the actual engine. The greater the cylinder condensation the larger is the saving possible. As a rough approximation the steam consumption is reduced about 1 per cent for every 10 deg. fahr. increase in superheat but the actual value depends upon the type and size of engine and the initial condition of the steam. In American practice superheat corresponding to a total steam temperature of 650 to 700 deg. fahr. appears to be the limit of commercial economy but in Europe temperatures as high as 900 deg. fahr. have been employed with apparent ultimate economy.

TABLE 76.

THEORETICAL EFFICIENCIES AND WATER RATES.

Rankine Cycle — Superheated Steam.

Initial Pressure 200 Lb. Per Sq. In. Abs.

Superheat, Deg. Fahr.	Efficiency.		Water Rate.	
	Condensing.*	Non-condensing.	Condensing.*	Non-condensing.
0	31.88	18.60	6.94	13.44
50	32.03	18.71	6.72	12.96
100	32.24	18.92	6.52	12.49
150	32.49	19.18	6.34	12.03
200	32.77	19.51	6.16	11.57
250	33.09	19.89	5.98	11.10
350	33.81	20.76	5.67	10.20
400	34.20	21.25	5.48	9.77
500	35.04	22.12	5.16	9.00

* Absolute back pressure 0.5 lb. per sq. in.

Table 83 gives test results for several different types of engine employing superheated steam. These figures may be compared with the performances of engines using saturated steam as given in Tables 81 and 82. A decided gain in economy is shown in favor of superheat for single-

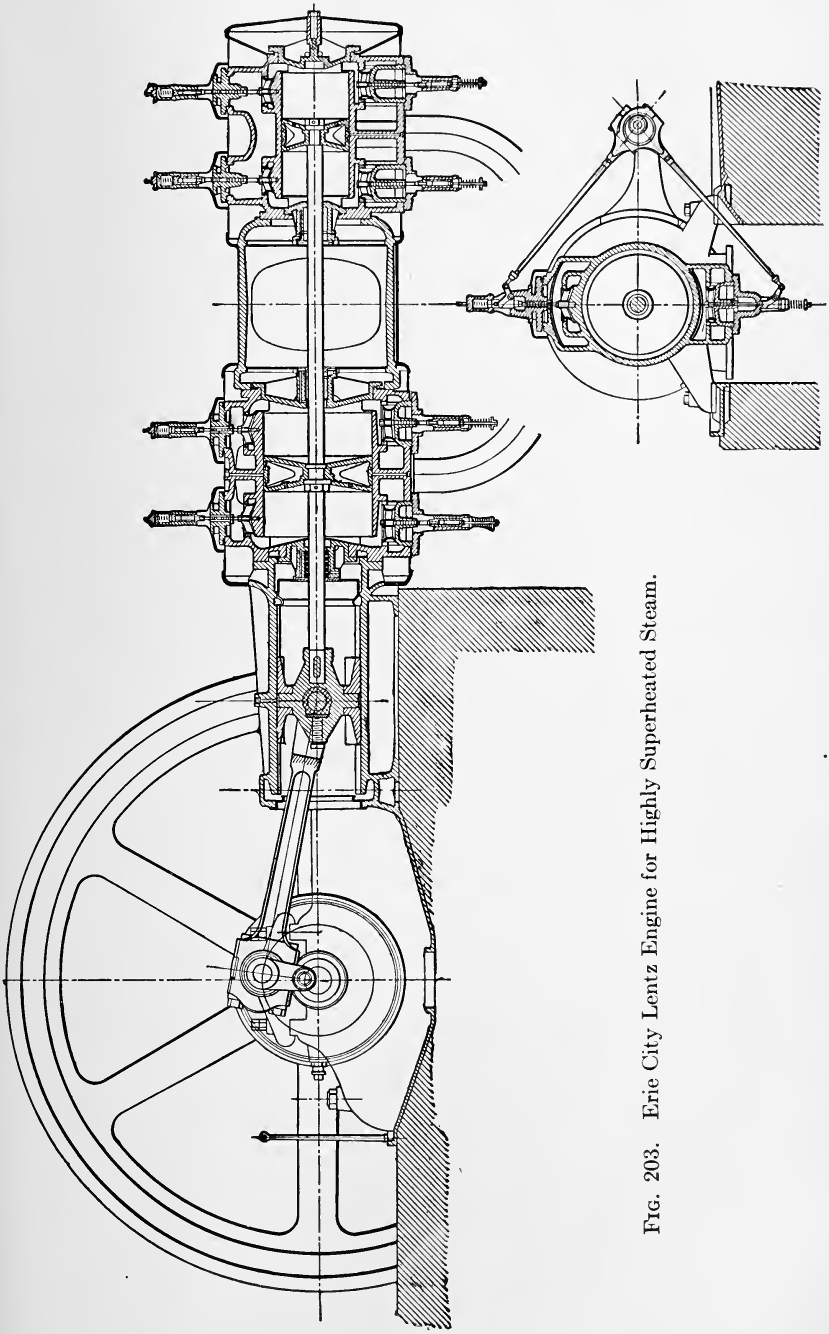


Fig. 203. Erie City Lentz Engine for Highly Superheated Steam.

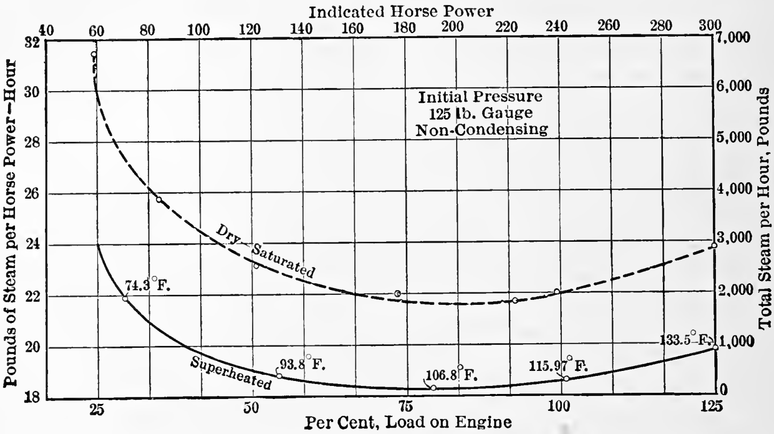
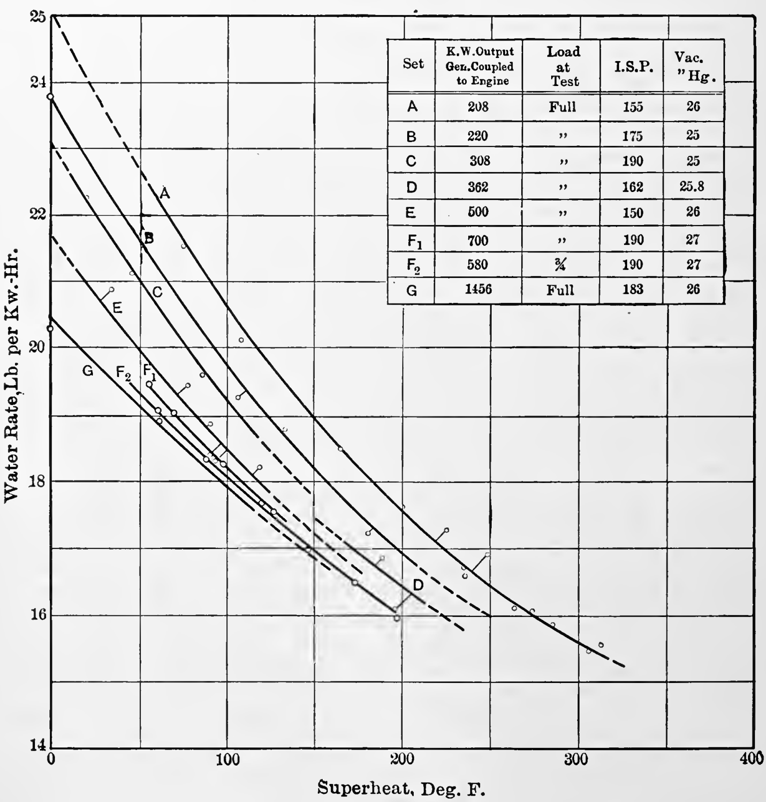


Fig. 204. Influence of Superheat on the Water Rate of a 16-inch by 22-inch Ideal Corliss Engine.



Set	K.W. Output Gen. Coupled to Engine	Load at Test	I.S.P.	Vac. "Hg.
A	208	Full	155	26
B	220	"	175	25
C	308	"	190	25
D	362	"	162	25.8
E	500	"	150	26
F_1	700	"	190	27
F_2	580	¾	190	27
G	1456	Full	183	26

Fig. 205. Effect of Superheat on Steam Consumption.

cylinder engines. With compound engines the advantage is not so apparent, while triple-expansion engines show the least gain. Tables 83 to 85 show the effect of superheating on simple, compound and triple-expansion engines. Some idea of the wonderful fuel economy effected in Europe with the use of highly superheated steam in connection with the so-called *locomobile* is gained from the results shown in Table 80. This type of engine has not yet been introduced to any extent in this country but it is only a matter of time when the cost of coal will advance to such a point as to preclude all but the more economical types of prime movers.

As far as steam consumption is concerned, all engines show greater economy with superheated than with saturated steam, but the thermal

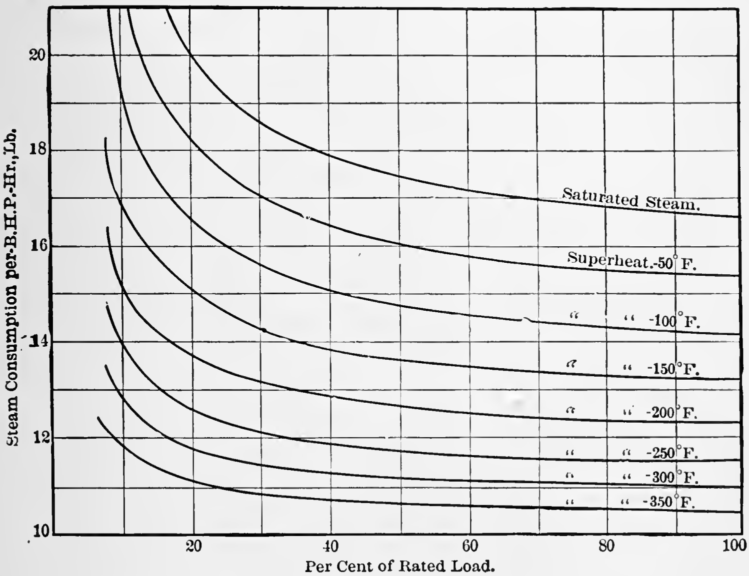


FIG. 206. Effect of Superheat on Steam Consumption.

gain is not so marked, and when the economy is measured in dollars and cents per developed horsepower, taking all things into consideration the gain is still further reduced and in some cases completely neutralized. First cost, maintenance, and disposition of the exhaust must all be considered in determining the ultimate commercial gain due to the use of superheated steam.

Fig. 205 gives the results of a series of tests made on a number of Belliss & Morcom engines using superheated steam. (Pro. Inst. of Mech. Engrs., March, 1905, p. 302.) The engines were from 200 to 1500 kilowatts capacity and were tested at full load. It is noticeable that the curves all converge to a single point and will meet at about 400 deg. fahr. The results show that if sufficient superheat is put into the steam all engines of whatever size are equally economical.

These curves though strictly applicable to the specific cases cited are more or less general and represent the influence of superheat on all types of piston engines.

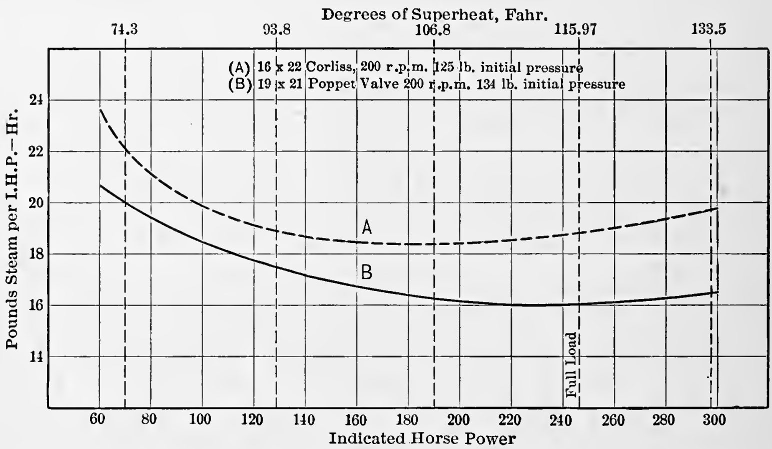


FIG. 207. Comparative Water Rates of a Corliss Four-valve and a Poppet Four-Valve, High-speed Engine.

183. Jackets. — If the walls of the cylinder are made double and the space between is filled with live steam under boiler pressure, the cylinder is said to be steam jacketed. The function of the jacket is to reduce initial condensation by maintaining the temperature of the internal walls as nearly as possible equal to that of the entering steam. The heat given up by the jacket steam, and the resulting condensation, is usually a smaller loss than would otherwise result from cylinder condensation. However, tests of numerous engines with and without steam jackets do not agree as to the conditions under which their use is profitable, the apparent gain ranging from zero to 30 per cent. According to Peabody, a saving of from 5 to 10 per cent may be made by jacketing simple and compound condensing engines, and a saving of from 10 to 15 per cent by jacketing triple expansion engines of 300 horsepower and under. On large engines of 1000 horsepower or more the gain, if any, is very small. (Peabody, "Thermodynamics," p. 400.)

Other things being equal, the smaller the cylinder and the lower the piston speed the greater is the value of the jacket. Experiments show no advantage in increasing the jacket pressure more than a few pounds above that of the initial steam in the cylinder, and it is usual to reduce the pressure in the jackets of the second and succeeding cylinders of multi-expansion engines. (Ripper, "Steam Engine," p. 170.)

To be effective, jackets should be well drained, kept full of live steam, and the water of condensation returned directly to the boiler.

Pumping engines and other slow-speed engines running at practically constant load are generally jacketed, but in street-railway work and in the majority of manufacturing plants carrying fluctuating load, jackets are not considered advantageous.

Whatever may be the actual economy due to jacketing, there is no question but that the jacket greatly influences the action of the steam in the cylinders, and whether beneficially or not depends upon the design and construction of the engine. Unless otherwise specified, manufacturers usually build their engines without jackets.

A revival of the steam jacket for small single cylinder engines is quite probable if the exceptional results obtained by Prof. E. H. Miller of the Massachusetts Institute of Technology on a Prosser-Fitchburg engine are maintained in practice. The engine is simple, non-condensing. The heads and barrel of the cylinders are jacketed with steam at throttle pressure. The cylinder has poppet valves, steam and exhaust, and is equipped with a double eccentric valve gear. A total steam consumption (steam dry and saturated at admission) of 20.46 lb. per i.hp-hr. was recorded, corresponding to a Rankine cycle ratio of 83 per cent. With steam superheated to 86.7 deg. fahr. the water rate was reduced to 16.59 lb. per i.hp-hr. corresponding to a Rankine cycle efficiency of 88.7 per cent. Rankine cycle ratios as high as 92.3 per cent are purported to have been realized in shop tests. See Table 78 for results of Prof. Miller's Tests.

Jacketing Applied to Steam Cylinders: Power, Mar. 18, 1913, p. 368.

184. Receiver Reheaters: Intermediate Reheating. — The receivers between the cylinders of multi-expansion engines are frequently equipped with heating coils, as illustrated in Fig. 453, the function of which is to superheat the exhaust steam before delivering it to the cylinder immediately following, with a view of reducing the losses occasioned by cylinder condensation. The coils are supplied with live steam under boiler pressure and may serve to evaporate a portion of the moisture or to actually superheat the steam supplied to the following cylinder. The question of the propriety of using reheaters is an open one, since reliable data relative to their use are meager and discordant. The conditions under which the few recorded tests were made are too diverse to warrant definite conclusions. Some show an appreciable gain in economy, others a decided loss. A reheater is of little value in improving the thermodynamic action of the engine, and is probably a loss unless it produces a superheat of at least 30 deg. fahr., and to be fully effective should superheat above 100 deg. fahr. (L. S. Marks, Trans. A.S.M.E., 25–500.) The effectiveness of the reheater will evidently be increased by the removal of the greater portion of the moisture from the exhaust steam before it enters the receiver. In the 5500-horsepower engine at the Waterside Station in New York it was shown that both jackets and reheaters, either together or alone, were practically valueless, throughout the working range of load. (Power, July, 1904, p. 424.) Many similar cases may be cited which show no gain in economy with the use of the reheaters. In all cases the reheater effects a great reduction in the condensation in the low-pressure cylinders, but the resulting gain, considering the condensation in the reheater coils, may be little, if any. On the other hand, with properly proportioned reheaters, the gain may be considerable and particularly with superheated steam. Practically all European engines operating with highly superheated steam are equipped with receiver-reheaters. In the locomobile type of engine plant the intermediate reheating is effected by heating coils placed in the path of the furnace gases. See also paragraph 194.

In triple-expansion pumping engines receiver-reheaters are found to effect an appreciable gain in economy, and practically all such engines are equipped with them. In electric traction work or where the load is a widely fluctuating one the reheater has been virtually abandoned. Apart from the consideration of fuel economy, all tests show a marked increase in the indicated power of the low-pressure cylinder (5 to 15 per cent), and to that extent it increases the capacity of the entire engine. (G. H. Barrus, Power, Sept., 1903, p. 516.)

Engine Reheaters: Mech. Engr., Dec. 23, 1910.

185. Compounding. — If the entire expansion instead of being effected in a single cylinder is allowed to take place in two or more cylinders the engine is said to be "compounded." The term "compound" without qualification, however, refers only to the two-cylinder arrangement. If expansion takes place in three stages the engine is known as a triple-expansion engine; similarly, the four-stage machine is called a quadruple expansion engine. When high-pressure steam is admitted into a single cylinder engine of the ordinary double-flow type and expansion is carried down to a comparatively low point a large portion is condensed by the metal surfaces; at the end of the stroke and during exhaust some of the water is re-evaporated, but the steam so formed is discharged without doing useful work. If the same weight of steam is expanded through the same pressure range in a compound engine, the temperature range in each cylinder will be less, initial condensation will be reduced and part of the heat lost in the first cylinder by leakage and clearance will do work in the second cylinder. The higher the temperature range the more pronounced will be the thermal economy effected by compounding. The number of stages is limited commercially because of the first cost, complexity, cost of lubrication, attendance and maintenance.

Cylinder ratios for high-speed single-valve compound engines vary from about 1 to $2\frac{1}{2}$ with 100 pounds pressure to about 1 to 3 with a pressure of 150 pounds, and for slow-speed condensing engines from 1 to 3 with 125 pounds pressure to about 1 to 4 with a pressure of 175 pounds. G. I. Rockwood recommends a ratio as high as 1 to 7, and a number of engines designed along this line have shown exceptional economy. For variable load operation two stages appear to give the best ultimate economy. In case of very large condensing engines the last stage consists of two cylinders because of the unwieldy and costly size of a single unit. For constant loads as in pumping stations and large marine installations three and four stages appear to be the best investment. The ratio of expansion for a multi-expansion engine is the ratio of the volume at release in the low-pressure to that at cut-off in the high-pressure cylinder. Commercially it is usually taken to be the product of the ratio of the volume of large to small cylinder divided by the fraction of the stroke at cut-off in the high-pressure cylinder. For example, a compound engine with cylinders 24-in., 48-in. by 48-in. cutting off at $\frac{1}{3}$ in the high-pressure cylinder has a nominal ratio of expansion of $4 \div \frac{1}{3} = 12$. The number of expansion at rated load in multi-expansion condensing engines varies widely, ranging from 10 to 33, with an average not far from 16.

The respective advantages and disadvantages of compounding may be tabulated as follows:

Advantages	Disadvantages
1. Permits high range of expansion.	1. Increased first cost due to multiplication of parts.
2. Decreased cylinder condensation.	2. Increased bulk.
3. Decreased clearance and leakage losses.	3. Increased complexity.
4. Equalized crank effort.	4. Increased wear and tear.
5. Increased economy in steam consumption.	5. Increased radiation loss.

186. Uniflow or Unaflow Engine. — A study of the Rankine cycle will show that the greater the pressure range between admission and release the greater will be the theoretical thermal efficiency. In the standard double-flow, unjacketed type of engine the actual thermal efficiency increases with the pressure range up to a certain maximum beyond

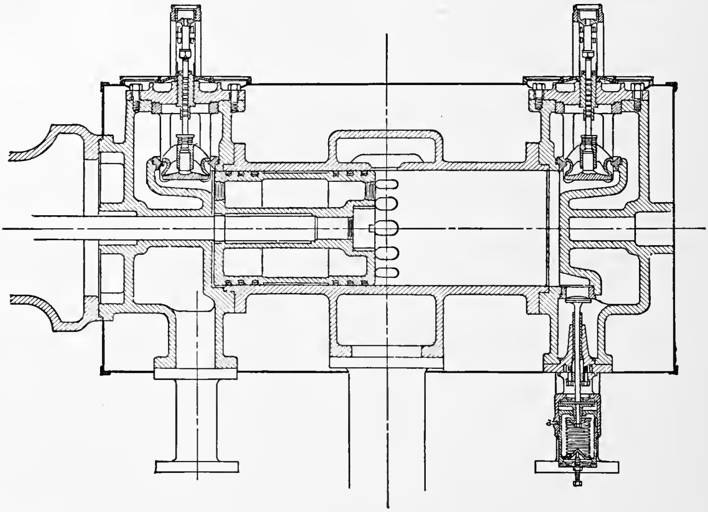


Fig. 208. Section through Cylinder of a Nordberg Uniflow Engine Showing Location of Cataract Relief Valve.

which increased leakage and cylinder condensation offset the theoretical gain. This maximum varies with the type and size of engine, number of cylinders, design of valve gear and other influencing factors.

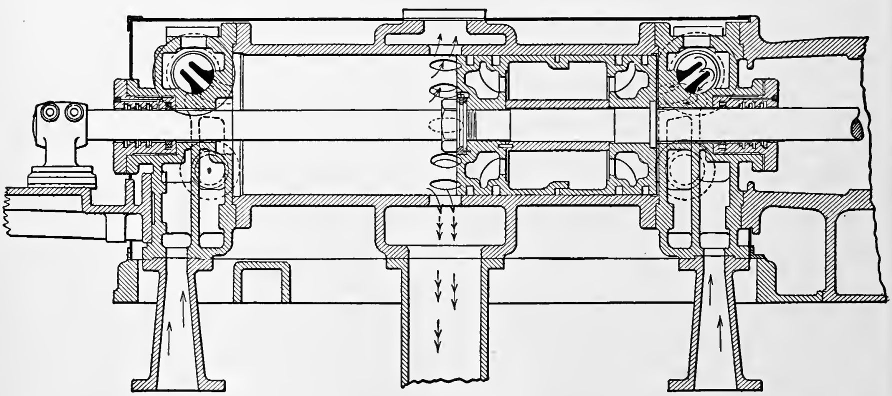


Fig. 209. Section through Cylinder of a C. & G. Cooper Co.'s. Uniflow Engine.

Cylinder condensation is increased with the pressure range because the cylinder head and other clearance surfaces are chilled by the exhaust steam so that at the beginning of the stroke a considerable portion of the incoming steam is condensed. With superheated steam an equivalent heat exchange takes place.

In the uniflow engine the steam enters at the end of the cylinder as in the double-flow type but it is exhausted from the center at the fur-

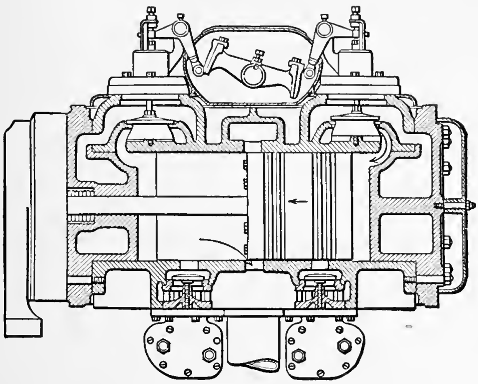


FIG. 210. Section through Cylinder of a Skinner "Universal" Uniflow Engine.

thest point from the heads. (See Fig. 208.) Consequently the cylinder heads are exposed to exhaust temperature only for the very small length of time that it takes the piston to uncover the ports. Furthermore, the heads are jacketed with live steam (and in some designs the entire

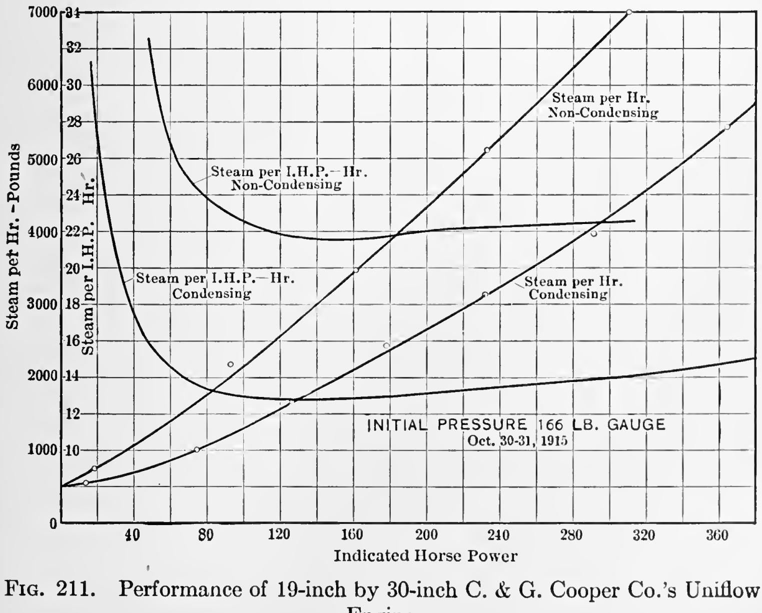


FIG. 211. Performance of 19-inch by 30-inch C. & G. Cooper Co.'s Uniflow Engine.

cylinder) so that on the return stroke the steam at exhaust temperature and pressure is compressed against the hot surfaces of the cylinder head; thus when the admission valve opens the incoming steam meets no cold surface and cylinder condensation is largely prevented. The result is that a wide pressure and temperature range can be allowed in a single cylinder with good economy. In fact the heat consumption of a uniflow engine operating condensing is equal to that of a compound Corliss.

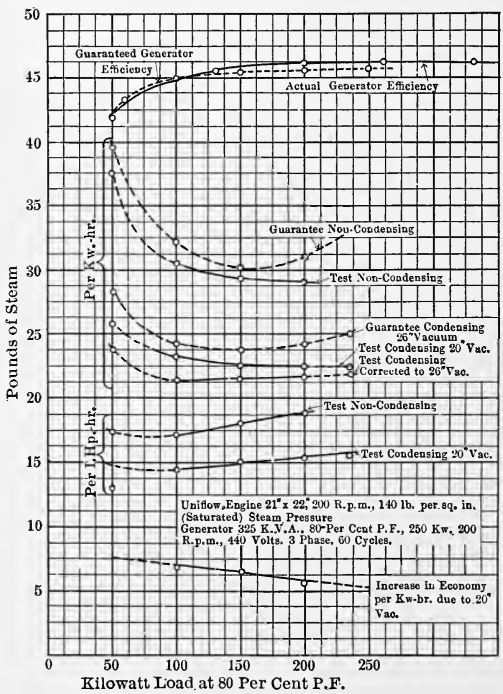


FIG. 212. Guarantee and Test Performance of a 21-inch by 22-inch Uniflow Engine.

For ordinary non-condensing service the economy is no better than that of a high-grade single-cylinder poppet-valve engine for the same operating conditions. This is due primarily to the harmful effects of the excessive pressure resulting from compression or to the methods employed for reducing this pressure. In the typical uniflow engine of low clearance volume, compression begins as soon as the piston covers the exhaust ports (corresponding to approximately 90 per cent of the stroke), and for moderate or low vacua the resulting pressure at the end of compression is equal to or less than that at admission, but with high back pressure as in non-condensing service it may be greatly in excess. To prevent this excessive rise in compression for non-condensing service American manufacturers either increase the clearance volume (Ames Stumpf "Unaflow" engine and C & G. Cooper Co. "Uniflow" engine) or employ an auxiliary valve which delays compression (Universal "Unaflow" engine). For high initial pressures the clearance volume may be reduced with a resulting increase in economy. In case the vacuum is lost when operating condensing, compression pressure may be relieved by an adjustable snifting valve (Fig. 208) or by means of the auxiliary valve mentioned above. Some idea of the economy

TABLE 77.

OPERATING PERFORMANCE OF A 33-IN. BY 36-IN. C. & G. COOPER CO.'S UNIFLOW ENGINE.

1. Engine, C. & G. Cooper Co., rated output 640 i.hp. or 440 kw. at switchboard.
2. Cylinder 33 in. by 36 in. Piston rod, head end, 5½ in. diameter; crank end, 5⅞ in. diameter.
3. Clearance, head end 4 per cent; crank end 3.8 per cent.
4. Boilers, Heine Boiler Co., rated at 200 hp. each. 2000 sq. ft. heating surface. Dean shaking grates hand fired. Grate area 39.6 sq. ft.
   Auxiliaries, Dean Bros. Surface condenser and pumps. 1400 sq. ft. cooling surface in condenser.
5. Date, April 25 and 26, 1914.

6. No. of run	1	2	3	4	5	6	7	8	9
7. Duration, min.	191	62	112	71	103	53	60	52	30
8. Barometric pressure:									
(a) in. of mercury	29.34	29.35	29.36	29.40	29.48	29.48	29.50	29.50	29.50
(b) lbs. per sq. in.	14.39	14.40	14.40	14.43	14.48	14.48	14.49	14.49	14.49
9. Steam pressure in supply pipe, lb.	161.60	162.50	163.90	161.20	158.50	154.20	145.60	125.30	127.50
10. Vacuum referred to 30 in. barometer:									
(a) at engine, in.	23.48	22.67	22.55	20.98	23.56	19.15	23.26	......	....
(b) in condenser, in.	24.61	24.34	24.18	23.77	24.18	23.27	23.32	24.35	23.53
11. Temperature of steam at engine, deg. fahr.	408.5	422.1	415.0	423.1	410.2	423.1	412.1	381.3	373.3
12. Superheat at engine, deg. fahr.	37.2	50.4	42.8	52.3	40.3	55.3	48.5	28.3	19.1
13. Temperature of condensed steam in measuring tanks, deg. fahr.	106	110	113	122	96	126	92	83	....
14. Temperature of cooling water entering condenser, deg. fahr.	76.9	77.2	73.4	72.2	74.5	70.2	77.2	......	....
15. Temperature of cooling water leaving condenser, deg. fahr.	88.6	88.4	87.4	91.5	82.9	90.7	79.6	......	....
16. Temperature of engine room, deg. fahr.	80	78	80	80	79	81	86	......	....
17. Steam used by engine during run, lb.	26201	10041	20156	16086	10070	16068	4070	2002	362
18. Steam used by engine per hr., lb. per hr.	8230	9722	10798	13594	5866	18190	4070	2280	725
19. R.p.m. of engine	124.96	124.43	123.92	123.47	125.52	124.23	124.50	124.04	124.20
20. Piston speed of engine, ft. per min.	749.76	746.58	743.52	740.82	753.12	747.00	745.38	744.24	745.20
Power as Measured at Switchboard.									
21. Volts	242.5	243.7	243.7	244.5	243.7	244.7	244.7	245.7	....
22. Amperes	1048	1212	1354	1583	794	1959	535	270	....
23. Kilowatts by wattmeter, kw.	440.8	513.3	555.1	669.2	336.3	815.5	239.2	123.2	....
24. Steam used by engine under actual conditions of operation, lb. per kw-hr.	18.67	18.94	19.45	20.31	17.44	22.31	17.02	18.57	....
Heat Data.									
25. Heat units in each lb. of steam supplied, B.t.u.	1108	1111	1106	1103	1110	1093	1111	1098	....
26. Total units supplied per hr. per kw., B.t.u. per kw-hr.	20680	21060	21510	22400	19360	24400	18900	20380	....
27. Thermal efficiency ratio between heat equivalent of kw. at the switchboard and heat units supplied in the steam per kw.	16.5	16.2	15.9	15.3	17.6	14	18	16.8	....
28. Heat units which would be obtained by perfect (adiabatic) expansion from initial to final pressure per lb., B.t.u.	227	272	271	258	275	243	267	267	....
29. Heat units per kw., B.t.u. per kw.	5170	5150	5270	5250	4800	5430	4550	4960	....
30. Rankine cycle ratio, per cent	66.2	66.3	64.8	65.1	71.2	63.0	75.0	68.9	....

effected by the American types of uniflow engine is shown in Fig. 212. In Europe the uniflow engine has been developed to a very high point of efficiency and exceptional heat economies have been recorded. Aside from the high efficiency in a single cylinder a characteristic feature of the uniflow engine is the capacity for heavy overloads and low underloads with a flat water rate curve (Fig. 211).

The cylinder diameter of the uniflow engine is larger than that of an equivalent single cylinder non-condensing engine, but it is smaller than the low-pressure cylinder of an equivalent compound engine.

It is difficult to predict the extent to which the uniflow engine will replace the double-flow type, but if the claims of the builders are substantiated it will prove a formidable competitor of both the compound piston engine and turbine at least for sizes ranging between 200 and 2000 horsepower.

187. Use of Binary Vapors. — A consideration of the Carnot or Rankine cycles shows that theoretically the efficiency of the steam engine may be increased by raising the temperature of the steam supplied or by lowering the temperature of the exhaust, that is to say, by increasing the range. Superheated steam development has practically determined the upper limit, and economical practice indicates a vacuum of about 26 inches, corresponding to 126 deg. fahr., as the average lower limit for most efficient results from a commercial standpoint.

In the binary-vapor engine the working range has been considerably increased by substituting a highly volatile liquid, as sulphur dioxide, for the water which is ordinarily used as the cooling medium in the surface condenser.

The SO_2 in condensing the exhaust steam is itself vaporized and the vapor, under a pressure of about 175 pounds per square inch, used expansively in a secondary reciprocating engine. The exhausted SO_2 is discharged into a surface condenser in which it is liquefied by cooling water much the same as in refrigerating practice and used over and over again. Referring to Fig. 213, which illustrates diagrammatically a binary-vapor engine at the Royal Technical High School, Berlin: A, B, and C are the three steam cylinders of an ordinary triple-expansion engine and D the SO_2 cylinder. All four cylinders drive a common crank shaft E. F is a high-pressure surface condenser which acts as a vaporizer for the SO_2 and a condenser for the steam. G is a surface condenser which serves to condense the SO_2 vapor. H is a liquid SO_2 tank. The operation is as follows: Highly superheated steam enters the high-pressure steam cylinder at I and leaves the low-pressure cylinder at J, just as in any steam engine. The exhaust steam enters chamber *F* and is condensed by the liquid SO_2 passing through the coils. The condensed steam and entrained air are removed from the chamber by a suitable air pump. The steam in condensing gives up its latent heat to the liquid SO_3 and causes it to vaporize. The SO_2

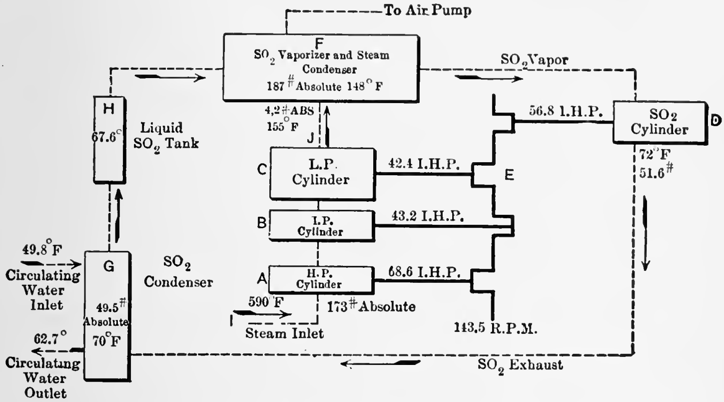


FIG. 213. Diagram of Binary-vapor Engine.

vapor passes from the coils in chamber *F* to the SO_2 engine *D* and performs work. The exhausted SO_2 vapor flows from cylinder *D* to chamber *G*, and is condensed by cooling water flowing through a series of tubes. The liquid SO_2 is collected in liquid tank *H* and thence is pumped into the coils in vaporizer *F*. The approximate temperatures and pressures at different points of the cycle are indicated on the diagram.

A number of experiments made by Professor E. Josse in the laboratory of the Royal Technical High School of Berlin on an experimental plant of about 200 horsepower gave some remarkable results. A few of the tests made with highly superheated steam gave the following average figures:

I.hp. (steam end)	146.4
Steam consumption per i.hp-hour	12.8
I.hp. (SO_2 end)	52.7
Percentage of power of SO_2 engine	35.9
Steam consumption per i.hp-hour of combined engine	9.43

When operating under the most satisfactory conditions a performance of 8.36 pounds of steam per i.hp-hour was recorded, corresponding to a heat consumption of 158.3 B.t.u. per minute. While this is an exceptional performance better results have been obtained with the uniflow engine and high-grade poppet-valve engine of the double-flow type. The binary-vapor engine has not proved to be a commercial success because of the high first cost and high maintenance charge.

SO_2 does not attack the metal surface of the engine unless combined with water, in which case sulphurous acid is formed. There is, however, no danger from this cause, since the SO_2 being under greater pressure effectually prevents leakage of water into the SO_2 system.

The SO_2 cylinder requires no other lubrication than the SO_2 itself, which is of a greasy nature.

Properties of SO_2: Trans. A.S.M.E., 25–181. *Binary-vapor Engines:* Jour. Frank. Inst., June, 1903; Elec. World and Engr., Aug. 10, 1901; U. S. Cons. Reports, No. 1139, Sept. 14, 1901; Engr. U. S., Aug. 1, 1903; Sib. Jour. of Eng., March, 1902.

188. Types of Piston Engines. — A general classification of the various types of engines used in steam power plant operation is unsatisfactory because of the overlapping of the various groups and the following modifications of a chart devised by Hirshfeld and Ulbricht ("Steam Power," p. 92) is merely offered as a general summary of the different nomenclatures used in connection with this class of prime mover.

- Rotative speed basis
  - High speed
  - Medium speed
  - Low speed
- Ratio of stroke to diameter basis
  - Short stroke
  - Long stroke
- Longitudinal axis basis
  - Vertical
  - Inclined
  - Horizontal
- Cylinder Arrangement
  - Single cylinder
  - Tandem compound
  - Cross compound
  - Duplex
  - Angle compound
- Valve gear basis
  - Slide valve
    - D-slide valve
    - Balanced slide valve
    - Multiported slide valve
    - Piston valve
  - Corliss valve
  - Poppet valve
    - Drop cut-off
    - Positively operated
- Steam expansion basis
  - Single expansion or single engine
  - Multi-expansion engine
    - Compound
    - Triple
    - Quadruple
- Steam flow basis
  - Double flow
  - Uniflow
- Crank mechanism basis
  - Standard
  - Back acting
  - Trunk
  - Oscillating
- Operating basis
  - Initial pressure
    - High pressure
    - Medium pressure
    - Low pressure
  - Back pressure
    - Condensing
    - Non-condensing

No attempt will be made to describe the various types as outlined in this chart further than that incident to the discussion of their relative merits for power plant service.

189. High-speed Single-valve Simple Engines. — This style of engine is made in sizes varying from 10 to 500 horsepower. The cylinder dimensions vary from 4-in. by 5-in. to 24-in. by 24-in. and the rotative speed from 400 to 175 r.p.m.

When ground is limited or costly and exhaust steam is necessary for heating or manufacturing purposes, the high-speed non-condensing engine is most suitable for horsepowers of 200 or less, being compact, simple in construction and operation, and low in first cost. For sizes larger than this the compound or uniflow engine may prove a better investment, except in cases where fuel is very cheap or large quantities of exhaust steam are to be used for manufacturing purposes.

Small high-speed engines are seldom operated condensing, since the gain due to reduction of back pressure is more than offset by the extra cost of the condenser and appurtenances.

Engines are ordinarily rated at about 75 per cent of their maximum output. For example, a 12-in. by 12-in. non-condensing engine running at 300 r.p.m., with initial steam pressure of 80 pounds gauge, is normally rated at 70 horsepower, though it is capable of developing 90 horsepower at the same speed.

The steam consumption of high-speed single-valve non-condensing engines at full load ranges from 26 to 50 pounds per indicated horsepower-hour, depending upon the size of the unit and the conditions of operation. An average for good practice is not far from 30 pounds. With superheated steam a steam consumption as low as 18 pounds per horsepower-hour has been recorded.

Table 81 gives the steam consumption of a number of single-valve high-speed engines running condensing and non-condensing, and Fig. 215 shows some of the results for different loads. The steam consumption is fairly constant from 50 per cent of the rated load to 25 per cent overload, but for earlier loads the economy drops off rapidly. The desirability of operating the engine near its rated load is at once apparent. The curves show a marked economy in favor of the larger cylinders, but the engines are not of the same make, and the conditions of operation are somewhat different.

The most economical cut-off for a simple engine is about one-third to one-fourth stroke when running non-condensing, and about one-sixth when running condensing.

The performances given in Table 81 are exceptional. It is not advisable to count on a better steam consumption for this type of engine than 30 to 35 pounds of steam per i.hp-hr.

The curves in Fig. 214 give the performance of a modern, high-grade, unjacketed, 15-in. by 14-in., high-speed, single-valve, simple, non-condensing engine at various ratings. It is not likely that this type and size of engine can be designed to better the results shown in the curves for the given conditions.

In general, when the requirements for exhaust steam are in excess of the

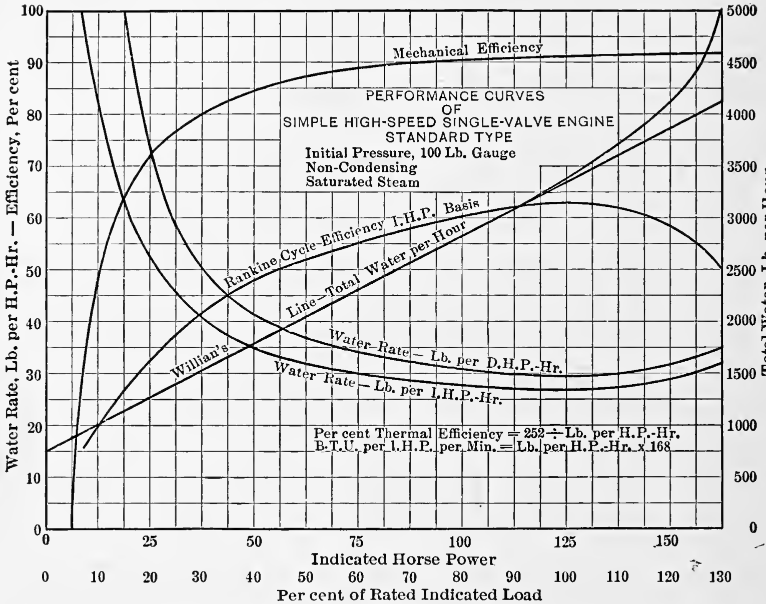


Fig. 214. Characteristic Performance Curves of a High-grade, Single-cylinder, Single-valve, Non-condensing Engine.

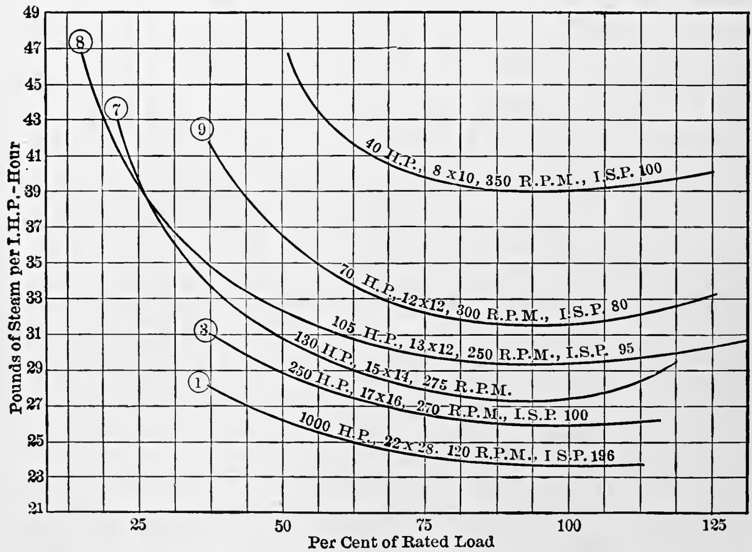


Fig. 215. Typical Economy Curves of High-speed, Single-valve, Non-condensing Engines. Saturated Steam.

steam consumption of a simple non-condensing engine a high-grade economical engine is without purpose.

190. High-speed Multi-valve Simple Engines. — The steam distribution in a single-valve engine may give good economy for a very small range in load but may be far from satisfactory for a wide range. This must necessarily be so since admission, cut-off, release, and compression are all functions of one valve, and any change in one results in a change of the others. To obviate the limitations of the single valve, many builders design engines with two or more valves. With a two-valve engine cut-off is independent of the other events, and with four valves all events are independently adjustable. In addition to the flexibility of the valve gear, the chief feature of the four-valve engines lies in the reduction of clearance volume which is made possible by placing the valves directly over the ports. The valves may be of the common slide-valve, or rotary type. As a class, four-valve engines are more economical than those having a less number of valves. The advantages and disadvantages of the four-valve over the single-valve engines may be tabulated as below.

Advantages.	Disadvantages.
1. Better steam distribution.	1. Increased number of parts.
2. Better regulation.	2. Increased first cost.
3. Reduced clearance volume.	3. Requires greater attention.
4. Less valve leakage.	
5. Better economy.	

The steam consumption of a high-speed Corliss non-condensing engine at full load varies from 21 to 27 pounds of saturated steam per i.hp-hr. (pressure 125–140 lb. gauge) with an average not far from 25 pounds. With superheated steam the water rate may run as low as 17 lb. per i.hp-hr. The poppet-valve type appears to be more economical in steam consumption than the Corliss, and a water rate for saturated steam as low as 18.9 lb. per i.hp-hr. has been recorded. A very high degree of superheat can be used with the poppet-valve type and water rates as low as 16 lb. per i.hp-hr. (initial pressure 150 lb. gauge, superheat 250 deg. fahr.) are not unusual. The high-speed, four-valve engine is usually operated non-condensing. Rankine cycle efficiencies over 80 per cent have been realized with both saturated and superheated steam. An exceptional record for a condensing unit is reported by Lentz. With steam at 461 lb. abs. initial pressure and steam temperature of 1018 deg. fahr. a 100-hp. Lentz unjacketed simple engine developed an indicated horsepower on a steam consumption of 5.67 lb. per hour.

Fig. 216 gives a comparison between a single-valve and a four-valve (Corliss type) high-speed engine, using saturated steam, and though the engines differ slightly in size, the conditions of operation were comparable and the marked gain in economy of the latter over the former is apparent. Both performances are exceptional, and a 10 to 15 per cent greater steam consumption may be expected in average good practice.

As a general rule single-valve simple engines do not exceed 500 horsepower in size for stationary work, whereas 1000 horsepower is not an uncommon size for the multi-valve type.

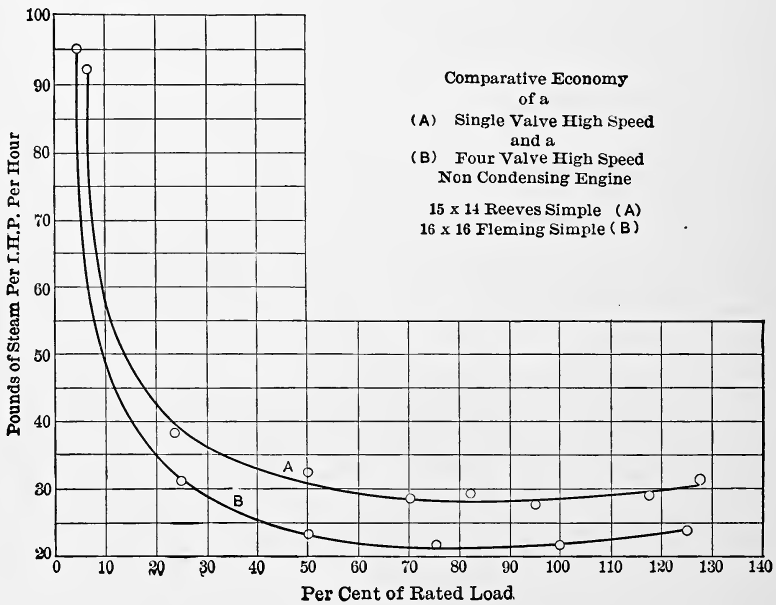


Fig. 216.

191. Medium and Low-speed Multi-valve Simple Engines. — A comparison of tests of high- and low-speed single-valve engines irrespective of design and construction shows the former as a class to be less economical than the latter. With four-valve engines there is no such disparity, and the high-speed type has shown just as good economy as the slow-speed class.

Of the various types of simple, low- or medium-speed, four-valve engines the poppet-valve appears to be the more economical in heat consumption, but so much depends upon the grade of workmanship that general comparisons are apt to lead to error. A comparison of the steam consumption of a high-speed, four-valve Corliss and a four-valve poppet engine, non-condensing, is shown in Fig. 207. The size and initial pressure are somewhat in favor of the poppet-valve mechanism so that the results are not strictly comparable but the exceptional economy of both types is apparent from the curves.

The following table taken from the report of Prof. Edw. F. Miller of the Massachusetts Institute of Technology gives the results of a Fitchburg Prosser single-cylinder, four-valve, jacketed, non-condensing engine which establishes a record for a small simple machine of the double-flow type using saturated steam.

TABLE 78.

ECONOMY TESTS OF A 15-IN. BY 24-IN. FITCHBURG-PROSSER ENGINE.

Non-condensing.

Test No.	1	2	3
Barometer, inches	29.5	29.32	29.32
Boiler pressure gauge, lb	124.5	121.6	101.5
Degrees superheat, fahr	86.7	0.999*	0.999*
R.p.m.	81.05	82.04	80.09
Indicated horsepower	54.84	48.74	54.39
Steam† per i.hp-hr	16.59	19.07	20.46
B.t.u. per i.hp. per minute	291.0	317.0	341.0
Rankine cycle ratio, per cent	81.05	82.04	80.09

* Quality. † Includes jacket condensation.

The low-speed multi-valve single-cylinder unit ranges in size from 50 to 3000 horsepower with cylinders varying from 12-in. by 30-in. to 48-in. by 72-in. The smaller sizes with trip gear operate at 90 to 120 r.p.m. and the larger at 50 to 100 r.p.m. Without trip gear, speeds of 150 r.p.m. are not uncommon but at this speed they are usually classified as high-speed engines.

A few exceptional performances of this type of engine for saturated steam are given in Table 81. For results with superheated steam see Table 83.

192. Compound Engines. — It should be borne in mind that the principal object of compounding is to permit the advantageous use of high pressures and large ratios of expansion and consequently this type of engine need not be considered for pressures lower than 125 lb. per sq. in. gauge. This does not signify that 125 lb. is the limiting pressure for compounding; on the contrary, compound condensing engines with initial pressures as low as 90 lb. have shown better heat economy than simple engines of the same capacity, but the thermal gain for these low pressures is usually more than offset by fixed charges and other practical considerations. In general, compounding increases the steam economy at rated load from 10 to 25 per cent for non-condensing engines and from 15 to 40 per cent for condensing engines. Compound engines range in size from the 100-hp. tandem, single-valve, automatic, high-speed, non-condensing unit to multi-valve, cross-compound condensing units of 4000 hp. or more. Compound engines have been built, and are still operating, up to 10,000 hp. rated capacity but the steam turbine has practically superseded the piston engine for sizes larger than 2000 hp. High-grade compound engines of the full poppet-valve type with superheated steam are more economical in steam consumption at rated load than steam turbines of the same capacity, but first cost, size, maintenance and attendance are decidedly in favor of the

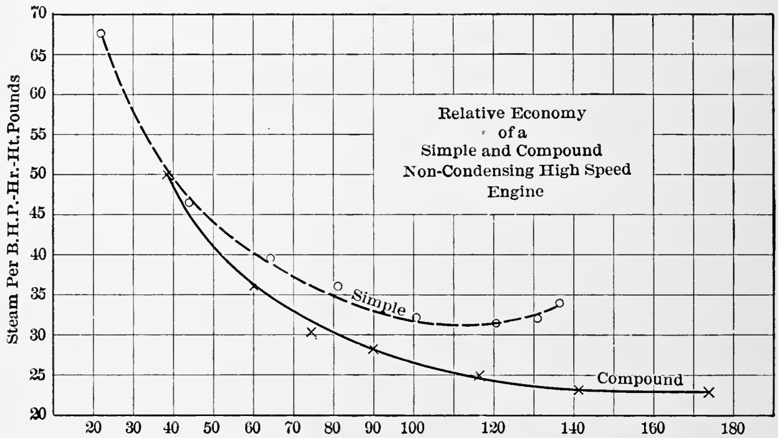


FIG. 217. Comparison of a Simple and Compound Slide-valve Engine.

turbine, at least for sizes over 2000 hp. Low rotative speed and reversibility, however, are points in favor of the engine, but the former may be offset by the turbine in connection with suitable reduction gearing.

With saturated steam the water rate of the standard type of single-valve compound non-condensing engine ranges from 22 to 27 lb. per i.hp-hr. at rated load. Since this type of engine permits of only a moderate amount of superheat the water rate with superheated steam is seldom less than 20 lb. per i.hp-hr. Condensing under a standard vacuum of 26 inches reduces the water rate approximately 20 per cent.

The four-valve compound non-condensing engine has a full load water rate, with saturated steam, ranging from 17 to 22 lb. per i.hp-hr., and with superheated steam an economy as low as 12 lb. per i.hp-hr. has been recorded. Rankine cycle efficiencies as high as 83 per cent have been realized for both saturated and superheated steam.

So much depends upon the initial pressure, degree of vacuum and initial temperature that general figures for condensing practice are without purpose. A few special cases are listed in Tables 81 and 86. With saturated steam the best performances are in the neighborhood of 75 per cent of the theoretical Rankine cycle efficiency, while with highly superheated steam 90 per cent of the Rankine cycle efficiency has been realized.

A number of exceptional performances are illustrated in Figs. 218 to 221.

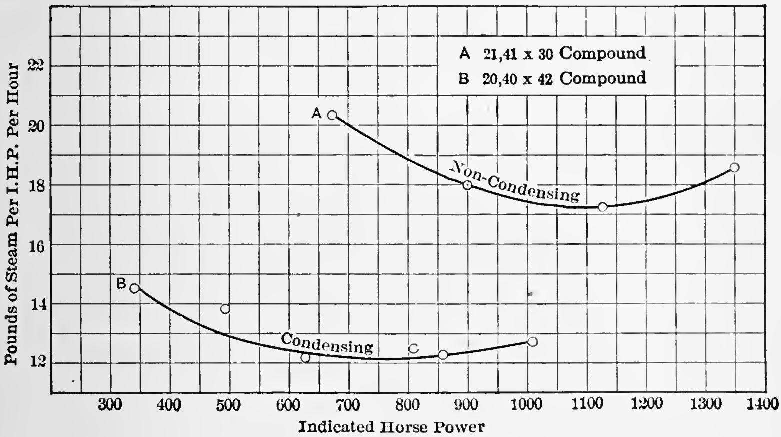


FIG. 218.

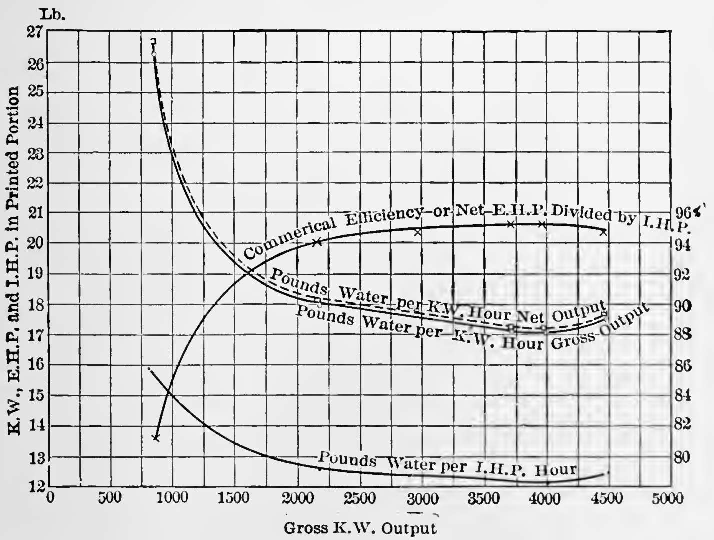


FIG. 219. Economy Test of the 5500-horsepower Three-cylinder Compound Engine and Generator.

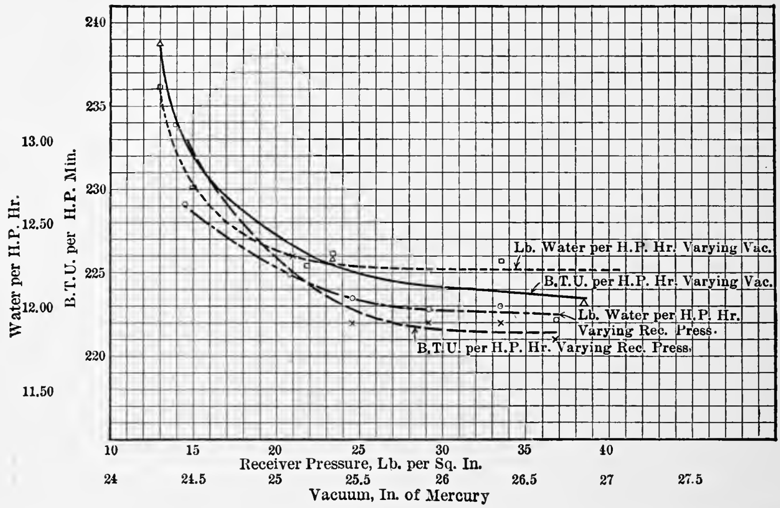


FIG. 220. Performance of 5500-horsepower Engine under Variable Receiving Pressures and at Different Vacua.

193. Triple and Quadruple Expansion Engines. — Triple and quadruple expansion engines are still in use where the load is practically constant, as in marine and pumping-station practice, but have been abandoned in street-railway work where the load fluctuates widely in favor of the steam turbine or the two- or three-cylinder compound. Some idea of the economy effected by triple-expansion pumping engines may be gained from Table 79. A 1000-hp. Nordberg quadruple expansion engine driving an air compressor at the power plant of the Champion Copper Co. is credited with a water rate of 11.23 lb. of saturated steam per i.hp-hr., initial pressure 257 lb. gauge. This engine operates in the "regenerative cycle" (see paragraph 460), and the steam consumption is equivalent to 169.3 B.t.u. per i.hp. per minute and the actual thermal efficiency 25.05 per cent.*

194. The Locomobile. — Although classified under "steam engines" the term "locomobile" applies to the complete power plant and not to the engine only. In Europe this type of plant has been developed to a high degree of efficiency, and with very high superheat steam consumptions as low as 6.95 lb. per i.hp-hr. have been recorded, corresponding to a coal consumption of 0.75 lb. coal per brake hp-hr. The American type of locomobile is not designed for superheat above 250 deg. fahr. and the best economies are in the neighborhood of 1 lb. of coal per brake hp-hr.

* Trans. A.S.M.E., vol. 28, p. 221.

TABLE 79.

ECONOMY OF MODERN VERTICAL TRIPLE-EXPANSION PUMPING ENGINES.

(Official Trials.)

Date of Test.	Type.	Location.	Rated Capacity, Millions of U.S. Gallons.	Initial Gauge Pressure.	Duty.		Dry Steam per I.h.p. Hour.
					Per Thousand Lb. of Dry Steam.	Per One Million B.t.u.	
5- 2-09	Holly	Louisville, Ky.......	24	155.1	*195.0	164.5	*9.64
3-10-10	Holly	Frankfort, Pa.......	20	180.2	184.4	......	......
4-29-10	Holly	Albany, N. Y.......	12	153.0	182.1	......	......
10-14-09	Holly	Brockton, Mass.....	6	150.0	170.0	......	......
12- 5-07	Holly	Cleveland, Ohio.....	2.5	149.6	164.6	148.8	11.51
5- 2-00	Allis	Boston, Mass.......	30	185.5	178.5	163.9	10.33
2- 4-06	Allis	St. Louis, Mo.......	20	140.6	181.3	158.8	10.66
2-26-00	Allis	St. Louis, Mo.......	15	126.2	179.4	158.1	10.67
1-15-10	Allis	Milwaukee, Wis.....	12	124.6	175.4	151.0	10.82

* 109 degrees F. superheat at throttle.

Date of Test.	Type.	R.P.M.	Water Actually Pumped, Millions of U.S. Gallons 24 Hr.	Net Head Pumped Against, Lb. per Sq. In.	Indicated Horse Power.	Developed Horse Power.	Thermal Efficiency Per Cent. I.h.p.
5- 2-09	Holly	24.0	24.111	90.0	925.7	879.4	22.54
3-10-10	Holly	20.1	21.219	95.7	.....	817.0	.....
4-29-10	Holly	22.3	12.193	139.5	.....	726.0	.....
10-14-09	Holly	40.1	6.316	130.6	.....	334.0	.....
12- 5-07	Holly	62.3	2.142	180.7	158.7	151.9	19.13
5- 2-00	Allis	17.7	30.314	61.0	801.5	747.8	21.63
2- 4-06	Allis	16.5	20.070	104.0	859.2	839.6	20.92
2-26-00	Allis	16.4	15.121	127.0	801.6	726.3	21.00
1-15-10	Allis	20.4	12.430	121.0	673.0	618.0	20.25

Fig. 221 shows a longitudinal section through a Buckeye-mobile, illustrating a well-known American design of locomobile. The entire plant is self-contained and requires very little floor space. The engine, of the compound center crank type, is set upon the boiler with cylinders projecting into the "smoke-box" so as to minimize piping and radiation losses. Steam is generated in an internally fired tubular boiler at a pressure of 225–275 lb. per sq. in. gauge and is superheated to a total temperature of 600–700 deg. fahr. Exhaust steam from the high-pressure cylinder is reheated by an auxiliary superheater (adjoining the main superheater) before it enters the low-pressure cylinder. The feed water is heated by an economizer or reheater placed in the breeching. The condenser is of the jet type and is provided with a rotary air pump.

Surface condensers are installed where conditions necessitate this type. All auxiliaries are driven by the main engine. Buckeye-mobiles are made in nine sizes ranging from 75 to 600 horsepower, rated capacity,

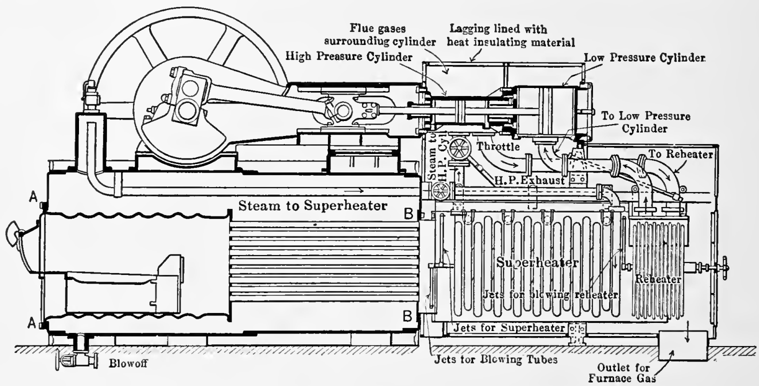


FIG. 221. Longitudinal Section through a Buckeye-mobile.

for belt drive or gearing. For direct-connected electric service the sizes range from 50 to 400 kilowatts.

These small plants give over all economies reached only by large central stations.

Fig. 222 shows the performance of a 150-hp. Buckeye-mobile under

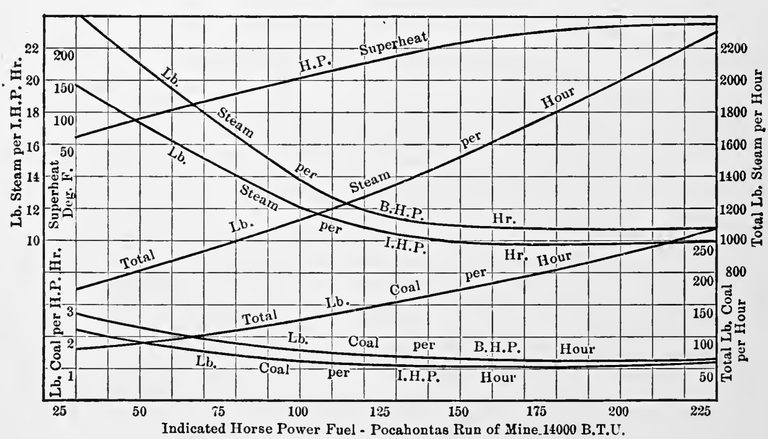


Fig. 222. Economy Test of 150-horsepower Buckeye-mobile.

various load conditions. Initial pressure 220 lb. gauge, vacuum 25 in. referred to 30 in. barometers.

The remarkable economy effected in Europe is shown in Table 80.

195. Rotary Engines. — The rotary engine differs from the reciprocating engine in that the piston, or equivalent, rotates about the

TABLE 80.

A REMARKABLE ENGINE PERFORMANCE.*

200, 400 × 400 mm. Locomobile.

(7.8, 15.7 × 15.7 in.)

Number of Test.	Initial Pressure, Lb. per Sq. In. Abs.	Condenser Pressure, Lb. Abs.	Steam Temperatures, Deg. Fahr.				R.p.m.
			Entering High-pressure Cylinder.	Leaving High-pressure Cylinder.	Entering Low-pressure Cylinder.	Final Feed Water.	
Condensing with Intermediate Superheating.							
1	220	1.47		Saturated.		242	236
2	227	1.17	712	377	462	212	241
3	220	1.17	718	367	460	206	242
4	221	1.17	806	426	530	221	246
5	220	1.17	842	469	538	...	243
6	220	1.17	872	520	...	241	243
Non-condensing without Intermediate Superheating.							
7	220	....	832	....	462	289	237
8	220	....	856	....	505	284	238
9	221	....	878	....	527	284	242
10	220	....	869	....	572	257	241
11	220	....	817	....	525	248	241
12	221	....	878	....	568	259	241

* Compiled from Zeit. des Ver. deut. Ingr., June, 1911.

No. of Test.	I.Hp.	D.Hp.	Mechanical Efficiency, Per Cent.	Steam Consumption, Pounds.		Coal Burned, Lb. per D.Hp-hr.	Heat Consumption B.t.u. per I.Hp. per minute.*
				Per I.Hp-hr.	Per D.Hp-hr.		
Condensing with Intermediate Superheating.							
1	112.5	103.2	91.6	13.98	14.19	1.59	260
2	138.4	132.8	96.0	8.51	8.87	1.00	198
3	140.3	131.4	93.5	8.33	8.90	1.00	195
4	140.4	133.4	95.0	7.68	8.06	0.96	186
5	138.8	132.5	95.5	7.24	7.56	0.87	175
6	141.8	134.0	94.5	7.15	7.56	0.86	175
Non-condensing without Intermediate Superheating.							
7	61.5	49.3	78.0	11.22	14.43	1.65	262
8	83.8	74.0	88.0	10.60	11.84	1.17	249
9	111.0	98.5	88.0	9.95	11.38	1.12	237
10	129.9	120.8	93.0	10.00	10.88	1.07	238
11	140.4	132.2	94.0	10.68	11.34	1.12	248
12	142.1	132.4	93.0	9.93	10.66	1.05	235

* Above ideal feed-water temperature corresponding to exhaust pressure.

cylinder axis. Its operation is entirely different from that of the steam turbine; in the rotary engine the static pressure of the steam actuates the piston and in the turbine the momentum of the steam is imparted to the rotating element.

Over 2200 patents have been issued to date on rotary engines but not a single machine has yet been able to compete with the reciprocating engine as regards steam economy. The advantages of the rotary engine are many and for this reason innumerable inventors have been exerting their skill in the development of this type of prime mover, but unfortunately the impracticability of satisfactorily packing the rubbing surfaces has more than offset the advantages and the commercially successful machine is yet to be found.

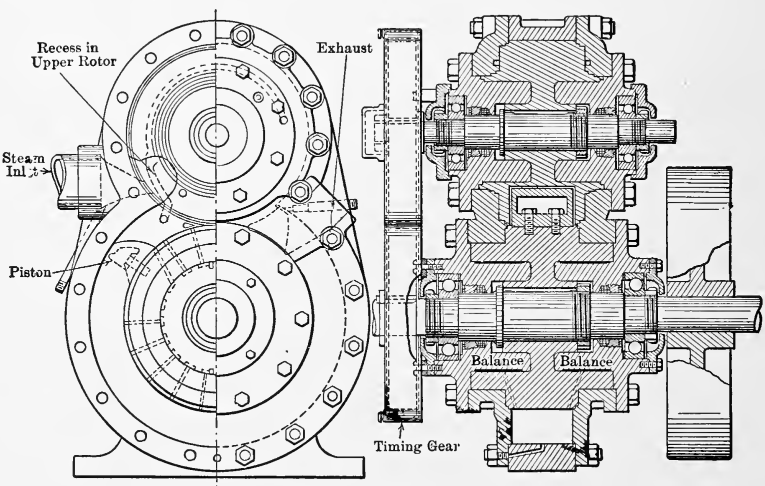


Fig. 223. Herrick Rotary Engine.

The writer has tested out various types of rotary steam engines, and the best has been but a poor competitor of the ordinary grade of reciprocating mechanism.

One of the most successful rotary engines is illustrated in Fig. 223. The device consists essentially of two rotors in rolling contact, the upper one containing a recess which serves as a steam inlet and allows the piston on the lower rotor to pass, while the lower one contains the piston and transmits the power to the shaft. In fundamental principle it is not unlike many other rotary engines in that the power is applied directly to the shaft by the expansion of steam behind a rotary piston. The synchronous movement of the two rotors is maintained by means of two timing gears on the far side of the casing. The curves in Fig. 224 are based upon the tests made by Professor Pryor of Stevens Institute of a 20-horsepower engine of this design, initial pressure 150 pounds gauge, atmospheric exhaust, steam dry and saturated.

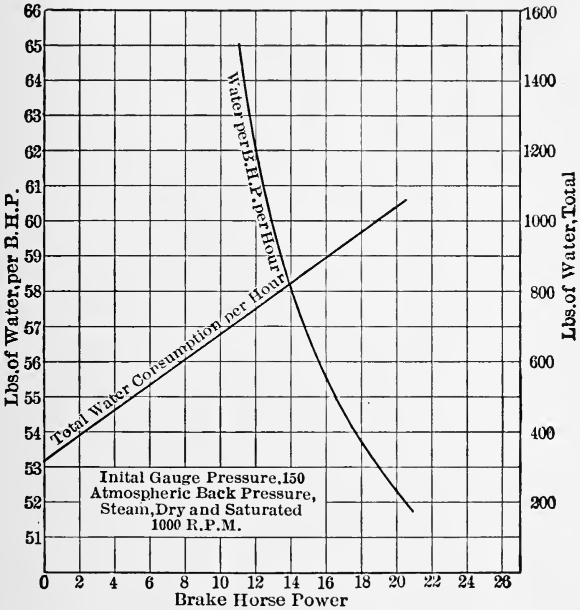


FIG. 224. Performance of Rotary Engine.

196. Throttling vs. Automatic Cut-Off. — The action of the governor in the throttling engine is shown by the superposed indicator cards (Fig. 225) taken between zero or friction load and maximum load. The effect of throttling is to reduce the pressure during admis-

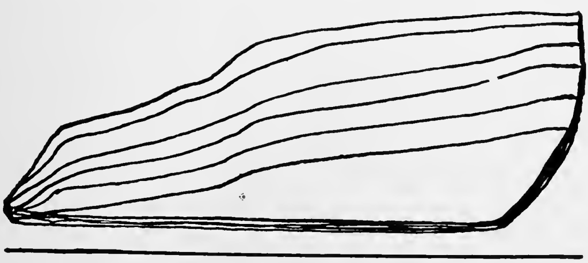


FIG. 225. Typical Indicator Cards. High-speed Throttling Engine.

sion, but does not change the point of cut-off or other events of the stroke. The steam may be partially dried or even superheated by throttling, thus tending to reduce cylinder condensation. Initially dry saturated steam at a pressure of 125 pounds gauge would be superheated about 12 degrees in expanding through a throttle to 90 pounds, or if it contained initially 2 per cent moisture would be perfectly dried in expanding to 40 pounds. Friction through the valve also tends to dry the steam. Thus with very light loads the superheat may be appreciable. The possible gain due to decreased cylinder condensation is to some extent offset by incomplete expansion. The best efficiency for a given load is realized by a proper compromise between cut-off and initial pressure. Experiments made by Professor Denton (Trans. A.S.M.E., 2–150) on a 17-in. by 30-in. non-condensing double-valve engine showed the most economical results with $\frac{1}{4}$ cut-off for 90 pounds pressure, $\frac{1}{3}$ cut-off for 60 pounds, and $\frac{45}{100}$ for 30 pounds. The average throttling engine does not give close regulation, the governor usually lacking sensitiveness. Tests show the economy to be better than that of the automatic engine on light loads, and the crank effort more uniform.

The indicator cards shown in Fig. 226 were taken from a single-valve high-speed automatic engine operating between friction load and maximum load. The mean effective pressure is adjusted to suit the load by the automatic variation in the cut-off, the initial pressure

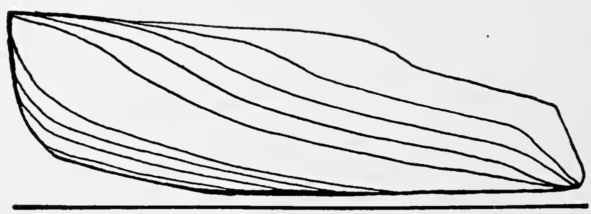


Fig. 226. Typical Indicator Cards. High-speed Automatic Engine.

remaining the same. Since the cut-off is controlled by the action of the governor on the single valve, all other events of the stroke are likewise changed. With a four-valve engine the variation in cut-off does not affect the other events.

The chief advantage of the automatic over the throttling engine lies in its sensitive regulation, and while, in general, it gives a lower steam consumption than the throttling engine, this is probably in most cases due to superior construction and not to the method of governing.

The following performances of a Belliss 250-horsepower high-speed condensing engine fitted with both automatic and throttling governing devices give results decidedly in favor of the throttling engine. (Pro. Inst. of Mech. Engrs., 1897, p. 331.)

	Automatic Cut-Off.				Throttling.			
Percentage of load	100	62.5	33	25	100	62.5	33	25
Electrical horsepower	213	132	77.8	53	213	132	77.8	53
Steam per i.hp-hour	22.5	22.9	28.5	34.3	21	21.7	25.6	28.4

Some of the comparative advantages and disadvantages of the automatic and throttling engines are as follows:

Advantages.

AUTOMATIC.

1. Sensitiveness of regulation.
2. Increased ratio of expansion.
3. Low terminal pressures.

THROTTLING.

1. Low first cost.
2. Crank effort more uniform.
3. Reduced cylinder condensation.
4. Simplicity of regulating device.

Disadvantages.

AUTOMATIC.

1. Increased cylinder condensation.
2. Greater variation in crank effort.
3. Complicated valve gear.
4. Low economy at very early loads.

THROTTLING.

1. Low ratio of expansion.
2. High terminal pressure.
3. Low initial pressure at early loads.

Fig. 227 shows the relative steam consumption of an engine under the same conditions of load when controlled by variable expansion and by throttling. Suppose this engine to be altered in capacity so that the m.e.p. referred to the low-pressure piston is about 32, then the steam consumption with the throttling governor will be as shown by straight line A. This shows that between 32 and 12 pounds m.e.p. very little is gained by a variable expansion, and below that load the throttled governor is the more economical. (Power, Feb. 21, 1911, p. 301.)

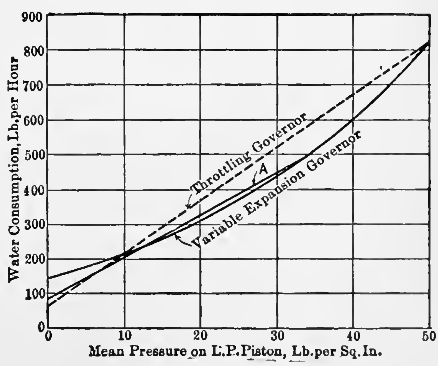


FIG. 227. Throttling *vs.* Automatic Cut-off.

197. Selection of Type. — Modern operating conditions are so diversified and at the same time so specialized that the selection of the type best suited for a proposed installation is an increasingly difficult problem. That engineers are not agreed as to the best practice is evidenced by the different types of engines selected for practically identical operating conditions. General rules are without purpose since each particular installation is a problem in itself. Floor space, capacity, cost of fuel, water rate, steam pressure, water supply, load characteristics, exhaust steam requirements, size of foundation, vibration, first cost, attendance and maintenance all govern the selection of type. The principal factor governing the size of units is the station load curve or rather, load curves. Where these load curves are known the problem is a comparatively simple one, but when they must be assumed as is generally the case with a new project, it is largely a matter of experience.

How to Select Prime Movers for Industrial Electrical Electric Generating Plants: Eng. Mag., Aug., 1916, p. 705.

Economic Selection of Prime Movers: Power, Oct. 12, 1915, p. 511.

198. Cost of Engines. — The cost of engines like any other commodity, varies with the price of raw material, cost of labor, design and grade of workmanship. Even a list of current prices is subject to discount incident to competition. Consequently data of this nature can only be of a very general nature and must be used accordingly. Specific figures should be obtained from builders if accurate comparisons are to be made between the various types for actual installation work. Prices per rated horsepower range from \$4.00 to \$25.00 and per pound from 5 to 15 cents. The more economical engines are generally higher priced. The following rules are based on average costs and should not be used except for rough estimates.

Simple high-speed engine. . . . . $C = 300 + 8 \times$ i.hp.
Compound high-speed engines. . . . . $C = 1000 + 15 \times$ i.hp.
Simple low-speed engines. . . . . $C = 1000 + 10 \times$ i.hp.
Compound low-speed engines. . . . . $C = 2000 + 13 \times$ i.hp.

Other rules given in this connection by different authorities are as follows:

Simple high-speed engines. . . . . $C = 435 + 6.5 \times$ i.hp.
Simple non-condensing Corliss. . . . . $C = 700 + 10 \times$ i.hp.
Compound high-speed Corliss. . . . . $C = 500 + 10.5 \times$ i.hp.
Compound Corliss engines. $C = 1800 + 13.6 \times$ i.hp.

C = cost in dollars, f.o.b. shipping point.
i.hp. = rated indicated horsepower.
Cost of setting high-speed engine $= 60 + 0.75 \times$ i.hp.
Cost of setting low-speed engine $= 500 + 1.3 \times$ i.hp.

TABLE 81. EXAMPLES OF STEAM-ENGINE ECONOMY. SIMPLE ENGINES, DOUBLE FLOW, SATURATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions, Inches.	Indicated Horsepower.	Initial Pressure, Lb. Gauge.	Back Pressure, Lb. Per Sq. In. Abs.	M.E.P.	R.P.M.	Lb. Dry Steam Per I.Hp-hr.	B.T.U. Per I.Hp. Per Min.	Thermal Efficiency, Per Cent.	Rankine Cycle Ratio, Per Cent.	Mechanical Efficiency, Per Cent.
					Single-valve, Non-condensing.								
1	Willans	Peabody, Thermodynamics	14 x 6	33.6	122.0	Atmos.	....	400	26.0	439	9.7	60.0	.....
2	Willans	Peabody, Thermodynamics	14 x 6	16.5	36.3	Atmos.	....	393	42.8	709	6.0	65.5	.....
3	Locomotive Purdue	Trans. A.S.M.E., Vol. 14, p. 826	17 x 14	399.0	110.0	Atmos.	54.0	136	24.97	421	10.1	65.7	.....
4	Westinghouse Standard	Shop test	20 x 16	257.0	100.0	Atmos.	36.0	275	26.19	440	9.6	65.2	.....
5	Buffalo Forge	Elec. World, Sept. 1, 1904, p. 407	12 x 12	121.0	124.0	Atmos.	58.5	302	27.5	464	9.2	57.5	.....
6	Reeves	Elec. World, Oct. 1, 1904, p. 587	15 x 14	120.0	114.0	Atmos.	35.0	275	28.0	472	9.0	55.4	92.5
7	Ames	Eng. Record, July 6, 1901, p. 7	13 x 12	105.0	95.0	Atmos.	52.0	250	29.1	489	8.7	60.0	.....
8	Ames	Meyer, Steam Power Plants, p. 56	17 x 16	248.0	100.0	Atmos.	50.0	270	26.0	437	9.7	65.9	.....
9	Locomotive, No. 1499, Penn. System	Locomotive Tests, 1904, at Louisiana Exposition	22 x 28	975.0	196.0	Atmos.	75.6	120	23.4	397	10.7	55.6	91.0
10	Buffalo Forge	Elec. World, Sept., 1904, p. 407	12 x 12	74.0	79.3	Atmos.	35.6	304	30.6	512	8.3	62.1	95.7
					Single-valve, Condensing.								
11	Willans	Peabody, Thermodynamics	14 x 6	33.2	70.0	1.0	....	383.0	22.2	413	10.3	40.9	.....
12	Reeves	Elec. World, Oct. 1, 1904, p. 587	15 x 14	140.0	114.0	3.2	41.0	275.0	26.0	466	9.1	39.6	95.8
13	Buffalo Forge	Elec. World, Sept. 10, 1904, p. 407	12 x 12	86.0	80.0	3.0	40.5	310.0	27.5	494	8.6	40.4	95.0
14	Piston Valve	Barrus, Engine Tests, p. 95	$18\frac{1}{2}$ x 30	204.6	69.3	2.6	38.1	29.3	27.15	485	8.7	40.9	.....
					Four-valve Non-Condensing								
15	Corliss, Jacketed	Peabody, Thermodynamics	21.6 x 43.3	237.0	103.5	Atmos.	42.1	62.7	21.5	361	11.7	79.5	.....
16	Fleming	Prof. Carpenter, June 28, 1905, at Cornell University	19 x 19.0	217.0	120.5	Atmos.	39.0	205.9	22.46	378	11.2	70.4	.....
17	Fleming	Prof. Spangler, June 6, 1905, at University of Pennsylvania	16 x 16.0	132.0	125.4	Atmos.	39.1	210.0	22.24	375	11.3	70.2	95.0
18	Fitchburg-Prosser	Prof. E. F. Miller, 1916	15 x 24	48.7	121.6	Atmos.	....	82.0	19.1	317	13.4	82.6	.....
19	Ideal Corliss	Power, Mar. 4, 1913	16 x 22	240.0	150.0	Atmos.	59.0	200.0	20.7	350	12.1	70.2	.....
20	Nordberg Poppet Valve	Shop test, 1916	15 x 18	123.0	130.0	Atmos.	38.0	200.0	18.8	318	13.4	80.0	90.8
					Four-valve, Condensing.								
21	Corliss, Jacketed	Peabody, Thermodynamics	21.6 x 43.3	155.0	103.8	1.2	32.0	60.0	16.5	307	13.8	52.8	.....
22	Poppet Valves, Jacketed	Zeit. d. V. D. Ing., Aug., 1905, p. 1310	22.6 x 45.0	262.0	79.0	1.36	30.2	47.6	15.0	276	15.4	62.8	.....
23	Gridiron Valves	Barrus, Engine Tests, p. 101	$34\frac{1}{2}$ x 60.0	613.0	82.3	1.0	37.2	60.0	18.5	344	12.3	46.0	.....
24	Corliss	Barrus, Engine Tests, p. 118	32 x 60.0	554.0	67.5	2.9	38.2	59.1	19.45	348	12.1	58.8	.....
25	Slide Valves	Barrus, Engine Tests, p. 88	18 x 30.0	213.0	67.0	2.2	33.7	165.0	22.0	400	10.6	48.6	.....
26	Corliss	Peabody, Thermodynamics	18 x 48.0	145.0	96.0	1.5	30.9	76.0	19.4	358	11.9	47.7	.....

TABLE 82.—EXAMPLES OF STEAM-ENGINE ECONOMY.

MULTIPLE EXPANSION ENGINES, SATURATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions.	Cylinder Ratio.	Horse Power.	Initial Press., Lbs. Gauge.	Back Press., Lbs. Abs.	R.P.M.	M.E.P. Referred to L.P. Cyl.	Temp. Feed Water, Deg. F.	Lbs. of Steam per I.H.P. Hr.	B.T.U. per I.H.P. per Min.	Thermal Efficiency, per Cent.	B.T.U. per I.H.P. Min., Perfect Eng.	Efficiency Ratio, per Cent.	Mech. Efficiency, per Cent.	Lbs. of Coal per D.H.P. Hr.
	Quadruple Expansion.																
1	Nordburg Pumping Engine, Wildwood, Pa.	Eng. News, May 4, 1899, p. 280.	19½, 29, 49½, 57½ x 42		712	200	0.9	36.5	35.5	310.8	12.26	186	22.8	138	74.2	93.0	1.12
	Triple Expansion.																
2	Allis Pumping Engine, Chestnut Hill, Boston.	Eng. News, Aug. 23, 1900, p. 125.	30, 56, 87 x 66	1:3½:8.4	801	185	0.85	17.2	23.4	155	10.33	196	21.63	138	70.5	*93.3	1.09
3	Allis Pumping Engine, Bissell's Point, St. Louis.	Power, May 1906, p. 299........	34, 62, 94 x 72	1:3.3:7.6	865	140	1.2	16.5	20.8		10.59	201.4	21.06	151	75.0	*97.4	
4	Holly Pumping Engine, Spot Pond, Boston.	Eng. News, Nov. 14, 1901, p. 371.	22, 41, 62 x 60	1:3½:8	464	150	1.05	24.8	20.5	157.9	11.01	203.4	20.85	142	70.0	*96.5	1.11
5	Sulzer Mill Engine, Augsburg.	Zeit. d. V.D.I., May 16, 1896, p. 534.	29.9, 44.5, 2(51.6) x 78.7	1:2.2:5.9	1823	134	1.8	56.2	19.5	122	11.33	208	20.40	158	76.0		1.19
	Compound, Condensing.																
6	Allis-Chalmers Engines, New York Subway.	Power, Feb., 1906, p. 115.	2(42), 2(86) x 60	1:4.2	7365	175	2.2	75	27.9	130	11.96	220	19.2	159	72.4	†93.0	
7	Cross-Compound Corliss, Atlantic Mills, Providence.	Am. Elecn., June, 1903, p. 260.	16, 40 x 48	1:6¼	500	170	0.8	80	20.5		11.20	222	19.0	141	63.5		1.20
8	Leavitt Pumping Engine, Louisville, Ky.	Trans. A.S.M.E., Vol. 16, p. 169.	27, 54 x 120	1:4	643	137	0.95	18.6	24.9	100	12.20	222	19.0	150	67.6	93.0	
9	Rice & Sargent Corliss, Amer. Sugar Refinery, Brooklyn.	Trans. A.S.M.E., Vol. 24, p. 1274.	20, 40 x 42	1:4	627	151	0.85	121	19.4	121	12.10	222.7	19.0	143	64.3		
10	Fleming Four-Valve.	Trans. A.S.M.E., Vol. 25, p. 212.	15, 40⅓ x 27	1:7.3	348	150	2.0	152	13.0	126	12.33	225.8	18.7	162	71.7		
11	Williams Vertical, New York Navy Yard.	Power, Oct., 1903, p. 583.......	19, 34 x 30	1:3¼	340	100	2.0	150	16.5	126	12.60	229	18.5	175	76.5		
12	Tandem-Compound Corliss. ...	Barrus, Eng. Tests, p. 185......	18, 44 x 72	1:6.4	6893	145.2	1.5	60.3	20.6	116	12.70	234	18.1	157	67.0		
13	Edison Waterside Sta., N. Y...	Power, July 1904, p. 424.......	43½, 75.3 x 60	1:6.02	5442	185	1.5	76.3	26.5	116	11.93	221	19.2	150	68.0	†95.2	
	Compound, Non-Condensing.																
14	Ball & Wood Co. Corliss, W. Albany Sta., N. Y. C. & H. R. R.	Test by Company Engineers. ..	21, 41 x 30	1:3.8	1125	175	Atmos.	120	47.0	Assumed for Comparison, 212° F.	17.17	291	14.5	229	78.8		
15	Willans.	Peabody, Thermodynamics. ...	10, 14 x 6	1:2	39.6	165.0	Atmos.	401	42.4		19.2	328	12.8	234	71.5		
16	Willans.	Peabody, Thermodynamics. ...	10, 14 x 6	1:2	33.0	113.9	15.7	402			21.4	358	12.4	297	83.0		
17	Ball Engines, Chicago Public Library.	Eng. Record, Aug. 6, 1898, p. 206.	12, 20 x 13	1:2.8	187.5	166.8	Atmos.	271	34.8		21.14	357	11.7	233	65.5	93.5	
18	Westinghouse Marine..	Power, Aug., 1903.	17, 27 x 24	1:2¼	540	148	Atmos.	210	37.1		19.3	326	13.0	244	75.0	†87.5	
19	Skinner Cross-Compound......	Power, July, 1906.	16, 27 x 18	1:2.84	375	130	Atmos.	226	31.0		21.14	355	11.9	255	72.0	91.5	
20	Buffalo Tandem-Compound. ..	Elec. World, May 23, 1903, p. 897.	12, 18 x 10	1:2¼	121	128	Atmos.	271	35.0		22.3	376	11.2	258	68.5	93.0	
21	Reeves Vert. Cross-Compound.	Eng. Record, July 1, 1905, p. 24.	12, 20 x 14	1:2.77	185	150	Atmos.		56.0		20.9	354	11.9	242	68.5	92.0	
22	Cross-Comp'd, 4 Slide Valves.	Barrus, Eng. Tests, p. 181......	17½, 28 x 48	1:2.58	486.7	1[illegible]5.3	Atmos.	99	32.9		21.59	362	11.7	260	71.9		
23	4-Cylinder Compound Locomotive, No. 2512 Penn. System.	Tests made at Louisiana Exposition, 1904.	14.2, 23.7 x 25.2	1:2.78	495	210	Atmos.	80	55.0		18.6	316	13.4	216	68.5	91.0	

TABLE 83. EXAMPLES OF STEAM-ENGINE ECONOMY. SUPERHEATED STEAM.

Index.	Kind of Engine.	References.	Cylinder Dimensions, Inches.	Cylinder Ratio.	Horsepower.	Initial Pressure, Lb. Gauge.	Back Pressure, In. Abs.	R.P.M.	Lb. Steam Per I.Hp-hr.	B.T.U. Per I.Hp. Per Min.	Thermal Efficiency, Per Cent.	Rankine Cycle Ratio, Per Cent.	Superheat, Deg. Fahr.	Steam Temp., Deg. Fahr.
			Triple Expansion.											
1	Binary Vapor Eng., Royal High School, Berlin.	Jour. Franklin Inst., Dec., 1902, p. 456.	....	....	211	143	4.5	143.5	8.60	158.3	26.8	....	221	590
2	Sulzer, Four-cylinder	Eng. News, Oct. 2, 1902, p. 259	32, 47, 58 x 59	1: 2.03: 3.27	2860	173	2 0	85	8.97	187.7	22.6	73.5	230	606
3	Sulzer, Three-cylinder	Zeit. d. V. D. I., Aug., 1905, p. 1353	15.5, 25.4, 37.5 x 25.6	1: 2.64: 5.9	549	166	2.4	144.4	10.00	207.3	20.4	68.5	229	602
4	Sulzer, Three-cylinder	Engr., Lond., May 25, 1900, p. 546	34, 46, 61 x 51	1: 2.08: 3: 22	2940	167	1.6	82.5	9.58	204.0	20.8	66.7	264	637
5	Worthington Pumping Eng., Central Park, Chicago.	Eng. News, May 26, 1904, p. 487	....	....	646	146.8	1.6	18.6	10.00	196.6	21.6	72.5	87	451
6	Riedler Pumping Engine, Chicago Ave. Sta., Chicago.	Engr. U. S., Nov. 15, 1907, p. 1092.	15, 29, 48 x 48	....	590	170	2.6	62	9.73	196.5	21.6	69.3	166	542
			Compound.											
7	Cole, Marchent and Morley, Cross-comp., Jacketed.	Engr., London, June, 1905, p. 546	21, 36 x 36	1: 2.94	145.5	114.5	1.72	100.7	8.58	176.1	24.0	82.0	202	548
8	Van den Kerchove, Tandem, Heads Jacketed.	Amer. Elecn., May, 1903, p. 217	12.8, 22 x 33.4	1: 2.97	212	131	2.2	127	8.99	194.8	21.7	73.0	342	699
9	Van den Kerchove, Tandem, Heads Jacketed.	Amer. Elecn., May, 1903, p. 217	12.8, 22 x 33.4	1: 2.97	217	129.5	2.2	127	10.75	218.2	19.4	69.1	183	539
10	Easton & Co., Tandem-compound.	Amer. Elecn., Apr., 1903, p. 178	15, 24 x 48	1: 2.67	239	120	2.0	140	9.00	187.0	22.6	78.5	240	590
11	Rice and Sargent, Melbourne Mills, Pa.	Trans. A.S.M.E., Vol. 25, p. 278	16, 28 x 42	1: 3.06	920	142	4.0	102	9.56	188.3	22.5	72.2	296	658
12	McIntosh and Seymour, Edison Co., So. Boston.	Trans. A.S.M.E., Vol. 25, p. 491	29, 60 x 56	1: 4.3	2202	158	4.8	98	11.21	209	20.2	79.8	92	462
13	Cross-compound, Cylinders Jacketed.	Barrus, Eng. Test, p. 202	18, 48 x 48	1: 7.3	659	143	3.4	80	11.89	223.5	19.0	73.5	40	402
14	Sulzer, Tandem-compound	Eng. News, Oct. 2, 1902, p. 259	26.8, 47.2 x 67	1: 3.1	788	116	2.8	65	9.68	207.2	20.3	71.5	343	690
15	White, Auto. Engine	Trans. A.S.M.E., Vol. 28	3, 6 x 4.5	1: 4.0	40	426	Atmos.	850	11.96	244.0	17.4	68.0	316	766
16	Nordberg Cross-compound	U. S. Metal Refining Co.	19, 44 x 42	1: 5.4	620	155	3.87	100	11.01	212.0	20.0	75.3	76.1	444
			Simple.											
17	Poppet-valve, Condensing	Zeit. d. V. D. I., Aug., 1905, p. 1310.	16.3 x 39.4	....	123	145	1.4	81.1	16.70	326.2	13.0	46.0	73.8	424
18	Poppet-valve, Condensing	Zeit. d. V. D. I., Aug., 1905, p. 1310.	16.3 x 39.4	....	20	145	1.5	81.2	14.70	307.4	13.8	47.6	226.2	576
19	Poppet-valve, Non-condensing	Zeit. d. V. D. I., Aug., 1905, p. 1310.	16.3 x 39.4	....	123	145	Atmos.	81.5	16.10	307.5	13.8	79.0	254.3	604
20	Ideal Corliss	Power, Mar. 4, 1913	16 x 22	....	190	125	Atmos.	200	18.3	328	13.0	78.5	107	459
21	Erie City Lentz	F. W. Dean, 1913	19 x 21	....	248	133	Atmos.	206	16.1	286	14.8	87.5	92.7	450

TABLE 84.

STEAM AND COAL SAVING IN A SIMPLE NON-CONDENSING ENGINE OF 250 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 12 atm. = 177 lb.; temp. of sat. steam 369 deg. F.; cut-off 20 per cent; piston speed 10 ft. per sec.; slide or piston valve; change of cut-off effected by valve gear.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant. I.H.P. Variable.							Cut-Off Variable. I.H.P. Constant.								Remarks.
			I.H.P.	At Engine.			At Boiler.			I.H.P.	At Engine.			At Boiler.			Cut-off, per Cent.	
				Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat Steam. Steam, per Cent.	Saving over Sat Steam. Fuel, per Cent.	Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.		Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam. Steam, per Cent.	Saving over Sat. Steam. Fuel, per Cent.	Steam Consump. per I.H.P. Hour, Lb.	Saving over Sat. Steam. Feed Water, per Cent.	Saving over Sat. Steam. Fuel, per Cent.		
	Deg. F.	Deg. F.																
None..........	0	369	250	27.00	......	......	29.00	......	....	250	27.00	......	......	29.00	......	......	20.00	Indirect; superheater in boiler
75 to 150° F. Low:																		
Indirect.....	103	472	235	21.00	22.0	18.5	21.40	27	23	250	21.05	22.0	18.5	21.47	26.0	22.5	20.88	Direct; superheater separately fired
Direct.......	103	472	235	21.00	22.0	10.0	21.40	27	14	250	21.05	22.0	9.5	21.47	26.0	14.0	20.85	
150 to 225° F. Medium:																		
Indirect.....	162	531	230	19.85	26.5	20.5	20.10	30	27	250	20.00	25.5	20.0	20.40	29.5	24.0	21.35	
Direct.......	162	531	230	19.85	26.5	21.5	20.10	30	19	250	20.00	25.5	11.0	20.40	29.5	15.5	21.35	
225 to 290° F. High:																		
Indirect.....	234	603	222	18.50	31.0	23.0	18.85	35	31	250	18.75	30.5	22.0	19.13	34.0	26.0	21.75	
Direct.......	234	603	222	18.50	31.0	15.0	18.85	35	22	250	18.75	30.5	13.5	19.13	34.0	18.0	21.75	

TABLE 85.

STEAM AND COAL SAVING IN A COMPOUND ENGINE, CONDENSING, OF 250 I.H.P. WITH SUPERHEATED STEAM AT DIFFERENT TEMPERATURES.

Press. 10 atm. = 142.23 lbs.; temp. of sat. steam 354 deg. F.; cut-off 6 per cent; piston speed 10 ft. per sec.; automatic cut-off; 4 poppet or piston valves per cyl.

Kind of Superheat.	Degrees of Superheat.	Temperature of Steam.	Cut-off Constant. I.H.P. Variable. I.H.P.	At Engine. Steam Consump. per I.H.P. Hour, Lb.	At Engine. Saving over Sat. Steam. Steam, per Cent.	At Engine. Saving over Sat. Steam. Fuel, per Cent.	At Boiler. Feed Water per I.H.P. Hour, Lbs.	At Boiler. Saving over Sat. Steam. Feed Water, per Cent.	At Boiler. Saving over Sat. Steam. Fuel, per Cent.	Cut-off Variable. I.H.P. Constant. I.H.P.	Variation of Cut-off, per Cent.	At Engine. Steam Consump. per I.H.P. Hour, Lbs.	At Engine. Saving over Sat. Steam. Steam, per Cent.	At Engine. Saving over Sat. Steam. Fuel, per Cent.	At Boiler. Feed Water, per. I.H.P. Hour, Lbs.	At Boiler. Saving over Sat. Steam. Feed Water, per Cent.	At Boiler. Saving over Sat. Steam. Fuel, per Cent.	Remarks.
	Deg. F.	Deg. F.																
None	0	354	250	14.72	......	......	15.73	......	....	250	6.00	14.72	......	......	15.73	......	......	Indirect; superheater in boiler
75 to 150 deg. F.																		
Low: Indirect	130	484	225	12.25	15.0	11.0	12.50	21	17	250	6.95	12.50	15.0	11.5	12.70	20.0	16.0	Direct; superheater separately fired
Direct	130	484	225	12.25	15.0	1.5	12.50	21	8	250	6.95	12.50	15.0	1.5	12.70	20.0	7.0	
150 to 225 deg. F.																		
Medium: Indirect ..	202	556	215	11.60	21.0	15.0	11.81	25	21	250	7.50	11.75	20.0	14.0	11.98	24.5	18.5	
Direct	202	556	215	11.60	21.0	5.5	11.81	25	12	250	7.50	11.75	20.0	4.5	11.98	24.5	9.5	
Double: (1) Indirect.																		
(2) Direct ..	202	556	205	10.70	27.5	18.0	10.93	31	24	250	8.00	10.85	26.0	17.0	11.09	30.0	21.0	
(1) Direct ..																		
(2) Indirect.	202	556	205	10.70	27.5	16.5	10.93	31	23	250	8.00	10.85	26.0	15.0	11.09	30.0	19.5	
225 to 290 deg. F.																		
High: Indirect.....	274	628	205	10.70	27.5	19.0	10.93	31	27	250	8.00	10.85	26.0	17.5	11.09	30.0	21.5	
Direct.......	274	628	205	10.70	27.5	10.0	10.93	31	17	250	8.00	10.85	25.0	8.5	11.09	30.0	13.0	
Double: (1) Indirect																		
(2) Direct .	274	628	198	10.27	30.0	18.5	10.50	34	26	250	8.50	10.51	28.5	16.0	10.75	32.0	20.5	
(1) Direct .																		
(2) Indirect	274	628	198	10.27	30.0	17.0	10.50	34	25	250	8.50	10.51	28.5	15.0	10.75	32.0	19.0	
290 to 360 deg. F.																		
Very high: Indirect	338	692	198	10.27	30.0	20.5	10.50	34	28	250	8.50	10.51	28.5	18.5	10.75	32.0	22.5	
Direct .	338	692	198	10.27	30.0	11.5	10.50	34	20	250	8.50	10.51	28.5	9.0	10.75	32.0	13.5	

TABLE 86.

WATER RATES OF PISTON ENGINES AT VARIABLE LOADS. SATURATED STEAM.*

Indicated Horse-power.	Pounds Steam per Indicated Horsepower Hour at			
	Full Load.	Three-quarter Load.	One-half Load.	One-quarter Load.
A. Automatic Single-cylinder Non-condensing. Initial Pressure 125-Lb. Gauge: Cut-off ¼.				
80	29.42	29.93	31.66	37.08
100	28.96	29.40	31.04	36.00
125	28.47	28.84	30.42	35.10
150	28.12	28.46	29.95	34.38
200	27.51	27.81	29.25	33.26
300	26.64	26.75	27.97	31.68
B. Medium-speed Four-valve Non-condensing. Initial Pressure 130-lb. Gauge: Cut-off ¼.				
200	23.45	23.05	24.75	35.00
350	23.03	22.54	24.07	33.79
500	22.61	22.06	23.45	32.73
650	22.24	21.67	22.92	31.71
800	22.00	21.40	22.57	30.91
900	21.90	21.31	22.44	30.48
C. Automatic Tandem-compound Non-condensing. Initial Pressure 140-lb. Gauge: Cylinder Ratio 4 to 1: Cut-off ¼.				
100	24.04	25.06	29.54	43.84
150	22.94	23.82	28.00	41.40
200	22.36	23.19	27.19	40.46
250	21.98	22.75	26.65	39.13
350	21.48	22.18	25.96	38.01
450	21.27	21.92	25.61	37.45
D. Same as *C* but Condensing. Vacuum 26-in.				
150	20.25	21.51	26.00	37.83
300	19.10	20.12	24.10	33.90
400	18.55	19.45	23.11	32.07
500	18.15	18.93	22.44	30.90
600	17.92	18.67	22.08	30.20
700	17.83	18.55	21.91	29.95
E. Four-valve Medium-speed Compound Non-condensing. Initial Pressure 150-lb. Gauge: Cylinder Ratio 4 to 1: Cut-off ¼.				
300	19.71	21.74	26.82	39.54
450	19.20	21.10	25.90	37.80
600	18.91	20.66	25.20	35.46
750	18.74	20.41	24.73	35.31
850	18.68	20.32	24.55	34.81
950	18.66	20.30	24.48	34.47
F. Same as *E* but Condensing. Vacuum 26-in.				
300	15.42	15.30	17.26	24.00
500	14.74	14.60	16.45	22.47
700	14.29	14.10	15.78	21.30
900	13.97	13.76	15.34	20.48
1100	13.73	13.51	15.02	19.94
1300	13.56	13.33	14.83	19.66
1500	13.49	13.23	14.71	19.52

* Guaranteed performance of a well-known line of high-grade piston engines.

PROBLEMS

1. A 40-hp. non-condensing piston engine uses 500 lb. of saturated steam per hour when running idle and 1600 lb. per hour when operating at full load; initial pressure 115 lb. abs. Draw the unit water rate curve assuming that the total water rate follows the "Willans" straight-line law.

2. A 15-inch by 18-inch poppet-valve engine uses 18.8 lb. steam per i.hp-hr. at rated load, initial pressure 145 lb. absolute; back pressure 0 lb. gauge; initial quality 99 per cent; release pressure 4 lb. gauge; mechanical efficiency at rated load 91 per cent. Required (on both i.hp. and br.hp. basis):

a. Heat consumption per hp-hr.
b. Thermal efficiency, per cent.
c. Rankine cycle ratio, per cent.
d. Cylinder efficiency, per cent.

3. The Rankine cycle ratio of a compound poppet-valve engine is 90 per cent at full load; initial pressure, 150 lb. abs.; temperature of steam at admission, 450 deg. fahr.; back pressure 16.1 lb. abs. Calculate the full load water rate, lb. per i.hp-hr.

4. If the exhaust from the engine in Problem 3 is used for heating purposes, required the full load water rate, lb. per i.hp-hr. chargeable to power.

5. A simple engine indicates 160 horsepower on a dry steam consumption of 31 lb. per i.hp-hr.; initial pressure 130 lb. abs., back pressure 0 lb. gauge. By shortening the cut-off, and by reducing the back pressure to 4-inch mercury (referred to a 30-inch barometer) the water rate is reduced to 22 lb. per i.hp-hr., the load remaining the same. If the condensing equipment requires 10 per cent of the steam supplied to the engine for its operation, required the net gain or loss in heat consumption per i.hp-hr. due to condensing.

6. Which is the more economical from a heat consumption standpoint, a simple non-condensing engine using 26 lb. dry steam per i.hp-hr., initial pressure 100 lb. absolute, or a compound condensing engine using 12 lb. steam per i.hp-hr., initial pressure 290 lb. abs., superheat 350 deg. fahr., back pressure 2-inches mercury? Which is the more perfect of the two?

See also Problems at end of Chapters XXII–XXIV.

CHAPTER X

STEAM TURBINES

199. Classification. — The development of the steam turbine during the past decade has been truly remarkable. So rapid has been the growth that many turbines representative of the best practice four years ago are virtually obsolete to-day. Because of the almost radical changes from year to year it is practically impossible to keep the descriptive features of a textbook strictly in accord with current practice, and the subject matter must necessarily be of a general nature.

Steam turbines are now being used for driving alternating-current generators, turbo-compressors, pumps, blowers and marine propellers, and, by means of gearing to furnish power for reciprocating air compressors, rolling mills and other classes of slow-speed machinery. Although the reciprocating engine will probably continue to be an important factor in the power world for years to come, its field of usefulness is being gradually limited by the steam turbine. The steam turbine has found favor chiefly on account of its low first cost, low maintenance cost, small floor-space requirements and low cost of attendance.

A general classification of steam turbines is unsatisfactory because of the overlapping of the various groups, and the following chart is offered merely as a guide in arranging a few well-known turbines according to the fundamental principles involved in their operation.

Steam Turbines	Impulse	Single Velocity.	De Laval.	Single-pressure Stage.
		Multi-velocity Stage.	Terry. Sturtevant. Curtis (Small Type). Westinghouse	
		Single-velocity Stage.	Kerr. De Laval. Rateau.	Multi-pressure Stage.
		Multi-velocity Stage.	Curtis.	
	Reaction.	Multi-velocity Stage.	Westinghouse-Parsons. Allis-Chalmers-Parsons.	
	Combined Impulse and Reaction.	Multi-velocity Stage.	Westinghouse	

As shown in the preceding chart, all turbines may be divided into three general classes, (1) impulse, (2) reaction, and (3) combined impulse and reaction, though strictly speaking all turbines depend more or less upon both impulse and reaction for their operation.

Impulse Type:

In the impulse type the steam is expanded by suitable means and the heat given up by the pressure drop imparts velocity to the jet itself. The jet impinges against the vanes of a rotating wheel and gives up its kinetic energy to the wheel. If the entire pressure drop takes place in one set of nozzles and the resulting jet is directed against a single wheel the turbine is classified with the *single-stage single-velocity group.* The velocity of the jet is very high, from 2000 to 4000 feet per second, and for satisfactory economy the peripheral velocity of the wheel must also be very high, from 700 to 1400 feet per second. The De Laval "Class A" turbine is the best-known example of this group.

If the entire pressure drop takes place in a single set of nozzles and a single wheel is to be used at a comparatively low speed satisfactory economy may be effected by *compounding the velocity.* That is, the jet issuing from the nozzle at a very high velocity is reflected back and forth from the vanes on the rotor to a series of fixed reversing buckets until all of the available kinetic energy of the jet has been imparted to the wheel. The Terry single-stage turbine is representative of this group.

Low peripheral velocity and high efficiency may be obtained by *pressure compounding;* that is, expansion takes place in a series of successive nozzles instead of one nozzle. Only a part of the available heat energy is converted into kinetic energy in each set of nozzles. For each set of fixed nozzles there is a corresponding rotor. This type of turbine is to all intents and purposes a series of single-velocity impulse turbines placed side by side. The Kerr turbine is representative of this group.

By *compounding both velocity and pressure* we have the multi-velocity and pressure type of which the Curtis turbine is the best-known example.

Reaction Type:

In the reaction type the conversion of potential to kinetic energy takes place in the moving blades as well as in the fixed blades. Only a very small portion of the heat energy imparts velocity in the first set of fixed blades or nozzles. The jet issuing from this set of nozzles impinges against the first set of moving blades at a velocity substantially that of the moving blades so that it enters them without impulse. The moving blades are proportioned so that partial expansion takes place within them and the resulting increase in velocity exerts a *reaction* upon the moving blades. The expansion is very gradual and a large number of alternately fixed and revolving blades are necessary to effect complete expansion. Because of the small pressure drop in each stage low peripheral velocities are possible with high over-all efficiency. The Westinghouse and Allis-Chalmers designs of the Parsons turbine are the best-known examples of this type.

Combined Impulse and Reaction Type:

In this class the high-pressure elements are of the impulse type and the low-pressure elements of the reaction type. The Westinghouse-Parsons double-flow high-pressure turbine is typical of this class and is virtually a combination of the Curtis and Parsons designs. Several European impulse turbines as recently designed are fitted with reaction blades adjacent to the nozzles, showing the tendency to merge the different fundamental types.

Turbines may be classified according to the service for which they are intended, as

High-pressure non-condensing,
High-pressure condensing,
Low-pressure,
Mixed-pressure,
Bleeder.

Each of these types is discussed later on in the chapter.

Recent Developments in Steam Turbine Practice: Mech. Engr., Jan. 26, 1912.
The Present State of Development of Large Steam Turbines: Jour. A.S.M.E., May, 1912.
The Steam Turbine: Engng., Dec. 29, 1911.
Status of the Small Steam Turbine: Power, Jan. 2, 1912.

200. General Elementary Theory. — A given weight of steam at a given pressure and temperature occupies a certain known volume and contains a known amount of heat energy. If the steam is permitted to expand to a lower pressure without receiving additional heat or giving up heat to surrounding bodies it is capable of doing a certain amount of work which will be the same whether the expansion takes place in the cylinder of a reciprocating piston engine, a rotary piston engine, or the nozzles and blades of a steam turbine.

Let W = weight of steam, lb. per sec.,
E = energy given up by 1 lb. of steam, ft-lb.,
P_1 = initial pressure, lb. per sq. in. abs.,
P_n = final pressure, lb. per sq. in. abs.,
H_1 = initial heat content per lb., B.t.u.,
H_n = final heat content per lb., B.t.u.

Then the heat drop, or heat available for doing useful work, is

$$W\,(H_1 - H_n) \text{ B.t.u.} \tag{152}$$

If the steam expands against a resistance, as, for example, the piston of a reciprocating engine, the energy given up in forcing the piston forward may be expressed

$$E_1 = 777.5\,W\,(H_1 - H_n) \text{ ft-lb.} \tag{153}$$

If the steam expands within a perfect nozzle the energy will be given up in imparting velocity to the steam itself, thus:

$$E_2 = W\,\frac{V_1^2}{2\,g} \text{ ft-lb.,} \tag{154}$$

in which

V_1 = velocity of the jet in feet per second.

If the velocity of the jet is retarded to V_n feet per second, as by placing a series of vanes in its path, then the energy given up to the vanes (neglecting all losses) is

$$E = W\,\frac{V_1^2 - V_n^2}{2\,g}. \tag{155}$$

If the kinetic energy is completely absorbed by the vanes (neglecting all losses), then $V_n = 0$ and the energy given up is

$$E_3 = W\,\frac{V_1^2}{2\,g}. \tag{156}$$

But $E_1 = E_3$. Hence,

$$777.5\,W\,(H_1 - H_n) = W\,\frac{V_1^2}{2\,g},$$

from which

$$V_1 = 223.8\,\sqrt{H_1 - H_n}.^* \tag{157}$$

If there are n pressure stages, then the theoretical stage velocity is

$$V_1' = 223.8\sqrt{\frac{H_1 - H_n}{n}}. \tag{158}$$

The jet issuing from the nozzle is capable of exerting an *impulse* equal to F upon any object in its path, thus:

$$F = \frac{W V_1}{g} \text{ lb.} \tag{159}$$

If A = the area of cross section of the jet in square feet, and γ = weight of steam, pounds per cubic foot, then $W = \gamma A V_1$, or

$$F = \frac{\gamma A V_1^2}{g} \text{ lb.} \tag{160}$$

* For most purposes it is sufficiently accurate to make 223.8 = 224.

The *reaction*, *R*, of the jet against the nozzles is equal in value and opposite in direction to the impulse, or

$$R = F = \frac{WV_1}{g} = \frac{\gamma A V_1^2}{g}. \tag{161}$$

The theoretical horsepower developed by a jet of steam flowing at the rate of one pound per second may be expressed

$$\text{Hp.} = \frac{E}{550} = \frac{V_1^2 - V_n^2}{2\,g \times 550}, \tag{162}$$

in which

V_1 = initial velocity of the jet, ft. per sec.,
V_n = final velocity of the jet, ft. per sec.

Steam consumption per horsepower hour:

$$W_1 = \frac{3600}{\text{Hp.}}. \tag{163}$$

Heat consumption, B.t.u. per horsepower, per minute:

$$= \frac{W_1\,(H_1 - q_n)}{60}, \tag{164}$$

in which

q_n = heat of the liquid corresponding to temperature of the exhaust P_n.

Impulse efficiency of the jet = equation (155) ÷ equation (156).

$$E_i = \frac{V_1^2 - V_n^2}{V_1^2}. \tag{165}$$

Thermal efficiency (Rankine cycle):

$$E_r = \frac{H_1 - H_n}{H_1 - q_n}. \tag{166}$$

Rankine cycle ratio:

$$E = \frac{2546}{W_1\,(H_1 - H_n)}. \tag{167}$$

Equations (152) to (167) are general and are applicable to all turbines of whatever make.

The more important types of turbines will be discussed separately and an application of above equations will be made in each specific case.

Heat Drop in Steam Turbines: Trans. A.S.M.E., Vol. 33, p. 325, 1911; Engr., Mar. 8, 1912; U.S. Bureau of Standards, Reprint No. 167, 1911.

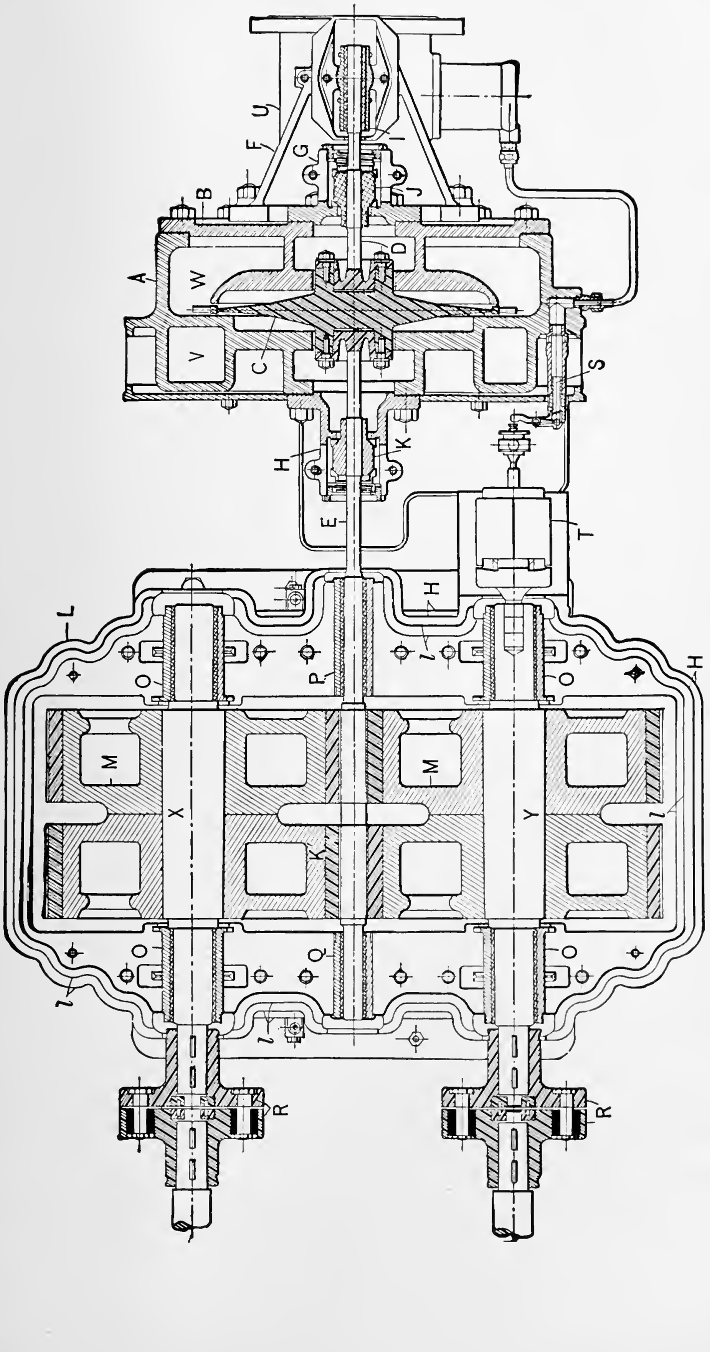


FIG. 228. De Laval Turbine, "Class A," Single Stage.

201. The De Laval Turbine. — Fig. 228 shows a section through a De Laval steam turbine and gear case and illustrates the principles of the single-stage "impulse" type. The turbine proper, to the right of the figure, consists of a high-carbon steel disk C fitted at the periphery with a single row of drop-forged steel blades and inclosed in a cast-steel casing. The disk is secured to a light flexible shaft and is of such a cross section that the radial and tangential stresses throughout its mass are of constant value. A flexible shaft is employed which allows the wheel to assume its proper center of rotation and thus to operate like a truly balanced rotating body.* The shaft is supported by three bearings, P, K, and I. I is self-aligning and carries the greater part of the weight of the disk. K is a flexible bearing, entirely free to oscillate with the shaft, and its only function is to seal the wheel casing against leakage. The power is transmitted through a steel helical pinion K' mounted on the extension of the turbine shaft E, to two large gears M, M at a reduction in speed of about 10 to 1. The blades, Fig. 229, are made with a bulb shank and fitted in slots milled in the rim of the wheel. The flanges, at the outer end of the blades, are brought in contact with each other and calked so as to form a continuous ring. The inlet and outlet angles of the blades are made alike and are 32 degrees for smaller sizes and 36 degrees for larger sizes.

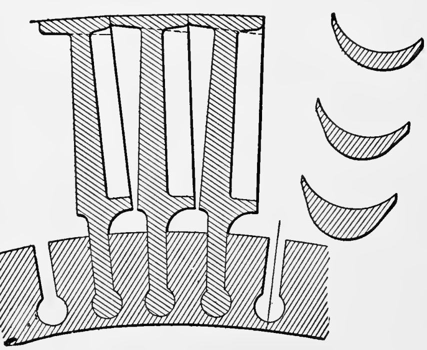


Fig. 229. De Laval Blades.

The operation is as follows: Steam enters the steam chest D, Figs. 228 and 230, through the governor (shown in detail in Fig. 231) and is distributed to the various adjustable nozzles, varying in number from 1 to 15 according to the size of turbine. In the earlier types the nozzles were uniformly distributed around the circumference, but in the later types are arranged in groups. As illustrated in Fig. 230 the nozzles are placed at an angle of 20 degrees with the plane of the disk. The steam is expanded adiabatically in the nozzles to the existing back pressure before it impinges at high velocity against the blades. After giving up its energy the steam passes into chamber W, Fig. 228, and out through the exhaust opening. Fig. 231 gives the details of the governor

* The shaft diameter for a 100-horsepower turbine is but 1 inch and for a 300-horsepower turbine approximately $1\frac{5}{16}$ inches.

and vacuum valves. Two weights *B* are pivoted on knife edges *A* with hardened pins *C* bearing on the spring *D*. *E* is the governor body, fitted in the end of the gear-wheel shaft *K*, and has seats milled for the knife edges *A*. The spring seat *D* is held against pins *A* by spiral

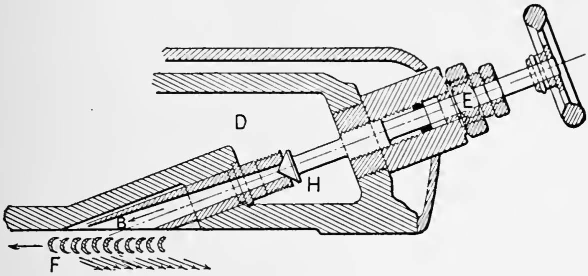


FIG. 230. De Laval Nozzle.

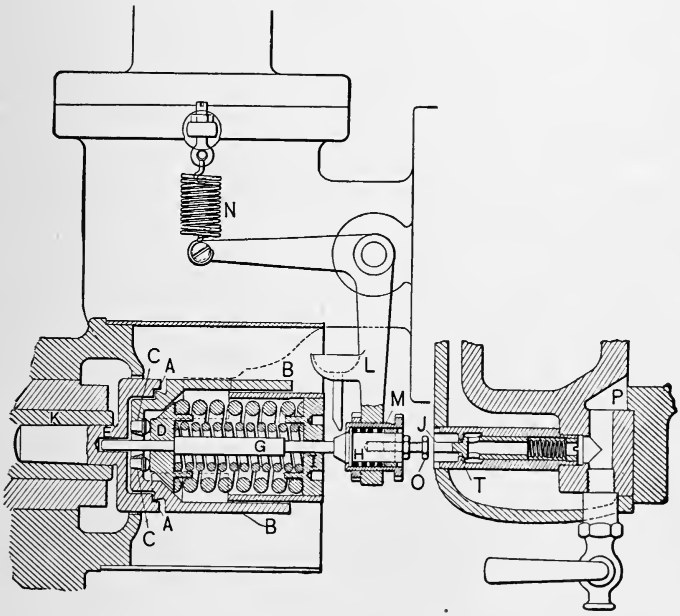


FIG. 231. De Laval Governor for Single-stage Turbine.

concentric springs, the tension on which is adjusted by a milled nut *I*. When the speed exceeds the normal, centrifugal force causes the weights to fly outward and overcome the resistance of the springs. This pushes pin *G* against bell crank *L*, which in turn closes the double-seated valve, thus throttling the supply of steam. To prevent racing in case the load is suddenly removed the vacuum valve *T* is added to the governor mechanism. Its operation is as follows: The governor pin G actuates the plunger H under normal conditions without moving the plunger relative to the bell crank. In case the load is suddenly removed, centrifugal force pushes pin G against bell crank L until it reaches its extreme position and the valve is nearly closed and little steam enters the turbine. If this does not check the speed, plunger G overcomes the resistance of spring M, and H moves relative to L, and its adjustable projection O presses against valve stem T and allows air to rush into the turbine through passage P.

The power of the turbine depends upon the number of nozzles in action, and these can be opened or closed by a hand wheel on each. Each nozzle performs its function as perfectly when operating alone as when operating in conjunction with others.

De Laval turbines of the single-stage geared type are made in sizes ranging from 17 to 700 horsepower, condensing and non-condensing, and are designed to regulate within an extreme variation of 2 per cent from no load to full load. The speeds vary from 10,600 r.p.m. for the largest size to 30,000 r.p.m. for the smallest, the gearing reducing these to 900 and 3000 r.p.m., respectively, at the shaft. The diameter of the wheel varies from 4 inches in the smallest turbine to 30 inches in the largest, thus giving peripheral velocities of from 520 to 1310 feet per second.

The single-stage geared type just described is no longer manufactured by the De Laval Co. and the multi-velocity stage machine is used in its place.

This company also manufactures a multi-pressure impulse turbine.

Both of these types are described further on.

202. Elementary Theory. — Single-wheel Single-stage Turbine. — The maximum theoretical power developed by a jet of steam flowing through a nozzle is dependent only upon the *weight* of steam flowing per unit of time and the *initial velocity*. Therefore the higher the initial velocity for a given rate of flow the greater will be the power developed and the higher the efficiency.

The maximum *weight* of steam discharged through a nozzle of any shape and for a given initial pressure is determined by the *area* of the narrowest cross section or *throat.*

To obtain the maximum *velocity* at the exit or *mouth,* for a given rate of flow, the nozzle should be proportioned so that expansion to the external pressure into which the nozzle delivers shall take place within the nozzle itself. If expansion in the nozzle is incomplete, sound waves will be produced and there will be irregular action and loss of energy. On the other hand, if expansion in the nozzle is carried below that of the external pressure at the mouth, sound waves will be produced with subsequent loss of energy even greater than in the former case.

Experimental and mathematical investigations indicate that the pressure at the narrowest section of an orifice or the throat of a nozzle through which steam is flowing falls to approximately 0.58 of the initial absolute pressure (with resultant velocity of about 1400 to 1500 feet per second) and any further fall in pressure must take place beyond the narrowest section. Thus for back pressures greater than 0.58 of the initial (conveniently taken as $\frac{3}{5}$), maximum exit velocity may be obtained from orifices of nozzles of uniform cross section or with sides *convergent.* For back pressure less than 0.58 of the initial the nozzle must first *converge* from inlet to throat and then *diverge* from throat to mouth in order to obtain maximum velocity. Without the divergent portion of the nozzle the jet will begin to spread after passing the throat, and its energy will be given up in directions other than that of the original jet.

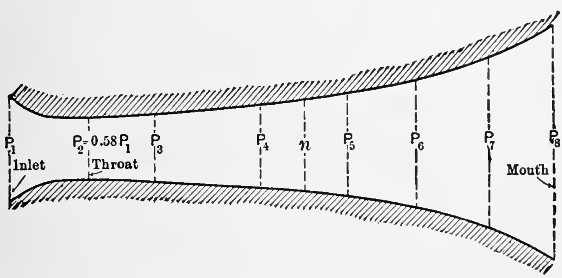


FIG. 232. Theoretically Proportioned Expanding Nozzle.

Fig. 232 shows a section through a theoretically proportioned expanding nozzle. The cross section of the tube at any point n may be calculated by means of equation

$$A_n = \frac{WS_n}{V_n}, \tag{168}$$

in which

A_n = area in square feet,
W = maximum weight of steam discharged, pounds per second,
S_n = specific volume of the steam at pressure P_n.

For wet steam $S_n = x_n u_n + \sigma$,

in which

x_n = quality of steam at pressure P_n after adiabatic expansion from pressure P_1,
u_n = specific volume of saturated steam at pressure P_n,
σ = volume of 1 lb. of water corresponding to pressure P_n. This quantity is very small compared with that of the steam and may be neglected.

For superheated steam, see Mollier diagram, paragraph 451.

V_n = velocity of the jet, feet per second.

V_n may be determined from equation (157):

$$V_n = 223.8 \sqrt{H_1 - H_n}.$$

By substituting H_n = heat content corresponding to pressure $P_n = 0.58\ P_1$ in equations (157) and (168) the area at the throats may be readily determined. The cross-sectional area for other points in the tube may be determined in a similar manner by assigning values of H_n corresponding to the various pressures.

In case of a perfect nozzle $H_1 - H_n$ represents the heat given up toward producing velocity by adiabatic expansion from pressure P_1 to P_n. In the actual nozzle the frictional resistance of the tube serves to increase its dryness fraction, but in doing so it decreases the amount of energy the steam is capable of giving up towards increasing its own velocity. If y one-hundredths of the heat $H_1 - H_n$ is utilized in overcoming frictional resistance, then the resulting velocity will be

$$V = 223.8 \sqrt{(1 - y)\,(H_1 - H_n)}. \qquad (169)$$

The quality of the steam after expanding to P_n against the resistance will be higher by an amount

$$I_n = \text{increase in quality} = \frac{y\,(H_1 - H_n)}{r_n}, \qquad (170)$$

in which

r_n = heat of vaporization at pressure P_n.

The curves in Fig. 233, calculated by means of equations (157) and (168), show the relationship between velocity, quality, pressure, and kinetic energy for all points in a theoretically perfect nozzle expanding one pound of dry steam per second from an initial absolute pressure of 190 pounds to a condenser pressure of one pound.

The curves in Fig. 234 are based upon the experiments of Gutermuth (Zeit. d. Ver. Ingr., Jan. 16, 1904) and show the effect of a few shapes of nozzles and orifices on the actual weight of steam discharged for various rates of initial and final pressures, the smallest section of the tube remaining constant.

The nozzles of most commercial types of steam turbines are made with straight sides as in Fig. 230, so that only the area at the mouth need be determined in addition to that at the throat in order to lay out the shape of the tube.

Equations (157) and (168) are general and are applicable to steam of any quality, wet, dry, or superheated.

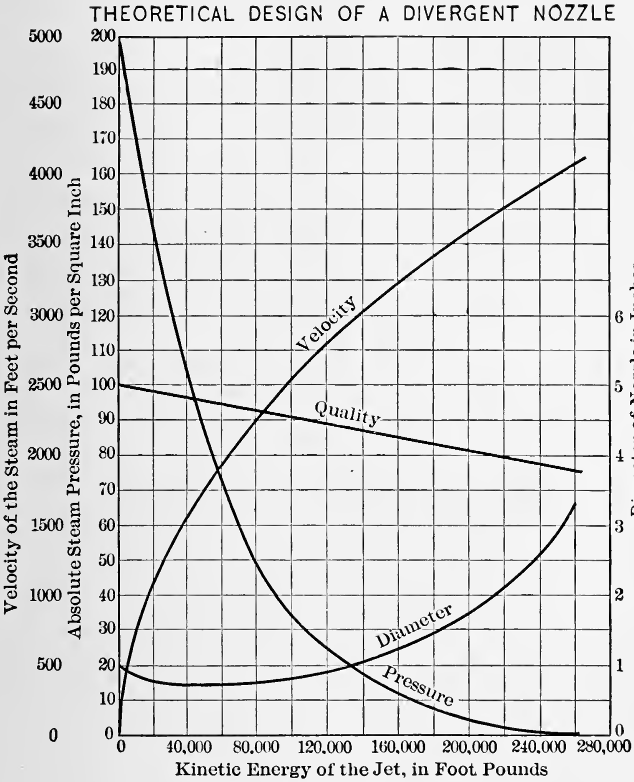


FIG. 233.

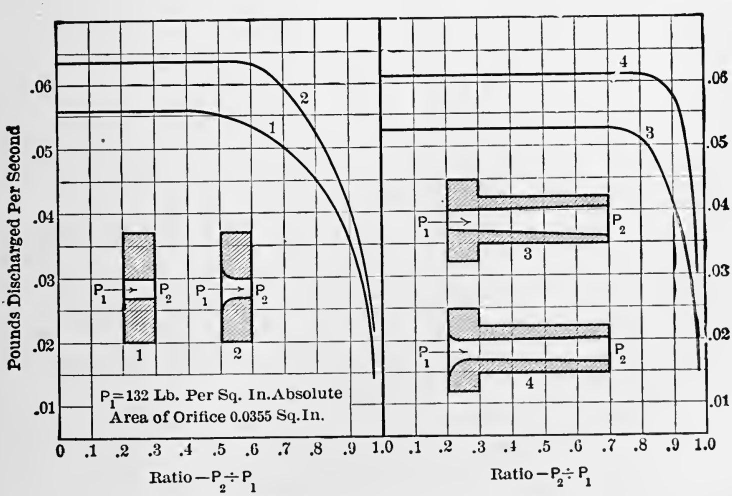


FIG. 234. Flow of Steam through Nozzles.

The diameter at the throat may be calculated within an error of 1 to 2 per cent for the range of pressures usually encountered by means of Grashof's formula.

For dry or wet steam when $P_n = < 0.58\, P_1$

$$w' = 60\, a_0\, P_1^{0.97} \sqrt{x_1}. \tag{171}$$

For superheated steam when $P_n = 0.58\, P_1$

$$w' = 60\, a_0\, P_1^{0.97} \div (1 + 0.00065\, t_s), \tag{172}$$

in which

w' = actual weight of steam discharged, lb. per hr.,
a_0 = area of the throat, sq. in.,
P_1 = initial absolute pressure, lb. per sq. in.,
x_1 = initial quality,
t_s = degree of superheat, deg. fahr.

For back pressures higher than the critical or $P_n > 0.58\, P_1$ the fundamental equation (157) offers the simplest solution. Approximate results for this condition may be obtained by multiplying equations (171) and (172) by a factor K

$$K = 2.182 \sqrt{c\,(1 - 1.19c)}, \tag{173}$$

in which

$c = 1 - (P_n \div P_1)$.

When a divergent nozzle having an actual expansion ratio r (= mouth area ÷ throat area) is used for steam pressure having a ratio R (= mouth area ÷ throat area for pressure ratio P_n/P_t) a percentage nozzle mouth error is introduced of a value $c_1 = 100\,(r - R) \div r$, which may be positive or negative. Table 87 gives the velocity efficiency or ratio of probable actual exit velocity to the theoretical velocity for various nozzle mouth errors, assuming the correctly proportioned nozzle to have a velocity efficiency of 97 per cent.

TABLE 87.

Nozzle mouth error, c_1	−40	−30	−20	−10	0	10	15	20	25	30
Velocity efficiency, per cent	93.5	94.8	95.9	96.7	97	96.7	96.3	95.3	93.6	90.6

When the actual expansion ratio of the nozzle is greater than required, the nozzle is said to be overexpanded; when smaller, underexpanded. From Table 87 it appears that it is preferable to have a nozzle underexpanded than overexpanded.

Moyer ("The Steam Turbine," 1st Edition, p. 40) states that the ratio of the area of a correctly proportioned nozzle at the throat A_0 to the area at any point a_n is very nearly proportional to the ratio of the pressure at point a_n to the initial pressure, or

$$\frac{a_0}{a_n} = \frac{P_i}{P_n}. \tag{174}$$

The entrance to the tube is rounded by any convenient curve.

The length of the tube may be roughly approximated by the following formula:

$$L = \sqrt{15\, a_0}, \tag{175}$$

in which

L = length between the throat and mouth, in inches,
a_0 = area at the throat, square inches.

Practice shows that the cross section of a nozzle, whether circular, elliptical, square, or rectangular (the latter with rounded corners), has very little influence on the efficiency, provided the inner surfaces are smooth and the ratio of the area at the throat to that of the mouth is correctly proportioned. The *velocity* efficiency of a properly proportioned nozzle with straight sides is about 95 to 97 per cent, corresponding to an *energy* efficiency of 92 to 94 per cent, so that it is not considered worth while to attempt to follow the more difficult exact curves.

Example 30. Find the smallest cross section of a frictionless conically divergent nozzle for expanding one pound of steam per second from an absolute initial pressure of 190 pounds to an absolute back pressure of 2 pounds and find six intermediate cross sections where the pressures will be 70, 30, 14.7, 8, 4, and 2 lb. respectively. Compare the velocity and energy of the jet issuing from this nozzle with those of an actual nozzle in which 10 per cent of the heat energy is lost in friction.

From steam and entropy tables we find the values of H, x, u, for absolute pressures corresponding to 190, $0.58 \times 190 = 110$, 70, 30, etc., lb. per square inch as follows (theoretical nozzle):

	H.	x.	u.	$S = xu$.
$P_1 = 190$	1197.3	1.00	2.406	2.406
$P_2 = 110$*	1152.6	0.960	4.047	3.885
$P_3 = 70$	1117.9	0.932	6.199	5.775
$P_4 = 30$	1057.2	0.887	13.75	12.27
$P_5 = 14.7$	1011.3	0.857	26.78	22.95
$P_6 = 8$	947.8	0.834	47.26	39.29
$P_7 = 4$	935.6	0.810	90.4	73.2
$P_8 = 2$	899.3	0.788	173.1	137.0

* $P_2 = 0.58\, P_1$ (= pressure at throat).

If entropy tables or charts are not available, values H_1 to H_8 and x_1 to x_8 must be calculated. (See Chapter XXII.)

The different quantities for the theoretical nozzle will be calculated for the exit pressure $P_n = P_8 = 2$ lb. per sq. in. absolute.

$$\begin{aligned} V_8 &= 223.8 \sqrt{H_1 - H_8} \\ &= 223.8 \sqrt{1197.3 - 899.3} \\ &= 3865 \text{ feet per second.} \end{aligned}$$

$$\begin{aligned} E_8 &= 778\,(H_1 - H_8) \\ &= 778\,(1197.3 - 899.3) \\ &= 232{,}000 \text{ foot-pounds.} \end{aligned}$$

$$\begin{aligned} A_8 &= \frac{WS}{V} \\ &= \frac{1 \times 137}{3865} \\ &= 0.0353 \text{ square foot.} \end{aligned}$$

$$\begin{aligned} d_8 &= \sqrt{\left(\frac{144 \times 4}{\pi}\right) A} = 13.56 \sqrt{A} \\ &= 13.56 \sqrt{0.0353} \\ &= 2.54 \text{ inches.} \end{aligned}$$

$$\begin{aligned} {}_8 &= \frac{WV_8}{g} \\ &= \frac{3865}{32.2} \\ &= 120 \text{ pounds.} \end{aligned}$$

THEORETICAL NOZZLE.

Quantity		V Ft. per Sec.	E Ft.-Lb.	A Sq. Ft.	d Inches.	F Pounds.
Formula		(73)	(72)	(76c)		(74)
Pressures	110	1,496	34,767	.00259	0.693	46.4
	70	1,995	61,853	.00269	0.702	61.98
	30	2,650	107,485	.00461	0.919	82.3
	14.7	3,053	144,742	.00745	1.1	94.8
	8	3,339	173,207	.0119	1.46	103.7
	4	3,624	203,968	.0202	1.92	112.5
	2	3,865	232,000	.0353	2.54	120.0

In the actual nozzle these values will be modified because of the frictional losses. Thus, for $P_n = 2$ lb.,

$$\begin{aligned} V_8 &= 223.8 \sqrt{(1 - y)\,(H_1 - H_8)} \\ &= 223.8 \sqrt{(1 - 0.1)\,(1197.3 - 899.3)} \\ &= 3667 \text{ ft. per sec.} \end{aligned}$$

$$E_8 = 778\,(1 - 0.1)\,(1197.3 - 899.3) = 208{,}800 \text{ ft-lb.}$$

$$x_8' = x_8 + I_8 = x_8 + \frac{y\,(H_1 - H_8)}{r_8}$$
$$= 0.788 + \frac{0.1\,(1197.3 - 899.3)}{1021}$$
$$= 0.788 + 0.029$$
$$= 0.817.$$
$$A_8 = \frac{W x_8' u_8}{V_8}$$
$$= \frac{0.817 \times 173.1}{3667}$$
$$= 0.0386 \text{ sq. ft.},$$

from which

$$d_8 = 2.66 \text{ in.}$$
$$F_8 = \frac{WV_8}{g} = \frac{3668}{32.2} = 114 \text{ lb.}$$

These various factors for all given pressures have been calculated in a similar manner and are as follows:

ACTUAL NOZZLE.

Quantities		V Ft. per Sec.	E Ft.-Lb.	x'.	A Sq. Ft.	d Inches.	F Ft.-Lb.
Pressures	110	1,420	31,317	.9658	.00275	0.711	44.1
	70	1,893	55,632	.9414	.00286	0.723	58.8
	30	2,515	98,257	.9026	.00493	0.951	78.12
	14.7	2,894	130,050	.876	.0080	1.2	98.8
	8	3,168	155,858	.856	.0127	1.53	98.4
	4	3,438	183,581	.836	.0220	2.01	106.8
	2	3,667	208,800	.817	.0386	2.66	114.0

Many of these values may be determined directly from the *Mollier* or total heat-entropy diagram as described in Chapter XXII; in fact, the Mollier diagram has to all intents and purposes supplanted the steam tables in this connection. For superheated steam the diagram is extremely useful in avoiding laborious calculations.

Fig. 235 gives a diagrammatic arrangement of the blades in a single-stage De Laval turbine. The nozzle directs the steam against the blades with *absolute* velocity V_1 and at an angle α with the plane of the wheel XX. Since the wheel is moving at a velocity of u feet per second, the velocity v_1 of the steam *relative* to the wheel is the resultant of V_1 and u. The angle β_1 between v_1 and XX will be the proper blade angle at entrance. If the blade curve makes this angle with the direction of motion of the wheel no shock will be experienced when the steam enters the blades. For convenience in construction the exit angle β_2 is made the same as the entrance angle β_1. Neglecting frictional losses in the blade channels the *relative* exit velocity will be $v_2 = v_1$, and the *absolute* velocity V_2 is the resultant of v_2 and u. The impulse exerted by the jet in striking the vanes is $\frac{W}{g} v_1$, and its component in the direction of motion is $\frac{W}{g} v_1 \cos \beta_1 = \frac{W}{g} (V_1 \cos \alpha - u)$. As the jet leaves the vanes the impulse is $-\frac{W}{g} v_2 \cos \beta_2 = -\frac{W}{g} (V_2 \cos \gamma + u)$.

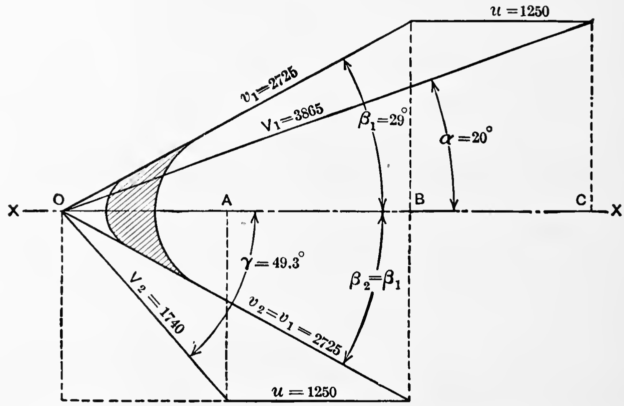


FIG. 235. Velocity Diagram. Ideal Single-stage Impulse Turbine.

The *total pressure* acting on the vanes, or the actual driving impulse, is

$$P = \frac{W}{g} \{V_1 \cos \alpha - u - [-(V_2 \cos \gamma + u)]\}$$

$$= \frac{W}{g} (V_1 \cos \alpha + V_2 \cos \gamma). \tag{176}$$

Equation (176) may also be expressed

$$P = \frac{W}{g} \cdot 2 (V_1 \cos \alpha - u). \tag{178}$$

The resultant *axial force* or *end thrust* is

$$F = \frac{W}{g} (V_1 \sin \alpha - V_2 \sin \gamma). \tag{179}$$

Evidently if $\alpha = \gamma$ and $V_1 = V_2$ there will be no end thrust, since $V_1 \sin \alpha - V_2 \sin \gamma$ will be zero.

The *work done* is

$$Pu = \frac{W}{g} u (V_1 \cos \alpha + V_2 \cos \gamma), \tag{180}$$

or, using equation (178) in place of (176),

$$Pu = \frac{W}{g} \cdot 2u (V_1 \cos \alpha - u)$$

$$= \frac{W}{g} \cdot 2 (uV_1 \cos \alpha - u^2). \tag{181}$$

By making the first derivative equal to zero

$$\frac{d}{du}\left\{\frac{W}{g}\,2\,(uV_1\cos\alpha - u^2)\right\} = V_1\cos\alpha - 2\,u = 0,$$

or

$$u = \tfrac{1}{2}\,V_1\cos\alpha.$$

That is, for *any nozzle angle* α the work done, Pu, has its greatest value when $u = \frac{1}{2}\,V_1\cos\alpha$ or $\gamma = 90$ degrees, whence

$$Pu = W\,\frac{V_1^2}{2\,g}\cos^2\alpha. \tag{182}$$

The work for *any initial velocity* V_1 becomes a maximum when $\alpha = 0$ and $u = \frac{1}{2}\,V_1$. *This condition can only occur for a complete reversal of jet and zero final velocity.* Substitute $\alpha = 0$ and $u = \frac{1}{2}\,V_1$ in equation (181).

$$Pu = \frac{WV_1^2}{2\,g},$$ which is necessarily the same as equation (156).

In the actual turbine the various velocities will be less than those as obtained on account of the frictional resistance in the blades, and the velocity diagram should be modified accordingly.

Example 31. Lay out the blades (theoretical and actual) for the nozzle in the preceding example, assuming that the jet impinges against the wheel at an angle of 20 degrees and that the peripheral velocity is 1250 feet per second.

Theoretical Case:

Lay off $V_1 = 3865$ feet per second in direction and amount as shown in Fig. 235 and combine it with $u = 1250$ feet per second; this gives v_1, the relative entrance velocity, as 2725 feet per second, and β, the entrance angle, as 29 degrees.

Lay off $v_2 = v_1$ at an angle $\beta_2 = \beta_1$ and combine with u; this gives V_2, the *absolute* exit velocity, as 1740 feet per second.

The theoretical energy available for doing work is

$$E = \frac{W}{2\,g}\,(V_1^2 - V_2^2)$$

$$= \frac{1}{64.4}\,(3865^2 - 1740^2) = 185{,}000 \text{ foot-pounds.}$$

The difference between 232,000 and 185,000 = 47,000 foot-pounds is evidently the kinetic energy lost in the exhaust due to the exit velocity.

The pressure exerted by the steam on the buckets is

$$P = \frac{W}{g}\,(V_1\cos\alpha + V^2\cos\gamma)$$

$$= \frac{1}{32.2}\,(3865 \times 0.9397 + 1740 \times 0.65166)$$

$$= 148 \text{ pounds.}$$

The theoretical impulse efficiency is

$$\frac{V_1^2 - V_2^2}{V_1^2} = \frac{3865^2 - 1740^2}{3865^2} = 0.797.$$

The theoretical horsepower per pound of steam flowing per second is

$$\text{Hp.} = \frac{185{,}000}{550} = 336.$$

Theoretical steam consumption per horsepower-hour is

$$\frac{3600}{336} = 10.7 \text{ pounds.}$$

Actual Case:

Proceed as in the theoretical case, using the actual absolute velocity $V_1 = 3865\sqrt{1-y} = 3865\sqrt{1-0.10} = 3667$ feet per second in place of the theoretical value $V_1 = 3865$. Lay off $V_1 = 3667$ at an angle of 20 degrees as before and combine with $u = 1250$, Fig. 236.

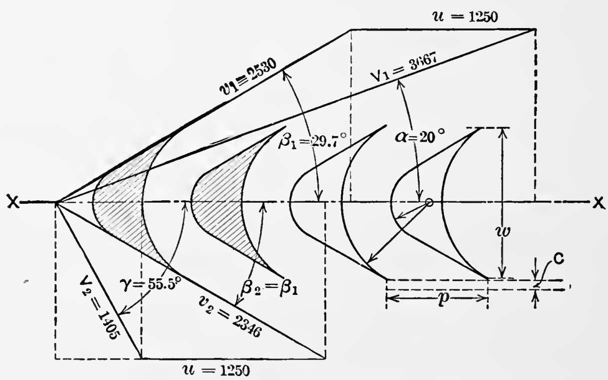


FIG. 236. Velocity Diagram as Modified by Friction Losses.

The resultant $v_1 = 2530$ is the velocity of the jet relative to the wheel, and the entrance angle β is found to be 29.7 degrees. The relative exit velocity v_2 will be less than v_1 because of the blade friction.

Assume the loss of energy ϕ between inlet and exit of the blades to be 14 per cent; then, since the velocity varies as the square root of the energy,

$$\begin{aligned} v_2 &= v_1\sqrt{1-\phi} \qquad (183)\\ &= 2530\sqrt{1-0.14} \\ &= 2346 \text{ feet per second.} \end{aligned}$$

The resulting absolute velocity V_2 is found from the diagram to be $V_2 = 1405$ feet per second.

Since the loss of energy in the nozzle is

$$\frac{V_1^2 - (1-y)\,V_1^2}{2\,g}, \qquad (184)$$

and that in the blade

$$\frac{v_1^2 - (1-\phi)\,v_1^2}{2\,g}, \qquad (185)$$

the remaining energy, deducting both losses in the nozzle and the blades, is

$$\frac{W}{2g}(V_1^2 - yV_1^2 - \phi v_1^2 - V_2^2) \qquad (186)$$

$$= \frac{1}{64.4}(3865^2 - 0.1 \times 3865^2 - 0.14 \times 2530^2 - 1405^2)$$

$$= 164{,}200.$$

The losses due to windage, leakage past the buckets and mechanical friction must be deducted from these figures to give the actual energy available for doing useful work. Assuming a loss of 15 per cent due to this cause, the work delivered is

$$0.85 \times 164{,}200 = 139{,}570 \text{ foot-pounds.}$$

The efficiency in the ideal case was found to be 0.797 and the available energy 185,000 foot-pounds.

The efficiency, deducting the loss due to friction, etc., is

$$\frac{139{,}570}{185{,}000} \times 0.797 = 0.60.$$

The horsepower delivered is

$$\frac{139{,}570}{550} = 254.$$

Steam consumption per horsepower-hour is

$$\frac{3600}{254} = 14.2 \text{ pounds.}$$

The heat consumption, B.t.u. per horsepower per minute is

$$\frac{14.2\,(1197.3 - 94)}{60} = 260.$$

Assuming the revolutions per minute to be 10,000, the mean diameter of the wheel to give a peripheral velocity of 1250 feet per second is

$$\frac{1250 \times 60}{10{,}000 \times 3.14} = 2.39 \text{ feet, or } 28.6 \text{ inches.}$$

The determination of the height and width of vanes, clearance between nozzles and blades, etc., are beyond the scope of this work and the reader is referred to the accompanying bibliography.

The ratio of exit to inlet velocity is called the blade or bucket velocity coefficient. Table 88 gives the values of this coefficient for the usual shape of impulse turbine blades. The values include all losses between the nozzle mouth and entrance to the exhaust opening. (Marks' Mechanical Engineers' Handbook, p. 984.)

TABLE 88.

Velocity relative to blades, ft. per sec.	200	400	600	800	1000	1500	2000	2500	3000	4000
Blade velocity coefficient	0.953	0.918	0.888	0.863	0.841	0.801	0.774	0.754	0.739	0.716

Blade Design for De Laval Turbines: Moyer, "Steam Turbine," Chap. IV; Power, Mar. 17, 1908, p. 391.

Flow of Steam through Nozzles: Jour. A.S.M.E., Mid. Nov., 1909, April, 1910, p. 537; Engineering, Feb. 2, 1906; Engr. Lond., Dec. 22, 1905; Eng. Rec., Oct. 26, 1901; Power, May, 1905; Eng. News, Sept. 19, 1905, p. 204.

Design of Turbine Disks: Engr. Lond., Jan. 8, 1904, p. 34, May 13, 1904, p. 481.

Turbine Losses and their Study: Jour. El. Power and Gas, March 9, 1912.

Critical Velocity of Shafting: Jour. A.S.M.E., June, 1910, p. 1060; Power, Sept., 1903, p. 484.

203. Terry Non-condensing Turbine. — Fig. 237 shows a section through a single-stage Terry turbine, illustrating an application of the *single-stage impulse* type with two or more *velocity stages.* This "compounding" of the velocity permits of much lower peripheral velocities

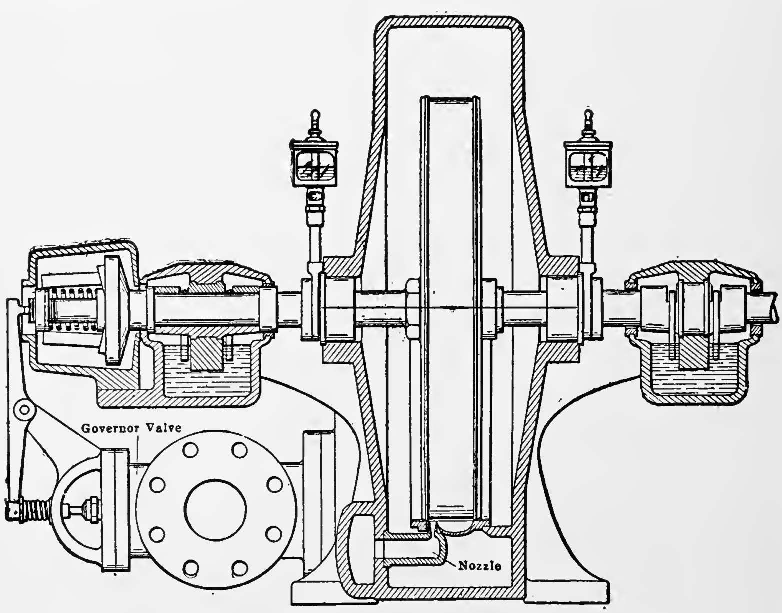


FIG. 237. Section through Single-stage Terry Steam Turbine.

than with the single-velocity type. The rotor, a single wheel consisting of two steel disks held together by bolts over a steel center, is fitted at its periphery with pressed-steel buckets of semi-circular cross section. The inner surface of the casing is fitted with a series of gun-metal reversing buckets arranged in groups, each group being supplied with a separate nozzle. The steam issuing from nozzle N, at very high velocity, Fig. 238, strikes one of the buckets, B, on the wheel, and since the velocity of the buckets is comparatively low, is reversed in direction and directed into the first one of the reversing chambers. The chamber redirects the jet against the wheel, from which it is again deflected; this is repeated four or more times until the available energy has been absorbed by the rotor. Terry turbines are made in a number of sizes varying from 5 to 800 horsepower, and operate at speeds varying from

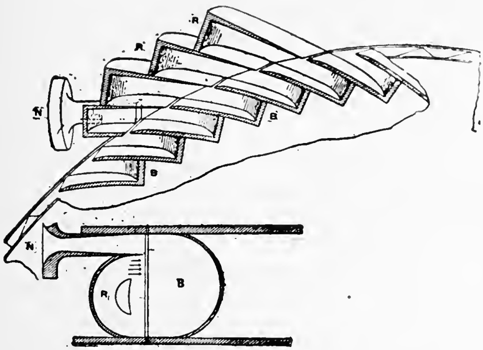


FIG. 238. Arrangement of Buckets and Reversing Chambers in a Terry Steam Turbine.

210 feet per second in the smaller machine to 260 feet per second in the larger. These low speed limits compared with the speed of single-stage De Laval turbines are made possible by the application of the velocity-stage principle in the use of the reversing buckets. The

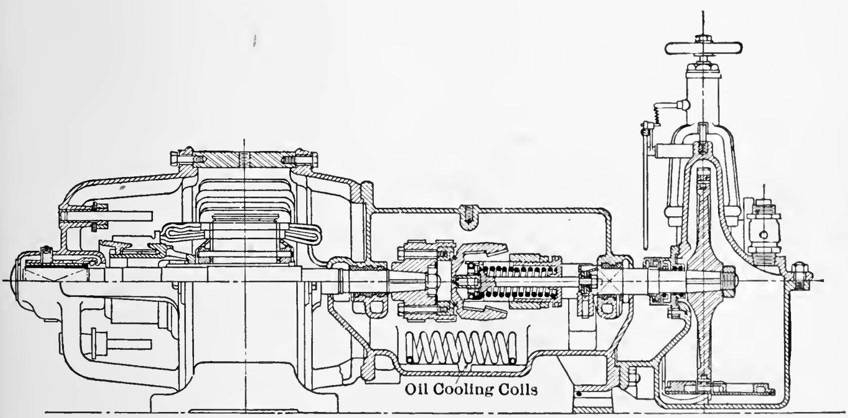


FIG. 239. Westinghouse Impulse Turbine Connected to Generator through Reduction Gearing.

rotor of the smaller machine is 12 inches in diameter and runs at 3800 r.p.m., and that of the larger, 48 inches, running at 1250 r.p.m. Since the flow of steam into and from the buckets is in the plane of the wheel there is no end thrust.

Non-condensing Terry turbines are all of the single-stage type.

204. Westinghouse Impulse Turbine. — The Westinghouse impulse turbine which is constructed in various sizes ranging from 10 to 800 horsepower is similar in basic principle to the Terry turbine. The rotor consists of a single wheel on the periphery of which are located blades of nickel steel. In the non-condensing unit the steam is expanded in a single nozzle and is directed upon the rotor where its energy is partially absorbed in work. From the rotor it is deflected to the reversing member and is directed on the wheel a second time when the remaining energy is finally extracted. For condensing service this reversing operation is repeated a second time making three passes through the wheel before the steam is exhausted to condenser. By the use of two separate nozzles, large and small, proportioned to suit the load conditions, relatively as good efficiency is obtained at half-load as at rated load. The economy of a single wheel turbine is vitally affected by its operating speed, the higher the speed up to a peripheral velocity of less than half that of the jet the better will be the heat economy. On the contrary, moderate-speed generators offer a better efficiency than high-speed generators. This type of unit has been designed so that the steam turbine and its accompanying generator may operate at their best speed through the medium of reduction gears. In fact, all builders of high-speed turbines are equipped to furnish reduction gears with their units, and the general tendency is toward the incorporation of gearing in all types under 1000 kw. rated capacity.

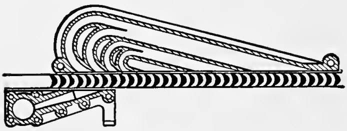


FIG. 240. Developed Section through Nozzle Blades and Reversing Chamber, Westinghouse Impulse Turbine.

205. Elementary Theory.—Single-wheel Multi-velocity-stage Turbine.— Fig. 241 gives the theoretical velocity diagram for a single-pressure stage Terry Turbine. Since the entire heat drop takes place in the nozzle the initial velocity of the jet OA is the same as with the single-stage De Laval turbine and may be calculated by means of equation (157). OA represents the absolute velocity of the jet, OC the peripheral velocity and AOC the angle of the nozzle. CB is the component, parallel to the line of the jet, of the resultant of AO and OC. DC, in line with and equal in length to CB, combined with the peripheral velocity DE gives EC, the absolute velocity of the steam as it leaves the first set of rotating buckets. O_1F, parallel to OA and equal in length to EC, represents the velocity of the steam as it enters the first stationary or reversing bucket. JG is the component of the resultant of O_1F and O_1J in line with the jet. The resultant IJ of HJ $(= JG)$ and HI represents the velocity of the steam as it leaves the rotary buckets the second time. This construction is repeated through all velocity stages. The final exit velocity of the steam as it issues from the moving buckets is WY. The energy converted into useful work is

$$\frac{W}{2\,g} = \left(\overline{OA}^2 - \overline{WY}^2\right).$$

In the actual turbine friction losses would reduce the length of the velocity lines and increase the amount of energy rejected in the exhaust. The construction of the velocity diagram as modified by friction is similar to that described in paragraph 202, Fig. 235.

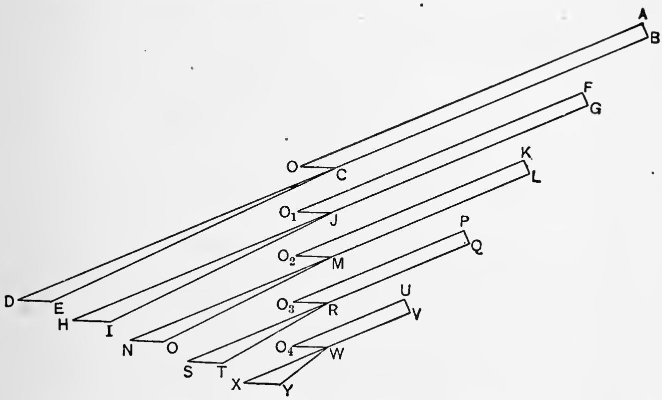


Fig. 241. Theoretical Velocity Diagram, Terry Turbine.

206. De Laval Velocity-stage Turbine.—Fig. 242 shows a section through a "Class C" De Laval steam turbine illustrating an application of the *single-pressure, multi-velocity* stage type in which the velocity is compounded by a series of wheels and reversing intermediates instead of having the jet redirected upon a single rotor. In this type of turbine the steam is completely expanded in a single set of nozzles from initial to terminal pressure just as in the single-wheel geared type. The jet from the nozzles impinges against the first row of moving blades or vanes and gives up part of its energy. It leaves the moving blades at a reduced velocity and is reversed in direction by the first set of stationary vanes. The latter redirect the jet against the second set of moving vanes where a further absorption of energy takes place and the velocity is again lowered. This process is repeated until the steam leaves the last row of moving vanes at practically zero velocity. The wheels are of forged steel and are fitted at the periphery with nickel bronze blades similar in design to those of the single-stage geared type. The guide vanes are similar in form to the moving vanes and are attached in a like manner to a steel retaining ring. The governor is of the throttling type. In the smallest machine the governor weights are attached directly to the main shaft and in the larger machines it is actuated through speed reduction gearing. The emergency governor is independent of the main governor and closes a butterfly valve in the steam inlet opening when a predetermined speed is exceeded. This type of turbine is

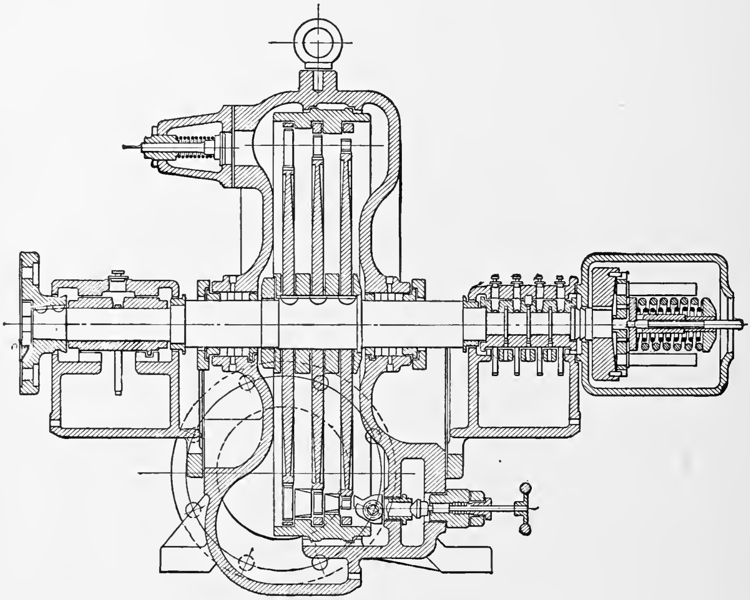


FIG. 242. De Laval Velocity-stage Turbine.

constructed in sizes ranging from 1 to 1500 horsepower and at speeds ranging from 3600 to 6000 r.p.m. By means of reduction gearing any desired lower speed may be obtained.

The velocity diagram may be constructed in a manner similar to that of any single-pressure stage of the Curtis turbine as described in paragraph 212.

207. Kerr Turbine. — Fig. 243 shows a longitudinal section through an eight-stage Kerr steam turbine illustrating the *compound-pressure* or *multi-cellular* group of the impulse type. The rotor consists of a series of steel disks, mounted on a rigid steel shaft. A series of drop-forged steel buckets is secured to the periphery and riveted in dovetailed slots as shown in Fig. 244. The tips of the buckets are riveted to a shroud ring, thereby insuring a rigid and positive spaced construction. The stator is made up of a number of arched cast-iron diaphragms with circular rims tongued and grooved, and bolted to steam-end and exhaust-end castings. The nozzles are formed by walls within the diaphragm and thin Monel metal vanes die-pressed into shape and cast

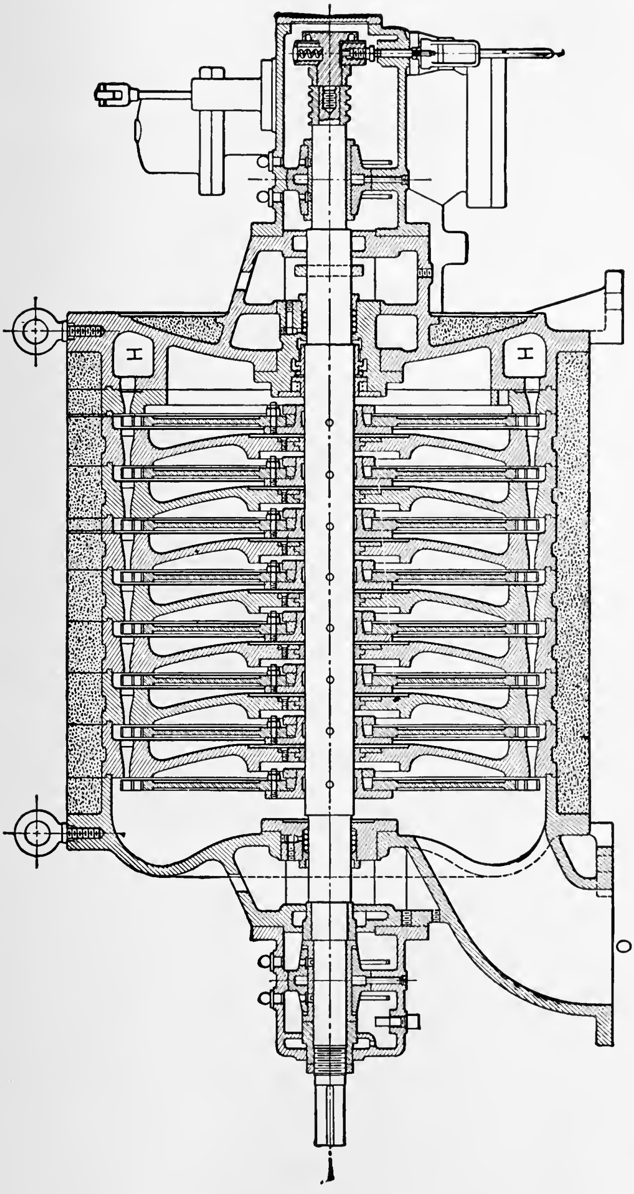


FIG. 243. Section through an Eight-stage Kerr Turbine.

into the diaphragm. One set of nozzles and one wheel constitute a stage and the expansion is usually carried out in from six to ten stages, depending upon the condition of operation.

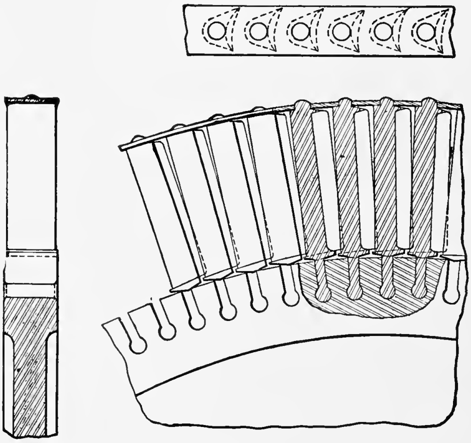


Fig. 244. Bucket Fastening, Kerr Turbine.

The operation is as follows: Steam enters the turbine through a double-beat balanced poppet valve, the stem of which is connected through levers to the governor, to the circular cored space H, H extending around the steam "end casting." This space acts as an equalizer and insures uniform admission to the first set of nozzles. Partial expansion takes place through the first set of nozzles and the kinetic energy is imparted to the rotor through the medium of the vanes. Steam leaves the buckets at a very low velocity and is again expanded through the second set of nozzles in the diaphragm. This process is repeated in each stage and exhaust steam leaves the turbine at O.

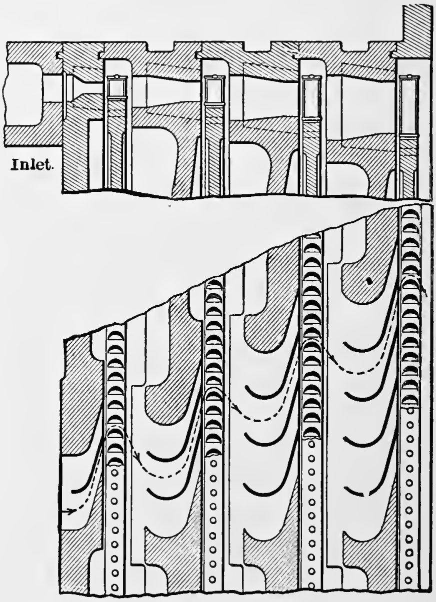


Fig. 245. Arrangement of Vanes and Nozzles, Kerr Turbine.

Fig. 246 illustrates the principles of the oil relay governor as applied to the larger sizes of turbines driving alternators. Referring to Fig. 246: rotation of the turbine shaft is transmitted through worm gear and governor spindle to weights, W, W'. Centrifugal force throws these weights outward about suspension points A and A', overcoming the resistance of the spring. The movement of the spring is transmitted through lever L to relay plunger P and admits oil pressure (about 30 pounds per square inch) to piston S and in this manner throttles admission valve *V*. Similarly, a downward movement of the relay plunger stem releases oil pressure and opens the admission valve.

Floating lever *L* is connected to the admission valve stem through secondary lever *M* so that the movement of the steam valve returns

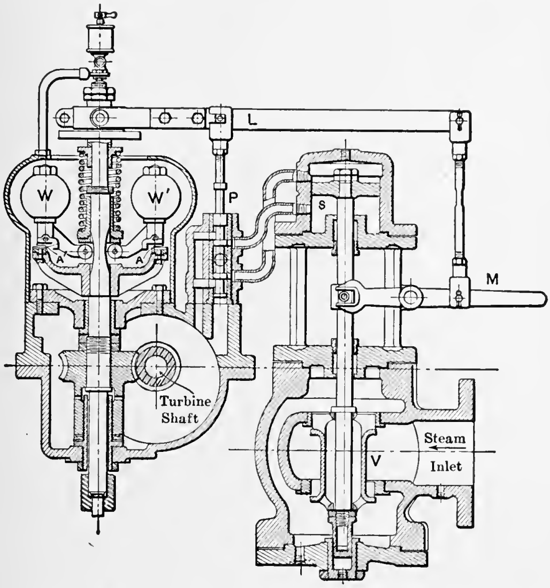


FIG. 246. Oil Relay Governor, Kerr Turbine.

the relay plunger to its central position. This equalizes the pressure on top and bottom of the main piston *S* and arrests its movement, thereby maintaining a fixed opening for a given speed. A suitable emergency valve automatically cuts off the steam supply in case the speed exceeds a predetermined amount.

A spring-loaded governor of the centrifugal type mounted directly on the turbine shaft is used to control the smaller sizes of turbines.

Kerr turbines are constructed horizontally and vertically and in various sizes ranging from 5 to 2500 horsepower, and are designed to operate all classes of pumps, blowers and generators. The rotative speed varies from 2000 to 4000 r.p.m., depending upon the service for which the turbines are intended. By means of gearing any lower speed may be obtained.

208. De Laval Multi-stage Turbine. — This is of the multi-cellular type and is constructed with single velocity stages or with two velocity stages for each pressure stage. The increase in the cross-sectional area of the passages required by the expansion of the steam as it proceeds through the turbine is effected by lengthening the blades, reducing the diameters of the wheels correspondingly and increasing the bore of the casing. (In the Kerr turbine the blades are lengthened and increased in width from the high-pressure to the low-pressure stages and the steam passages are increased in size but the outside diameter of the rotor remains the same.) The bearings are of rigid construction arranged for water cooling. Labyrinth packing is used between stages and combined labyrinth and carbon-ring packing at the steam and exhaust ends of the casing. Air leakage into the turbine is prevented by introducing live steam between the two outer carbon rings. The governor is of the throttling type and is mounted upon a vertical shaft driven through worm gearing by the main turbine shaft. These machines are constructed in various sizes ranging from 50 to 15,000 horsepower. The maximum speed of the smaller machines is about 7500 r.p.m.

209. Elementary Theory. — Multi-pressure Single-velocity-stage Turbine. In the frictionless or ideal turbine the velocity issuing from each nozzle or pressure stage is dependent upon the heat drop in the nozzle. If there are n stages the heat drop per stage will be $\frac{1}{n}$ of the total heat drop. Since there are no friction losses in the ideal turbine the total heat drop is

$$H_1 - H_n$$

and the heat drop per stage

$$\frac{H_1 - H_n}{n}.$$

The stage velocity or initial velocity of jet from each nozzle is

$$V = 224\sqrt{\frac{H_1 - H_n}{n}}.$$

The pressure, specific volume and quality of the steam in each stage may be determined by subtracting $\frac{H_1 - H_n}{n}$ from the heat content of the preceding stage and finding the corresponding quantities from temperature-entropy tables or diagrams.

Thus, an eight-stage turbine operating non-condensing at 190 pounds initial absolute pressure would show about the following conditions. (All friction and leakage losses neglected and final velocity in each stage assumed to be zero.)

$$H_1 = 1197.3 \text{ B.t.u. per pound.}$$

$$H_n = 1012.5 \text{ B.t.u. per pound.}$$

$$\text{Total heat drop} = H_1 - H_n = 1197.3 - 1012.5 = 184.8.$$

$$\text{Heat drop per stage} = \frac{184.8}{8} = 23.1.$$

$$\text{Stage velocity} = 224\sqrt{23.1} = 1080 \text{ feet per second.}$$

Stage.	Heat Content.	Pressure, Lb. Abs.	Quality, Per Cent.	Specific Volume Cu. Ft. per Lb.
Admission.	1197.3	190	100	2.41
1	1174.2	145	97.9	3.04
2	1151.1	109	95.9	3.93
3	1128.0	80	94.0	5.14
4	1104.9	58	92.2	6.77
5	1081.8	42	89.6	8.96
6	1058.7	30	88.8	12.07
7	1035.6	21	87.3	16.33
8	1012.5	Atmospheric	85.8	22.55

In the actual turbine only 50 to 75 per cent of the heat theoretically available is transformed into useful work. A small portion is lost by gland leakage, radiation and bearing friction and the balance has been retransformed from kinetic energy into potential energy by eddying, fluid friction and blade leakage. The efficiency of each stage is less than that of the turbine as a whole since the increase in heat content due to friction, etc., is available for transformation into useful work in the succeeding stages. To find the actual pressure condition in each stage allowing for the various losses, it is necessary to correct the theoretical quantities for these losses. See "Energy and Pressure Drop in Compound Steam Turbines," by F. E. Cardullo, Proc. A.S.M.E., Feb., 1911, and paper read by Prof. C. H. Peabody, Proc. Society of Naval Architects and Marine Engineers, June, 1909. Consult also, "The Steam Turbine Expansion Line on the Mollier Diagram and a Short Method of Finding the Reheat Factor," by Edgar Buckingham, Bul. No. 167, 1911, U. S. Bureau of Standards.

210. Terry Condensing Turbine. — The condensing units are of a composite design, namely, a high-pressure compound-velocity element similar to the non-condensing device and a series of single-velocity multi-pressure elements for the low-pressure end. A section through such a unit is shown in Fig. 247. It will be noted that the steam (slightly above atmospheric pressure) leaving the high-pressure element instead of passing directly into the low-pressure stages passes to the other end of the casing and returns through the low-pressure stages to the center. This arrangement maintains a pressure somewhat above atmospheric on the inside of both glands and prevents inward leakage of air. It also insures uniform temperature at both ends of the turbine casing. These units are built in sizes up to 750 kw.

211. Curtis Turbine. — A textbook description of the Curtis line of turbines must necessarily be of a very general nature because of the many changes effected from year to year. A detailed description of a

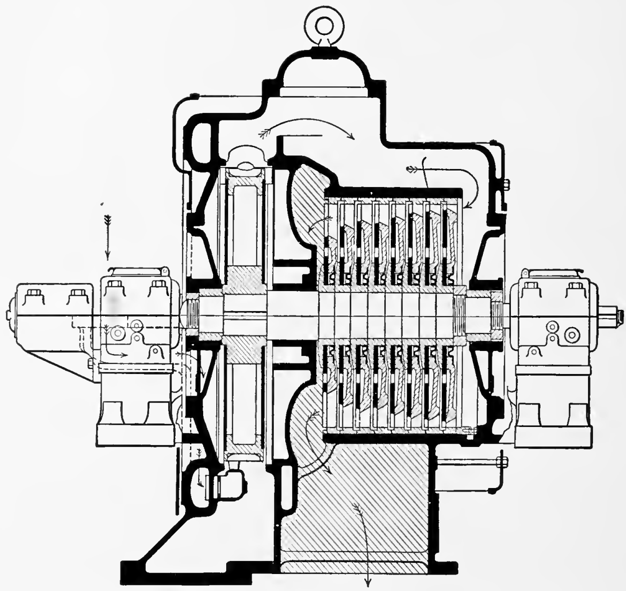


Fig. 247. Section through Terry Condensing Steam Turbine.

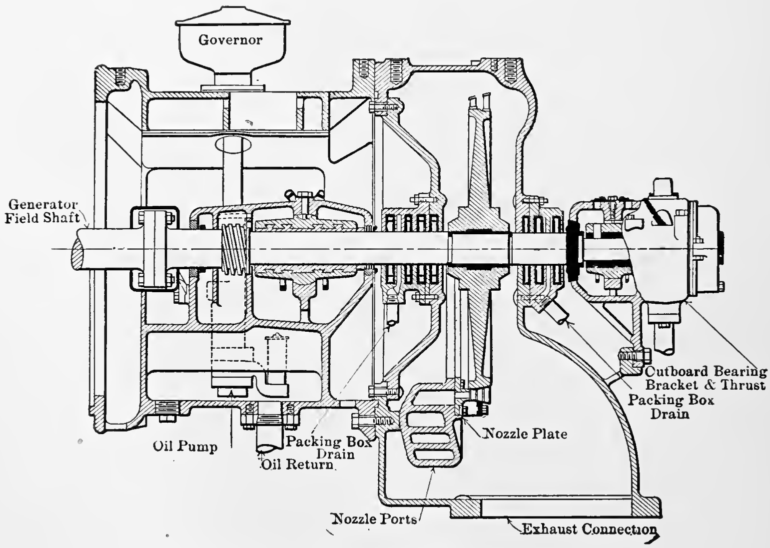


Fig. 248. Curtis Single-stage Turbine.

machine representing current practice may be obsolete in many respects when compared with that of a similar unit constructed a year later. In a general sense the basic principle is the same for all types and sizes, but the structural details, methods of governing, number of stages, and the like vary with the size and the service for which the turbine is intended. All Curtis turbines ranging from the very small direct-current machine to the huge turbo-alternator of 45,000 kilowatts rated capacity are of the impulse type. The small direct-current machines, Fig. 248, have a single-pressure stage, and two or three velocity stages and operate at approximately 5000 r.p.m. The large turbo-alternators have nine or

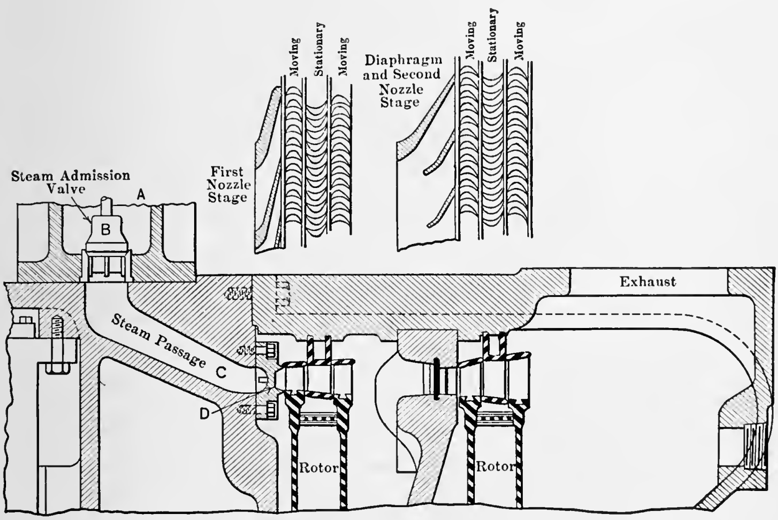


Fig. 249. Arrangement of Nozzle and Blades, Curtis Turbine.

more pressure stages contained in either a single cylinder casing, Fig. 251 or double cylinder casing, Fig. 256 and operate at 1800 r.p.m. The high-pressure stage in practically all sizes comprises a set of nozzles and a single wheel carrying two rows of buckets. The succeeding stages have but one row of buckets on a single wheel, except in the low-pressure element of the compound cylinder units where there are two wheels per stage, each with a single row of buckets. In the mixed pressure type there are two rows of buckets on each wheel and in the small, two-stage turbo-oil pump for circulating the main bearing oil there is but one velocity for each pressure stage. In all machines the steam flow is axial. In the single cylinder units the flow is unidirectional but the low-pressure member of the compound cylinder machine is arranged for double flow, that is, the steam enters at the center of the low-pressure turbine and flows through double stages on either side of the condenser as shown in Fig. 251.

A comparatively high initial velocity is given to the jet in each pressure stage by expansion in the nozzle and the energy is absorbed by successive action upon a series of moving and stationary vanes. Since expansion takes place only in the nozzles the pressure in any stage is the same on both sides of the wheel. The action of the steam is as follows (Fig. 249): Entering at A from the steam pipe it passes through one or more admission valves B into the bowls C. The number of admission valves depends on the load and their action is controlled by the governor. From bowls C the steam expands through nozzles D and impinges against the first row of moving blades and gives up part of its energy. The steam flowing from the first row of moving blades is reversed in direction by the adjacent stationary vanes and is redirected against the second set of moving blades where it gives up its remaining kinetic energy. From this stage the steam flows at reduced pressure through the nozzles of the second stage which are sufficient in number and size to afford the greater area required by increased volume. In expanding in these nozzles it acquires new velocity and gives up energy to the moving blades as before. This process is repeated through several additional stages.

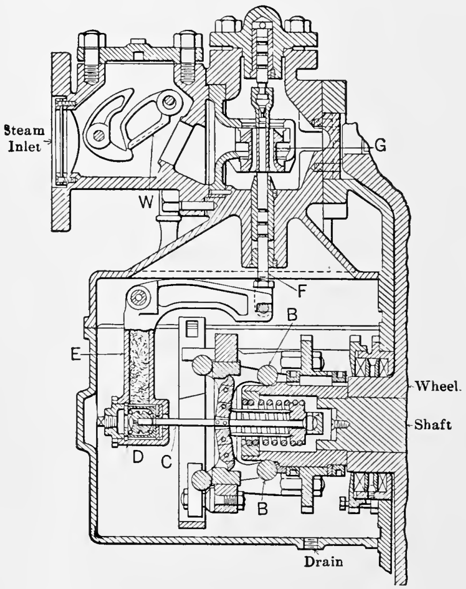


Fig. 250. Throttling Governor Mechanism for 25-kw., 3600-r.p.m., Direct-current Curtis Turbo-Generator.

The rotor consists of 1 to 13 or more steel disks mounted side by side on a horizontal shaft. In some of the earlier designs the shaft was mounted vertically but this construction has been discontinued. Buckets or vanes of nickel steel, monel metal or nickel bronze, according to the condition of the steam, are secured to the periphery by a dovetail-shaped root which fits snugly in a channel of the same section machined in the rim. The types of the vanes are tenoned and riveted

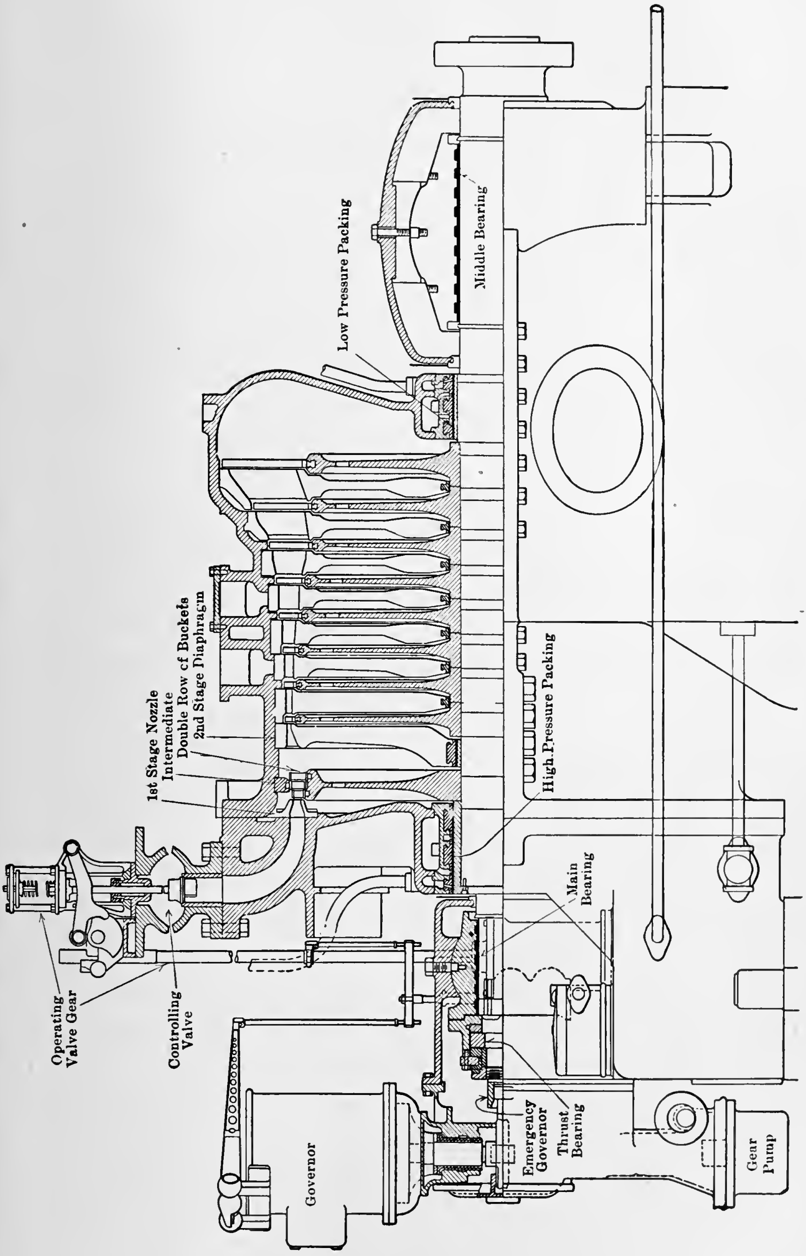


FIG. 251. Assembly of 12,500-kw., Single-cylinder Curtis Steam Turbine, 1800 r.p.m., 12,000 volts.

into a shroud ring. The stationary vanes are secured to the casing as illustrated in Fig. 249. Between the revolving wheels is a stationary steam-tight diaphragm which contains the nozzles through which the steam is expanded from the preceding stage. It will be seen from Fig. 249 that vanes and nozzles increase in size in succeeding stages as the

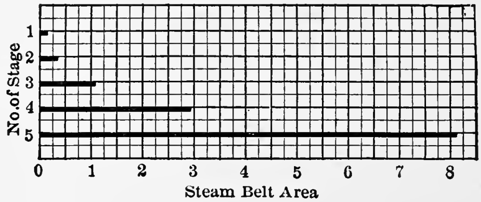


FIG. 252. Steam-belt Area in Five-stage Curtis Turbine.

pressure falls and the volume increases. The parts are so proportioned that the steam gives up approximately $1/n$ of its energy in each pressure stage, n representing the number of stages. The number of stages and the number of vanes in each stage are governed by the degree of expansion, the peripheral velocity which is practical or desirable, and by various conditions of mechanical expediency. The nozzles extend

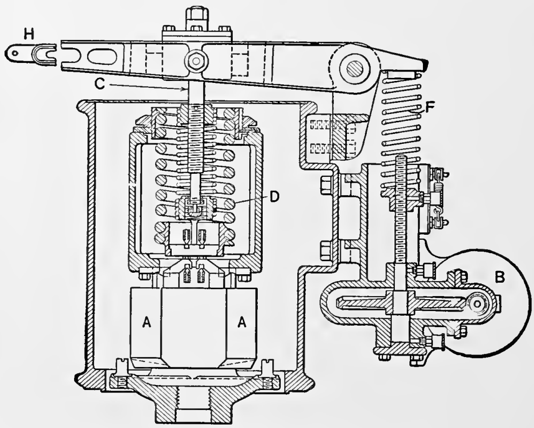


FIG. 253. Main Operating Governor.

around a relatively short arc in the periphery of the first stage and increase progressively in number until they extend around the entire wheel in the last stage. See Fig. 252.

In the smaller machines the speed is controlled by a centrifugal governor mounted on the end of the main shaft. The governor actuates a throttling valve of the balanced poppet-valve type. The larger sizes are controlled by an indirect or relay system. Fig. 250 shows an assembly of the simple throttling governor. The movement of governor weights B is transmitted through spindle C and thrust ball D to bell crank E, which in turn operates the valve stem F and double balanced poppet valve G. The valve is shown in wide open position. Two emergency governors of the clock-spring type are mounted on the outer ring of the main governor. These springs rest under tension against a stop and when the turbine exceeds 10 per cent of the normal

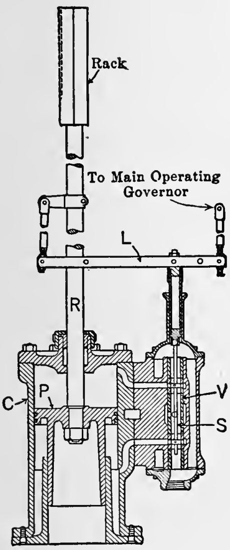


FIG. 254. Assembly of Hydraulic Cylinder.

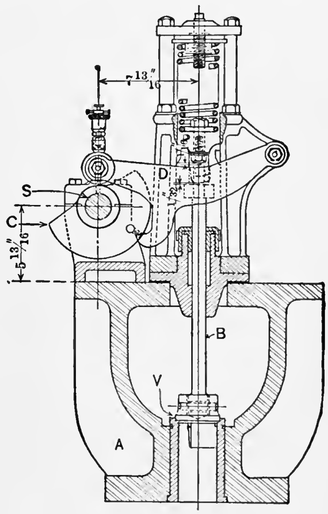


FIG. 255. Main Controlling Governor.

speed will strike a trigger and release flap valve W, thus shutting off the supply of steam.

The large turbo-alternators are controlled by a relay governor of the hydraulic type. This mechanism consists of a cylinder to which oil under pressure is fed through a pilot valve under the control of the main governor. The piston rod of the cylinder contains a rack which meshes with a pinion on a cam shaft and so rotates the shaft. The cams on the shaft lift the individual controlling valves as determined by the angular spacing of the cams. The general assembly is shown in Fig. 251 and the general details of the main governor, cylinder and one of the controlling valves in Figs. 253 to 255 respectively.

Referring to Fig. 253 speed regulation is accomplished by the balance maintained between the centrifugal force of moving weights AA and the static force exerted by spring D. The governor is provided with an auxiliary spring F for varying its speed when synchronizing, the

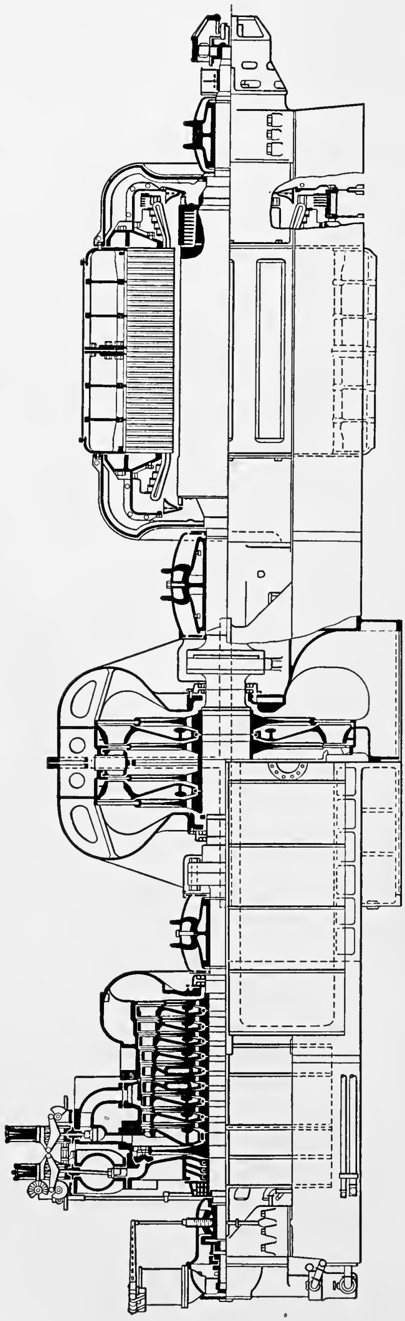


Fig. 256. 30,000-kw. Curtis Turbo-generator, Northwest Station, Commonwealth Edison Co.

tension of which is varied by a small pilot motor controlled from the switchboard. The movement of the governor weight is transmitted through rod *C* to arm *H* and by means of the latter to floating lever *L*, Fig. 254. This floating lever is pivoted on a clamp attached to the pilot valve stem *S*. The other end of the lever is connected by links to the piston rod *R* of the operating cylinder. A movement of governor arm displaces the small pistons of the pilot valve from their normal location in which they close the ports of the cylinder. This displacement causes oil to be admitted to the cylinder and the pressure of the oil operates the main piston. The piston rod opens and closes the controlling valves through the agency of the cam shaft and at the same time transmits its motion through the link system to the end of the floating lever and thus brings the pilot valve back to its normal position. Each position of the governor determines a definite position of the piston in the operating cylinder and consequently the opening of a definite number of controlling valves. The general details of one of the controlling valves is shown in Fig. 255.

The emergency governor or stop consists of a ring *R* (Fig. 257), unevenly weighted, attached to and revolving with the shaft. At normal speeds and less, the unbalanced ring is held concentric with the shaft by helical springs S. When the speed increases to 10 per cent above normal the centrifugal force of the unbalanced portion of the ring overcomes the spring tension, and the ring revolves eccentrically. In this position the ring strikes a trip finger and closes the main throttle valve which is of the balanced type.

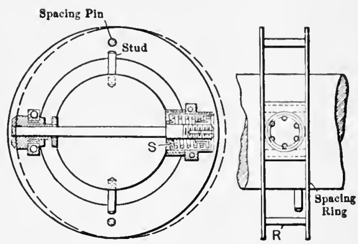


Fig. 257. Emergency Governor.

Another type of governor used on the smaller machines is shown assembled in Fig. 258. In this arrangement the main governor arm A, Fig. 259, actuates a small steam pilot valve. The latter in turn moves a piston on the stem of which are mounted several valve hangers designed to raise the various controlling valves successively with the upward travel of the piston. The valves are of the double-seated poppet type, annular in shape, free to move, but guided and controlled by the valve hangers. Referring to Fig. 259 with turbine and governor at rest and no steam bled to the pilot valve, the position of the various

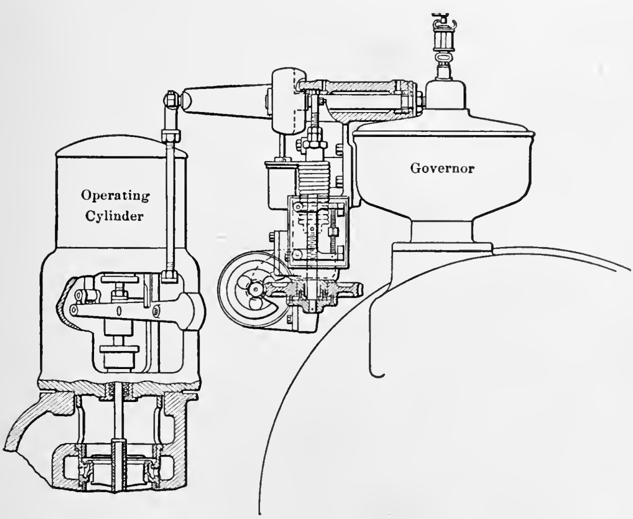


Fig. 258. Assembly of Governor and Operating Cylinders for Steam Relay Control.

levers is such that the pilot valve is in a position to admit steam to the under side of the piston for operating the main valves. With the opening of the throttle the turbine speeds up and the governor mechanism moves upward, the connection to floating lever L moves downward and the latter fulcruming on pin *B* moves the pilot valve *V* upward to a position which shifts the admission of steam from the under to the upper side of piston *P*, closing the main valves successively until the governor assumes a position of equilibrium. Movement of

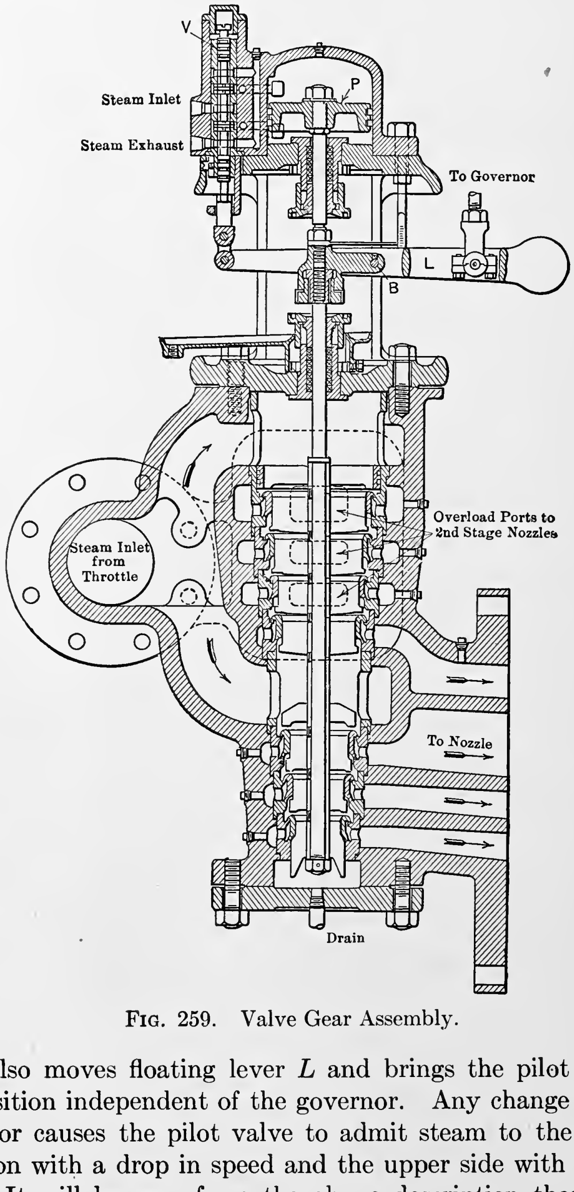


FIG. 259. Valve Gear Assembly.

piston *P* also moves floating lever *L* and brings the pilot valve in a neutral position independent of the governor. Any change in speed of the governor causes the pilot valve to admit steam to the under side of the piston with a drop in speed and the upper side with an increase in speed. It will be seen from the above description that throttling is practically eliminated since all but one of the valves is either in the wide open or shut position, and the over travel of the piston during a

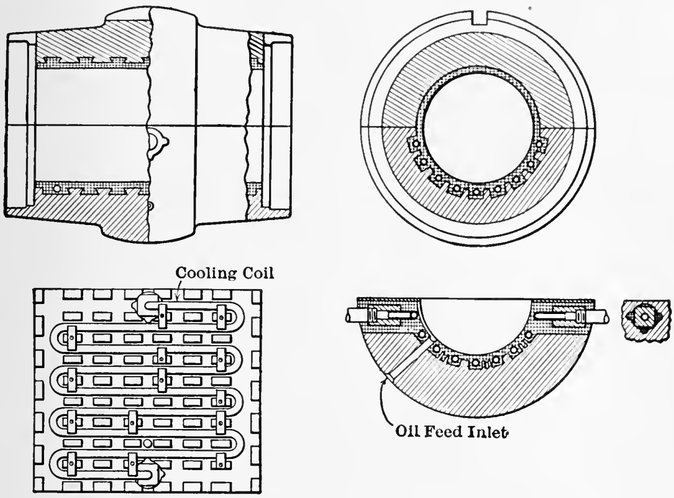


Fig. 260. Water-cooled Bearing, Horizontal Curtis Turbine.

small variation in load causes periodic opening and closing of the individual valve.

The main bearings as well as the governor are supplied with forced lubrication from an oil pump bolted to the main pillow block. The oil is cooled by a current of water flowing through the main bearing linings, the bottom halves of which are equipped with a number of copper coils imbedded in the babbitt as shown in Fig. 260. In some designs the oil pump is of the geared type while in others it is driven by a small independently operated steam turbine.

The general details of the main thrust bearing are shown in Fig. 259. The drawing is self-explanatory.

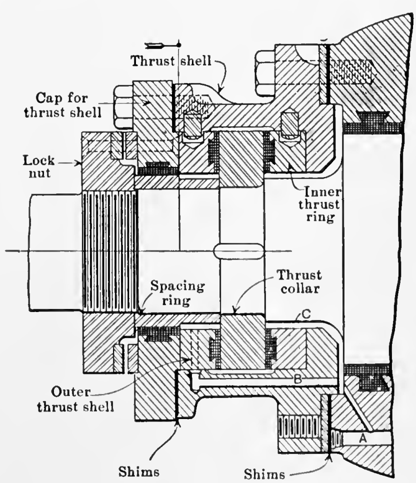


Fig. 261. Details of Thrust Bearing.

Fig. 262 shows a general assembly of the main shaft packing for the ends of the casing and for one intermediate. The packing rings are of carbon and are self-lubricating. Each ring is composed of three segments held against the shaft by radial springs. There are usually four rings in the high-pressure head, three in the low-pressure head and a single ring in the intermediates. For condensing service the heads and packing chambers are sealed with live steam to prevent

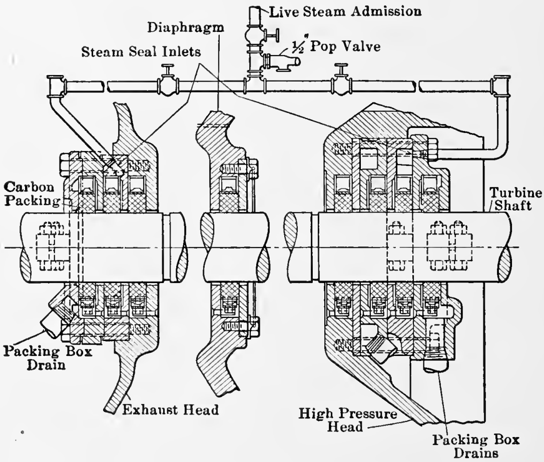


Fig. 262. Assembly of Packing.

air leakage into the casing when the pressure inside is less than atmospheric and to prevent the escape of steam when the pressure in the casing is greater than atmospheric.

Although there are numerous Curtis turbines of the Curtis vertical shaft type in successful operation this design has been discontinued and no attempt will be made to describe them. The mechanical valve employed in some of the earlier designs has also been abandoned.

212. Elementary Theory, Curtis Turbine. — Fig. 263 gives a diagrammatic arrangement of the blades and nozzles in the first stage of a two-stage Curtis turbine, each stage consisting of one set of nozzles and two moving and one stationary sets of blades.

Referring to the diagram: the steam is expanded in the first stage from pressure P_1 to P_2 and issues from the first set of nozzles with *absolute* velocity V_1, striking the first set of moving blades at an angle α with the line of motion of the wheel. The resultant v_1 of V_1 and the peripheral velocity u is the velocity of the steam *relative* to the vanes; and the angle β which the line v_1 makes with the line of motion of the wheel is the proper entrance angle of the blades for the first set. Neglecting friction the exit angle γ will be the same as the entrance angle β. The resultant of v_2, the exit velocity *relative* to the blade, and u, the peripheral velocity, is V_2, the *absolute* exit velocity.

Since the second set of blades is fixed and serves as a means of changing the direction of flow, the absolute velocity entering them is V_2. The angle δ formed by V_2 and the center line of the stationary blades is the proper entrance angle. Neglecting friction the absolute exit velocity will be $V_3 = V_2$, and the exit angle will be $\epsilon = \delta$. The steam flowing from the stationary blades strikes the second set of moving blades at an angle $\epsilon = \delta$ with *absolute* velocity V_3. Combining V_3 with the peripheral velocity u we get v_3, the velocity of the steam *relative* to the second set of moving blades. The angle θ, formed by v_3, and the line of motion of the wheel, is the proper entrance angle for the second set of moving blades. The resultant of v_4 $(= v_3)$ and u is V_4, the absolute exit velocity for the first stage.

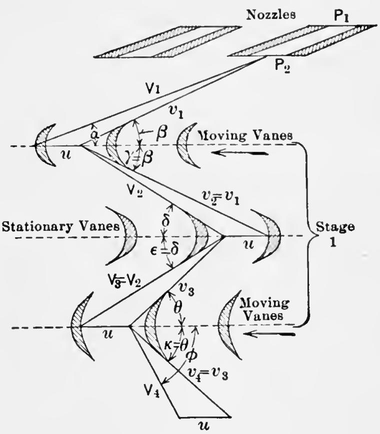


FIG. 263. Velocity Diagram, Curtis Turbine.

In the second stage the steam is expanded from pressure P_2 to that in the condenser and acquires initial velocity V_a, leaving the last bucket with residual velocity V_n. The theoretical velocities and blade angles for this stage may be found as above.

Example 32. A four-stage Curtis turbine develops 800 horsepower on a steam consumption of 12 pounds per horsepower-hour; initial pressure 150 pounds absolute, superheat 100 deg. fahr., back pressure 1.5 pounds absolute, peripheral velocity 450 feet per second, angle of the nozzle with the plane of rotation, 20 degrees. Each stage consists of two rotating elements and one stationary element. Compare the performance of the actual turbine with its theoretical possibilities.

Ideal Turbine:

For the sake of simplicity it will be assumed that the final velocity of each stage is zero and that the heat drop in the first set of nozzles is one-fourth of the total theoretical drop assuming adiabatic expansion.

From steam tables $H_1 = 1249.6$ B.t.u.

From entropy tables or Mollier diagram $H_n = 934.6$.

Total heat drop $= 1249.6 - 934.6 = 315$.

Heat drop in first stage $\frac{315}{4} = 78.75$.

The velocity of the jet in the first stage is

$$V_1 = 224\sqrt{78.75} = 1985 \text{ feet per second.}$$

By laying off this initial velocity in direction and amount and combining it with the peripheral velocity as in Fig 263, the absolute velocities V_2 and V_3 may be readily obtained.

The kinetic energy absorbed in the first set of moving blades, per pound of steam, is

$$E_1 = \frac{1}{64.4}(V_1^2 - V_2^2)$$

$$= \frac{1}{64.4}(1985^2 - 1170^2) = 39{,}930 \text{ foot-pounds per second,}$$

and in the second set of moving blades

$$E_2 = \frac{1}{64.4}(V_3^2 - V_4^2)$$

$$= \frac{1}{64.4}(1170^2 - 670^2)$$

$$= 14{,}280 \text{ foot-pounds per second.}$$

The total energy converted into useful work is

$$39{,}930 + 14{,}280 = 54{,}210 \text{ foot-pounds per second.}$$

Had the entire heat drop been utilized in doing work the total energy would be

$$\frac{1}{64.4} \times 1985^2 = 61{,}180 \text{ foot-pounds per second.}$$

The difference $61{,}180 - 54{,}210 = 6970$ represents the loss due to the residual velocity of the steam leaving the last bucket.

Since the steam is brought to rest before entering the second set of nozzles, the heat equivalent of this energy or $\frac{6970}{778} = 8.96$ B.t.u. increases the final heat content; thus

$$H_2 = 1249.6 - 78.75 + 8.96 = 1179.8 \text{ B.t.u.}$$

But a total heat drop per stage of 78.75 B.t.u. was assumed as a requirement of the problem and the final result obtained above shows it to be $78.5 - 8.96 = 69.54$. By trial and adjustment or by means of empirical formulas a value of H_2 may be obtained which will fulfill the given conditions. Such an analysis is beyond the scope of this book, and the reader is referred to Forrest E. Cardullo's article "Energy and Pressure Drops in Compound Steam Turbines," Trans. A.S.M.E., vol. 33, p. 325, 1911.

The remaining stages may be analyzed in a similar manner.

It should be borne in mind that in the actual turbine the velocity will be less than the theoretical on account of frictional resistances in the nozzles and blades and the heat content $H_1, H_2 \ldots H_n$ will be greater than that of the ideal mechanism. Radiation, leakage, windage and other losses must also be considered in determining actual conditions.

Neglecting the residual energy in the exhaust, the total heat drop $H_1 - H_n$ is available for doing useful work and the water rate of the ideal turbine is

$$W = \frac{2546}{H_1 - H_n} = \frac{2546}{315} = 8.1 \text{ pounds per horsepower-hour.}$$

Heat consumption per horsepower per minute

$$= \frac{8.1\,(1249.6 - 83.9)}{60} = 157 \text{ B.t.u.}$$

Thermal efficiency

$$E_r = \frac{1249.6 - 934.6}{1249.6 - 83.9} = 0.27.$$

Actual Turbine:

Steam used per hour $= 800 \times 12 = 9600$ pounds.

Steam used per second $= 9600 \div 3600 = 2.66$ pounds.

Horsepower developed per pound of steam flowing per second $= 800 \div 2.66 = 300$.

Kinetic energy converted into useful work:

$$300 \times 550 = 165{,}000 \text{ foot-pounds per second.}$$

Thermal efficiency

$$E_t = \frac{2546}{12\,(1249.6 - 83.9)} = 0.182.$$

Heat consumption, B.t.u. per horsepower per minute,

$$\frac{12\,(1249.6 - 83.9)}{60} = 233.$$

$$\text{Rankine cycle ratio} = \frac{E_t}{E_r} = \frac{0.182}{0.270} = 0.675.$$

213. Westinghouse Single-flow Reaction Turbine. — The Westinghouse-Parsons single-flow reaction turbine was one of the first reaction turbines in successful use in this country. This particular type of machine is no longer constructed, though the modern Westinghouse non-condensing reaction turbine and the high-pressure element of the large compound units are modifications of the conventional Parson design. The reaction turbine is always a multi-pressure-stage machine with small pressure drop per stage. Each stage consists of a stationary set of blades or nozzles and a row of rotating vanes or buckets. The stationary blades are inserted radially into circumferential grooves in the main cylinder or are carried in separate blade rings which are centered within the cylinder. The rotating vanes are mounted in rows on a steel barrel or drum and when in operating position revolve between the rows of fixed blades on the stator. Theoretically each set of stationary and moving buckets should either continually increase in height from one end to the other to accommodate the increasing volume of steam, or, with equal heights of blades the equivalent nozzle area should gradually increase. Practically, for constructive reasons, it is preferable to subdivide the stages into three divisions, each with a different pitch diameter, and to arrange each division into a number of groups. The blades in each group are of the same height and shape but are so assembled or "gagged" that the equivalent nozzle area gradually increases as the steam volume increases. The entire expansion is effected in the annular compartment between rotor and stator and resembles in effect a single divergent nozzle with the exception that the dynamic relationship of jet and vane is such as to secure a comparatively low velocity from inlet to outlet. The action of the steam on the blades is illustrated in Fig. 264. Steam is expanded in the first row of stationary blades from pressure P to P_1 and accelerates the jet. The velocity of the jet issuing from these stationary nozzles is such that steam enters the adjacent set of moving blades practically without impulse. The steam expands from pressure P_1 to P_2 in passing through the first set of moving blades and exerts a reactive force on the blades. The jet, with low residual velocity, is deflected from the moving blades to the entrance of the second set of stationary nozzles. In this second set of stationary nozzles the steam is expanded from pressure P_2 to P_3 and the jet strikes the second set of moving blades. This process is repeated in each element of the turbine, the steam expanding as it flows from element to element in its passage to the condenser. It will be seen that the rotating force is primarily due to reaction though there may be some impulse when the jet strikes the moving members. Since all reaction turbines are subject to an axial end thrust of the rotating parts due to the difference of steam pressures at each end of the drum, provision must be made for resisting this thrust. In the original Westinghouse-Parsons turbine this was effected by balancing pistons or "dummies" mounted on the rotor, running with close clearance to the casing, and of the same diameter as the overall diameter of each drum. Each dummy is then subjected to the same difference of pressure as the rotating drums by means of equalizing pipes. In the modern single-flow design there is but one balancing piston and the thrust is taken up by a suitably designed thrust bearing. A section through a modern Westinghouse

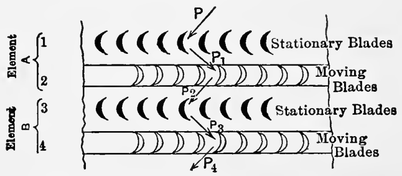


FIG. 264. Blade Arrangement, Reaction Turbine.

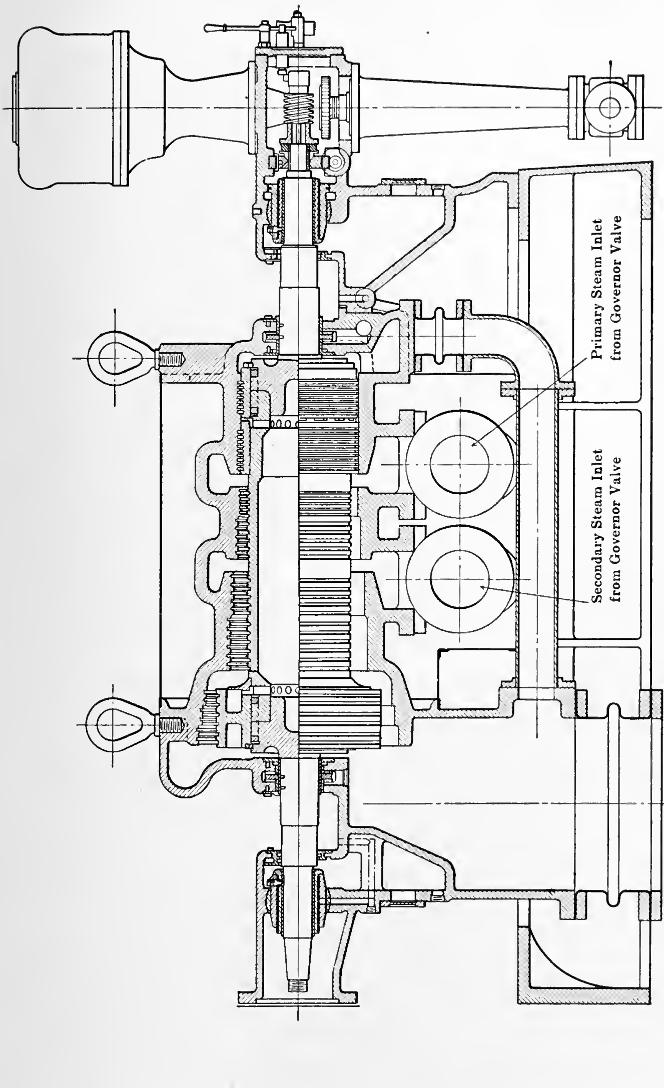


Fig. 265. Section Through a Westinghouse Non-condensing Reaction Turbine.

non-condensing reaction unit is illustrated in Fig. 265. It will be seen that grooves are cut in the surface of the dummy in which corresponding collars on the turbine mesh, although they run without actual metallic contact. The dummy is split and the openings lead to the interior of the spindle. Thus the steam leaking past the inner half of the balancing piston is conducted through the spindle to the low-pressure stages and does work in the low-pressure cylinder. An equilibrium pipe connects the chamber housing the balance piston with the exhaust chamber and serves to equalize the pressure at both ends

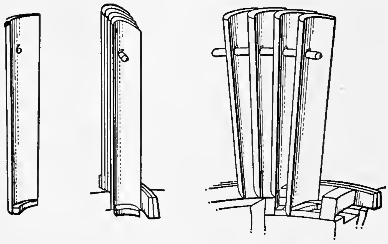


Fig. 266. Method of Fastening Reaction Blading.

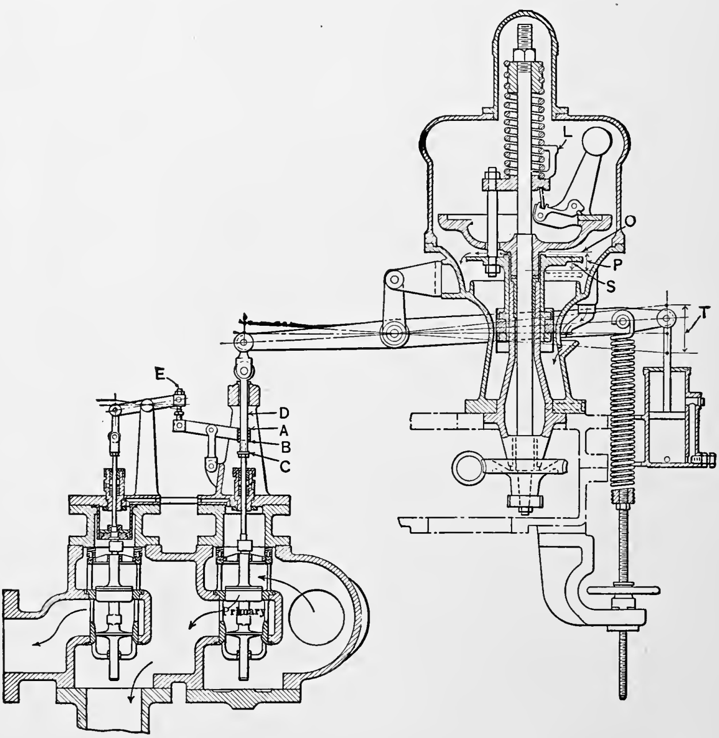


Fig. 267. Assembly of Governor Mechanism, Oil Relay Control.

of the spindle and at the same time permits steam leakage past the outer half of the piston to escape into the exhaust.

Fig. 266 shows the method of fastening the roots of the blades and of bracing the upper ends. It will be seen that dovetailed packing pieces placed between the blades and on top of the upset root provide an interlocking system with which no caulking is necessary. This arrangement makes it possible to replace the blade without mutilating the blade-carrying member.

Fig. 267 shows an assembly of the main governor mechanism. The movement of the governor weights is transmitted through suitable linkage to lever *L* which in turn actuates rocker *R*. Flat-faced cam *C* and vibrator rod *B* impart a slight but continuous reciprocating motion to lever *L* and thus overcome the friction of rest. Rocker *R* controls

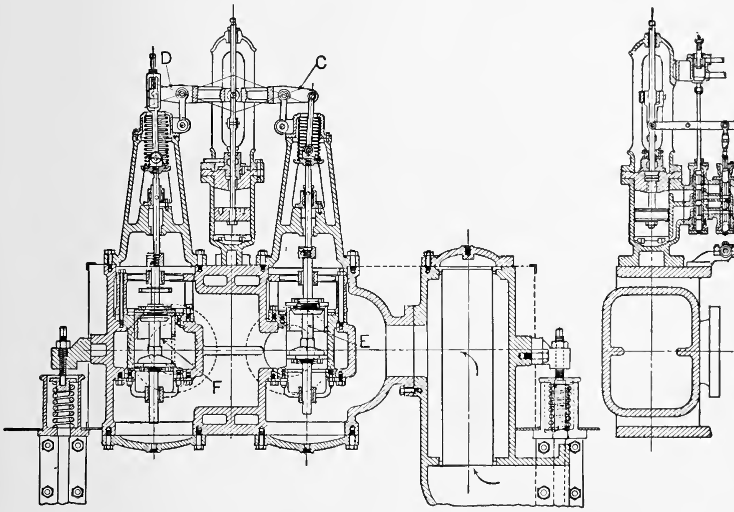


Fig. 268. Oil Relay Valve Gear.

a small pilot valve *V* which admits oil under pressure to or exhausts it from the admission valve operating cylinder. Fig. 268 shows the general details of the oil relay gear. Relay valve *A* is controlled by the governor and admits or exhausts oil from the operating cylinders. When oil is admitted to the operating cylinder raising the piston, the lever *C* lifts the primary valve *E*. The lever *D* moves simultaneously with *C* but on account of the slotted connection with the steam of the secondary valve *F* the latter does not begin to lift until the primary valve is raised to the point at which its effective opening ceases to be increased by further upward travel. The secondary valve admits steam to the intermediate section and enables the turbine to carry about 50 per cent more load than on the primary valve alone. A steam relay gear is also used with this type of turbine. The steam chest in this case contains only one valve — the primary valve — which is operated by the steam relay gear. All capacities in excess of that of the primary valve are carried by means of a hand-operated secondary valve. All turbines are equipped with an automatic governor stop which shuts off the main steam supply when the turbine speed exceeds a predetermined limit.

In the smaller sized machine running at 3000 r.p.m. or more, flexible bearings are employed to absorb the vibration incident to the *critical*

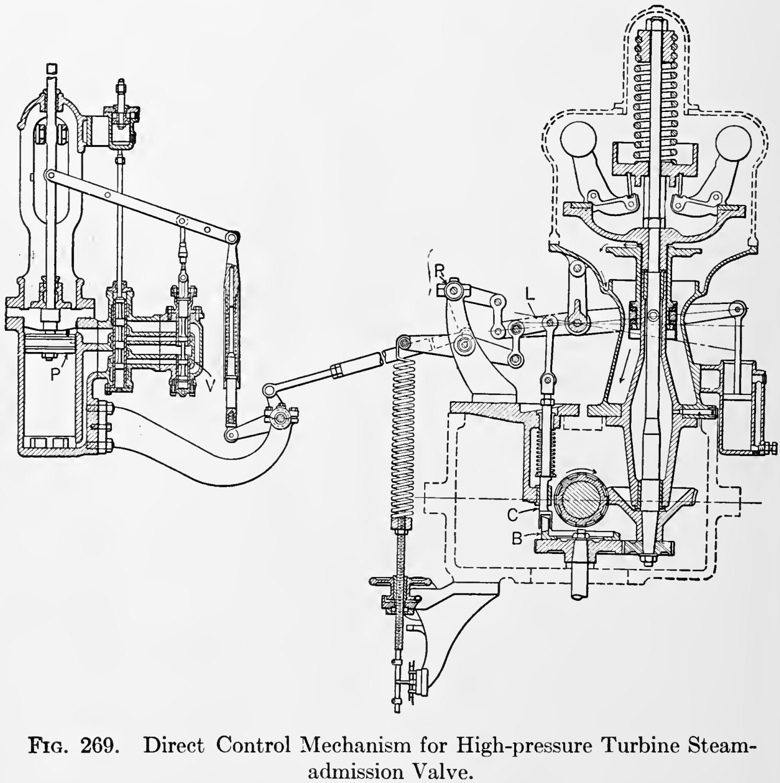


Fig. 269. Direct Control Mechanism for High-pressure Turbine Steam-admission Valve.

velocity. They consist of a nest of loosely fitting concentric bronze sleeves with sufficient clearance between them to insure the formation of a film of oil. In the larger and lower speed machines a split self-aligning babbitted bearing is used instead of the flexible bearing. Before the babbitt is run in, a large copper tube is placed in a groove cast in the shell. This tube receives the oil and delivers it to the top of the bearing.

A closed oiling system is maintained by means of a pump geared to the main shaft of the turbine. The oil, after it drains from the bearing, passes through a strainer into a collecting reservoir whence it is pumped through a cooler and back to the bearings. In turbines in which the oil-relay governing system is employed, and a higher pressure is maintained by the pumps, the comparatively small quantity of oil required for operating the valve mechanism passes to the relay cylinder, from which it exhausts to the cooler. In the larger machines an auxiliary oil pump is furnished for establishing a circulation with the turbine at rest.

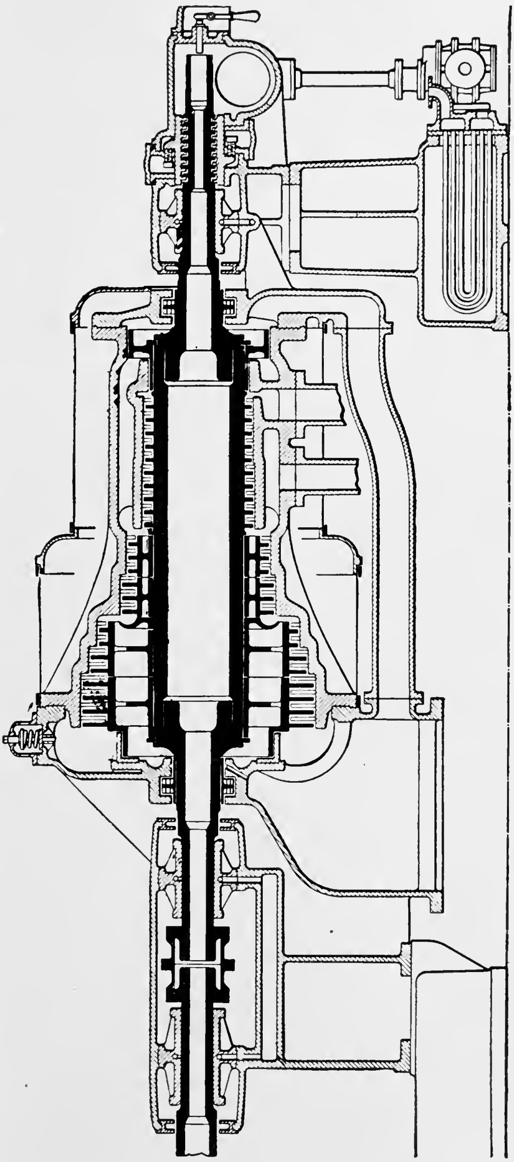


Fig. 270. Section through Allis-Chalmers Steam Turbine.

The glands on both the non-condensing and the condensing units are water sealed. This seal is effected by small bronze impellers fitted on either end of the turbine shaft and which revolve in annular chambers. Water is fed into these chambers and the centrifugal action of the impellers maintains a pressure which effectually seals the glands against air leakage into the casing or steam leakage into the atmosphere. The amount of sealing water required is very small. The grooves and mating collars on the balancing piston constitute a *labyrinth packing*.

214. Allis-Chalmers Steam Turbine. — Fig. 270 shows a section through an Allis-Chalmers standard steam turbine, which is of the

Parsons type but differs from the original Parsons machine and the Westinghouse-Parsons construction principally in manufacturing details. In the older Parsons type, three balance pistons are placed at the high-pressure end. In the Allis-Chalmers design, the larger piston is placed at the low-pressure end of the rotor, behind the last row of blades, the other two remaining at the high-pressure end. This construction permits of a smaller balance piston and allows a smaller working clearance in the high-pressure and intermediate cylinders. In the Allis-Chalmers turbine the roots of the blades are dovetailed and fitted into a foundation ring, and the tips are incased in a channel-shaped shroud ring, thereby insuring a rigid and positively spaced construction. The governor is of the Parsons type, except that the main valve and pilot valve are actuated by hydraulic instead of steam pressure. The bearings are of the self-adjusting ball-and-socket pattern and are kept "floating in oil" by a small pump geared to the turbine shaft. The oil is passed through a tubular cooler with water circulation after it leaves the bearings and is used over and over again.

215. Westinghouse Impulse-reaction Turbine. — With the exception of a non-condensing unit and the purely impulse type described in paragraph 204 all high-pressure single-cylinder turbines constructed by the Westinghouse Company are of the combined impulse and reaction types. A typical unit is illustrated in Fig. 271. There are two rows of moving blades or buckets upon the impulse wheel with an intermediate set of reversing blades, the operation being practically the same as in the first stage of the Curtis turbine. The drop in pressure in the nozzles is such that approximately 20 per cent of the total energy developed is absorbed by the impulse element. The steam discharged from the impulse element is expanded through the reaction elements in the usual manner. The substitution of the impulse element for the high-pressure section of reaction blading has no influence on the efficiency but results in a shorter machine and gives a more rigid design of rotor. From Fig. 271 it will be seen that the cylinder has been shortened not only by the substitution of the narrow impulse element for a comparatively wide section of reaction blading but also by the elimination of the intermediate balancing pistons or dummies as used in the conventional Parsons design. A further inspection of Fig. 271 will show that the glands on each end of the cylinder are subjected to exhaust pressure and that leakage of air into the turbine casing is prevented by the water-sealing device described in the preceding paragraph. Steam leakage past the balancing piston through the labyrinth packing escapes into the exhaust. Fig. 272 shows a section through a double-flow impulse-reaction turbine which differs from the

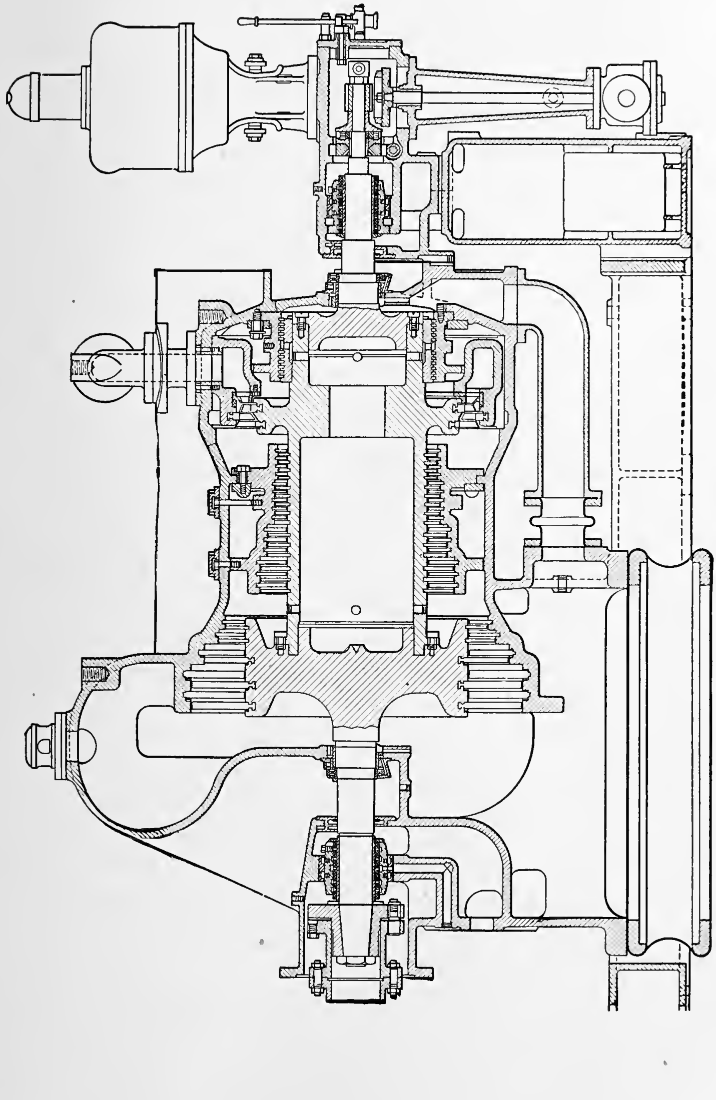


Fig. 271. Section through a Combined Impulse and Reaction Single-flow Westinghouse Turbine.

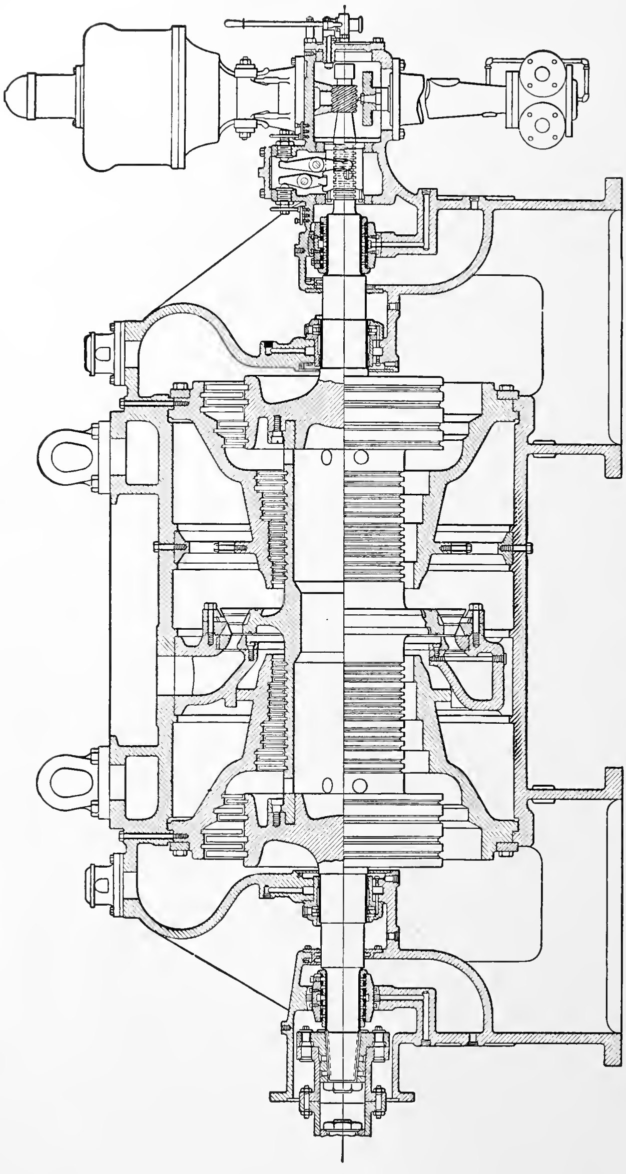


Fig. 272. Section through a Double-flow High-pressure Westinghouse-Parsons Condensing Turbine.

one illustrated in Fig. 271 by the use of reaction elements on either side of the impulse wheel. It will be seen from Fig. 272 that the double-flow is shorter than the single-flow machine by a length equivalent to that of the balance piston. Single-cylinder, double-flow impulse-reaction turbines have been constructed in sizes up to 22,000 kilowatts.

216. Westinghouse Compound Steam Turbines. — In a turbine maximum centrifugal stresses occur at the exhaust end where the large volume of steam requires the greatest blade area. In the high-pressure blading involving the use of high-density steam the best velocity ratio conducive to high economy cannot be met by the rotative speed as determined by the exhaust end. To avoid a compromise with its resulting reduced efficiency the expansion is carried out in two or more separate elements. All of the modern large turbines built by the Westinghouse Company are of the multi-cylinder type. The two-cylinder machines are arranged either tandem or cross compound. Three-cylinder cross-compound units have also been built and it is not unlikely that the four-cylinder construction may be used for very large units. The advantages of the multi-cylinder construction over a single cylinder of like capacity are as follows:

1. Smaller cylinder structure.
2. Lower temperature range within the cylinder.
3. Highest efficiency for each cylinder for the expansion of range involved.
4. Reduction in weight of the parts to be handled.
5. Possibility, in case of emergency, to operate either cylinder alone.

In the Westinghouse compound turbine the high-pressure element is practically a typical single-cylinder reaction turbine and the low-pressure element is a reaction turbine of the double-flow type. The high-pressure element of the 30,000-kw. turbine at the 74th Street Station of the Interborough Rapid Transit Company, N. Y., operates at 1500 r.p.m. and the low-pressure element at 750 r.p.m.

217. Elementary Theory.— Reaction Turbine.—Fig. 273 gives a diagrammatic arrangement of fixed and stationary blades in the first stage of a multi-stage reaction turbine. The steam enters the stationary blades at a comparatively low initial velocity and is there partially expanded and impinges against the moving blades at velocity V_1. In practice V_1 is made such that there is practically no impulse when the jet strikes the vanes. In passing through the moving vanes the steam is further expanded and leaves at absolute velocity V_2, exerting a reactive force on the rotor. The steam enters the second set of stationary blades with absolute velocity V_2 and is still further expanded to velocity V_3, and so on.

The energy imparted to the steam in the first set of stationary blades is

$$E_1 = \frac{W}{778}(H_1 - H_2) = \frac{W}{2g}V_1^2, \tag{187}$$

in which

H_1 = initial heat content, B.t.u. per lb.,
H_2 = heat content after expansion through the blades, B.t.u. per lb.,
W = weight of steam, lb. per sec.,
V_1 = velocity imparted to the jet by expansion.

The absolute spouting velocity $V_s = V_0 + V_1$, in which V_0 = entrance velocity to the fixed blades.

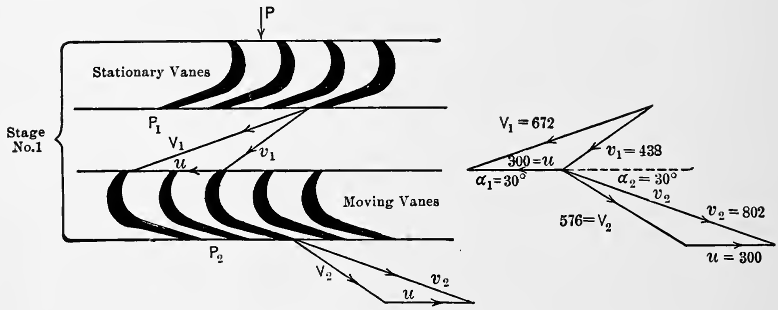


FIG. 273. Velocity Diagram, Westinghouse Reaction Turbine.

The energy imparted to the steam in the first set of moving blades is

$$E_2 = \frac{W}{2g}(v_2^2 - v_1^2), \tag{188}$$

in which

v_1 = relative velocity of steam entering the moving blades,
v_2 = relative velocity of steam leaving the moving blades.

The total energy available in the first stage is $E_s + E_2$, in which E_s = kinetic energy of the jet leaving the stationary vanes$\left(E_s = \frac{W}{2g}V_s^2\right)$.

The energy converted into useful work in this stage is

$$E = E_s + E_2 - \frac{W}{2g}V_2^2$$
$$= (V_s^2 + v_2^2 - v_1^2 - V_2^2)\frac{W}{2g}. \tag{189}$$

V_2 = absolute velocity of the steam leaving the moving blades. This residual velocity will also be the initial entrance velocity of the second stage.

Each stage may be analyzed in a similar manner.

Example 33. Construct the velocity diagram and calculate the work done per stage in a frictionless reaction turbine for the following conditions: Heat drop per stage = 18 B.t.u. per lb. of steam; peripheral velocity = 300 ft. per second; exit angle = 30 deg.; entrance velocity = zero.

The velocity imparted to the steam in the first set of stationary blades is

$$V_1 = 224\sqrt{\frac{18}{2}} = 672 \text{ ft. per sec.}$$

The spouting velocity is

$$V_s = V_1 = 672 \text{ ft. per sec.}$$

Lay off V_s in direction and amount and combine with $u = 300$, Fig. 273. The resultant is v_1, the velocity of the steam relative to the blades. The angle between v_1 and the line of motion of the wheel will be the entrance blade angle. From the diagram $v_1 = 438$. The energy given up by expansion in the moving blades is

$$E = 778 \times \frac{18}{2} = 7002 \text{ ft. per sec.}$$

Substituting $v_1 = 438$ and $E_1 = 7002$ in equation (188)

$$7002 = \frac{1}{64.4}\left(v_2^2 - \overline{438}^2\right),$$

$$v_2 = 802 \text{ ft. per sec.}$$

The resultant of v_2 and u is V_2, the residual velocity of the steam leaving the moving blades. From the diagram $V_2 = 576$.

The energy converted into work in the first stage is from equation (189)

$$E = \left(\overline{672}^2 + \overline{802}^2 - \overline{438}^2 - \overline{576}^2\right)\frac{1}{64.4}$$

$= 10,420$ ft. lb. per sec. for each lb. of steam flowing through the turbine.

In the actual turbine the various friction and leakage losses must be included in the calculation. Such an analysis is beyond the scope of this text and the reader is referred to the accompanying bibliography.

218. Exhaust-steam Turbines. — Low and Mixed Pressures. — In a general sense the reciprocating engine reaches its maximum overall economy at a vacuum of about 26 inches referred to a 30-inch barometer. Cylinder condensation and excessive size of low-pressure cylinder usually offset the reduction in steam consumption for vacua higher than 26 inches. When it is considered that an increase in vacuum from 26 to 29 inches (initial pressure 200 lb. abs.) increases the available energy about 22 per cent the loss in economy due to the inability of the engine to utilize the higher vacuum is at once apparent. The ability of the turbine to take care of large volumes of steam and to avoid cylinder condensation makes a high vacuum desirable for economical reasons. The gain possible by taking advantage of high vacua has brought about the exhaust-steam turbine. Since the installation of the low-pressure turbine connected to the 7500-kw. angle-compound engine at the 59th Street Station of the Interborough Rapid Transit Company of New York, exhaust-steam or low-pressure turbines have been installed in many plants. In this noteworthy installation the addition of the low-pressure turbine effected:

a. An increase of 100 per cent in maximum capacity of plant.

b. An increase of 146 per cent in economic capacity of plant.

c. A saving of approximately 85 per cent of the condensed steam for return to the boiler.

d. An average improvement in economy of 13 per cent over the best high-pressure turbine results guaranteed at that time.

e. An average improvement in economy of 25 per cent over results obtained by the engine units only.

f. An average unit thermal efficiency of 20.6 per cent between the limits of 6500 kw. and 15,500 kw.

The low-pressure turbine is installed between the exhaust of the low-pressure engine cylinder and the condenser as shown in Fig. 274. Running with the engine the low-pressure turbine generator carries a variable load without governor control. The turbine generator takes care of the speed by automatically taking such a load as will keep the frequency in unison with that of the engine-driven unit. The turbine is equipped with the usual emergency speed-limit attachment for cutting off the steam supply should the speed exceed a predetermined limit.

Although numerous examples may be cited showing great gains in both capacity and economy in existing reciprocating-engine plants by the addition of a low-pressure turbine, a combined reciprocating-engine low-pressure turbine unit would not be selected in place of a high-pressure turbine unit for a new plant. Combined units of large power will cost approximately $40 per kw. against $8 per kw. for the high-pressure turbine unit. The space requirements of the combined unit are much larger than that of the high-pressure turbine unit and the cost of attendance, supplies and maintenance is also greater. Besides, high-pressure turbine economy has been greatly improved and water rates considerably below that guaranteed at the time the low-pressure turbines were installed in the 59th Street Station are realized in current practice. There is no question as to the economy effected in adding an exhaust steam turbine to large non-condensing

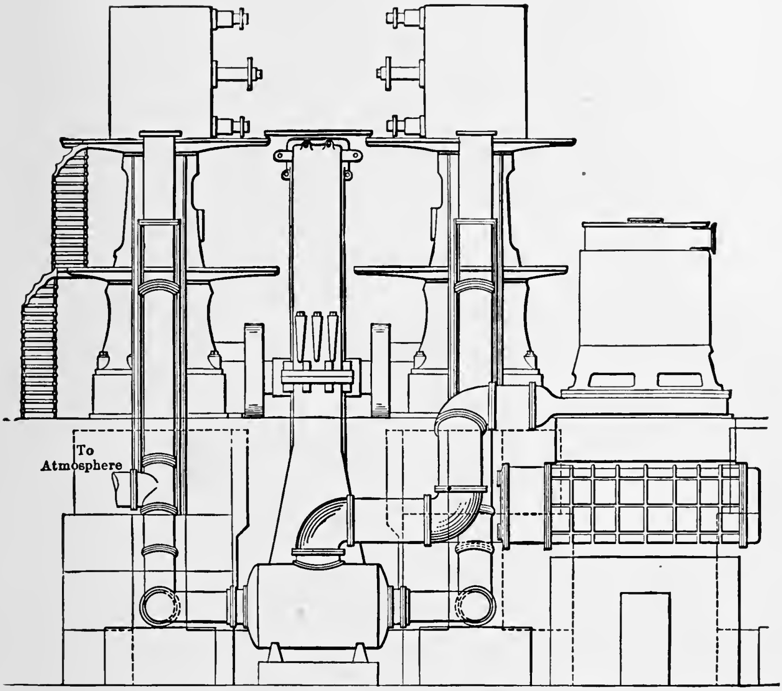


FIG. 274. Low-pressure Turbine Installation at the 59th Street Station of the Interborough Rapid Transit Company, New York.

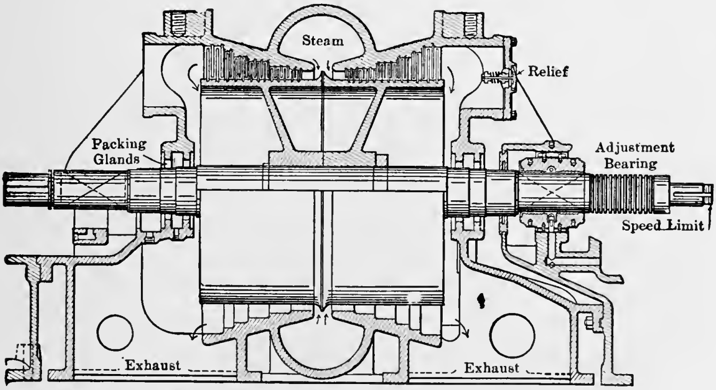


FIG. 275. Westinghouse Double-flow Low-pressure Turbine.

reciprocating engines, but with condensing engines it should be borne in mind that without increase in vacuum the addition of a low-pressure turbine will hardly warrant the extra expense.

Exhaust steam turbines may be divided into three classes, the division depending on the supply of low-pressure steam, viz., straight

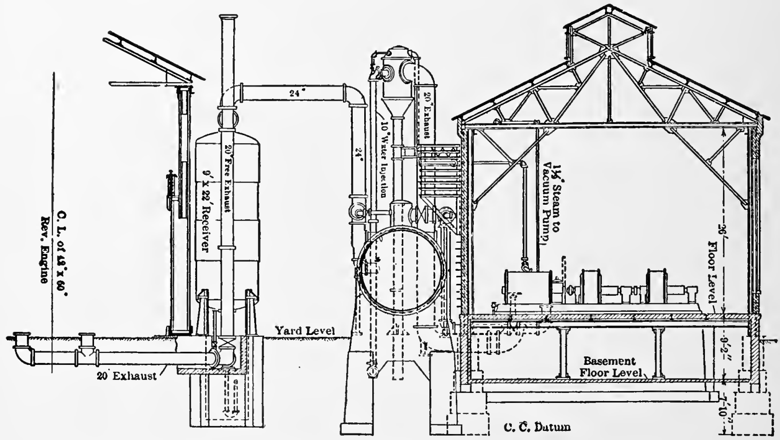


FIG. 276. Rateau Low-pressure Steam Turbine Installation.

low-pressure, mixed-pressure, and high-and-low-pressure turbines. Low-pressure turbines use exhaust steam only and are installed where there is an ample supply of low-pressure steam to carry the load at all times. Mixed-pressure turbines carry the full load (1) on all low-pressure steam, (2) all high-pressure steam, (3) any proportion of high-and-low-pressure steam at the same time. High-and-low-pressure turbines can carry the load on either high-pressure or low-pressure steam, but are not arranged to carry the load on both high- and low-pressure steam at the same time.

Low-pressure turbines may be installed so as to receive exhaust steam from a number of engines and other steam-actuated appliances, all of which exhaust into a common main or receiver, or they may be installed so as to receive the exhaust from one engine only.

Low-pressure turbines are frequently installed in connection with regenerator *accumulators*, to rolling-mill engines, steam hammers, and other appliances using steam intermittently, and have proved to be paying investments. The generator accumulator is intended to regulate the intermittent flow of steam before it passes to the turbine. The steam collects and is condensed as it enters the apparatus and is again vaporized during the time when the exhaust of the engines diminishes or ceases.

The regenerator usually consists of a cylindrical boiler-steel shell divided into two similar chambers by a central horizontal diaphragm, Fig. 277. In each compartment are a number of elliptical tubes A, each of which is perforated with a number of $\frac{3}{4}$-inch holes. The spaces surrounding the tubes and, under certain conditions, the tubes themselves are filled with water to a height of about four inches above the top of the upper tubes. Baffle plate B serves to separate the entrained moisture from the steam. The operation is as follows: Exhaust steam enters the apparatus at N, passes to the interior of the elliptical tubes, and escapes into the steam space through the perforations and thence

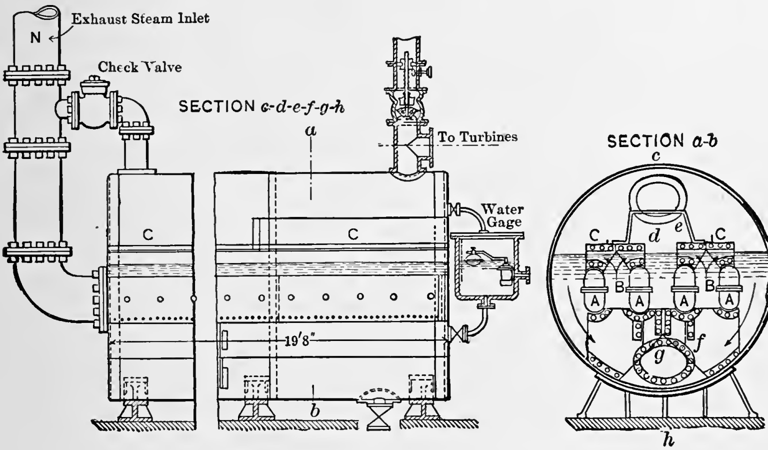


FIG. 277. Rateau Regenerator Accumulator.

to the turbine. When the supply of steam from the main engine ceases, the pressure in the regenerator decreases, the water liberates part of the heat it has absorbed and a uniform flow of low-pressure steam is given off. The continued demand of the turbine reduces the pressure in the accumulator and causes the steam still retained in the tubes to escape, thereby maintaining the circulation of the water (indicated by arrowheads) and facilitating the liberation of steam. Suitable valves regulate the limits of pressure in the accumulator and prevent the return of water to the main engine.

In the size normally installed this type of accumulator will furnish a sufficient supply of steam for four minutes with exhaust entirely cut off. If the period is longer than four minutes it becomes necessary to admit live steam. Low-pressure turbines develop one electrical horsepower-hour on a steam consumption of about 30 pounds with initial pressure of 15 pounds absolute and a back pressure of 1.5 pounds absolute. Fig. 278 gives the performance of a typical Westinghouse low-pressure turbine for various vacua, initial pressure 15 pounds absolute.

The weight of water W required to operate the low-pressure turbine for a given period with a predetermined temperature drop may be calculated from the relationship

$$W = \frac{tsr}{q_1 - q_2}, \tag{190}$$

in which

t = maximum number of minutes the exhaust supply may be entirely cut off,

s = water rate of the turbine, pounds per minute,

r = mean latent heat at regenerator pressure,

q_1 = heat of the liquid corresponding to maximum temperature of water in regenerator, deg. fahr.,

q_2 = heat of the liquid corresponding to minimum temperature of water in regenerator, deg. fahr.

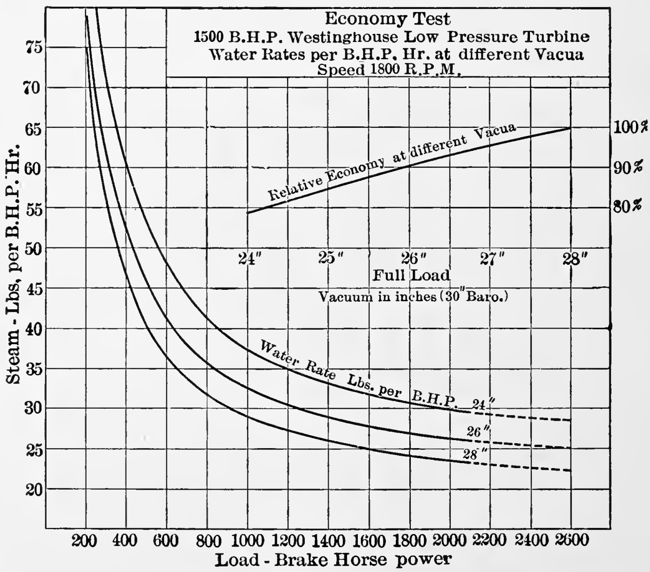


FIG. 278. Performance of Westinghouse Low-pressure Turbine.

If the regenerator is to absorb M pounds of exhaust steam in t minutes as in case of a sudden flux of exhaust the weight of water W_1 required is

$$W_1 = \frac{Mr}{q_1 - q_2}. \tag{191}$$

Example 34. Determine the weight of water to be stored in a regenerator to operate a 500-horsepower exhaust steam turbine for five minutes if the steam supply is entirely cut off; pressure drop 17 to 14 pounds absolute, turbine water rate 30 pounds per horsepower-hour.

$$t = 5, \quad s = \frac{500 \times 30}{60} = 250, \quad r = \frac{965.6 + 971.9}{2} = 968.8,$$

$$q_1 = 187.5, \quad q_2 = 177.5,$$

$$W = \frac{5 \times 250 \times 968.8}{187.5 - 177.5} = 121{,}100.$$

If the regenerator is to absorb 2000 pounds of the exhaust steam in five minutes during a period of sudden flux,

$$W_1 = \frac{2000 \times 968.8}{187.5 - 177.5} = 193{,}760.$$

Theory of Steam Accumulators and Regenerative Processes: F. G. Gasche, Proc. Eng. Soc. Wes. Penn., Dec., 1912, p. 723.

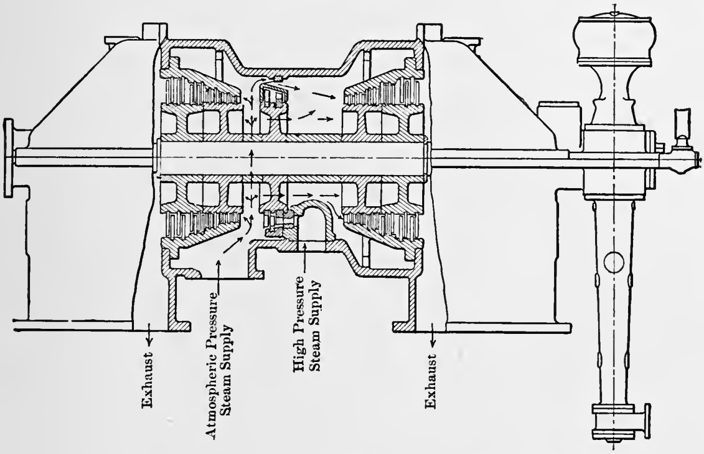


Fig. 279. Westinghouse Mixed-pressure Turbine.

In the mixed-pressure turbine the transition from all low pressure to all high pressure, through all the conditions intermediate between these extremes, is provided for automatically by the turbine governor; a deficiency of low-pressure steam causes the high-pressure nozzles to open automatically. With this arrangement it is not necessary for purposes of economy to proportion exactly the low-pressure turbine to the amount of exhaust steam available, but within limits it may be made as large as the load demands.

Mixed-pressure turbines have been constructed in single units as large as 10,000 kw.

The high- and low-pressure turbine is used when there is a sufficient supply of low-pressure steam to carry the load for a long period, say three or four months, and when for a similar period only high-pressure steam is available. When designed for this pressure range the turbine does not operate at maximum efficiency at either the high- or the low-pressure condition. For this reason it is doubtful whether this arrangement results in better overall economy than two separate units, a high- and a low-pressure turbine.

219. Advantages of the Steam Turbine. — The principal advantages of the steam turbine are: (1) low first cost; (2) low maintenance and attendance; (3) economy of space and foundation; (4) absence of oil in condensed steam; (5) freedom from vibration; (6) uniform angular

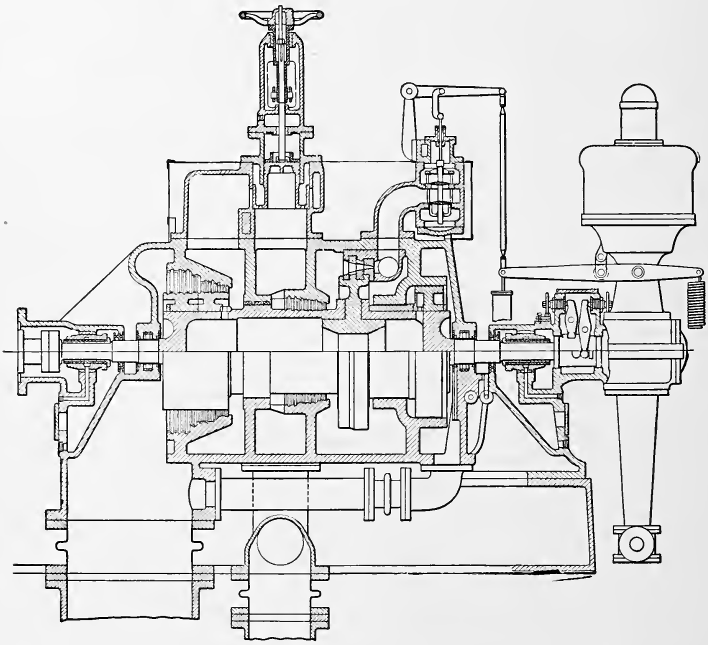


Fig. 280. Section through a Westinghouse Bleeder Type Turbine.

velocity, and (7) high efficiencies for large variations in load. The reciprocating engine is well adapted for pumping stations, compressor plants, hoisting engines, and the like, requiring low angular velocity, and for reversing service, but its place is being rapidly taken by the steam turbine for alternating-current dynamos, centrifugal pumps and blowers requiring high angular velocity. The recent development of high-efficiency speed-reduction gearing makes it possible for the turbines to compete with the engines for low-speed work. In fact the geared turbine is rapidly replacing the engine for low-speed work.

First Cost. — Because of the various uses to which turbines are applied and on account of the extreme variation in design general rules for approximating the cost of turbines are without purpose. Values based on rated capacity vary within such wide limits that average figures are apt to lead to serious error. In a general sense steam turbines are lower in first cost than steam engines of equivalent rated capacity irrespective of size. Specific figures are given in Chapter XVIII.

Maintenance and Attendance. — Although composed of a large number of parts as compared with a reciprocating engine of the same capacity, there are few moving parts and rubbing surfaces. The only contact between rotor and stator is in the main bearings, and the problem of lubrication is therefore a simple one. The absence of pistons, stuffing boxes, dish pots, etc., reduces the cost of maintenance and attendance to a minimum and limits the possibility of leakage. See Chapter XVIII for specific figures.

Economy of Space and Foundation. — The floor space required by practically all types of turbines is considerably less than the space requirements of piston engines. Vertical three-cylinder compound Corliss engines of the New York Edison type require the least floor space of any large slow-speed reciprocating engines, but take up about twice the space of a Parsons turbine installation of the same size. With non-condensing high-speed engines the comparative economy in space is less marked. The average space occupied by turbine units is approximately $\frac{1}{5}$ less than that of engine units of equivalent capacity, but specific cases may be cited in which the ratio varies widely from the average. In the modern central station the actual space reduction per kilowatt of plant rating is much less than that referred to the prime mover only because of the tendency toward less crowded conditions.

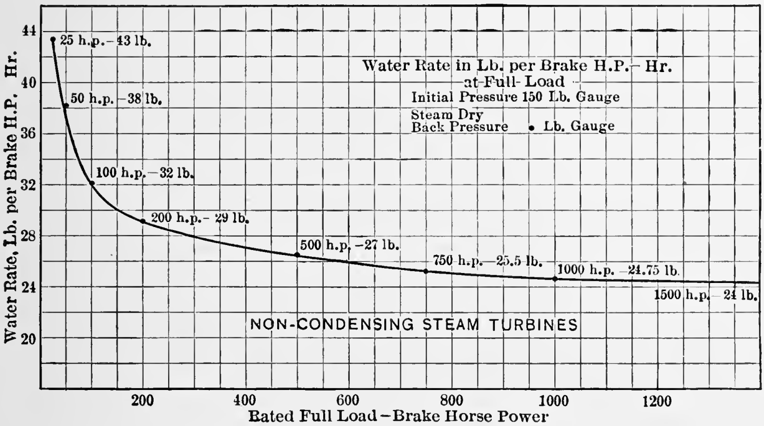
The weight of the steam turbine is very small compared with a reciprocating engine of the same horsepower. The New York Edison engine and generators weigh more than eight times as much as a turbine installation of equal capacity. The turbine, for this reason, and also because of the total absence of vibration, requires a relatively light foundation. In many instances the foundation consists of steel beams with concrete arches sprung between them resting upon the floor, and the basement underneath may be used for the condenser instead of the massive foundation required for the reciprocating engine. Engines are seldom constructed in sizes above 5000 horsepower, whereas single turbine units of 30,000 kw. are not uncommon and a turbine 60,000 kw. normal capacity is now being installed in the 74th Street Station of the Interborough Rapid Transit Co., N. Y.

Absence of Oil in Condensed Steam. — As the steam turbine requires no internal lubrication, oil does not come in contact with the steam, and the condensed steam from the surface condensers is available for boiler-feeding purposes without purification. In many cases the reuse of condensed steam effects a large saving in cost of feed water and in expense for maintenance and cleaning of boilers. The amount of entrained air is reduced to a minimum and consequently the work of air pumps is lessened.

Regulation. — The variable pressure at the crank pin of a reciprocating engine necessitates the use of a heavy flywheel to keep the instantaneous angular fluctuation within practical limits. In the steam turbine the motion is purely rotary and a flywheel is not necessary. In the former there are always instantaneous variations in velocity during each revolution, even with constant load, while in the latter the speed is practically constant. A number of published tests of Parsons and Curtis turbines show an average fluctuation of 2 per cent from no load to full load and 3 per cent from no load to 100-per-cent overload. Although closer regulation than this is possible, it is not deemed necessary, particularly in alternating-current work where a comparatively wide range is desirable for parallel operation.

Overload Capacity. — The overload capacity of any prime mover depends entirely upon the designation of the rated load. The maximum economy of the average piston engine lies between 0.7 and full load, and for this reason the *rated* load refers usually to this maximum economical load. Evidently if the engine is rated under its maximum possible output it is capable of *overload.* Under the existing system of rating the average piston engine is capable of operating with overloads of 25 to 50 per cent. According to the old rating the steam turbine was capable of overloads ranging from 100 to 200 per cent and much confusion arose in determining the station load factor. Current turbine practice gives as the normal rating the maximum continuous load which can be carried for 24 hours when under control of the primary valves. Through the agency of the secondary valves overloads of 50 per cent or more are possible. The steam economy of the turbine is superior to that of the engine for overloads. Since all modern turbines are designed for a point of best steam consumption somewhere regardless of what their rating may be, the *actual* rating means little.

220. Efficiency and Economy of Steam Turbines. — A general comparison of the water rates of piston engines and steam turbines is very unsatisfactory because of the wide range in operating conditions. In a general sense the piston engine is more economical in the use of steam than the turbine for non-condensing service and the reverse is true for high-pressure, high-vacuum condensing service. Condensing engines of the uniflow or poppet-valve type have shown superior economy (under favorable conditions) to the turbine for sizes up to 3000 horsepower and in some instances up to 5000 horsepower but heat economy is only one of the many factors entering into the ultimate cost of power. For sizes over 3000 horsepower the turbine is in a class of its own and piston engines above this size are seldom found in modern stationary practice. A comparison of the curves in Fig. 215, showing typical economy curves of high-speed single-valve non-condensing engines, and of Fig. 283, showing the performance of non-condensing steam turbines, is somewhat in favor of the piston engine, the difference decreasing as the size of unit increases. A similar comparison of the performance curves



Corrections for fractional loads.—Increase full load water rate as follows: $\frac{1}{2}$—20%; $\frac{3}{4}$—8%; 1—0%; $1\frac{1}{4}$—5%.

Corrections for initial pressures.—175 lb. deduct 3%; 200 lb. deduct 5%; 125 lb. add 5%; 100 lb. add 10%; 75 lb. add 20%.

Corrections for increased back pressure.—Add for each lb. back pressure 200 lb.—1%; 175 lb.—$1\frac{1}{4}$%; 150 lb.—$1\frac{1}{2}$%; 125 lb.—2%; 100 lb.—$2\frac{1}{2}$%; 75 lb.—3%.

Correction for superheat.—Subtract 1% for each ten degrees superheat up to 200 degrees.

FIG. 281. Average Water Rates of High-grade Small Non-condensing Steam Turbines.

of compound single-valve, single-cylinder four-valve, and compound four-valve non-condensing piston engines with those of steam turbines of the same size show marked increase in economy in favor of the piston engine. For sizes between 2000 and 6000 horsepower there is little difference between the steam economy of the very best grade of piston engine and that of the turbine. Piston engines above 10,000 horsepower have not been built for stationary practice, hence a comparison with the turbine for larger sizes is impossible. The Manhattan type at the 74th Street Station of the Interborough Rapid Transit Company represents the largest piston engines (7500 kw.) ever constructed for central station service. The heat consumption of these engines is considerably more than that of the modern turbo-generator of the

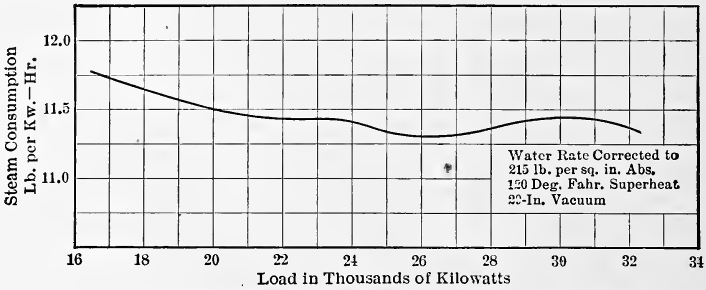


FIG. 282. Performance of 30,000-kw. Westinghouse Compound Turbine, Interborough Rapid Transit Co.

same capacity. Tables 81 and 85 give the general conditions of operation and the steam consumption of exceptionally good piston engines of various sizes and types and Table 89 similar data of first-class tur-

TABLE 89.

PERFORMANCE OF THE MODERN STEAM TURBINE AT RATED CAPACITY.

(Manufacturer's Guarantee.)

Index.	Make of Turbine.	Rated Capacity.	R.P.M.	Initial Pressure, Lb. Abs.	Back Pressure, Inches.	Superheat, Deg. Fahr.	Lb. Steam Per Kw-hr.*	Rankine Cycle, Ratio.
1	Westinghouse	500 Kw.	3600	165	2.0	0	19.8	†53.0
2	"	1,000 "	3600	165	2.0	0	18.1	†58.0
3	"	5,000 "	1800	165	2.0	0	16.0	†65.6
4	"	15,000 "	1800	215	1.5	125	12.6	†71.0
5	"	30,000 "	1200	235	1.0	200	10.65	†75.0
6	"	45,000 "	1200	215	1.0	200	10.65	†76.0
7	Curtis	500 "	3600	215	2.0	0	18.5	†54.0
8	"	1,000 "	3600	215	2.0	0	17.5	†57.1
9	"	5,000 "	1800	215	2.0	125	14.3	†64.6
10	"	15,000 "	1800	215	2.0	125	12.5	†74.0
11	"	30,000 "	1200	215	2.0	125	12.2	†75.8
12	"	45,000 "	1200	215	2.0	125	11.9	†77.6
13	Kerr	25 Hp.	3600	165	At.	0	‡43.0	‡31.0
14	"	50 "	3600	165	At.	0	‡38.0	‡34.2
15	"	100 "	3600	165	At.	0	‡32 0	‡40.6
16	"	200 "	3600	165	At.	0	‡29.0	‡44.9
17	"	500 "	3600	165	At.	0	‡27.0	‡48.1
18	"	750 "	3600	165	At.	0	‡25.5	‡51.0
19	"	1,000 "	3600	165	At.	0	‡24.75	‡57.5
20	"	1,500 "	3600	165	At.	0	‡24.0	‡59.1

* Lower water rates than these have been guaranteed for higher pressures and superheat and for lower back pressures. The guaranteed water rate for a 20,000-kw. Curtis turbo-generator at the River Plant of the Buffalo General Electric Company is 10.6 lb. per kw-hr. for initial pressure, 265 lb. abs., back pressure 1 in., superheat 250 deg. fahr.

† Based on electrical horsepower.

‡ Based on developed horsepower.

bines. A study of these tables will show that the choice must be based on other factors than the steam consumption. In a general sense the piston engine is superior to the turbine for high back pressures, slow rotative speeds, reversing service and heavy starting torques, while the turbine has practically superseded the piston engine for large central station units and for auxiliaries requiring high rotative speed. Recent tests of geared turbines show exceptionally high efficiency for sizes as large as 10,000 hp., and it is not unlikely that the turbine equipped with this device will offset the low rotative speed factor of the piston engine.

If the tests of steam turbines and piston engines could be made at some standard initial pressure, back pressure and quality or superheat, then a comparison could readily be made, but both types of prime movers are designed to give the best results for special operating conditions, and any marked departure from these conditions will result in loss of economy. It is frequently desired, however, to make a comparison between the economy of the different machines, and the following methods are in vogue:

(1) Steam consumption under assumed conditions.

(2) Heat consumption per unit output per minute above the ideal feed-water temperature.

(3) Rankine cycle ratio.

Steam Consumption under Assumed Conditions (*Standard Correction Curves*): This method for comparing engines or turbines or both is best illustrated by a specific example:

Example 35. Compare the full-load performance of a 125-kilowatt direct-connected piston engine with that of a 125-kilowatt turbogenerator with operating conditions as follows:

	Steam Consumption, Lb. per Kw-hour.	Initial Pressure, Lb. Absolute.	Vacuum, Inches of Hg.	Superheat, Deg. Fahr.
Engine....................	25.0	160	25.5	0
Turbine	22.7	110	28.0	125

Manufacturers of steam turbines have provided correction curves as illustrated in Fig. 284, showing the influence of varying vacuum, superheat and pressures on the steam consumption.* From curve B, we find that the steam consumption of the turbine should be decreased 2.5 pounds to give the equivalent at 160 pounds initial pressure; from curve A it should be increased 2.5 pounds to give the equivalent at

* These curves are drawn to a much larger scale than the reproduction given here.

25.5 inches of vacuum, and from curve C it should be increased 2.5 pounds to give the equivalent at 0 degree superheat. The full-load steam consumption for the turbine under the engine conditions is therefore $22.7 - 2.5 + 2.5 + 2.5 = 25.2$ pounds per kilowatt-hour.

The *ratio* method is also used in this connection, thus: The full-load steam consumption at 160 pounds pressure, curve B, Fig. 284, is multiplied by the ratio $\frac{25}{27.5}$ to give the equivalent consumption at 110 pounds (25 is the steam consumption at 160 pounds and 27.5 the consumption at 110 pounds). Similarly, the correction ratio to change the consumption at 28 inches of vacuum to 25.5 is $\frac{25.5}{25}$, and to correct 125 deg. fahr. superheat to 0 deg. fahr. is $\frac{25}{22.5}$.

SUMMARY.

$$\text{Pressure correction } \frac{25}{27.5} = 0.91 = -9 \text{ per cent.}$$

$$\text{Vacuum correction } \frac{27.5}{25} = 1.10 = 10 \text{ per cent.}$$

$$\text{Superheat correction } \frac{25}{22.5} = 1.11 = \underline{11} \text{ per cent.}$$

Net correction 12 per cent.

Corrected steam consumption $= 22.7 + 0.12 \times 22.7 = 25.4$ pounds per kilowatt-hour.

The ratio method is generally used if the difference between the corrected steam consumption and that of the correction curves for the same conditions is greater than 5 per cent ("The Steam Turbine," Moyer, p. 128).

This ratio method for correcting steam consumption at full load may be used without appreciable error for half to one and one-half load and is the only practical method for quarter load. (Engineering, London, March 2, 1906.)

Heat Consumption:

The heat consumption B.t.u. per unit output per minute above the ideal feed-water temperature may be expressed

$$\frac{W(H_1 - q_2)}{60}.$$

For the case cited above

$$\text{Engine,} \quad \frac{25\,(1194.1 - 98)}{60} = 455 \text{ B.t.u.}$$

$$\text{Turbine,} \quad \frac{22.7\,(1264.2 - 70)}{60} = 451 \text{ B.t.u.}$$

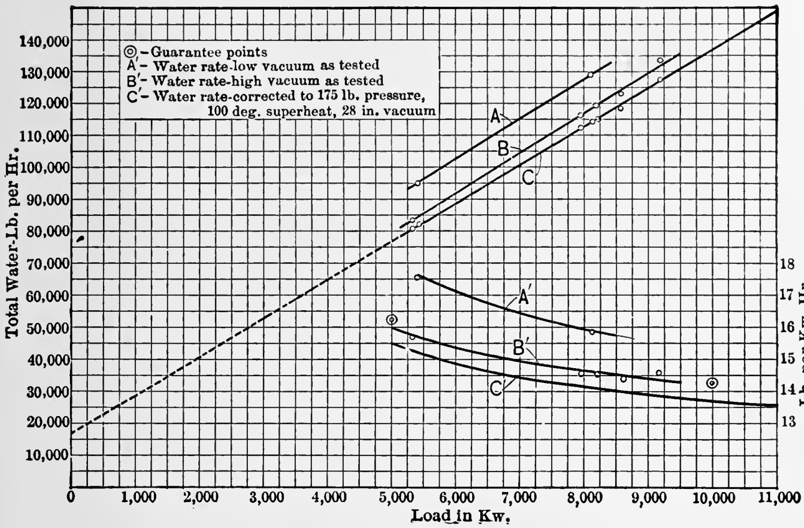


FIG. 283. Performance of 10,000-kilowatt Westinghouse Double-flow Turbine, City Electric Co., San Francisco, Cal.

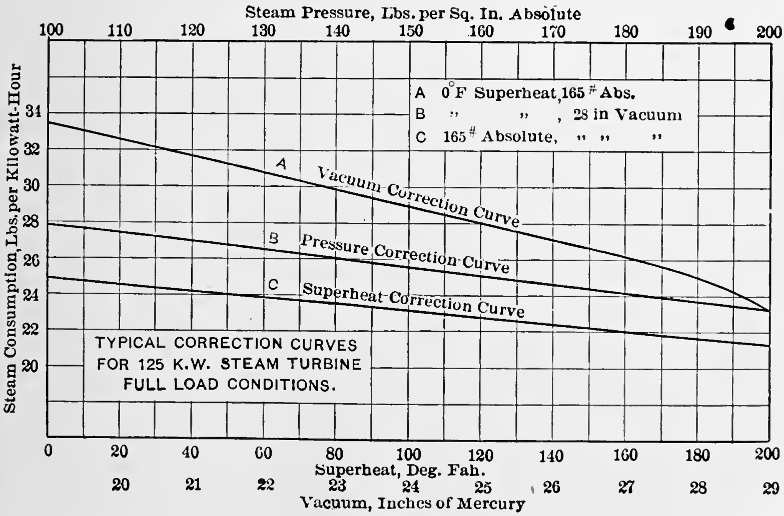


FIG. 284.

Rankine Cycle Ratio:

The Rankine cycle ratio, or the extent to which the theoretical possibilities are realized, may be expressed

$$E_r = \frac{2546 \times 1.34}{W\,(H_1 - H_2)}.$$

For the case cited above

Engine, $$\frac{2546 \times 1.34}{25\,(1194.1 - 915)} = 0.49.$$

Turbine, $$\frac{2546 \times 1.34}{22.7\,(1264.2 - 915.3)} = 0.43.$$

In the assumed case the turbine is the more economical in heat consumption, but the engine is the more perfect of the two as far as theoretical possibilities are concerned.

221. Influence of Superheat. — Theoretically, the gain in heat economy due to the use of superheat is the same for all prime movers of whatever type. In the actual mechanism the rate of improvement with increase

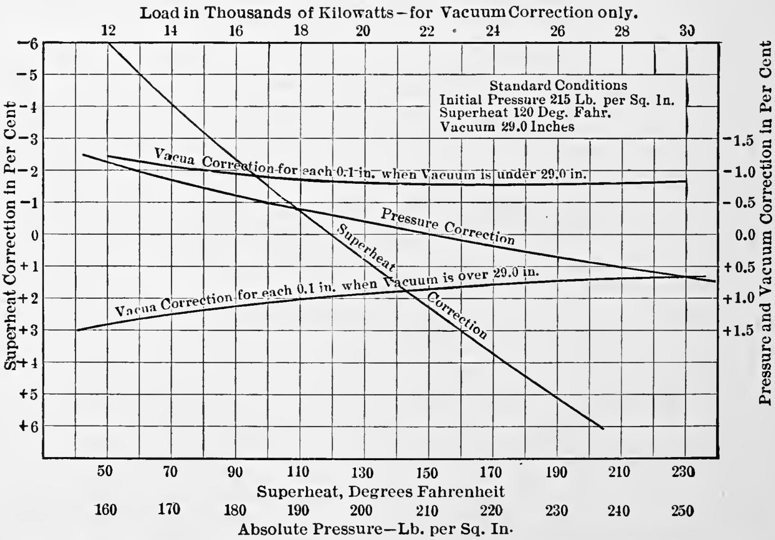


FIG. 285. Correction Factors for 30,000-kilowatt Westinghouse Compound Turbines.

of superheat differs with type and size. The gain in the reciprocating engine is due mainly to the reduction of cylinder condensation while in the turbine the improvement in economy is due primarily to the reduction in "windage" and other friction losses. In both types the actual gain is much greater than in the perfect mechanism for pure adiabatic expansion. In the ideal or frictionless high-pressure condensing turbine an increase of 35 deg. fahr. superheat effects an increase of about 1 per cent in thermal efficiency. In the actual turbine the steam consumption is improved 1 per cent for every 6 to 14 deg. fahr. superheat. The advantage of superheat is greater with non-condensing than with condensing units and is even more marked with low-pressure units. In average practice the maximum temperature of the steam is approximately 500 deg. fahr., but in large central stations temperatures

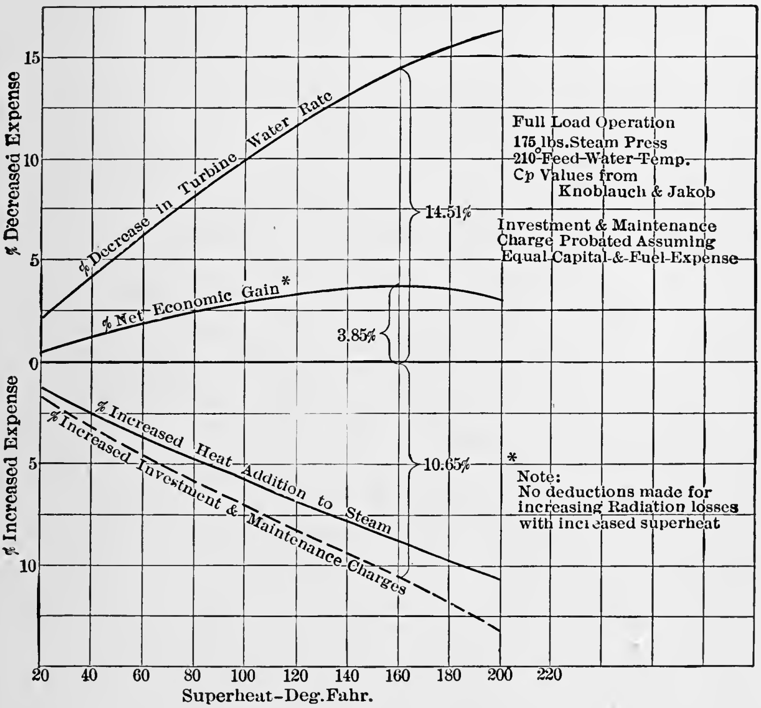


FIG. 286. Influence of Superheat on Overall Economy of Operation.

of 600 degs. are not uncommon and a number of recent installations are designed for total steam temperature of 700 deg. fahr. The General Electric Company is prepared to design Curtis turbines for temperatures as high as 800 deg. fahr. should the demand warrant such high temperatures. The higher the initial temperature the greater will be the investment cost and a point is eventually reached where the increased fixed charges will offset the gain in heat economy. This is illustrated in Fig. 286, the curves of which though based on a specific case are applicable in general principle to all cases. The influence of superheat on the economy of a 30,000-kw. turbo-generator is illustrated in Fig. 285.

222. Influence of High Initial Pressure. — The theoretical gain due to increase in initial pressure has been discussed in paragraph 179. In view of the marked improvement in heat economy for very high initial pressures it is only a matter of time when pressures far above the present maximum will be a matter of everyday practice. It seems that the only difficulty in the way of very high pressures is the boiler. There is no particular mechanical obstacle in designing the turbine since it simply means the use of heavier parts for the extremely high pressure end and a somewhat increased cost of construction. Because of the increased density of high-pressure steam the windage and friction losses would be correspondingly increased in the high-pressure stages and the dew point with its attending losses would be advanced farther up in the turbine. Just what will be the economical limit cannot be predicted with any degree of certainty. In case of machines of 20,000-kw. capacity or more intended for operation in large power houses when the units are running at or near their most efficient load, considerable investment is warranted for the sake of a comparatively small actual gain in heat consumption and consequently the tendency is toward high pressures and temperatures. For example, a number of recent installations call for a working pressure of 290 lb. per sq. in. abs. with superheat of 275 deg. fahr. corresponding to a temperature of 590 deg. fahr. The new turbine for the Joliet station of the Public Service Company of Northern Illinois is designed for 365 lb. absolute pressure with superheat of 225 deg. fahr. Designs are now being perfected for pressures as high as 500 pounds and even higher pressures have been considered.

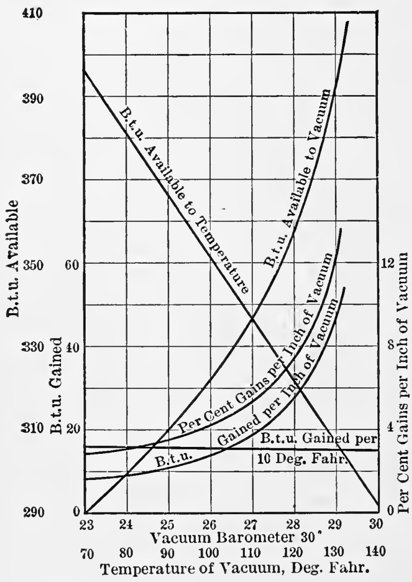


FIG. 287. Distribution of Heat during Adiabatic Expansion; Initial Pressure 15 lb. Absolute; Saturated.

223. Influence of High Vacua. — The possible economy of the reciprocating engine is greatly restricted by its limited range of expansion. Cylinders cannot be profitably designed to accommodate the rapid increase in the volume of steam when expanded to very low pressures. For example, the specific volume of 1 pound of steam under a vacuum of

29 inches (referred to a 30-inch barometer) is about 667 cubic feet or nearly double its volume under a vacuum of 28 inches. Usually the exhaust is opened at a pressure of 6 or 8 pounds absolute and consequently a large proportion of the available energy is lost. The lower vacuum in the exhaust pipe, therefore, serves only to diminish the back pressure and does not affect the completeness of expansion. Even if it were practical to expand to 1 pound absolute, the increased condensation in the reciprocating engine would probably offset any gain due to expansion unless the steam were highly superheated. A study of a number of tests of reciprocating engines shows but a slight improvement in overall plant economy due to increasing the vacuum beyond 26 inches. Tests of steam turbines show a decrease in steam consumption of about 5 per cent for each inch of vacuum between 25- and 27-inch vacuum, 6 per cent between 27- and 28-inch and 8 to 12 per cent between 28- and 29-inch. These values are approximate only since the influence of vacuum on the steam consumption varies greatly with the type and size of turbine.

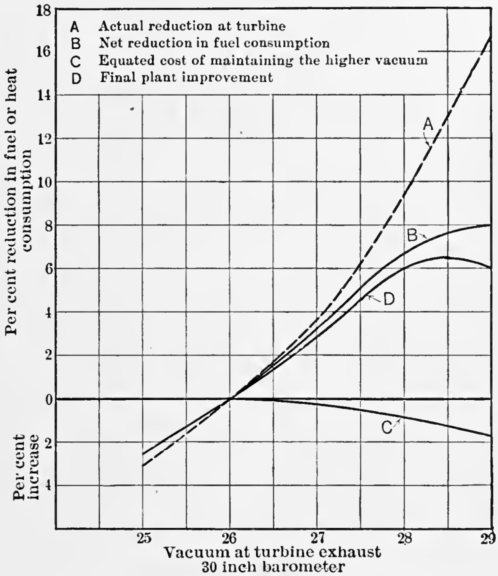


FIG. 288. Influence of Vacuum on Cost of Power.

Since the volume of the steam increases very rapidly with the decrease in back pressure the corresponding capacity and power required by the air and circulating pumps becomes proportionately larger. There is consequently a point where the improvement in steam economy fails to exceed the increased power demanded by the auxiliaries. This is illustrated graphically in Fig. 288. The values in Fig. 288 refer to a specific case only but the general principle is the same for all conditions. In the older types of condensing equipment the cost of maintaining the vacuum above 27 inches, referred to a 30-inch barometer, increased very rapidly with the increase in vacuum. In the modern plant vacua amounting to 97 per cent of the theoretical maximum (as determined by the temperature of the cooling water) are readily maintained with excessive cost. This influence of vacuum on the economy of a 30,000-kw. turbo-alternator is shown in Fig. 285.

224. Tesla Bladeless Turbine. — Fig. 289 shows a section through a 200-horsepower experimental turbine designed by Nikola Tesla. It consists of a rotor composed of 25 steel disks (each $\frac{1}{32}$ inch thick and arranged on the shaft so that the length of the shaft covered by the disks is approximately 3.5 inches) revolving in a plain cylindrical casing. There are no guide plates or vanes and the viscosity and adhesion of the steam is depended upon for driving the rotor instead of impulse and reaction as in the standard type of turbine. Steam flows from the circumference to the center, and, when the rotor is at rest, flows by a short curved path, as indicated by the line in the end view, across the

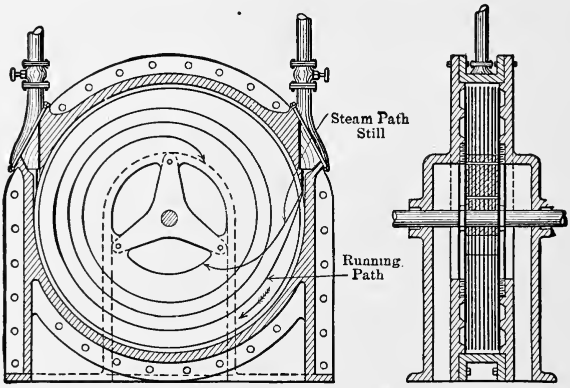


Fig. 289. Tesla Bladeless Turbine.

face of the disk. When the rotor is up to speed the steam passes to the exhaust in a spiral path from 12 to 16 feet in length. Since the direction of rotation is determined solely by the direction of the entering jet it is only necessary to change the direction of the latter to effect complete reversal of the rotor. Mr. Tesla states that a 200-horsepower turbine of this type has attained a performance of 38 pounds per horsepower hour, initial pressure 125 pounds gauge, atmospheric exhaust, 9000 r.p.m. (Prac. Engineer, U. S., Dec., 1911, p. 852.) The space occupied by this unit is only 2 feet by 3 feet and 2 feet high and the weight of the engine alone is 2 pounds per horsepower developed. This device has never been commercialized.

225. "Spiro" Turbine. — Fig. 290 gives the general details of a high-speed rotary steam engine which has been erroneously classified by its builders as a turbine. It consists essentially of a pair of herringbone gears revolving in a twin cylindrical casing. Steam enters space a, Fig. 295, through ports pp and presses upon the gear teeth, driving them forward. The volume is increased from that indicated at *a* to that shown at *b, c, d, e,* and *f* and the energy produced is the product of the pressure and volume. Exhaust occurs when the ends of the grooves in which the action lies pass the line of contact so that

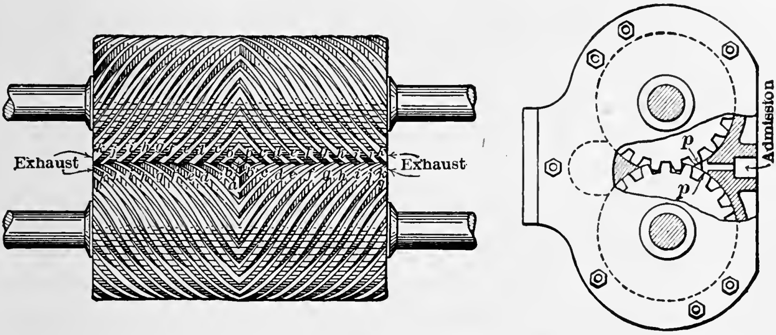


FIG. 290. The "Spiro" Turbine.

they are no longer closed by the teeth of the opposite gear. The load may be varied by throttling or by cutting off the steam supply. The "Spiro" is built in various sizes ranging from 1 to 200 horsepower and operates at 2000 to 3000 r.p.m. The following tests give an idea of the economy effected by this type of motor. (Power, Feb. 6, 1912, p. 188.)

	Test 1.	Test 2.
Boiler pressure, pounds gauge	120	130
Inlet pressure, pounds gauge	101.5	115
Back pressure, pounds gauge	Atmos.	Atmos.
Horse power developed	25.3	151
R.p.m.	2450	2710
Steam, pounds per horse-power hour	53.2	31.8

PROBLEMS.

1. Steam expands adiabatically in a frictionless nozzle from an initial pressure of 200 lb. per sq. in. absolute, superheat 200 deg. fahr., to a back pressure of 1 in. absolute, weight discharged 7200 lb. per hr.; required:

a. Velocity of the jet at the throat.
b. Maximum spouting velocity.
c. Diameter of the throat.
d. Diameter of the mouth.
e. Quality of the steam at the mouth.

2. If the jet in Problem 1 impinges tangentially against a set of moving vanes and leaves them with residual velocity of 500 ft. per second, required:

a. Velocity of the vanes, neglecting all friction and leakage losses.
b. Horsepower imparted to the rotor.

c. Pressure exerted against the vanes.
d. Impulse efficiency of the jet.
e. Water rate, lb. per hp-hr.

3. Same conditions and requirements as in Problems 1 and 2 except that the energy-efficiency is 94 per cent and the loss of energy between inlet and exit of the vanes is 15 per cent.

4. If the nozzle in Problem 1 is to be used in connection with a multi-pressure steam turbine, required the theoretical number of stages necessary for a peripheral velocity of 500 ft. per sec. Jet impinges tangentially against the rotor and all of the available energy is absorbed in driving the rotor.

5. A single-stage impulse turbine (De Laval type) develops 200 hp. under the following conditions: Initial pressure 153 lb. abs., back pressure 4 in. abs., superheat 50 deg. fahr., water rate 14.4 lb. per hp-hr., nozzle angle 20 deg., peripheral velocity of the rotor 1200 ft. per sec. Required:

a. Thermal efficiency.
b. Rankine cycle ratio.
c. B.t.u. per hp. per minute.

6. Construct the theoretical velocity diagram for the conditions in Problem 5 and sketch in the blade outlines.

7. Construct the theoretical velocity diagram for a 750-hp., 2-stage Curtis turbine operating under the following conditions: Initial pressure 175 lb. abs., superheat 150 deg. fahr., back pressure 2 in. abs., Rankine cycle efficiency 65 per cent, nozzle angle 20 degs., peripheral velocity 500 ft. per sec. Each stage consists of two rotating elements and one stationary element.

8. Construct the velocity diagram and calculate the work done per stage in a frictionless reaction turbine for the following conditions: Heat drop per stage, 16 B.t.u. per lb. of steam, peripheral velocity to be the maximum theoretically possible for the given conditions, exit angle 30 degs., entrance angle 0.

9. Determine the weight of water to be stored in a regenerator to operate a 1000-hp. exhaust steam turbine for 6 minutes if the steam supply is entirely cut off; pressure drop 15 to 12 lb. abs., turbine water rate 28 lb. per hp-hr.

CHAPTER XI

CONDENSERS

226. General. — The primary object of condensing is the reduction of back pressure although the recovery of the condensate may be of equal importance. If a given volume of saturated steam be confined in a closed vessel abstraction of heat will result in condensation of part of the vapor with a corresponding drop in temperature and pressure. The greater the amount of heat abstracted the greater will be the amount condensed and the lower will be the temperature and pressure. All of the vapor can never be condensed in practice since this would necessitate a lowering of the temperature to absolute zero or 492 degrees below the fahrenheit freezing point; consequently, the pressure can never be reduced to zero. With water as the cooling medium the minimum temperature to which the vapor can be reduced is 32 deg. fahr. corresponding to a pressure of 0.0886 lb. per sq. in. or 0.1804 in. of mercury. This represents, therefore, the lowest condenser pressure possible in practice. Condensing results in reduction of pressure only when the vapor is contained in a closed vessel. Thus if the vessel is open to the atmosphere heat abstraction will result in condensation but the pressure will not fall below that of the atmosphere.

The standard atmospheric pressure at sea level and at latitude 45 degrees is 14.6963 lb. per sq. in., corresponding to a mercury column 29.921 inches in height, temperature of the mercury 32 deg. fahr. For any other temperature there will be a corresponding height of column because of the expansion or contraction of the mercury. Steam tables are based on a standard pressure of 29.921 inches of mercury at 32 deg. fahr. and for this reason it is convenient to transfer the observed barometer and mercurial vacuum gauge readings to the 32-degree standard.

The mercury column correction for any change in temperature may be closely approximated by the equation

$$h = h_1 [1 - 0.000101 (t_1 - t)], \tag{192}$$

in which

h = height of mercury column corrected to temperature t,
h_1 = observed height of mercury column,
t_1 = observed temperature of mercury column,
t = temperature to which column is to be referred.

Example 36. If the height of mercury in a vacuum gauge is 28.52 inches, temperature 80 deg. fahr., and the barometer column is 29.85 inches in height, temperature 62 deg. fahr., transfer the readings to the 32-degree standard.

For the barometer:

$$h = 29.85\,[1 - 0.000101\,(62 - 32)]$$
$$= 29.77.$$

For the vacuum gauge:

$$h = 29.52\,[1 - 0.000101\,(80 - 32)]$$
$$= 28.37.$$

Absolute back pressure = 29.77 − 28.37 = 1.40.

Vacuum referred to 32-deg. standard = 29.92 − 1.40 = 28.52.

In condenser work it is common practice to refer the reading of the vacuum gauge to a 30-inch barometer, in which case it is necessary to increase the standard temperature of the mercury to such a figure as will increase the height of the barometer from 29.921 to 30 inches; viz., 58.15 deg. fahr. Thus, if the barometer and vacuum gauge readings are corrected to a temperature of 58.15 deg. fahr. the difference between the figures will give the absolute pressure in inches of mercury at 58.15 deg. fahr., and if the difference is subtracted from 30 inches the result will give the inches of vacuum referred to a 30-inch barometer. According to A.S.M.E., 1915 Power Code, a 30-inch barometer refers in round numbers to a standard atmosphere with mercury at an ordinary temperature of 78 degrees.

Example 37. Height of mercury in vacuum gauge 28.52 inches, temperature of mercury 80 deg. fahr., barometer 29.85 inches, temperature 42 deg. fahr.; determine the vacuum referred to a 30-inch barometer.

For the vacuum gauge

$$h = 28.52\,[1 - 0.000101\,(80 - 58.15)]$$
$$= 28.46.$$

For the barometer

$$h = 29.85\,[1 - 0.000101\,(42 - 58.15)]$$
$$= 29.9.$$

Absolute pressure in inches of mercury at temperature 58.15 deg. fahr. = 29.9 − 28.46 = 1.44.

Vacuum referred to 30-inch barometer = 30 − 1.44 = 28.56.

According to Dalton's Laws: (1) The mass of a given kind of vapor required to saturate a given space at a given temperature is the same whether the vapor is all by itself or associated with vaporless gases; (2) the maximum tension of a given kind of vapor at a given temperature is the same whether it is all by itself or associated with vaporless gases; (3) in a mixture of gas and vapor the total pressure is equal to the sum of the partial pressures. The final pressure P_c is therefore the combined pressure of the air P_a and that of the water vapor P_v, or, assuming complete saturation,

$$P_c = P_a + P_v. \tag{193}$$

According to the laws of Boyle and Charles the volume, pressure and temperature relation of an ideal gas is

$$\frac{PV}{T} = \text{constant } (= 53.34 \text{ for dry air}) \tag{194}$$

in which

P = absolute pressure of the air, lb. per sq. ft.,
V = volume of one pound, cu. ft.,
T = absolute temperature, deg. fahr.

Since 1 lb. per sq. ft. = 0.016 in. of mercury at 32 deg. fahr., equation (194) may be conveniently expressed

$$\frac{P_aV}{T} = 0.755 \tag{195}$$

in which

P_a = absolute pressure, in. of mercury.

By means of equations (193) and (195) all problems involving a saturated mixture of air and water vapor may be readily solved. See Chapter XXV for a discussion of the properties of dry, saturated and partially saturated air.

Example 38. If the absolute pressure in a condenser is 4 inches of mercury and the temperature of the air-vapor mixture is 100 deg. fahr., required the percentages of air by weight in the mixture.

From steam tables the pressure of vapor corresponding to a temperature of 100 deg. fahr. is 1.93 inches of mercury.

Hence, from equation (193),

$$\begin{aligned} P_c &= P_a + P_v, \\ 4 &= P_a + 1.93, \\ P_a &= 2.07. \end{aligned}$$

Let v = volume of the condenser chamber, cubic feet.

Then $0.00285\,v$ = weight of vapor in the chamber (0.00285 = density of water vapor at 100 deg. fahr.), and

$0.08635 \times \dfrac{2.07}{29.92} \times \dfrac{460 + 0}{460 + 100}\,v = 0.00491\,v$ = weight of dry air in the chamber. (0.08635 = density of air at 0 deg. fahr. and 29.92 inches of mercury pressure.)

The total weight of the mixture is

$$0.00285\,v + 0.00491\,v = 0.00776\,v,$$

and the percentage of air in the mixture is

$$\frac{0.00491\,v}{0.00776\,v} = 0.632 \text{ or } 63.2 \text{ per cent.}$$

Curve *A*, Fig. 291, shows the influence of vacuum on the percentage of air in the air-vapor mixture for a constant air pressure of 0.1 in.

Curve *B*, Fig. 291, shows the difference between the temperature of saturated vapor corresponding to the total pressure in the condenser to that of the actual vapor for various vacua with constant air pressure of 0.1 in.

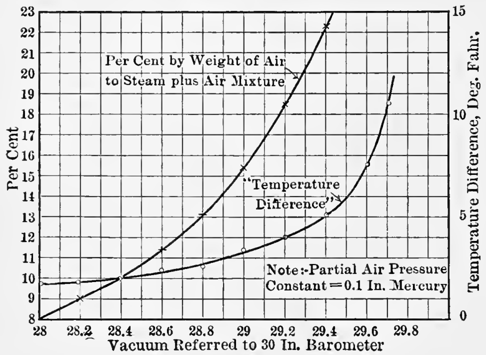


FIG. 291. Percentage of Air in Mixture and Difference of Temperatures Corresponding to Total Pressure and that Actually Existing, for Constant Air Pressure of 0.1 in.

For data pertaining to the amount of air carried into condensers see paragraphs 306–9.

Example 39. If the temperature within a condenser is 110 deg. fahr. and there is entrained with the steam 0.2 of a pound of air per pound of steam, required the maximum degree of vacuum obtainable.

One pound of saturated steam at a temperature of 110 deg. fahr. occupies a volume of 265.5 cu. ft. The corresponding vapor tension is 2.589 in. of mercury. This must also be the volume occupied by 0.2 pound of air mixed with it and the temperature of the air is that of the vapor (110 deg. fahr.). Then from equation (195),

$$P_a = \frac{0.755 \times (110 + 460)}{265.5 \div 0.2} = 0.322 \text{ in. of mercury.}$$

From equation (194)

$$\begin{aligned} P_c &= P_a + P_v \\ &= 0.322 + 2.589 = 2.911 \text{ in. of mercury.} \end{aligned}$$

And the vacuum

$$= 29.921 - 2.911 = 27.01 \text{ in. of mercury.}$$

If no air were present the maximum vacuum would be

$$29.921 - 2.589 = 27.332 \text{ in. of mercury.}$$

The lower the temperature of the vapor the greater will be the influence of the air, thus, if the temperature in the preceding problem were 80 deg. fahr. the pressure of the air would be 0.306 and that of the vapor would be 0.505. The ill effects from air entrainment at low vacua are apparent.

Air in Condensers: Power, Feb. 29, 1916, p. 291; June 13, 1916, p. 834, Mar. 14, 1916, p. 376: Elec. Wld., July 8, 1916, p. 84; Jour. A.S.M.E., Feb., 1916, p. 190.

A condenser is a device in which the process of condensation and subsequent removal of the air and condensed steam is continuous, the degree of vacuum obtained depending upon the tightness of valves and joints, the quantity of entrained air, and the temperature to which the condensed steam is reduced.

The degree of vacuum may be expressed in different ways. (1) Excess of the atmospheric pressure over the observed vacuum. For example, a 26-inch vacuum implies that the pressure of the atmosphere is 26 inches of mercury above the pressure in the condenser. (2) Per cent of vacuum, by which is meant the ratio of the observed vacuum to the atmospheric pressure. Thus, with the barometer standing at 30 inches, a vacuum of 26 inches may be expressed as $100 \times \frac{26}{30} = 86.6$ per cent vacuum. This method of expression gives an idea of the efficiency of the condensing system. For example, the degree of vacuum indicated by 26 inches would be 93 per cent with a barometric pressure of 28 inches but only 84 per cent when the barometer reads 31 inches. (3) Absolute pressure. Thus a 26-inch vacuum referred to a 30-inch barometer would be indicated as a pressure of $30 - 26 = 4$ inches absolute, or 1.99 pounds per square inch.

The place of measurement of the vacuum should be stated since the lowest back pressure will be found at the air-pump suction, a higher pressure in the body of the condenser and the highest at the prime mover exhaust nozzle.

227. Effect of Aqueous Vapor upon the Degree of Vacuum. — The futility of attempting to better the vacuum by exhausting the vapor is best illustrated by a specific example.

Example 40. Required the volume of aqueous vapor to be withdrawn per hour from a condenser operating under the following conditions, in order that the vacuum may be increased one pound per square inch: Temperature of discharge water 125 degrees; corresponding vapor tension 4 inches of mercury; barometer 30 inches; relative vacuum 26 inches; horsepower 100; steam consumption 20 pounds per horsepower-hour; cooling water 25 pounds per pound of steam condensed.

$$100 \times 20 \times 25 = 50{,}000 \text{ pounds of cooling water per hour.}$$
$$= 833 \text{ pounds of cooling water per minute.}$$

Now to increase the vacuum one pound per square inch, approximately 2 inches of mercury, the temperature of the water must be lowered to 102 deg. fahr., that is, $833\ (125 - 102) = 19{,}159$ B.t.u. must be abstracted from the water in one minute, or $\frac{19{,}159}{1030} = 18.6$ pounds of water to be evaporated per minute. (1030 = average heat of vaporization of water under 26 to 28 inches of vacuum.) Now, one pound of vapor at 102 to 125 deg. fahr. has an average volume of 270 cubic feet.

Therefore $18.6 \times 270 = 5022$ cubic feet of vapor must be exhausted per minute to increase the vacuum from 26 to 28 inches, which while not impossible is manifestly impracticable for condenser practice.

In the Westinghouse-Leblanc refrigerating system cooling is effected by the withdrawal of aqueous vapor by means of an air pump.

228. Gain in Power due to Condensing. — The advantages to be gained by decreasing back pressure may be most readily illustrated by the following example:

Example 41. A non-condensing engine taking steam at a pressure of 100 pounds absolute and cutting off at one-quarter stroke will have, theoretically, a mean effective pressure on the piston of 44.6 pounds per square inch, the back pressure being 14.7 pounds per square inch absolute. If the engine exhausts into a condenser against a 26.5-inch vacuum (1.7 pounds absolute) the mean effective pressure will be increased to 44.6 + (14.7 − 1.7) = 57.6 pounds per square inch, resulting in a gain in power which may be expressed

$$\text{Hp.} = \frac{P_r A S}{33{,}000}, \tag{196}$$

in which

Hp. = horsepower gained,
P_r = reduction in back pressure, pounds per square inch,
A = area of the piston in square inches,
S = piston speed in feet per minute.

If P = mean effective pressure on the piston when running non-condensing, the percentage of increase of power may be expressed

$$\text{Per cent} = 100 \frac{P_r}{P}. \tag{197}$$

In the above example the percentage of power gained would be

$$100 \frac{13}{44.6} = 29.2 \text{ per cent.}$$

The actual gain due to the use of the condenser would be much less than this, depending upon the type of engine and conditions of operation.

TABLE 90.

PRESSURE OF AQUEOUS VAPOR IN INCHES OF MERCURY FOR EACH DEGREE F.

(Marks and Davis.)

	0°	1°	2°	3°	4°	5°	6°	7°	8°	9°
30°.........	......	......	.180	.188	.195	.203	.212	.220	.229	.238
40°.........	.248	.257	.268	.278	.289	.300	.312	.324	.336	.349
50°.........	.362	.376	.390	.405	.420	.436	.452	.468	.486	.503
60°.........	.522	.541	.560	.580	.601	.622	.644	.667	.690	.714
70°.........	.739	.764	.790	.817	.845	.873	.903	.964	.996	1.03
80°.........	1.03	1.06	1.10	1.13	1.17	1.21	1.25	1.30	1.33	1.37
90°.........	1.42	1.46	1.51	1.55	1.60	1.65	1.71	1.76	1.81	1.87
100°.........	1.93	1.98	2.04	2.11	2.17	2.24	2.30	2.37	2.44	2.51
110°.........	2.60	2.66	2.74	2.82	2.90	2.99	3.07	3.16	3.25	3.34
120°.........	3.44	3.53	3.63	3.74	3.84	3.95	4.06	4.17	4.28	4.40
130°.........	4.52	4.64	4.76	4.89	5.02	5.16	5.29	5.43	5.58	5.73
140°.........	5.88	6.03	6.18	6.34	6.51	6.67	6.84	7.02	7.20	7.38

With steam turbines the advantage gained by reduction of back pressure is more marked than with the reciprocating engine, though theoretically the same for the same range of expansion. Initial condensation, leakage past valves, and other sources of loss prevent a reciprocating engine from benefiting from a good vacuum to the same extent as a turbine. See paragraph 223.

Referring again to the example given above, if the steam is cut off at about one-sixth stroke, the work done when running condensing will be the same as when running non-condensing and cutting off at one-quarter. Theoretically the steam consumption will be decreased nearly in proportion to the reduction in cut-off. Generally speaking, a condensing engine will require from 20 to 30 per cent less steam for a given power than a non-condensing engine. (See results of engine tests, paragraph 181.) This decrease in steam consumption is only an apparent one. If steam is used by the auxiliaries in creating the vacuum, the amount must be added to that consumed by the engine, unless the steam exhausted by the former is utilized to warm the feed water, in which case only the difference between the heat entering the auxiliaries and that returned to the heater should be charged against the engine. The power necessary to operate the condenser auxiliaries varies from one to six per cent of the main engine power, depending upon the type and conditions of operation.

In power plants where the exhaust steam is not used for heating or manufacturing purposes, the engines are almost invariably operated condensing, provided there is an abundant supply of cooling water. Even if the water supply is limited, it is often found to be economical to use some artificial cooling device, notwithstanding the high first cost and cost of operation of the latter.

Some of the considerations affecting the propriety of running condensing and the choice of condensing systems are taken up in paragraph 249.

229. Classification of Condensers. — The following is a classification of a few well-known condensers:

1. Jet condensers	Parallel current (*a*)	Standard low level	Worthington. Blake. Deane.
		Siphon	Baragwanath. Bulkley.
		Ejector	Schutte. Körting.
	Counter current (*b*)	Barometric	Weiss. Alberger.
		High vacuum	LeBlanc. Wheeler. Worthington.
2. Surface condensers	Water cooled (*a*)	Single-flow	Baragwanath.
		Double-flow	Wheeler.
		Multi-flow	Wainwright.
	Air cooled (*b*)	Forced draft	Fouche.
		Natural draft	Pennell.
	Evaporative (*c*)		Ledward.

Condensers may be divided into two general groups:

1. *Jet condensers*, in which the steam and cooling water mingle and the steam is condensed by direct contact, Figs. 292 to 300.

2. *Surface condensers,* in which the steam and cooling medium are in separate chambers and the heat is abstracted from the steam by conduction, Figs. 305 to 309.

Jet condensers may be further grouped into two classes, according to the direction of flow of the air and cooling water:

(*a*) *Parallel-current condensers,* in which the condensed steam, cooling water, and air flow in the same direction, collect at the bottom of the condenser chamber, and are exhausted by a suitable pump, Fig. 292.

(*b*) *Counter-current condensers,* in which the cooling water and condensed steam flow from the bottom of the chamber, while the air is drawn off at the top, Fig. 301.

Parallel-current condensers may be subdivided into three classes:

(1) *Standard condensers,* in which the cooling water, condensed steam, and air are exhausted by a vacuum pump, Fig. 292.

(2) *Siphon condensers,* in which the cooling water, condensed steam, and air are exhausted by a barometric column, Fig. 297.

(3) *Ejector condensers,* in which the condensed steam and air are exhausted by the cooling water on the ejector principle, Fig. 298.

Surface condensers may be classified according to the nature of the cooling medium as

(*a*) *Water-cooled condensers.*

(*b*) *Air-cooled-condensers.*

(*c*) *Evaporative condensers,* in which the condensation of the steam is brought about by the evaporation of a fine stream of water trickling on the surface of the tubes.

230. Standard Low-level Jet Condensers. — Fig. 292 shows a section through a Worthington jet condenser, illustrating the low-level type in which the condensing water is drawn into the apparatus by the vacuum. When the pump is started a partial vacuum is created in the suction chamber above the valves *H*, *H* in the cone *F*. As soon as sufficient air has been exhausted, cooling water enters at *B* with a velocity depending upon the degree of vacuum in chamber *F* and the suction head, and is divided into a fine spray by the adjustable serrated cone *D*. The spray mingles with the exhaust steam entering at *A* and both move downwards with diverse velocities. The steam gives up its heat to the water and condenses. The velocity of the steam diminishes in its downward path to zero, while the velocity of the water increases according to the laws of falling bodies. The condensed steam, cooling water, and air collect at the lower part of the condenser and are exhausted by the *wet air pump G*, from which they are forced through opening *J* to the hot well. The vacuum in chamber *F* will depend upon the vapor tension of the warm water in the bottom of the well, the amount of air carried along by the cooling water and steam, and the tightness of valves and joints. In case the water accumulates in the condenser cone F, either by reason of an increased supply or by a sluggishness or even stoppage of the pump, the condensing surface is

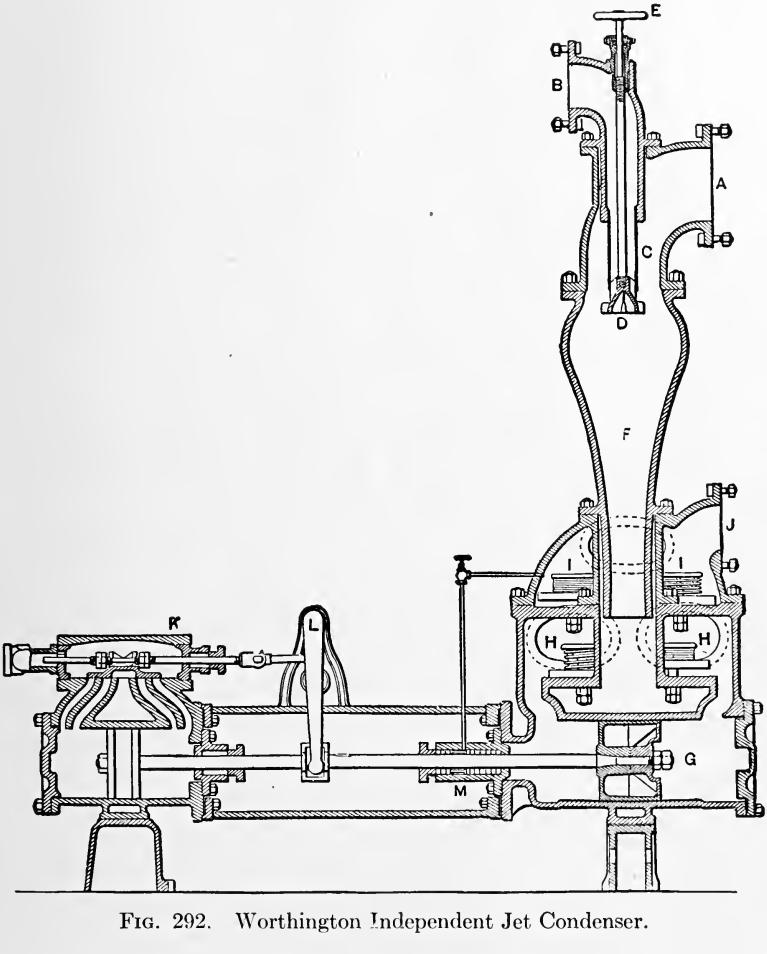


FIG. 292. Worthington Independent Jet Condenser.

reduced to a minimum, as soon as the level of the water reaches the spray pipe and the spray becomes submerged, and only a small annular surface of water is exposed to the exhaust steam. The vacuum is immediately broken, and the exhaust steam escapes by blowing through the injection pipe and through the valves of the pump and out the discharge pipe at J, forcing the water ahead of it; consequently flooding of the steam cylinder cannot occur. In starting up the condenser a partial vacuum for inducing a flow of injection water into the condenser chamber may be created by the pump if the suction lift is not too great. Many engineers, however, prefer to install a small forced injection or priming pipe the function of which is to condense sufficient steam to produce the necessary partial vacuum.

Fig. 293 shows a section through the condensing chamber and air pump of a Blake vertical jet condenser with an automatic vacuum-breaking device. The injection water enters at opening marked "injection" and flows through the adjustable "spray" nozzle in a fine spray,

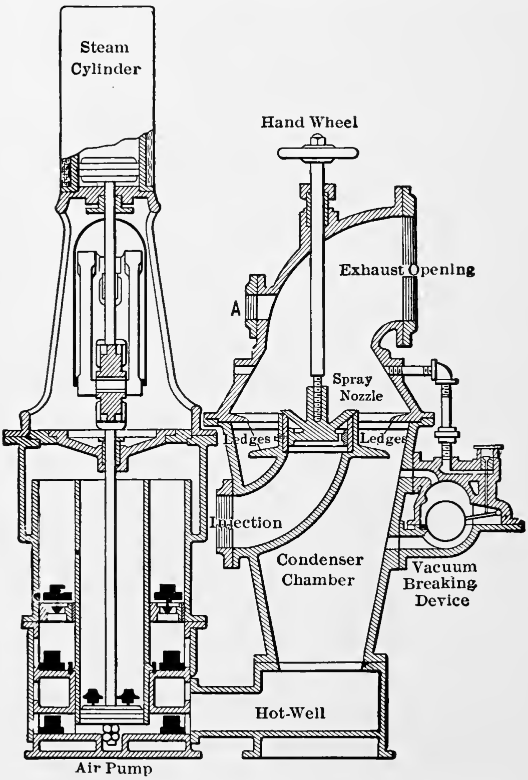


Fig. 293. Section through a Blake Jet Condenser.

at an angle of about 45 degrees, and impinges on the conical sides of the upper condenser chamber. The spray falls from the sides to the projecting ledges shown in the illustration. The ledges prevent the spray from falling directly to the bottom of the chamber and insure an efficient mingling of steam and cooling water. A perforated copper plate is substituted for the shelves when the force of the injection water is not sufficient to produce spray. The circulating water and condensed steam together with the non-condensable gases are drawn off at the bottom of the chamber. The vacuum-breaking device is shown at the right of the figure. When the rising water reaches the level of the float chamber, as in the case of an accidental stoppage of the air pumps, the float is raised and forces a check valve from its seat and allows an inrush of air to break the vacuum, thus preventing further suction of water into the condenser and consequent flooding of the engine. *A* is the forced injection or "priming" inlet used in starting up when the suction lift is considerable.

231. Injection Orifice. — The velocity of water entering a jet condenser, neglecting friction, may be determined from the equation

$$V = \sqrt{2\,gh}, \tag{198}$$

where

V = velocity of the water in feet per second,
g = acceleration of gravity = 32.2,
h = total head in feet.

If p = pressure below the atmosphere in pounds per square inch,
h_1 = distance in feet between the source of supply and the injection orifice,

then

$$h = 2.3\,p \pm h_1, \tag{199}$$

and equation (198) may be written

$$V = 8.025\,\sqrt{2.3\,p \pm h_1}. \tag{200}$$

If the supply is under pressure, h_1 is positive; if under suction, it is negative.

Example 42. What is the theoretical velocity of water entering a condenser with 26-inch vacuum (referred to 30-inch barometer); suction head 8 feet?

Here p = pressure in pounds per square inch, corresponding to 26 inches of mercury = 12.8 pounds per square inch.

$$\begin{aligned} h_1 &= 8. \\ V &= 8.025\ \sqrt{2.3 \times 12.8 - 8} \\ &= 37.1 \text{ feet per second} \\ &= 2226 \text{ feet per minute.} \end{aligned}$$

In proportioning the injection orifice in practice the maximum velocity of flow is assumed to be between 1500 and 1800 feet per minute, or, approximately, area of injection orifice in square inches = weight of injection water in pounds ÷ 650 to 780. ("Manual of Marine Engineering," Seaton, p. 204.) A rough rule gives area of orifice = area of low-pressure piston in square inches ÷ 250. (Seaton, p. 204.)

232. Volume of the Condenser Chamber. — According to Thurston the volume of a jet condenser should be from one fourth to one half that of the low-pressure engine cylinder. ("Steam Engine Manual," Thurston, II, 127.)

According to Hutton the volume should not be less than that of the air pump and should approximate three fourths that of the engine cylinder in communication with it.

233. Injection and Discharge Pipes. — In practice the diameter of the injection pipe is based on a velocity of 400 to 600 feet per minute and that of the discharge pipe of 200 to 400 feet per minute; the lower figures for pipes under 8 inches in diameter, the upper range for larger diameters.

(Atmospheric relief valves. — See paragraph 363.)

234. High-vacuum Jet Condensers. — The standard low-level jet condenser is not suitable for high vacua because of the limited air capacity

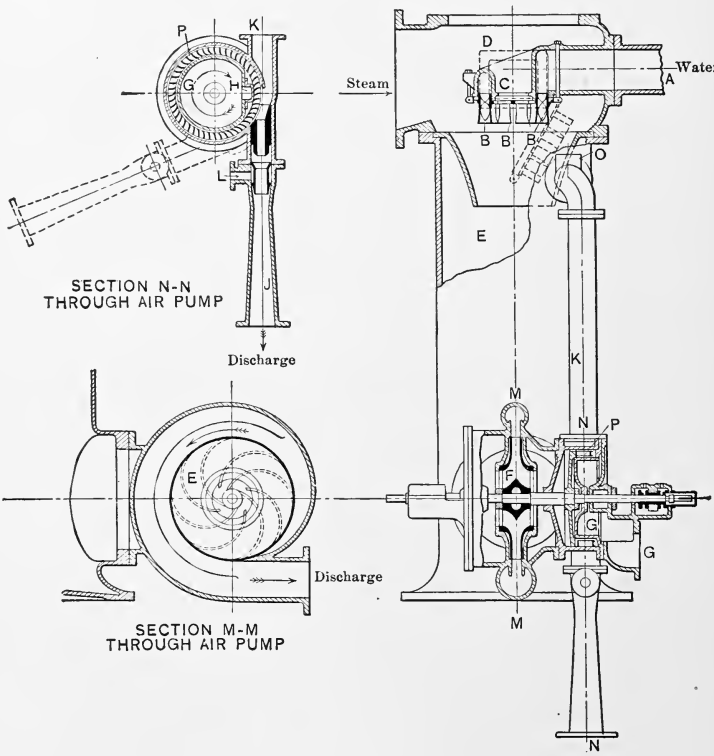


Fig. 294. Westinghouse-Leblanc Multi-jet High-vacuum Condenser System.

of the combined air and circulating-water pump. Even with a tight system considerable air is carried into the condenser with the circulating water and efficient removal of the air necessitates a larger pump capacity than is usually furnished with this type of condenser. Low-level jet condensers may be operated with a high degree of vacuum by equipping them with independent air and circulating pumps. Examples of this type of jet condenser are illustrated in Figs. 294 to 296. Referring to Fig. 294 which gives several views of the Leblanc type of condenser, steam enters the condensing chamber as indicated and meets the cooling water injected through spray nozzle C. The condensed steam and injection water fall to the bottom of the vessel and are removed by centrifugal pump M. Air saturated with water vapor is withdrawn by centrifugal air pump P through suction opening O. Referring to section NN through the air pump it will be seen that this device consists primarily of a reverse Pelton wheel in conjunction

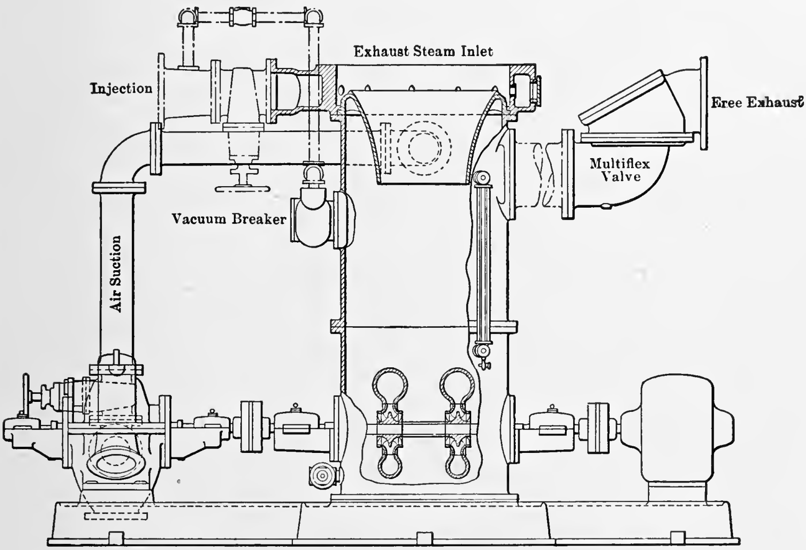


FIG. 295. C. H. Wheeler Low-level, High-vacuum Jet Condenser.

with an ejector. Sealing water is introduced through the branch indicated by dotted outline into the central chamber G from which it passes through port H. It is then caught up by the blades P of the Pelton wheel, which is rotated at a suitable speed, and ejected into the discharge cone in the form of thin sheets having a high velocity. These sheets of water meet the sides of the discharge cone and thus form a series of water pistons, each of which entraps a small pocket of air and forces it out against the atmospheric pressure. In passing through the air pump the sealing water receives practically no increase in temperature, hence the same water may be used over and over again. The air pump rotor and main pump runner are enclosed in a common casing mounted on the same shaft. There is a clear passage through the condenser and pump, so that, should the pump stop for any reason, air rushes into the condenser through the air pump and immediately breaks the vacuum. In starting up the condenser steam is turned into auxiliary nozzle *L*, section *NN*, for a few moments, thus creating sufficient vacuum to start the regular flow of water through the air pump.

Any type of air pump may be used in connection with a suitable circulating pump but the majority of low-head, high-vacuum jet condensers are equipped with the hydraulic type. Recent experiments

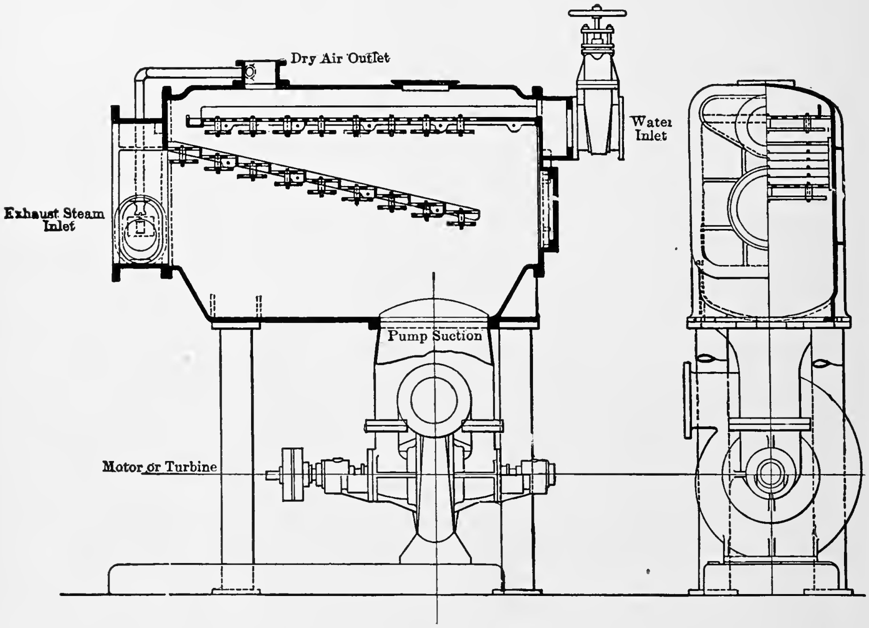


FIG. 296. Wheeler Low-level Centrifugal Jet Condenser.

indicate that the steam jet type of air ejector may supersede the mechanically-operated air pump within a very short time. (See paragraph 309.)

235. Siphon Condensers. — Fig. 297 shows a section through a Baragwanath siphon condenser, illustrating the principles of a parallel-current barometric condenser. The cooling water enters the side of the condenser chamber at *A* and passes downward in a thin annular sheet around the hollow cone *D*. The exhaust steam enters at *B* and is given a downward direction by the goose neck *C*. It flows through the nozzle *D* and is condensed within the hollow cone of moving water, the combined mass including the entrained air discharging through the contracted throat *E* at high velocity into the tail pipe *O*. The water column in the tail pipe must be enough to overcome the pressure of the atmosphere; i.e., it should be 34 feet or more above the surface of the hot well, otherwise water would rise within this pipe to a height corresponding to that of the barometer, which is approximately 34 feet for a barometric pressure of 30 inches of mercury. This is not strictly true when the condenser is in full operation, as the injector effect of the moving mass is sufficient to overcome several pounds pressure, and the tail pipe may be less than 34 feet, but to provide against any possibility of the water being drawn into the cylinder of the engine the length is made greater than 34 feet. The spray cone D is adjustable and admits of close regulation of the water supply without changing the annular form of the stream. The condensing water may be supplied under pressure or under suction. For lifts not greater than 15 feet no supply pump is necessary, the water being raised by the siphon action of the condenser. This condenser requires the same amount of cooling water per pound of steam as the standard jet condenser, and is capable of maintaining a vacuum of from 24 to 27 inches. A vacuum of $28\frac{1}{2}$ inches has been recorded for a condenser of this general type. (Trans. A.S. M.E., 26–388.) An atmospheric relief valve G is provided in case the vacuum fails from any cause, which will permit the steam to escape to the atmosphere.

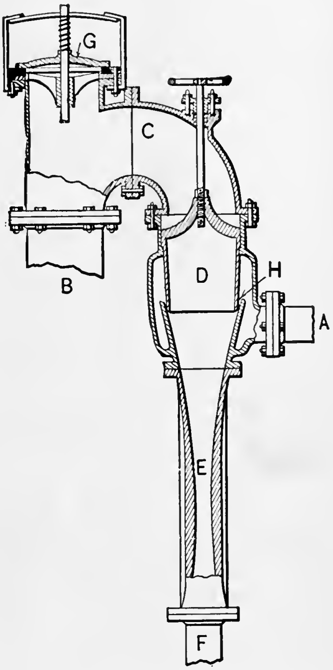


FIG. 297. Baragwanath Siphon Condenser.

The above type of condenser is adapted to very muddy cooling water, since no filtration is necessary beyond the removal of such solid matter as may clog up the annular space H.

Siphon Condensers, Discussion: Trans. A.S.M.E., Vol. 26, p. 388. *Siphon Condensers:* Electrical World, June, 1897, p. 818; Engr. U. S., Jan., 1906.

236. Size of Siphon Condensers. — The size of siphon is indicated by the diameter of the engine exhaust pipe.

Table 91 gives the sizes of barometric condensers as manufactured by prominent makers.

The diameter of the throat may be closely approximated by the empirical formula

$$\text{Diam. in inches} = 0.0077 \sqrt{Ww}, \qquad (201)$$

in which

W = weight of steam to be condensed per hour,
w = weight of water required to condense one pound of steam.

The maximum width of the annular opening for the admission of water may be obtained from the empirical formula

$$\text{Width in inches} = \frac{Ww}{39{,}550\, d}, \tag{202}$$

in which

d = diameter of the nozzle or bottom of the cone in inches.
W and w as in equation (201).

TABLE 91.

SIZE OF SIPHON CONDENSERS.

Steam to be Condensed.		Size Usually Furnished, Inches.	Steam to be Condensed.		Size Usually Furnished, Inches.
Pounds per Hour.	Pounds per Minute.		Pounds per Hour.	Pounds per Minute.	
2,000	33	5	8,000	133	10
3,000	50	7	10,000	166	12
4,000	66	8	15,000	250	14
5,000	83	9	20,000	333	14
6,000	100	9			

Vacuum 26 inches; barometer 30 inches.

237. Ejector Condenser. — Fig. 298 shows a section through a Schutte exhaust steam "induction" condenser, illustrating the principles of the ejector condenser in which the momentum of flowing water ejects the discharge without the aid of the circulating pump. Exhaust steam enters the ejector through the opening marked "exhaust," passes through a series of inclined orifices and nozzles at considerable velocity, and, meeting the cooling water in the inner annular chamber, is condensed. The cooling water is drawn in continuously through the opening marked "water," by virtue of the vacuum formed, and sufficient velocity is imparted to the jet to discharge the combined mass. of condensed steam, cooling water, and air against the pressure of the atmosphere.

The condenser should be installed vertically with three feet of pipe between the strainer and the head of the condenser and should be arranged as shown in Fig. 299. There should be a clear discharge of not less than two feet below the bottom flange of the apparatus to the level of the water in the discharge sump, or hot well. It is advisable that the end of the discharge pipe be sealed under water, unless there is a horizontal discharge main, and trap to water seal at the bend immediately under the condenser. Except with condenser of very large size a difference of level between supply and discharge of 30 feet will usually give the necessary pressure of water at the condenser with full allow-

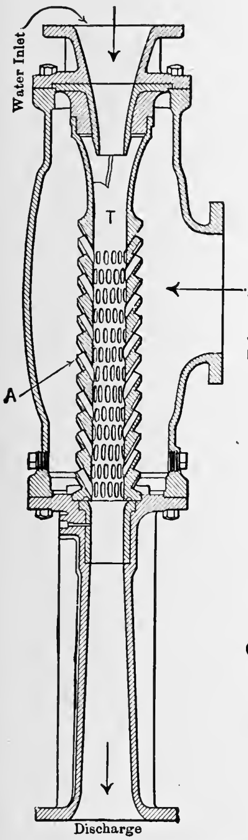


FIG. 298. Schutte Ejector Condenser.

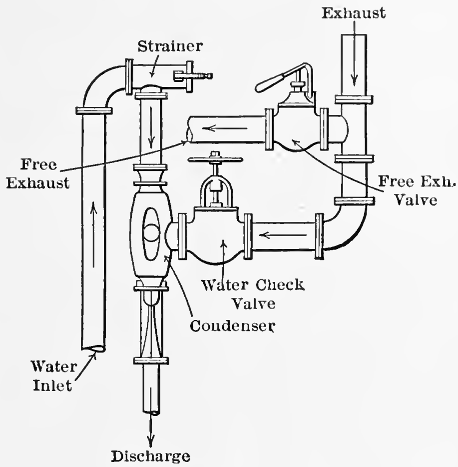


FIG. 299. Piping for Schutte Ejector Condenser.

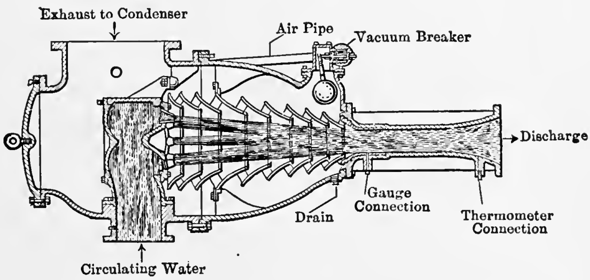


FIG. 300. Section through Condensing Chambers of Körting Multi-jet Condenser. Chamber Capable of Maintaining a Vacuum of 95 Per Cent of the Ideal without the Use of Air Pumps.

ance for friction losses. These condensers are made in all sizes conforming with exhaust pipe diameters of $1\frac{1}{2}$ to 24 inches. The same amount of cooling water is required as for jet condensing and vacua of 20 to 25 inches are readily obtained.

238. Barometric Condensers.* — Fig. 301 shows a section through a Weiss counter-current condenser, illustrating the principles of a barometric jet condenser. The cooling water enters the upper part

* The author has been informed that the word "Barometric" in connection with jet condensers is the registered trade mark of the Alberger Condenser Company.

of the condensing chamber *A* through pipe *N* and falls in cascades, as shown in the figure, to tail pipe *B*, from which it flows by gravity to the hot well. The exhaust steam enters chamber *A* through pipe *D*, and, coming in contact with the cold-water spray, is condensed. The air is exhausted from the top of the condenser by a dry vacuum pump through pipe *F*. In flowing to the pump the air passes upwards through the water spray and its temperature is lowered to that of the injection water, thereby reducing the volume to be exhausted. Any moisture passing over with the air is separated at *G* before reaching the air pump, and flows out through the small barometric tube *H*. The cooling water is forced to the condenser chamber through pipe *N* by any positive displacement pump, the actual head pumped against being the difference between the total height and that of a column of water corresponding to the degree of vacuum in the condenser. The main barometric tube or tail pipe *B* through which the water is discharged is 34 feet or more in length and is provided with a foot valve *C*. The counter-current principle permits a much higher temperature of hot well for the same degree of vacuum than does the parallel current, a hot-well temperature of 120 degrees and a vacuum of 27 inches being readily maintained. A small pipe *K* connecting the main condenser with the small barometric tube *H* insures at all times a sufficient quantity of water in the small auxiliary hot well to seal the tube. The water from this auxiliary hot well flows over a weir, as indicated, into a counterweighted bucket *M*, the latter having a hole in the bottom which allows the normal flow to escape. But in case a sudden heavy overload is thrown on the engines, and the adjustment is for a light load, the temperature of the discharge will reach the boiling point and an abnormal quantity of water will flow down the small barometric tube. This will cause the water to flow into the bucket much faster than the opening in the bottom can dispose of it;

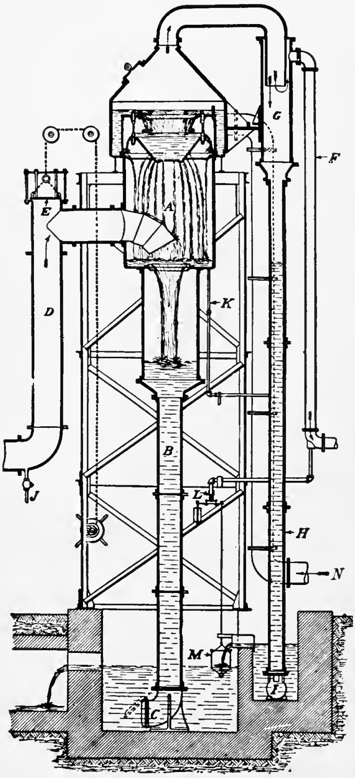


Fig. 301. Weiss Counter-current Condenser.

as a result the bucket will increase in weight and will open up a free-air valve L which reduces the vacuum two or three inches and raises the boiling point without "dropping" the vacuum entirely. E is the atmospheric relief valve.

Fig. 302 shows a section through the condensing chamber of an Alberger barometric condenser. In principles of operation the condenser is similar to the Weiss, but differs considerably in details. Exhaust steam enters at A and divides into two streams, one flowing directly to the inner chamber D, the other through the annular space E. Cooling water enters through B and is broken up into a fine spray by the serrated cone F, which is hung upon a long spring, thus automatically adjusting itself to the quantity of water entering the condenser. After condensing the exhaust steam in the inner cylinder the partly heated spray of cooling water in falling is brought in contact with the exhaust steam which enters through the annular space. This process permits of a high hot-well temperature without affecting the degree of vacuum. The air which is not entrained by the cooling water and carried down the tail pipe collects under the spray cone F and ascends through the tubular support of the cone into the air cooler. This air cooler is simply a small chamber in which the non-condensable gases are cooled by a small portion of the circulating water before they are withdrawn by the air pump. The circulating water used for the purpose is forced into the cooling chamber through pipe K and falls through serrated openings in the bottom to the condenser proper. The air enters the chamber through these same openings, and is withdrawn by the air pump. Surrounding the cooler is a separating space of large capacity to allow the subsidence of any entrained moisture before the air reaches the vacuum pump.

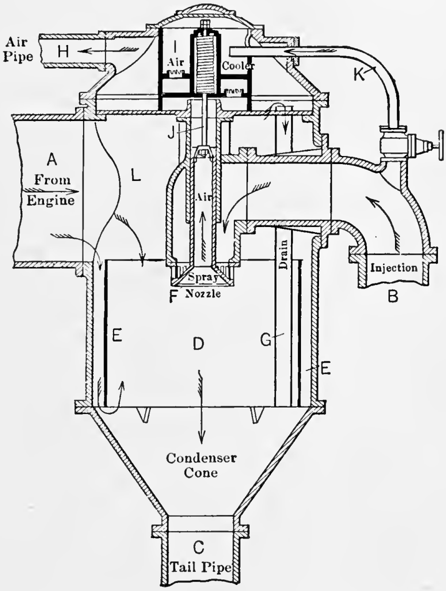


Fig. 302. Section through Condensing Chamber, Alberger Barometric Condenser.

Fig. 303 shows a section through a Tomlinson type B barometric condenser which differs from the conventional type in the addition of an overflow or auxiliary tail pipe. The main tail pipe takes care of the light loads and the overflow comes into service only on full loads and overloads. This arrangement reduces the quantity of circulating water required at light loads since it is not necessary to keep a large tail pipe filled with water as is the case with the single pipe design.

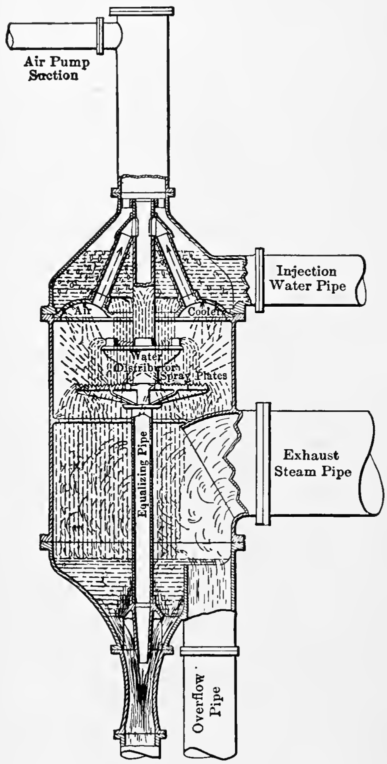


FIG. 303. Tomlinson Type B Barometric Condenser.

239. Condensing Water: Jet Condensers. — In a jet condenser the cooling water and exhaust steam mingle, and the degree of vacuum is a function of the final or discharge temperature; thus the quantity of cooling water required depends upon its initial temperature, the temperature of the discharge water, and the total heat of the steam entering the condenser. If the steam in the low-pressure cylinder at exhaust is dry and saturated, and there is no air entrainment the heat entering the condenser will correspond to the total heat of saturated steam at condenser pressure. This condition is not likely to occur in practice since exhaust steam usually carries considerable moisture and there will be more or less air entrained with it. Furthermore, the cooling water contains air in varying amounts so that the total amount of air entering the condenser may be considerable. Neglecting radiation and leakage the heat absorbed by the cooling medium must be equal to that given up by the steam and its air entrainment. The heat exchange may be expressed

$$R = \frac{H_m - q_2}{q_2 - q_0}, \tag{203}$$

in which

R = weight of injection water necessary to condense and cool one pound of air-vapor mixture,

H_m = heat content of the air-vapor mixture at condenser pressure, B.t.u. per lb. above 32 deg. fahr.,

q_2 = heat of liquid of the discharge water, B.t.u. per lb.,

q_0 = heat of liquid of the injection water, B.t.u. per lb.

In practice it is sufficiently accurate to neglect the influence of the air on the heat content of the exhaust steam and circulating water, and the mean specific heat of water under condenser conditions may be taken as unity so that equation (203) may be written,

$$R = \frac{H - t_2 + 32}{t_2 - t_0}, \tag{204}$$

in which

H = heat content of the exhaust, B.t.u. per lb. above 32 deg. fahr.,
t_2 = temperature of the discharge water, deg. fahr.,
t_1 = temperature of the injection water, deg. fahr.

It has been shown (paragraph 177) that

$$\begin{aligned} H &= H_i - H_r - \frac{2546}{w} \\ &= H_i - H_r - \frac{3412}{w_1}, \end{aligned}$$

in which

H_i = initial heat content of the steam entering the prime mover, B.t.u. per lb. above 32 deg. fahr.,
H_r = heat lost by radiation from the prime mover and exhaust piping, B.t.u. per lb. of steam admitted,
w = water rate, lb. per brake hp-hr.,
w_1 = water rate, lb. per kw-hr.

In a well-lagged piston engine with short connection to the condenser the loss by radiation varies from 0.3 to 2.0 per cent, but seldom exceeds 1 per cent of the total heat admitted, and in a turbine this loss is even less, and 0.5 per cent is a very liberal allowance. The temperature of the discharge water will approximate that of the vapor at its partial pressure. For air-free steam this will correspond to that of vapor at total condenser pressure. In high-vacuum jet condensers in which the air pressure is kept very low this depression of the hot-well temperature will range from 0 to 5 degrees below that of vapor at total condenser pressure, and in the ordinary low-vacuum condenser it may range from 15 to 25 degrees below. The influence of air entrainment for a specific case is illustrated in Fig. 291. The minimum weight of cooling water for air-free steam at various vacua is shown graphically in Fig. 304.

Example 43: Determine the amount of cooling water necessary per pound of steam for a standard low-vacuum jet condenser operating under the following conditions: Engine uses 16 lb. steam per brake hp-hr., initial pressure 140 lb. per sq. in. absolute, superheat 50 deg. fahr., vacuum 26 inches referred to a 30-inch barometer, temperature of injection water 70 deg. fahr.

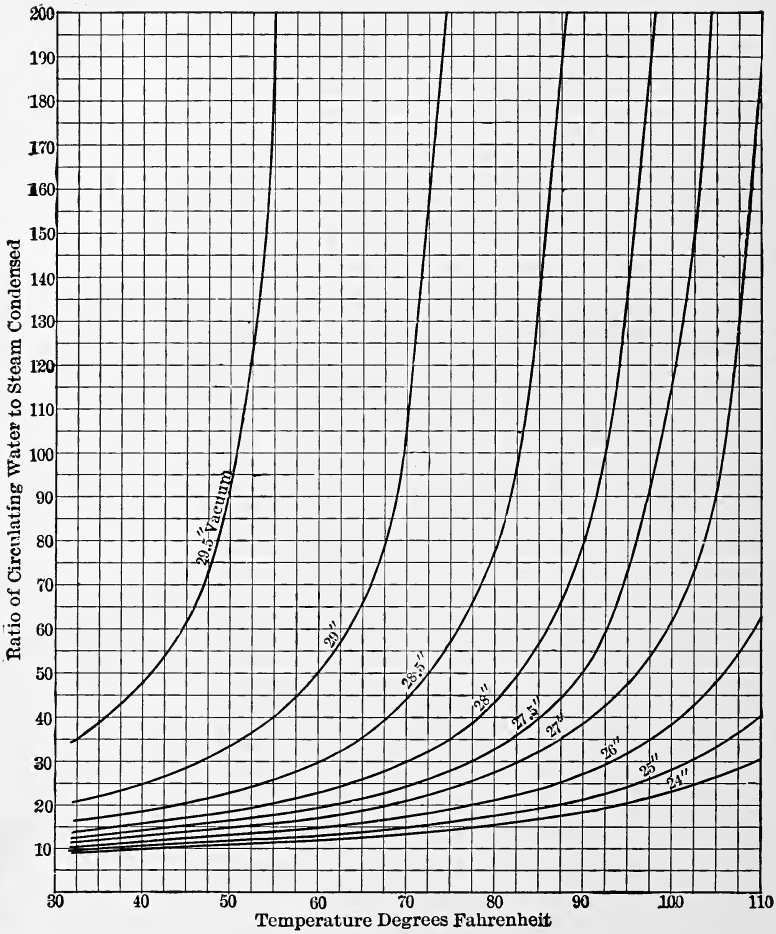


FIG. 304. Curves showing Minimum Ratio of Circulating Water to Steam Condensed for Various Initial Temperatures.

From steam tables $H_i = 1221$; assume $H_r = 1$ per cent of H_i, then

$$H = 1221 - 0.01\,(1221) - \frac{2546}{16} = 1050.$$

The temperature t_s of vapor corresponding to an absolute pressure of 4 inches = 126 deg. fahr.* Assume $t_2 = t_s - 15 = 111$.

* This is not the actual temperature in the condenser. The actual temperature will be that corresponding to the partial pressure of the vapor. For convenience in calculation the temperature in the condenser is assumed to correspond to that of the total pressure and the temperature depression of the hot well is then based on this hypothetical temperature. When the extent of air leakage and entrainment is known the actual temperature in the condenser may be readily calculated.

Substituting these values in equation (204),

$$R = \frac{1050 - 111 + 32}{111 - 70} = 24.3 \text{ lb.}$$

Example 44. Determine the amount of cooling water necessary per pound of steam for a high-vacuum jet condenser operating under the following conditions. Turbine uses 14 lb. steam per kw-hr., initial pressure 165 lb. per sq. in. absolute, superheat 120 deg. fahr., vacuum 29 inches referred to a 30-inch barometer, temperature of injection water 65 deg. fahr.

From steam tables, $H_i = 1262$; $t_s = 79$; assume $H_r = 0.005\,H_i$. (This is so small that it may be omitted, particularly in view of other assumptions which may be made.)

Then,

$$H = 1262 - 0.005\,(1262) - \frac{3412}{14} = 1012.$$

Assume

$$t_2 - t_s - 4 = 75$$

$$R = \frac{1012 - 75 + 32}{75 - 65} = 96.9 \text{ lb.}$$

240. Water-cooled Surface Condensers.—With the exception of the "standard" water-works condenser all water-cooled surface condensers are of the water-tube type, that is, the cooling water passes through the tubes. Fig. 305 shows a sectional elevation of the simplest type of surface condenser. It consists essentially of a cast-iron shell provided with two heads, into which a number of brass tubes are expanded. Exhaust steam fills the shell and flows around and between the tubes, while the cooling water is forced through the tubes by means of a circulating pump. The steam is condensed by contact with the tubes and drops to the bottom tube sheet from which it is exhausted by the

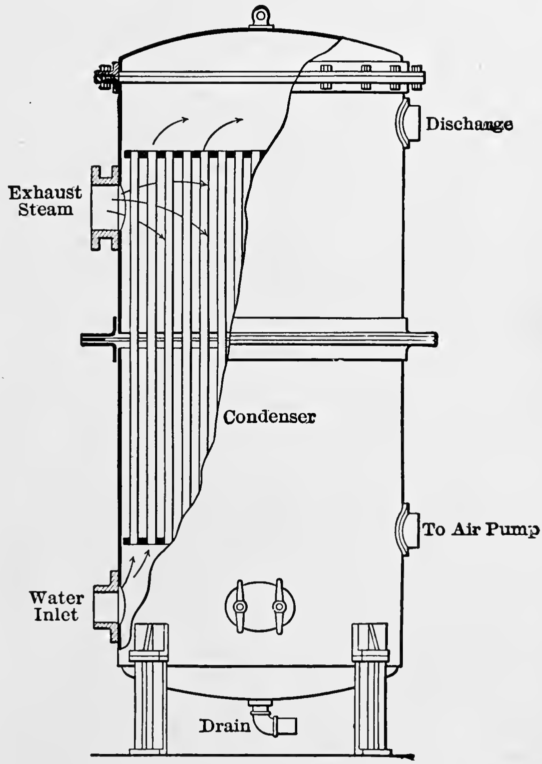


Fig. 305. Baragwanath Surface Condenser.

air pump. The circulating water flows through the tubes in one direction only, hence the name "single flow." To allow for unequal expansion of shell and tubes the two halves of the shell are provided with slightly thinner plates flanged outward, the flanges being bolted together with a spacing ring between them. This joint gives the shell, in the direction of its length, a certain amount of elasticity which is sufficient to allow for the greatest elongation of the tubes, without straining the tube sheet and causing leakage. This type of condenser is the least efficient of all since (1) the velocity of the water through the tubes is low; (2) the tubes are blanketed with a film of condensed steam which increases in thickness from top to bottom, and (3) air stagnates in the chamber opposite the air pump suction. The influence of these factors on the heat transmission is discussed in paragraph 242.

Fig. 306 shows a section through a "two-pass" condenser unit which is an improvement over the one just described, in that for a given temperature rise the velocity of the water through the tubes may be increased by doubling the length of its travel. In other respects, however, it is open to the same criticism as the single-flow device. The arrangement shown in Fig. 306 is not intended for high-vacuum work.

Replacing the combined air and condensate pump by independent pumps will result in higher vacua but the tube arrangement is not conducive to high efficiencies.

At the time of the introduction of the steam turbine it was discovered that a very high vacuum would improve turbine economies to an extent hitherto impossible when applied to reciprocating engines. This condition naturally created an era of development among the condenser designers. It became evident at once that the old types that were capable of creating a 26-inch or 27-inch vacuum would require considerable modification to maintain a vacuum of 29.0 or 29.5 inches. Any number of condensers have been designed which are capable of maintaining a vacuum of 29.0 inches referred to a 30-inch barometer, but that the art is still in an experimental stage is evidenced by the fact that each new installation differs from the preceding one even for practically identical operating conditions. Engineers are agreed that (1) steam should enter the condenser with the least practical resistance and the pressure drop through the condenser should be reduced to a minimum; (2) air should be rapidly cleared from the heat transmitting surfaces, collected at suitable places, freed from entrained water and removed at a low temperature with least expenditure of mechanical energy; (3) condensate should also be rapidly cleared from the heat transmitting surfaces, freed from air and returned to the boilers at the maximum practical temperature; (4) circulating water should pass

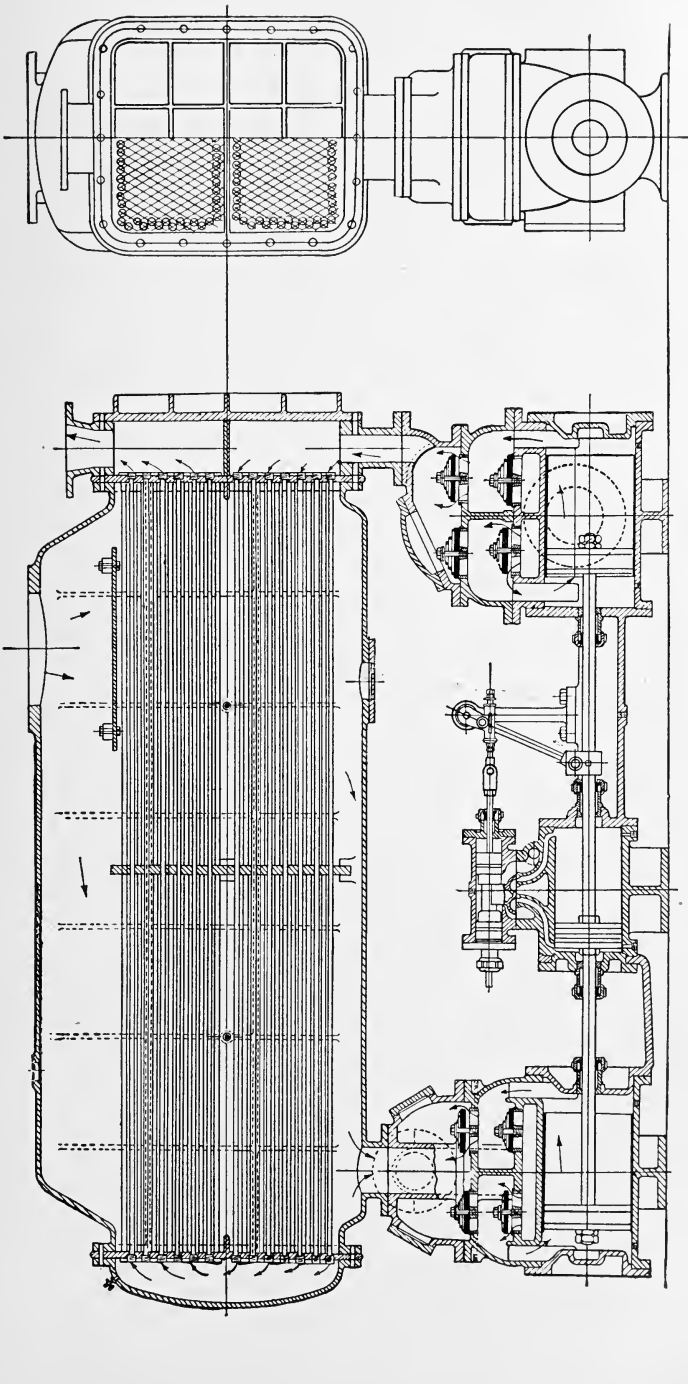


Fig. 306. Wheeler Admiralty Surface Condenser and Pumps.

through the condenser with least friction but at a velocity consistent with high efficiencies.

In the types of condensers described above the steam diminishes in value, due to condensation, as it passes over the tubes, hence the velocity decreases and becomes practically zero at the bottom of the vessel. The velocity of the entrained air also decreases in its passage through the condenser and becomes stagnate. By shaping the condenser as shown in Fig. 307, the original velocity may be maintained to the point of air offtake. In the modern type of high-vacuum condenser the same effect has been realized by establishing steam lanes, by means of differential tube spacing or by a combination of both as indicated in Fig. 308. The latest practice

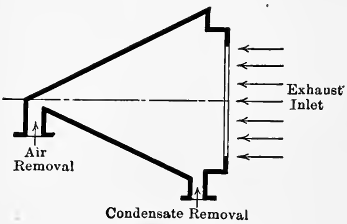


FIG. 307. Theoretically Correct Condenser Shape.

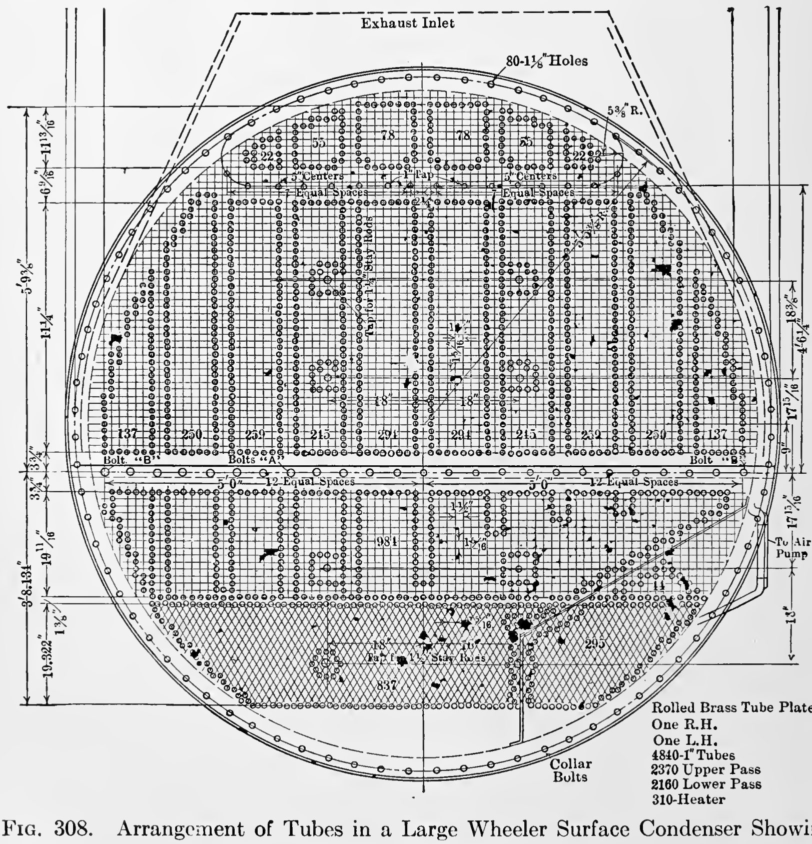


FIG. 308. Arrangement of Tubes in a Large Wheeler Surface Condenser Showing Steam Lane and Differential Spacing.

is in favor of the differential spacing, that is, the tubes are spaced evenly across the path of the steam, leaving no preferential lanes down which the steam can short circuit. This uniform spacing is maintained for a portion of the upper cooling surface, after which the

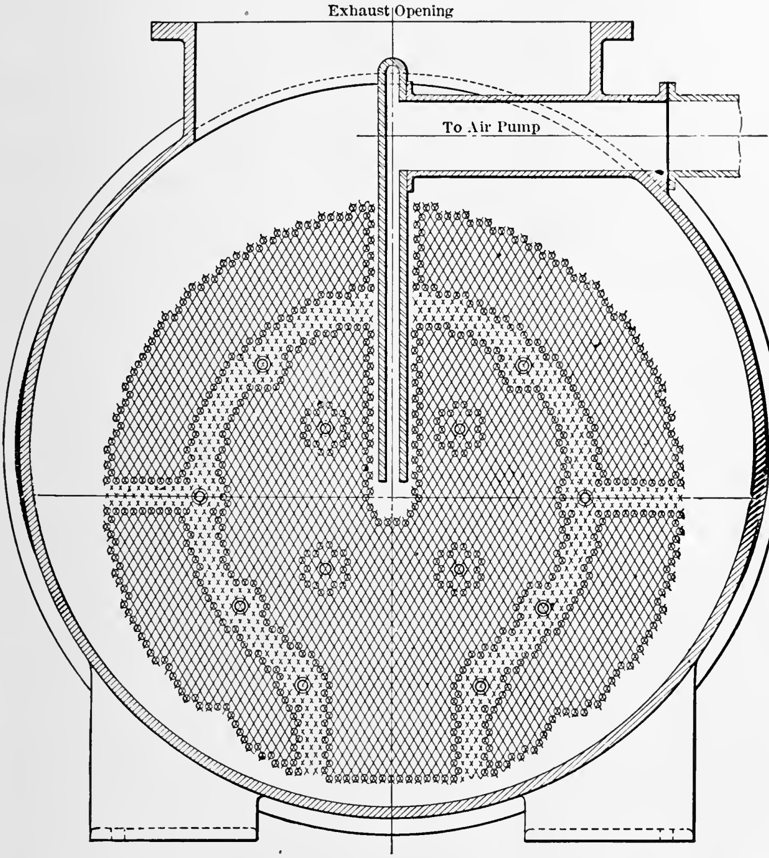


FIG. 309. Tube Arrangement in Westinghouse Radial Flow Surface Condenser.

distance between centers is gradually reduced. The tubes in the lower portion are arranged in diagonal rows in order to guide the air entrainment toward the vacuum pump suction.

241. Cooling Water: Surface Condensers. — Since the heat absorbed by the cooling water must equal that given up by the exhaust, neglecting radiation and leakage, the amount of cooling water may be determined as follows:

$$R = \frac{H_m - q_1}{q_2 - q_0}, \tag{205}$$

in which

q_1, q_2, and q_0 = heat of liquid of the condensate, discharge and inlet water, respectively, B.t.u. per lb. above 32 deg. fahr.

Other notations as in equation (203).

Neglecting the heat content of the air entrainment and assuming a constant mean specific heat of unity for water, equation (203) may be written

$$R = \frac{H - t_1 + 32}{t_2 - t_0}, \tag{205a}$$

in which

t_1 = temperature of the condensate, deg. fahr.

Other notations as in equation (204).

In the ordinary low-vacuum surface condenser the depression of the hot-well temperature, t_1, below that corresponding to the total pressure in the condenser may range from 10 to 25 deg. fahr. depending upon the amount of air entrainment and the pressure drop through the condenser. An average figure is 15 deg. fahr. The temperature of the discharge water t_2 may range from 10 to 25 deg. fahr. below that corresponding to the total pressure in the condenser.

The following empirical rule for determining the terminal difference between the temperature of the steam corresponding to the vacuum in the condenser and that of the circulating water discharge gives results agreeing substantially with current surface condenser practice

$$t_d = t - t_0, \tag{206}$$

in which

t_d = terminal difference, deg. fahr.,
t = temperature corresponding to saturated vapor pressure $(p_0 + B)$,
t_0 = initial temperature of the circulating water, deg. fahr.,
p_0 = pressure of saturated vapor corresponding to temperature t_0,
B = coefficient, as follows:

VALUE OF COEFFICIENT *B*.

Vacuum, In.	*B*.	Vacuum, In.	*B*.	Vacuum, In.	*B*.
1.00	0.20	1.75	0.35	3.00	0.50
1.25	0.25	2.00	0.40	3.50	0.60
1.50	0.30	2.50	0.45	4.00	0.70

Thus for $t_0 = 70$ and a 2-inch vacuum: $p = 0.739$, $B = 0.40$ t corresponding to $0.739 + 0.40$ $(= 1.139) = 83.0$ deg. fahr., whence $t_d = 83 - 70 = 13$ deg. fahr.

Example 45. (Low-vacuum condenser.) Required the weight of cooling water necessary to cool and condense one pound of steam under the following conditions: Engine uses 16 lb. steam per brake hp-hr., initial pressure 140 lb. per sq. in. absolute, quality 0.99, initial temperature of the cooling water 70 deg. fahr., vacuum 26 inches referred to a 30-inch barometer.

From the Mollier diagram or by calculation from steam tables H_i = 1185 (approx.), t_s = 126, H_r by assumption = 1 per cent of H_i.

From equation (145)

$$H = 1185 - 0.01 \times 1185 - \frac{2546}{16} = 1014.$$

Assume

$$t_1 = t_s - 15 = 111, \quad t_2 = t_s - 20 = 106 \text{ [see equation (206)].}$$

$$R = \frac{1014 - 111 + 32}{106 - 70} = 25.7 \text{ lb.}$$

With the modern high-vacuum surface condenser in connection with a practically air-tight system the temperature of the condensate will be from 0 to 5 degrees lower than that corresponding to saturated vapor at condenser pressure and the temperature of the discharge water will range from 2 to 10 degrees below that corresponding to the vacuum. The pressure drop through the condenser from exhaust inlet to air pump suction varies with the type and size of condenser and the rate of driving and ranges from 0.02 to 0.2 inch with an average at rated load of approximately 0.1 inch.

Example 46. (High-vacuum surface condenser.) Required the weight of cooling water necessary to cool and condense one pound of steam under the following conditions: Turbine uses 12 lb. steam per kw-hr., initial pressure 200 lb. per sq. in. absolute, superheat 150 deg. fahr., initial temperature of cooling water 70 deg. fahr., vacuum 28.5 inches referred to a 30-inch barometer.

From steam tables, H_i = 1283. Assume H_r = 0.5 per cent of H_i.

$$\text{From equation 145, } H = 1283 - 0.005\ (1283) - \frac{3412}{12} = 993.$$

Assuming a pressure drop of 0.1 inch, the probable absolute pressure in the condenser will be 30 − (28.5 + 0.1) = 1.4 in. The corresponding temperature of vapor at this pressure t_s = 89.5 deg. fahr.

Assume $t_1 = t_s - 4 = 85.5$, $t_2 = t_s - 8 = 81.5$.

81.7 × 12 = 980 ± per l

$$\text{Whence } R = \frac{993 - 85.5 + 32}{81.5 - 70} = 81.7 \text{ lb.}$$

242. Heat Transmission through Condenser Tubes. — Numerous investigations have been conducted on special laboratory apparatus and on condensers in actual service for determining the heat transmission through condenser tubes, but the laws based on these results have been far from harmonious. In steam engine practice where the vacua are comparatively low extreme refinement in design is unnecessary and simple empirical formulas for estimating the extent of cooling surface are sufficiently accurate. In modern high-vacuum practice, however, particularly for large turbo-generators where a fraction of an inch of change in vacuum greatly affects the economy of the prime mover, and where thousands of square feet of cooling surface are involved in a single unit the older empirical rules are apt to lead to serious error. Despite the tremendous advance in condenser design during the past few years the art is still largely a matter of experience and the best rules are subject to arbitrary assumptions.

In any type of surface condenser, neglecting radiation and leakage, the heat absorbed by the cooling water, SUd, must be equal to that given up by the exhaust $w_m (H_m - q_1)$ or,

$$SUd = w_m (H_m - q_1),^* \tag{207}$$

in which

S = extent of cooling surface, sq. ft.,
U = experimentally determined mean coefficient of heat transmission, B.t.u. per hour, per deg. fahr. difference in temperature, d, per sq. ft.,
d = mean temperature difference between that of the steam and of the circulating water, deg. fahr.,
w_m = weight of condensate, lb. per hr. plus the air entrainment,
H_m = heat content of the exhaust steam, moisture and air entrainment, B.t.u. per lb. above 32 deg. fahr.,
q_1 = heat of liquid of the condensate.

From equation (207)

$$S = \frac{w_m (H_m - q_1)}{Ud}. \tag{208}$$

In view of the liberal factor allowed in estimating the value of U and because of the uncertainty of the true value of d, the influence of the heat content of the air entrainment becomes negligible and equation may be written:

$$S = \frac{w (H - t_1 + 32)}{Ud}, \tag{209}$$

in which

w = weight of condensate, lb. per hr.,
H = heat content of the exhaust steam, B.t.u. above 32 deg. fahr.,
t_1 = temperature of the condensate, deg. fahr.

Since the heat absorbed by the cooling water is equal to that given up by the steam, equation (209) may also be stated

$$S = \frac{Q (t_2 - t_0)}{Ud}, \tag{210}$$

* This is on the assumption that the heat transfer is directly proportional to the mean temperature difference. See also equation (223).

in which

Q = total weight of cooling water, lb. per hour,
t_2 = temperature of the discharge water, deg. fahr.,
t_0 = temperature of the intake water, deg. fahr.

Considering first the coefficient of heat transfer, it must be remembered that the coefficient U, as used in equations 207–210, refers to the *mean* or average value for the *entire* surface and not the *actual* value, since the latter varies widely for different parts of the condenser. The actual value varies from more than 1000 for air-free vapor, in the first few rows of tubes (where the steam comes directly into contact with the cooling surface) to less than 50 in the bottom row (where the tubes may be practically blanketed with the condensed steam) and to 3 or less for tubes surrounded only by air. Tests by various investigators show that the actual value of U for a given temperature difference varies with

(*a*) material, thickness, size, shape and cleanliness of the tubes;
(*b*) velocity of water through the tubes;
(*c*) percentage of air on the steam side of the tubes;
(*d*) critical velocity of the water in the tubes;
(*e*) extent of water blanketing on the steam side of the tubes;
(*f*) viscosity of the circulating water.

Taking the material coefficient, m, of plain copper tubes as 1.00, under similar conditions the heat transfer for other materials is approximately

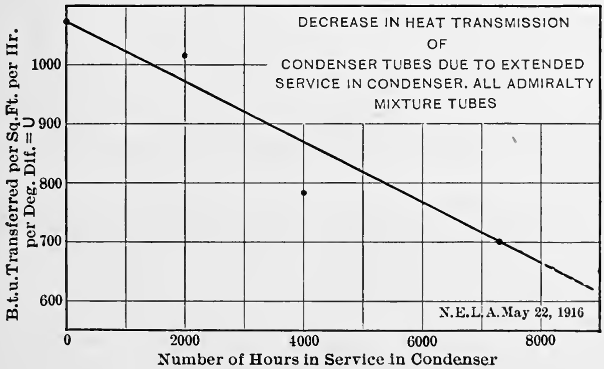


Fig. 310.

as follows: Admiralty brass 0.98, Muntz metal 0.95, tin 0.79, Admiralty lead line 0.79, Monel metal 0.74 and Shelby steel 0.63. No better material than Admiralty brass has been found and it is the standard for modern condenser practice. Corrosion, oxidation and pitting have a marked effect in reducing the heat transference and may lower the conductivity as much as 50 per cent. (See Fig. 310.) The cleanliness coefficient, c, is about 0.9 for such waters as New York or Chicago. The coefficient of heat transfer appears to decrease with the increase in diameter but since the one-inch tube, No. 18 B.W.G. is the most common in use this factor need not be considered for any other size.

The influence of the velocity of the water through the tubes on the coefficient of heat transfer is illustrated in Fig. 311 and Fig. 315. According to Orrok the value of U, other conditions remaining constant, varies approximately as the square root or six-tenths power of the velocity.* For the ordinary low-vacuum condenser the velocity through one-inch standard tubes seldom exceeds 3 ft. per second, whereas velocities as high as 10 ft. per sec. are not uncommon in the high-vacuum type. An average value for the latter is 8 ft. per sec. Except for a very low rate of flow (below that in average condenser practice) critical velocities need not be considered. For example, critical velocities for a one-inch No. 18 B.W.G. condenser tube are approximately as follows.*

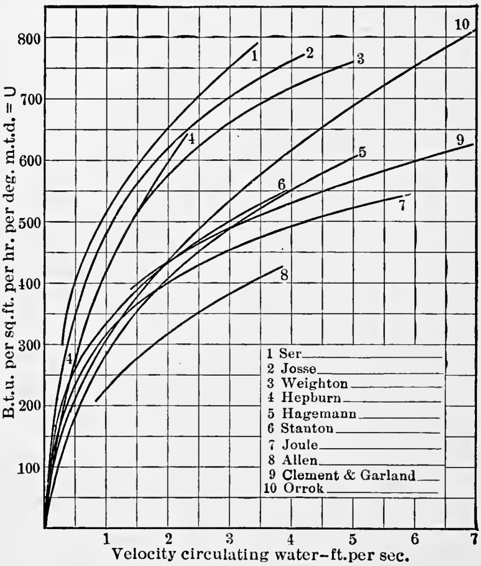


FIG. 311. Variation of Heat Transmission with Water Velocity.

t_m...	40	50	60	70	80	90	100	115	120	150
v_c....	0.50	0.42	0.36	0.32	0.28	0.25	0.22	0.19	0.17	0.14
v_a...	2.84	2.40	2.06	1.81	1.58	1.42	1.27	1.09	0.94	0.80

in which

$t_{m'}$ = mean temperature of the water, deg. fahr.,
v_c = the lower critical velocity, ft. per sec., below which all motion is stream-line unless disturbed artificially,
v_a = the high critical velocity above which all motion is turbulent.

* The proportioning of Surface Condensers, Geo. A. Orrok, Journal of A.S.M.E., Nov., 1916.

The effect of air on the heat transference is very marked as is shown in Fig. 314. The depression of the hot-well temperature below that corresponding to the vacuum may be reduced by good design. Certain designs of dry tube condensers may give hot-well temperatures somewhat higher than the average temperature in the condenser and tests have been reported on several other designs in which the depression was zero. Orrok's investigations show that air entrainment reduces the heat transference approximately according to the law $(p_v \div p_c)^2$, in which p_v = pressure of the vapor and p_c the total pressure in the condenser.

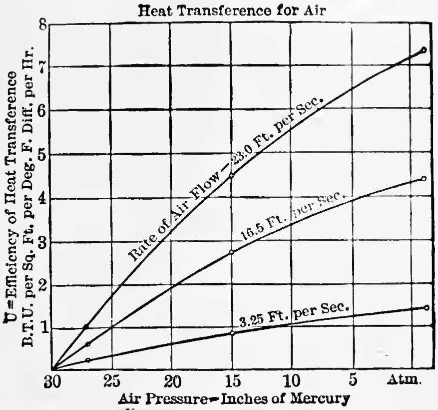


FIG. 312. Heat Transmission Steam to Air.

The value of $(p_v \div p_c)^2$ varies within wide limits, but for tight condensers with efficient air pumps it may be taken as 0.95.

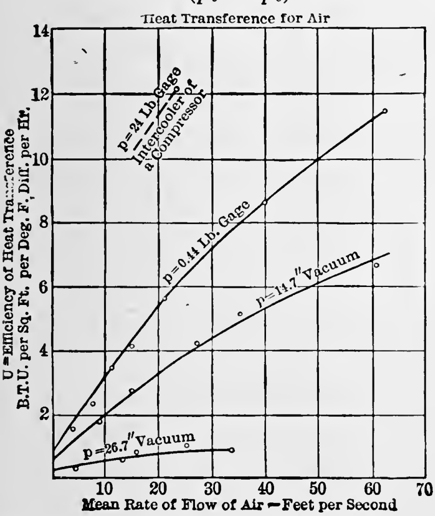


FIG. 313. Heat Transmission Steam to Air.

The reduction in heat transmission due to thickness of water film on both sides of the tubes has been expressed mathematically but it is customary in condenser design to include this factor in the assumed value of U.*

The coefficient of heat transmission increases with the mean temperature of the circulating water, that is, the warmer the water and the lower the vacuum the smaller will be the mean temperature head required to transmit practically constant amount of heat through the surface.

According to Orrok

$$U = kcpm \frac{v^{0.6}}{d^{\frac{1}{8}}}, \tag{211}$$

* Trans. A.S.M.E., Vol. 35, 1915, p. 67.

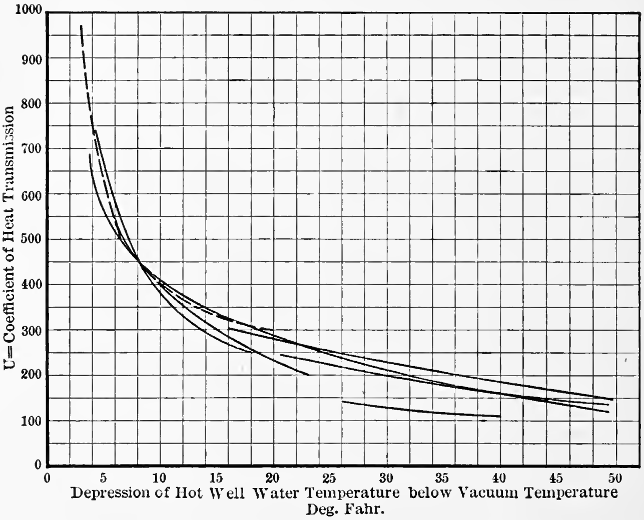


FIG. 314. Relation Between Coefficient of Heat Transfer and Temperature Depression.

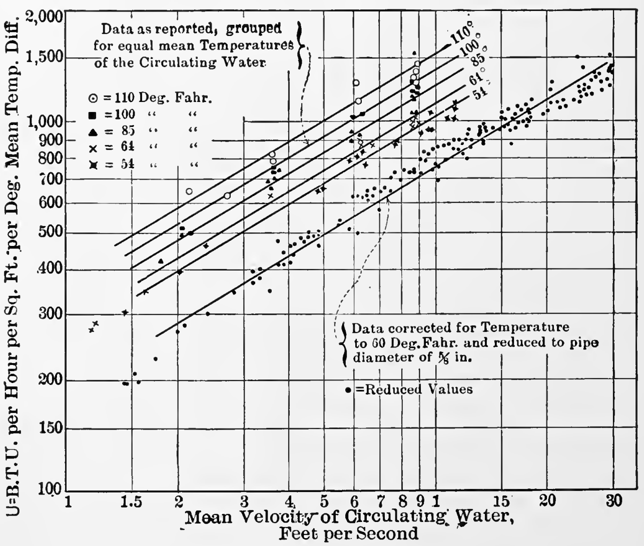


FIG. 315. Rate of Heat Transfer Versus Circulation Water Velocity. Results of Tests by Geo. A. Orrok, Trans. A.S.M.E., 1910.

in which

U = mean coefficient as previously defined and as used in connection with equations (209) and (210),
k = experimentally determined coefficient = 350 for average working conditions,
c = cleanliness coefficient,
p = air richness ratio = $(p_v \div p_c)^2$,
m = material coefficient,
v = velocity through the tube, ft. per sec.,
d = logarithmic mean temperature difference.

The following empirical rule gives values of U which agree substantially with current practice in condenser design

$$U = K\sqrt{v} \tag{212}$$

VALUE OF K FOR VARIOUS INITIAL TEMPERATURES OF CIRCULATING WATER.

Initial Temp., Deg. Fahr.	K.	Initial Temp., Deg. Fahr.	K.	Initial Temp., Deg. Fahr.	K.
40	141	60	192	80	212
45	160	65	198	90	218
50	175	70	203	100	220

Mean Temperature Difference. — It is definitely known that the quantity of heat passing through the cooling surface is proportional to some power of the temperature difference at any instant, but the instantaneous temperature difference is indeterminate, consequently it is necessary to establish an average or mean temperature difference for the whole period of thermal contact of the steam and circulating water.

If t_s = temperature of the steam or hot substance,
t = any momentary temperature of the circulating water,
t_0 = initial temperature of the circulating water,
t_2 = final temperature of the circulating water,
d = mean temperature difference,
Q = weight of circulating water, lb. per hr.,
S = extent of cooling surface, sq. ft.
U_1 = instantaneous value of the coefficient of heat transfer,
U = mean coefficient of heat transfer for the entire period of heat exchange.

All temperatures deg. fahr.

Then the heat transmitted per hour through the elementary surface dS is $U_1 (t_s - t)$.

the circulating water per hour is $Q\,dt$ (theoretically this should be $cQ\,dt$ in which c is the mean specific heat of the water, but for all practical purposes the value of c may be taken as unity).

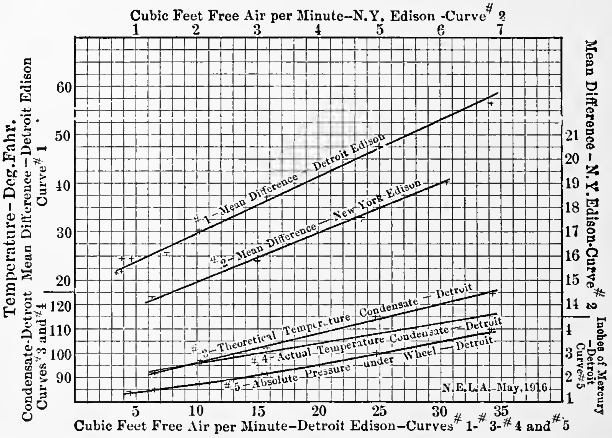


FIG. 316. Curves Showing Effect of Air Leakage on Condenser Efficiency.

These two quantities must be equal, or

$$U_1\,(t_s - t)\,dS = Q\,dt. \tag{213}$$

from which

$$dS = \frac{Q}{U_1}\,\frac{dt}{t_s - t}. \tag{214}$$

If the temperature of the steam is assumed to be constant t_s is independent of t, and if the heat transmitted per hour is assumed to be directly proportional to temperature difference, U is likewise independent of t and $U_1 = U$, therefore the relation between rise in temperature of the circulating water and the surface traversed becomes

$$S = \frac{Q}{U}\int_{t_0}^{t_2} \frac{dt}{t_s - t} \tag{215}$$

$$= \frac{Q}{U}\log_e \frac{t_s - t_0}{t_s - t_2}. \tag{216}$$

For the whole period of transfer,

$$SUd = Q\,(t_2 - t_0), \tag{217}$$

$$d = \frac{Q\,(t_2 - t_0)}{US} \tag{218}$$

Combining equations (216) and (218) and reducing,

$$d = \frac{t_2 - t_0}{\log_e \dfrac{t_s - t_0}{t_s - t_2}}. \tag{219}$$

This is known as the *logarithmic mean* temperature difference and is the one most commonly used in condenser design. The relation between temperature of the steam and that of the circulating water for this condition is shown graphically in Fig. 317.

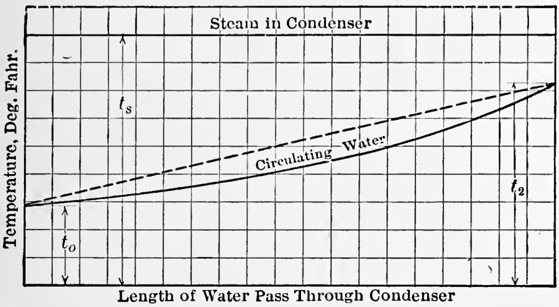


FIG. 317. Rise of Circulating Water Temperature in Condenser Tubes.

If the rise in temperature of the circulating water follows the dotted line *cd* the mean temperature difference may be expressed

$$d = t_s - \frac{t_2 + t_0}{2}. \qquad (220)$$

This is known as the *arithmetic mean* temperature difference and is used only for rough calculation or where other influencing factors can only be approximated.

If the quantity of heat transmitted per hour is proportional to the nth power of the instantaneous temperature difference, as appears to be the case in actual practice, and U is assumed to be constant

$$Q\,dt = U\,(t_s - t)^n\,dS \qquad (221)$$

Integrating and reducing,

$$S = \frac{Q}{U}\left[(t_s - t_0)^{1-n} - (t_s - t_2)^{1-n}\right]\frac{1}{1-n}. \qquad (222)$$

By assumption,

$$SUd^n = Q\,(t_2 - t_0). \qquad (223)$$

Therefore $$d = \left[\frac{(1-n)\,(t_2 - t_0)}{(t_s - t_0)^{1-n} - (t_s - t_2)^{1-n}}\right]^{\frac{1}{n}} \qquad (224)$$

This is known as the *exponential mean* temperature difference. Orrok's * experiments lead to a value of $n = 0.875$. Loeb † assigns a value of $n = 0.9$. Because of the uncertainty of the value of U it is sufficiently accurate for most purposes to take $n =$ unity, which results in the logarithmic mean temperature difference.

* Jour. A.S.M.E., Nov., 1916. † Jour. Am. Soc. Naval Engrs., Vol. 27, May, 1915.

Orrok * gives a general rule for high-vacuum surface condensers operating under favorable conditions which may be reduced to the form

$$S = \frac{Q}{40.3\ v^{0.6}}\left[(t_s - t_0)^{\frac{1}{8}} - (t_s - t_2)^{\frac{1}{8}}\right]. \tag{225}$$

Let d = outside diameter of the tube, in.,
n = number of tubes in each pass of the condenser.
l = length of water travel, or total tube length, ft.

Then, $$S = \frac{\pi d}{12}\, nl, \quad \text{whence} \quad l = 3.83\, S \div dn. \tag{226}$$

By simple arithmetical calculation it may be shown that

$$n = \frac{Q}{1233\, v\, (d - 2\, t)^2}, \tag{227}$$

in which t = thickness of the tube, inches.

Example 47. (Low-vacuum condenser.) Approximate the amount of cooling surface for a 1000-hp. compound engine operating under the following conditions: Water rate 16 lb. per hp-hr., initial steam pressure 140 lb. absolute, initial quality 0.99, inlet temperature of circulating water 70 deg. fahr., vacuum 26 in. referred to a 30-inch barometer.

In view of the absence of data, in this particular problem, for estimating the value of U with any degree of accuracy it is sufficiently accurate to assume the temperature of the steam in the condenser to be that of saturated vapor corresponding to the vacuum, and for the same reason the heat of the exhaust may be assumed to be that of saturated steam corresponding to the absolute pressure in the condenser.

The temperature of vapor t_s corresponding to an absolute pressure of 4 inches ($p_s = 30 - 26$) is 126 deg. fahr. and $H = 1114$ (approx.).

In the ordinary engine condenser considerable air will be carried with the steam into the condenser and the hot-well depression may range from 5 to 20 degrees; assume the depression to be 10 degrees, then $t_1 = t_s - 10 = 126 - 10 = 116$ deg. fahr.

Any value may be assumed for t_2 greater than t_0 and less than t_1. The nearer t_2 is to t_0 the greater must be the quantity of circulating water per lb. of condensate. On the other hand, the nearer t_2 is to t_0 the less is the mean temperature difference d, and hence the greater must be the cooling surface for a given weight of condensate. When water is cheap and the head pumped against is small t_2 may be given a lower value than when water is costly and the discharge head is large. In average engine condenser practice t_2 may range from 5 to 20 degrees below t_1; assume it to be 10 degrees, then $t_2 = t_1 - 10 = 116 - 10 = 106$ deg. fahr. Equation (206) gives $t_2 = 105.5$.

Because of the great latitude in assuming values of t_1 and t_2 it is sufficiently accurate to use the arithmetical mean, or

$$d = 126 - \frac{70 + 106}{2} = 38.0.$$

* Jour. A.S.M.E., Nov., 1916.

In engine practice a very liberal factor is allowed in assuming a value for U because of the possible reduction in heat transmission caused by the deposit of cylinder oil on the tubes and because of the air entrainment. For the usual engine type of condenser a safe value is $U = 300$. According to equation (212), $U = 300$ for $v = 2.25$ ft. per sec.

Substituting these values in equation (209) and reducing

$$S = \frac{16{,}000\,(1114 - 116 + 32)}{300 \times 38} = 1446 \text{ sq. ft.}$$

This corresponds to approximately 11.0 lb. of condensate per hr. per sq. ft. of tube surface. An average figure commonly quoted for engine condensers is 10 lb. of steam per hr. per sq. ft. of tube surface for 24–26 in. vacuum with 70-degree cooling water. A rough rule is to allow 2 sq. ft. of cooling surface per i.hp.

Example 48. (High-vacuum condenser.) Calculate the amount of tube surface required for a 10,000-kw. turbine operating under the following conditions: Water rate 12.0 lb. per kw-hr., initial absolute pressure 200 lb. per sq. in., superheat 150 deg. fahr., temperature of circulating water 70 deg. fahr., vacuum 28.5 inches referred to a 30-inch barometer, water velocity through tubes 8 ft. per sec. Cooling surface to consist of one-inch (18 B.W.G.) Admiralty tubes.

For maximum theoretical efficiency $t_2 = t_1 = t_s$. This condition is possible only for air-free vapor, perfect heat transmission, and no pressure drop between turbine nozzle and air pump suction. In the very latest designs the pressure drop between turbine nozzle and air pump suction seldom exceeds 0.2 in. The temperature of the condensate varies from $t_1 = t_s - 0$ to $t_1 = t_s - 4$ deg. fahr., and t_2 varies from $t_2 = t_s - 4$ to $t_s - 10$ deg. fahr. Assume a pressure drop of 0.2 in., $t_1 = t_s$ and $t_2 = t_s - 8$, then

$p_s = 30.0 - (28.5 + 0.2) = 1.3$ in. and the corresponding $t_s = 87.1$ deg. fahr.

For the given conditions $H = 993$ (see example 46).

$$\text{Then} \quad Q = \frac{12 \times 10{,}000\,(993 - 87.1 + 32)}{79.1 - 70.0} = 12{,}368{,}000 \text{ lb.}$$

$$d = \frac{79.1 - 70}{\log_e \dfrac{87.1 - 70}{87.1 - 79.1}} = 11.98 \text{ deg.}$$

Exponential mean gives $d = 11.97$ deg.

The condenser must be designed for the maximum load when the circulating water is at its highest temperature and a suitable factor should be allowed for dirty, oxidized tubes and the presence of undue amounts of air. For this reason a much lower value of U is assumed than is possible with everything in first-class shape. An average value for a velocity of 8 ft. per sec. is $U = 600$. According to equation (212) $U = 575$.

Substituting these values in equation (210),

$$S = \frac{12{,}368{,}000\,(79.1 - 70)}{600 \times 11.98} = 15{,}630 \text{ sq. ft.}$$

Corresponding to 1.56 sq. ft. per kw. of turbine rating. Surface condensers for large turbines have generally from 1.6 to 1.7 sq. ft. of condensing surface per kw. See Table 93.

Orrok's rule gives for this example

$$S = \frac{12,368,000}{40.3 \times 80.6}\left[(87.1 - 70)^{\frac{1}{8}} = (87.1 - 79.1)^{\frac{1}{8}}\right]$$
$$= 11,500 \text{ sq. ft.}$$

Taking the heat content of the steam as that of saturated steam at condenser pressure Orrok's rule gives S = 12,650 sq. ft. In fact, Orrok's rule is based on the assumption that the steam entering the condenser is saturated, an assumption which simplifies calculation and which is justifiable in view of the uncertainty of the true values of many of the factors entering into the problem.

TABLE 92.

TEST OF 50,000 SQ. FT. SURFACE CONDENSER, 74TH ST. STATION INTERBOROUGH RAPID TRANSIT CO.

(H. G. Stott and W. S. Finlay, Jr.)

Pressure at throttle, lb. abs	220
Temperature at throttle, deg. fahr	487
Superheat	97
Load, average kw	31,233
Exhaust vacuum, in	28.61
Exhaust pressure, in. abs	1.39
Corresponding temp. deg. fahr	89.4
Mean temp. diff. (log.)	12.9
Condensate, lb. per hr	357,060
B.t.u. per sq. ft. per hr. per deg. mean temperature difference	490
Air leakage, cu. ft. per min	16.88
Temp. of hot well, deg. fahr	86
Temp. intake water, deg. fahr	70.8
Temp. discharge water, deg. fahr	80.9
Temp. rise, deg. fahr	10.1
Circulating water, gal. per min	64,700
Ratio circulating water to condensate	91

TABLE 93.

MODERN SURFACE CONDENSER PROPORTIONS.

Size of Turbo-generator.	Tube Surface, Sq. Ft.	Sq. Ft. Tube Surface Per Kw.	Size of Turbo-generator.	Tube Surface, Sq. Ft.	Sq. Ft. Tube Surface Per Kw.
500	1,500	3.00–3.50	10,000	17,500	1.75–2.25
1000	2,750	2.75–3.25	15,000	25,000	1.67–2.00
2000	5,000	2.50–3.00	20,000	32,000	1.60
5000	10,000	2.00–2.50	35,000	56,000	1.60

The curves in Fig. 318 are based upon equation (209) with U = 300 and afford a simple means for determining the extent of cooling surface for different conditions of operation. For any other value of U multiply by 300 and divide by the new value of U.

Design and Proformance of Surface Condensers: Jour. A.S.M.E., Nov., 1916, p. 864; Power, Aug. 29, 1916, p. 300; Jour. A.S.M.E., Jan., 1916, p. 23; Jan., 1915, p. 546; Aug., 1915, p. 459.

SURFACE CONDENSER AIR PUMFS. — See paragraphs 308 to 310.

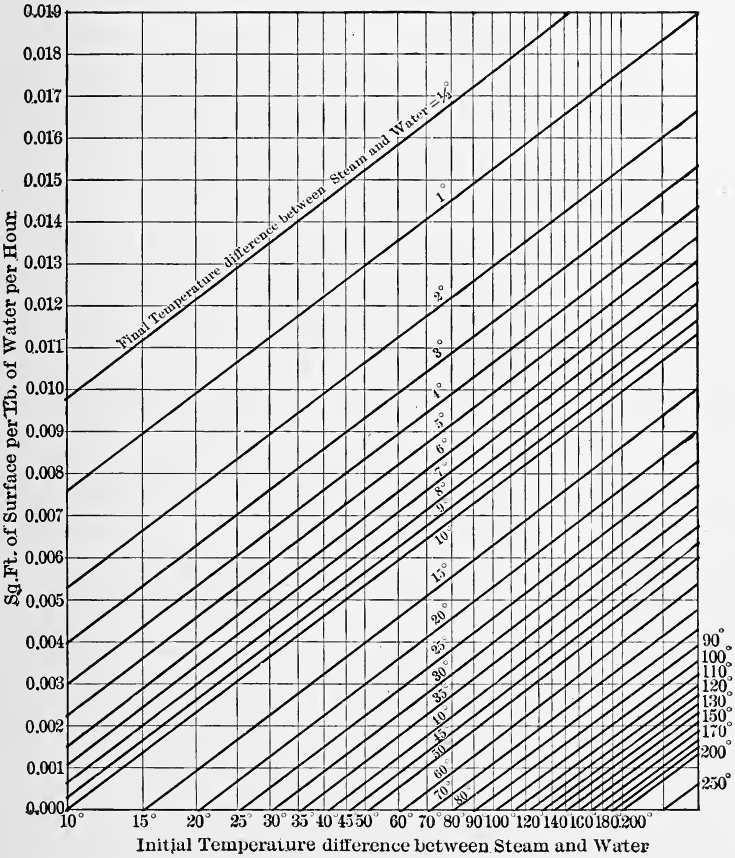


FIG. 318. Curves for Determining the Amount of Cooling Surface.

243. Dry-air Surface Condensers (Forced Circulation). — Where water is very scarce and the feed supply is reclaimed by condensing the exhaust steam, water-cooled condensers may be prohibitive in cost of operation, even when combined with cooling tower or other water-cooling device, since the latter involves a loss of water approximately equivalent to the amount of steam condensed, due to evaporation.

A notable installation of air-cooled surface condenser is that in an electric station of 2000-horsepower capacity in the city of Kalgoorlie, West Australia.* The condenser consists of a large number of narrow chambers constructed of thin corrugated sheet-steel plates spaced $\frac{1}{4}$ inch between centers. Each chamber has 1345 square inches of cooling surface. Fifty-one of these chambers are grouped into a compartment and 15 compartments constitute a section. Each section is equipped with three motor-driven fans 7 feet in diameter and running normally at 320 r.p.m. In all there are six sections, giving a total cooling surface of 45,000 square feet. The steam consumption of the main engines is 16 to 16.5 pounds per i.hp-hour at rated load. At full load the fans require 130 kilowatts, or approximately 10 per cent of the station output. The average vacuum obtained is about 18 inches throughout the year and ranges from 0 inches on very hot days to 22 inches in cooler weather. The following figures, based on actual observation, show the effect of temperature of the external air on the vacuum when condensing 32,000 pounds of steam per hour (the rated capacity of the condenser).

Temperature External Air, Degrees F.	Vacuum, Inches (referred to 30-Inch Barometer).	Temperature External Air, Degrees F.	Vacuum, Inches (referred to 30-Inch Barometer).
42.8	22	96.8	9.6
50	21.2	100.4	7.6
60.8	20	107.6	3.6
68	18.4	113	0
78.8	16		

Air-Cooled Surface Condensers: Engineering News, Oct., 1902, p. 271; ibid., Vol. 49, p. 203.

244. Quantity of Air for Cooling (Dry-air Condenser). — The volume of air, under atmospheric conditions, necessary to condense steam to any given temperature may be determined as follows:

Let H = heat content of the steam at condenser pressure,
t_s = temperature of the vapor in the condenser,
t_1 = temperature of the condensed steam,
t = temperature of the air entering condenser,
t_0 = temperature of the air leaving condenser,
V = volume of air in cubic feet necessary to condense and cool one pound of steam,
B = specific weight of air under atmospheric conditions,
C = mean specific heat of air under atmospheric conditions,
d = mean temperature difference between the air and steam,
S = cooling surface in square feet,
U = coefficient of heat transmission, B.t.u. per square foot per degree difference in temperature per hour.

* This condenser has been recently discarded since the cost of water has been greatly reduced.

Since the heat absorbed by the air must be equal to the heat given up by the steam, neglecting radiation, we have

$$VBC\,(t_0 - t) = H - t_1 + 32, \tag{228}$$

from which

$$V = \frac{H - t_1 + 32}{BC\,(t_0 - t)}. \tag{229}$$

For practical purposes C may be taken as the specific heat of dry air, the error due to this assumption being negligible even if the air is saturated with moisture.

Example 49. How many cubic feet of air are necessary to condense and cool one pound of steam under the following conditions: Vacuum 20 inches; temperature of entering air, leaving air, and condensed steam, 60, 110, and 140 deg. fahr., respectively?

Here $H = 1130$ (from steam tables),
$t_0 = 110,\ t_1 = 140,\ t = 60,\ C = 0.24,\ B = 0.075.$

Substituting these values in equation (229),

$V = \dfrac{1130 - 140 + 32}{0.075 \times 0.24\,(110 - 60)} = 1135$ cubic feet of air necessary to condense one pound of steam under the given conditions.

The proper area of cooling surface depends upon the value of the coefficient of heat transmission, which varies with the velocity and humidity of the air and character of the cooling surface. Accurate data are not available on this point.

A few experiments made at the Armour Institute of Technology gave values of $U = 10$ to 25 B.t.u. per hour, per square foot, per degree difference in temperature for air velocities of 500 to 4000 feet per minute. for corrugated-steel sheeting $\frac{1}{8}$ inch thick. Assuming these values of U for the above example, $S = 1.5$ square feet of cooling surface per pound of steam condensed per hour for air velocity of 4000 feet per minute, and $S = 3.7$ square feet for a velocity of 500 feet per minute.

245. Saturated-air Surface Condensers (Natural Draft).— Fig. 319 shows vertical and horizontal sections of a Pennel saturated-air surface condenser. The apparatus consists of an upright cylindrical shell containing a number of vertical 4-inch steel tubes through which air is drawn by natural draft. A centrifugal pump circulates about one half gallon of water per horsepower per minute from a cistern below the condenser. The water flowing over the upper tube sheet and then descending the tubes by gravity forms a film over their entire interior surface.

The condensing action is as follows: The current of exhaust steam entering the side of the shell at A is caused by suitable baffle plates to circulate among the tubes, and in condensing gives up its latent heat to the water film, which wholly or partially evaporates, saturating the ascending current of air at its own temperature. The upward current of hot vapor-laden air carries off the heat into the atmosphere. The cooling water which is not evaporated and lost to the atmosphere falls into the cistern below to be again taken up by the circulating pump, the water level in the cistern being kept constant by a float governing

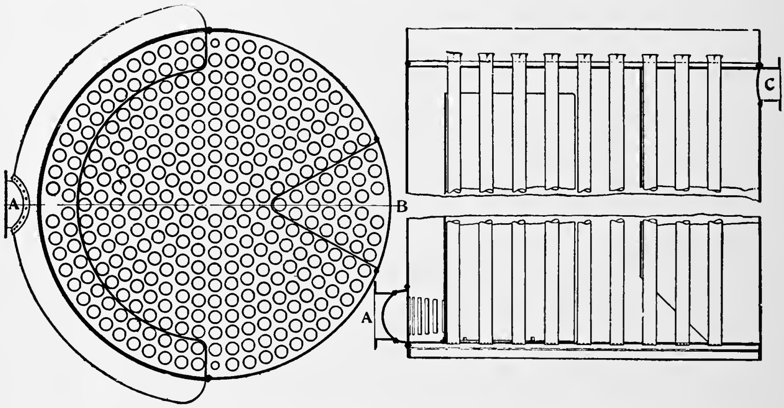


FIG. 319. Pennel Saturated-air Surface Condenser.

a valve on the supply pipe. The non-condensable gases collect at C, where they are removed by the dry-air pump, while the condensed steam is drawn off from the bottom tube sheet by the vacuum pump and discharged into the hot well. An excellent feature of this device is that the film of water on the cooling surface is secured without interference with the ascending air currents and also without the use of sprays through small orfices likely to become clogged with rust or sediment. Where the recovery of the condensed steam is essential and a high vacuum of secondary importance, condensers of this type have proved to be good investments on account of the low first cost.

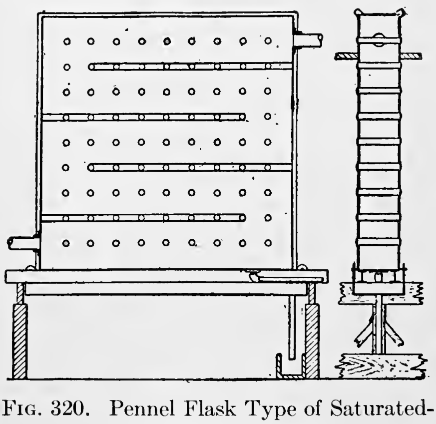


FIG. 320. Pennel Flask Type of Saturated-air Surface Condenser.

Table 94 gives the results of a test of a condenser of this type, taking steam from a 30-in. by 58-in. by 48-in. engine running at 45 r.p.m. (Power, December, 1903, p. 672; West. Elect., May 19, 1900, p. 323.)

Fig. 320 illustrates the Pennel "flask" type of atmospheric condenser. The exhaust steam enters below and follows the zigzag course bounded by the internal stay channels, condensing as it goes and driving before it the non-condensable gases to the outlet at the top. The condensed steam gravitates to the bottom and thence to the hot well. The top of the flask is trough shaped and causes the cooling water to flow down the sides of the flask in a thin stream. The portion of the cooling water not evaporated collects at the bottom of the flask and flows to the cooling-water reservoir.

TABLE 94.

TEST OF PENNEL SATURATED-AIR SURFACE CONDENSER.

Duration of trial	9 hours
Average steam pressure at engine by gauge	139.8 pounds
Average vacuum, mercury column	17.5 inches
Average temperature in condenser	123.7 deg. fahr.
Average temperature of circulating water	116.4 deg. fahr.
Average temperature of city water	52 deg. fahr.
Average temperature of outside air	62 deg. fahr.
Average temperature of saturated air	106 deg. fahr.
Average draft in stack of condenser	1.1 inches
Average humidity of outside air	67 per cent
Average amount of steam condensed per hour	7950 pounds
Average amount of circulating water used per hour	114,660 pounds
Average amount of city water used per hour	3462 pounds
Pounds of steam per pound of city water	2.3
Pounds of circulating water per pound of steam	14.4
Average horsepower of engine	569.7
Steam, pounds per i.hp-hr	13.95
Horsepower required to run air pumps	10.5
Horsepower required to run circulating pumps	3.0
Condensing surface, square feet	3900
Pounds of steam condensed per square foot surface per hour	2038
Barometer	28.58 inches
Vapor tension corresponding to 123.7 degrees	3.82 inches
Per cent of main engine steam used by auxiliaries	2.38

246. Evaporative Surface Condensers. — An evaporative surface condenser consists of a number of copper, brass, wrought- or cast-iron tubes arranged horizontally or vertically and connected to manifolds or chambers at each end. The exhaust steam passes through the tubes and a thin film of water is allowed to flow over the external surfaces. The cooling effect is brought about by the evaporation of part of the circulating water, and the general principle of operation is the same as that of the saturated-air condenser described above. Evaporation is sometimes hastened by constructing a flue over the tubes, thereby creating a natural draft, or by means of fans. With horizontal cast-iron tubes and natural draft, vacua from 23 to 27 inches are readily maintained with a cooling surface of approximately eight tenths square foot per pound of steam condensed per hour. With vertical brass tubes and fan draft 8 pounds of steam per hour per square foot of cooling surface is not an unusual figure. The amount of cooling water evaporated per pound of steam varies from eight tenths to one pound, depending upon the draft. The power necessary to operate the pumps and fans varies from 1 to 10 per cent of the total output of the plant. For an interesting discussion of evaporative condensers the reader is referred to the admirable article by Oldham in the Proceedings of the Institute of Mechanical Engineers, 1899, and reproduced as a serial in Engineering (London), April 28 to June 30, 1899. The following test of a vertical cast-iron tube evaporative surface condenser (Table 95) will give some idea of the performance of this type of condenser. This condenser consisted of two rows of 4-inch vertical cast-iron pipes connected at the top by *U* bends and at the bottom by cast-iron manifolds. A perforated iron trough distributes the water over the center of the bend and causes it to flow in a thin stream over the surface of the tubes. A wet-air pump is used for withdrawing the condensed steam and air. No fan is used for hastening evaporation.

See Chapter XXV, for evaporative surface condenser calculations.

Evaporative Condensers: Engr., Lond., May 5, 1889, pp. 432, 442, 447; Engineering, May 19, 1899, p. 661, June 2, 1899, p. 721, June 30, 1899, p. 861; Trans. A.S.M.E., 14–696; Power, Nov. 16, 1909; Prac. Engr. U. S., June, 1910, p. 346.

TABLE 95.

TEST OF A CAST-IRON, VERTICAL-TUBE, EVAPORATIVE SURFACE CONDENSER NATURAL DRAFT.

Date	Sept. 12	Sept. 13
Weather	Wet	Fine
Barometer	29.8	29.5
Temperature of air	?	60
Cooling surface, external	272	272
Duration of trial, minutes	99	115
Weight of steam condensed, pounds	800	800
Boiler pressure	60	60
Weight of water in circulation	1830	1830
Weight of fresh water added	600	640
Vacuum in condenser	23.36	24.1
Initial temperature of circulating water	117.5	113.9
Final temperature of circulating water	128.4	125
Temperature of "make up" water	58	58
Temperature of water in hot well	136.5	131.8
Weight of steam condensed per hour, pounds	485	427
Weight of water circulated per hour, pounds	6786	?
Weight of "make-up" water added per hour	364	334
Weight of steam condensed per square foot of cooling surface per hour	1.8	1.54
Weight of "make-up" water per pound of steam condensed, pounds	0.75	0.80

247. Location and Arrangement of Condensers. — In the modern power house one sees two general arrangements of condensers and auxiliaries:

1. The independent or subdivided system, in which each engine or turbine is provided with its own condenser, air and circulating pumps.

2. The central system, in which the condensers and auxiliaries are grouped together. Ordinarily one condenser suffices for all engines.

THE INDEPENDENT SYSTEM. — The condenser is usually placed close to and below the engine so that all condensation may gravitate

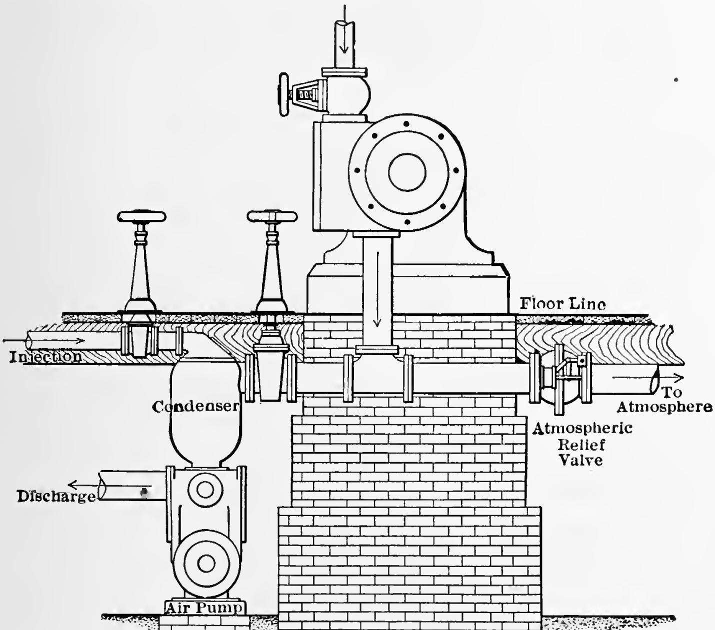


FIG. 321. Jet Condenser Located below Engine-room Floor.

into it. Figs. 321 and 327 show an application of this system with jet condensers. Here each condenser receives its supply of cooling water from a main injection pipe and discharges into a main overflow pipe. The exhaust pipe leading to the condenser is by-passed through a suitable atmospheric relief valve to a main free exhaust header so that the engine may operate non-condensing in case the vacuum breaks or the condenser is cut out. The chief feature of this arrangement is its flexibility, as each unit is complete in itself and independent of the others. By far the greater number of central stations are equipped with independent condensers.

Occasionally a jet condenser is located on the same level with the

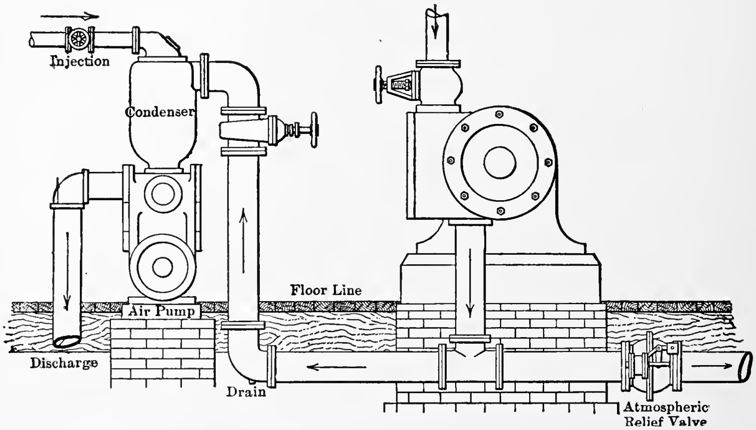


FIG. 322. Jet Condenser Located above Engine-room Floor.

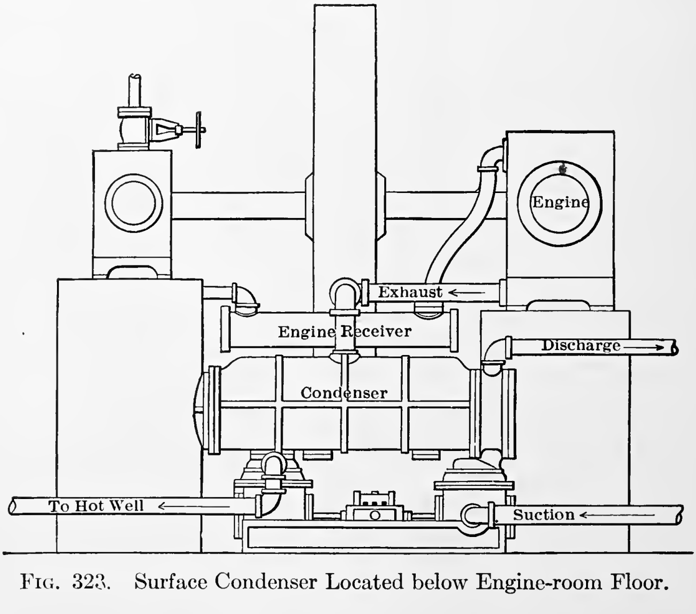


FIG. 323. Surface Condenser Located below Engine-room Floor.

engine or even above it, Fig. 322, but such a location should be avoided if possible, as it usually necessitates a larger number of bends and joints in the exhaust pipes than the basement arrangement, and increases the possibility of air leakage. If the exhaust pipe does not drain directly into the condenser, the lowest point in the piping should always be provided with a drip which should be opened when the engine is shut down, as condensation and leakage are apt to fill the pipe with water if the engine stands for any length of time. The end of the drip should be connected so that water cannot be drawn back through the drip pipe and into the engine cylinder. The length of exhaust pipe and particularly the number of bends between engine and condenser should be kept as small as possible, otherwise the engine may not derive the full benefit of the vacuum in the condenser. A case is recorded where the exhaust piping and appurtenances in connection with a 5000-horsepower engine caused a drop of several inches in vacuum between condenser and exhaust opening of the low-pressure cylinder. (National Engineer, December, 1906, p. 10.) The wet-air pump must always be located below the condenser chamber so that the condensation may gravitate to it.

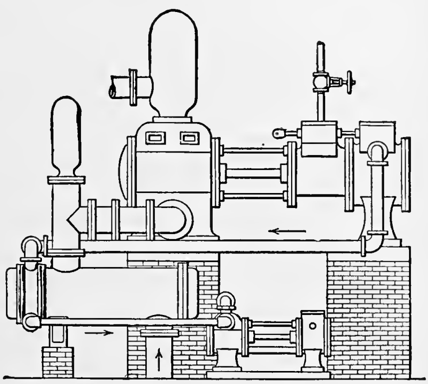


Fig. 324. Surface Condenser Installed in Connection with Pumping.

Fig. 323 shows the arrangement of a surface condenser with combined air and circulating pump in connection with a horizontal cross compound engine. The condenser and appurtenances are placed below the engine, thereby permitting the condenser to be closely connected to the engine.

Fig. 324 shows the arrangement of a surface condenser in connection with a pumping engine. The condenser is placed in series with the pump suction.

Several typical installations of surface condensers in connection with various forms of condenser auxiliaries are shown in Figs. 325 to 328.

Central Systems. — In the central condensing systems the condenser is located at any convenient point and the exhaust from all the engines piped to it. Any arrangement of condenser and auxiliary machinery may be adopted which will favor the lowest cost of installation and expense of operation. Except where continuity of operation

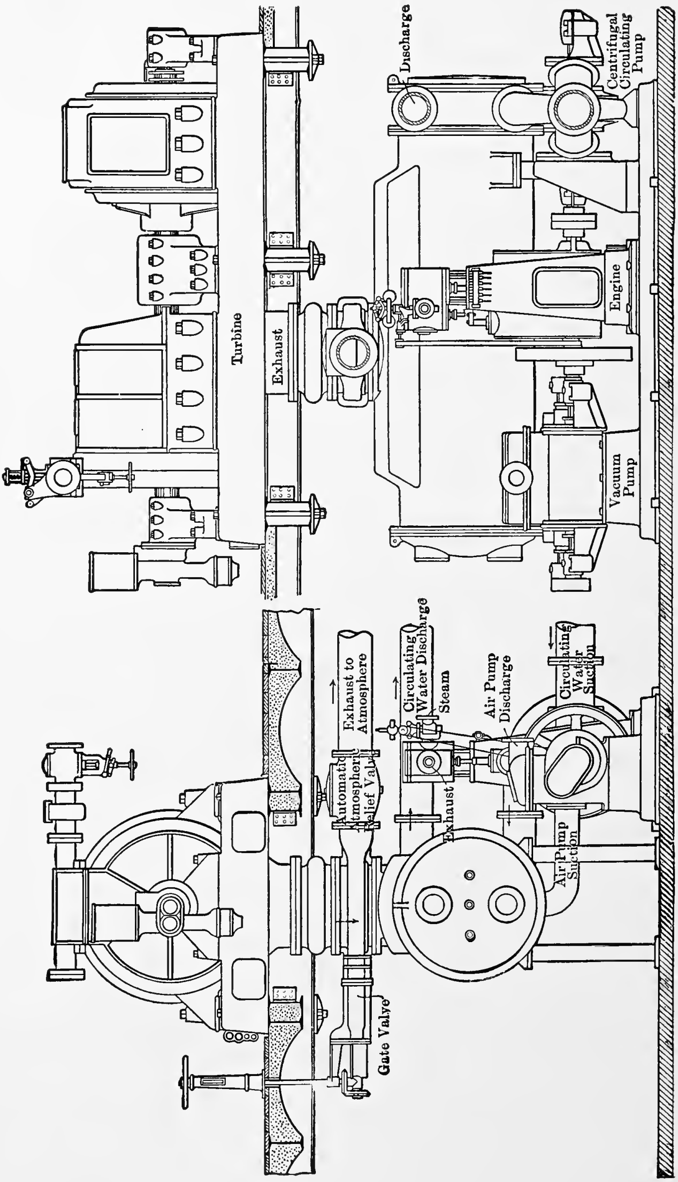


FIG. 325. C. H. Wheeler Company's High-vacuum System.

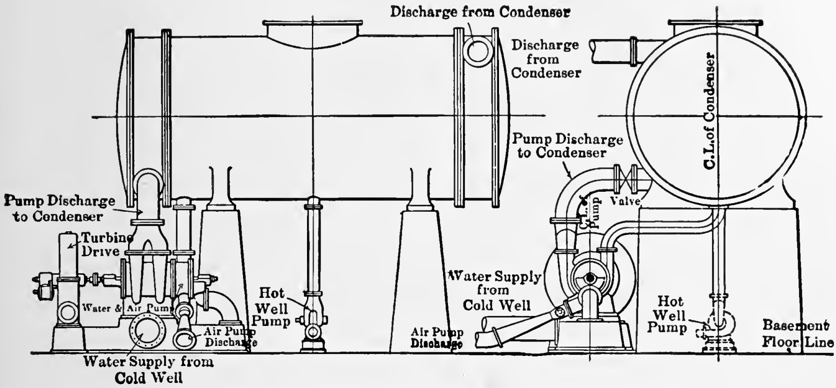


FIG. 326. Surface Condenser with Leblanc Pumps.

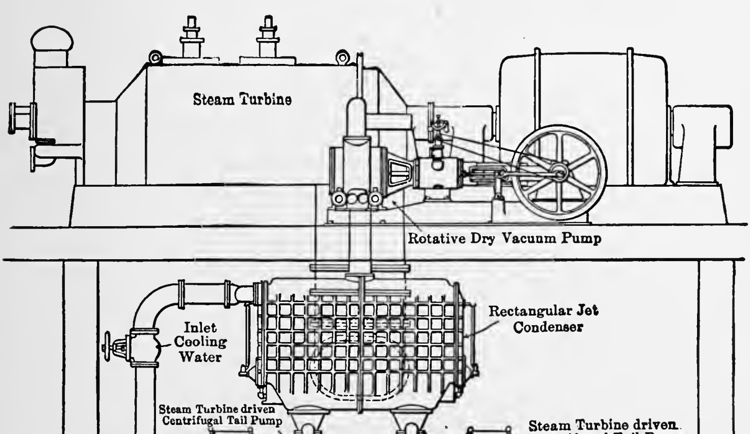


FIG. 327. Wheeler Rectangular Jet Condenser with Centrifugal Tail Pump and Rotative Dry Vacuum Pump in Connection with a 10,000-kilowatt Steam Turbine.

is absolutely essential, only one circulating pump and one air pump are installed. This reduces the number of auxiliary pumps and appliances to a minimum, with a consequent decrease in first cost and maintenance. With properly designed exhaust piping the condenser may be located at a considerable distance from the engine without undue loss of vacuum.

Central condensers have found great favor in power plants in which the individual units are subjected to extreme variations in load, as in

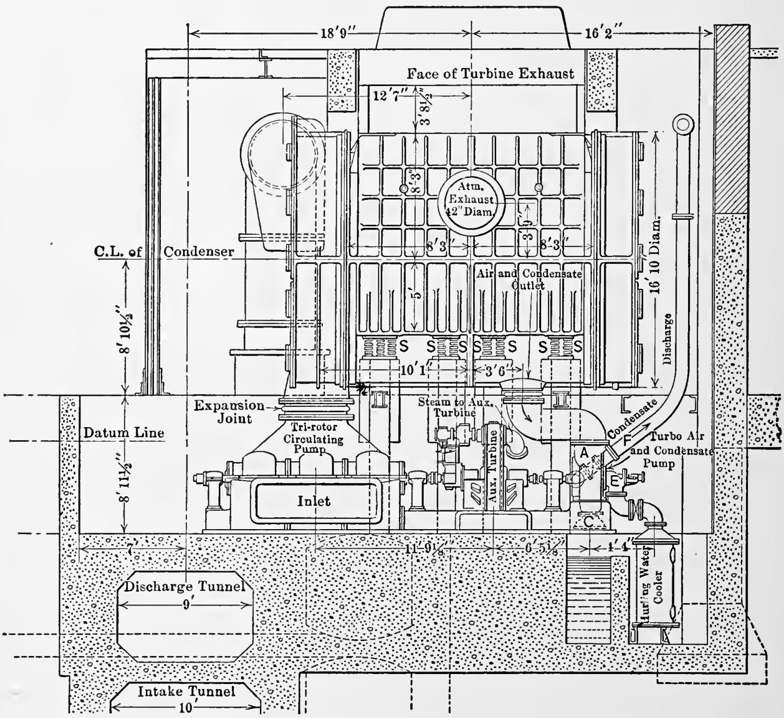


Fig. 328. Longitudinal Elevation of the 50,000 sq. ft. Condenser for the Commonwealth Edison Co.

rolling mills. At the works of the Illinois Steel Company, South Chicago, Ill., one condenser takes care of the steam from 15,000 horsepower of engines in the rail mill, and another condenses the steam from the 15,000 horsepower of engines in the Bessemer steel mill. A notable installation of this system in connection with street-railway work is in the power house of the Northwestern Elevated Company, Chicago, where a single condenser takes care of the exhaust steam of five engines, 11,000 horsepower in all. Fig. 330 shows the general arrangement of this installation.

For a comparison of the advantages and disadvantages of the independent and central systems see Engineering Magazine, October, 1900, p. 56; Engineering, London, June 23, 1899, p. 615; and Engineering, July 17, 1903.

Condenser Auxiliaries. — The various types of condenser auxiliaries and their power requirements are treated at length in paragraphs 307 to 314.

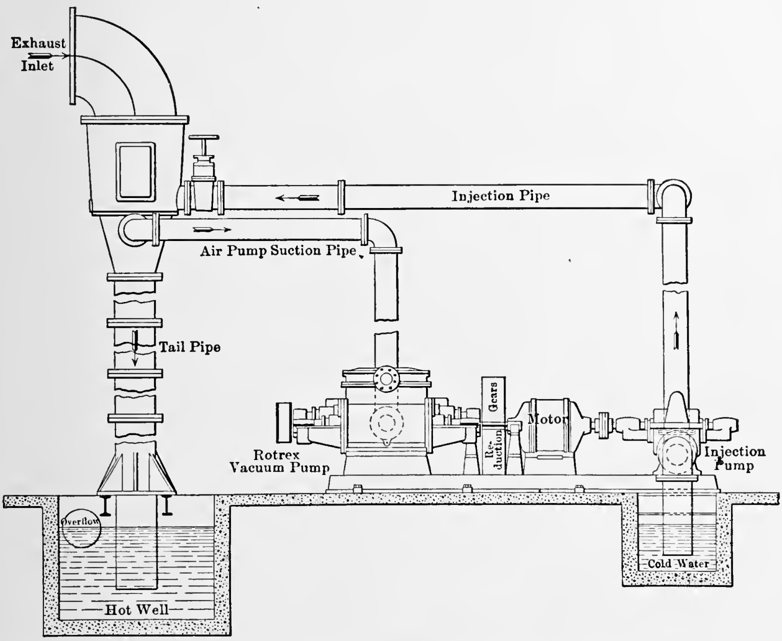


Fig. 329. Barometric Condenser with Centrifugal Water Injection and "Rotrex" Air Pump.

248. Cost of Condensers. — The curves in Fig. 331 compiled by A. R. Smith of the Construction Engineering Department, General Electric Company, show the approximate costs of condensers including their auxiliaries, f.o.b. factory. The average for each capacity of turbine was compiled from costs without regard to surface, quality of water, vacuum maintained and steam or electric drive. Actual cost may vary considerably from those shown on the curves, depending on local conditions and other special considerations.

The following figures give an idea of the relative costs of the different types of condensers and auxiliaries for a 1000-i.hp. plant using 20 pounds of steam per i.hp-hour at rated load, or a total of 20,000 pounds per hour. Vacuum to be maintained, 26 inches unless otherwise stated;

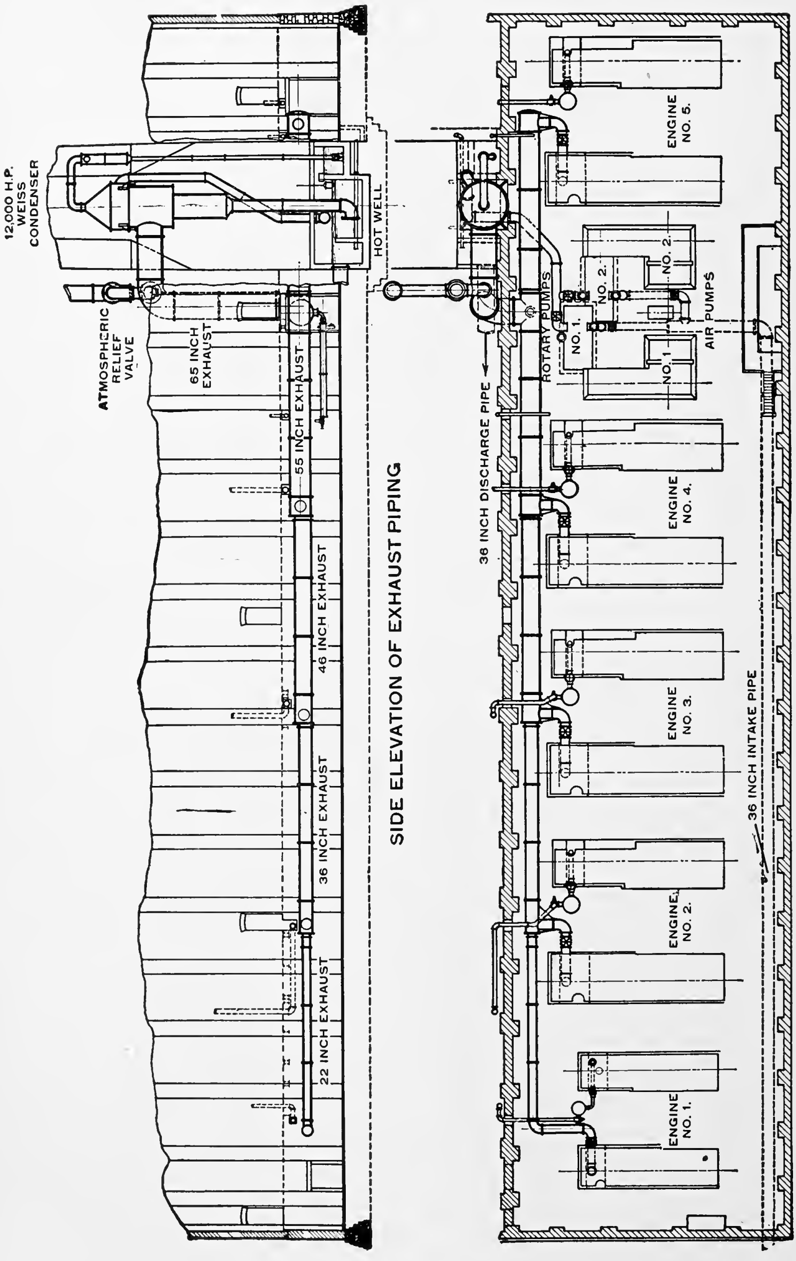


FIG. 330. Condenser and Exhaust Steam Piping at the Northwestern Elevated R.R. Power House, Chicago, Ill.

temperature of cooling water, 70 deg. fahr.; hot-well temperature, 105 to 120 deg. fahr.; distance between engine exhaust opening and mean level of intake well, 10 feet.

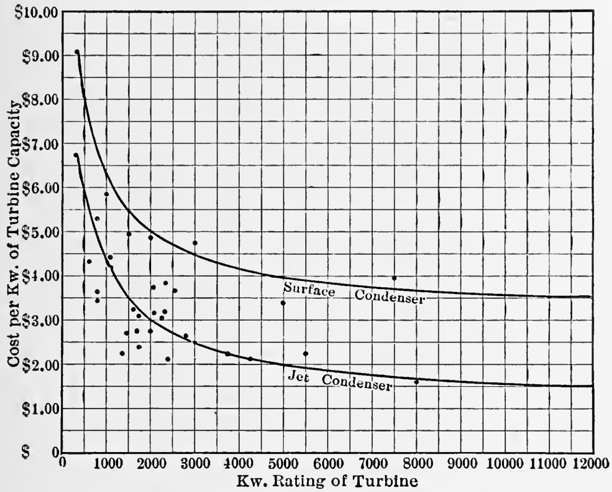


FIG. 331. Curves Showing Approximate Cost of Condenser Equipment per Kilowatt of Turbine Capacity.

Siphon Condensers.

1 16″ siphon condenser with 6″ centrifugal pump driven by 6″ by 6″ vertical engine	$800

Jet Condensers.

1 14″ by 22″ by 24″ jet condenser with single horizontal direct-acting pump	1335
1 16″ by 24″ by 18″ jet condenser with single vertical direct-acting pump	1620
1 14″ by 24″ by 18″ jet condenser with single vertical flywheel vacuum pump	1770
1 12″ by 17″ by 22″ by 25″ jet condenser, single horizontal direct-acting compound pump	2200

Barometric Condensers.

1 barometric condenser, 10″ by 16″ by 12″ horizontal single-cylinder rotative dry-air pump; 8″ horizontal volute centrifugal pump direct connected to 23-horsepower high-speed engine	2500
1 barometric condenser, 16″ by 16″ dry-air pump direct connected to 9″ by 16″ steam engine; positive rotary pump, for circulating cooling water, belted to above engine	4300

Surface Condensers.

1 surface condenser, 1025 square feet cooling surface, mounted over 7½″ by 14″ by 14″ by 12″ combined air and circulating pump	2100
1 surface condenser, 1025 square feet cooling surface, with 7½″ by 12″ by 12″ horizontal air pump, direct acting, and 6″ centrifugal pump driven by 5″ by 5″ engine	2300
1 surface condenser, 1025 square feet cooling surface; 5″ by 12″ by 10″ Edwards single-cylinder air pump and 6″ centrifugal pump driven by a 5″ by 5″ engine; maximum 28″, referred to 30″ barometer	2850
1 surface condenser, 1025 square feet cooling surface; 6″ by 8″ rotative dry-air pump; 6″ by 6″ Edwards wet-air pump and 6″ centrifugal pump driven by 5″ by 5″ engine; maximum vacuum 29″, referred to 30″ barometer (temp. cooling water 50 deg. fahr.)	3500

Westinghouse-Leblanc Jet Condenser.

1 jet condenser with turbine-driven pumps, 20,000 pounds steam per hour, 26″ vacuum, 70 deg. fahr. inlet water	2150
1 jet condenser with turbine-driven pumps, 20,000 pounds steam per hour, 29″ vacuum, 50 deg. fahr. inlet water	3275

In general the cost of complete condensing equipments installed and ready for operation will approximate as follows:

	Cost per Kilowatt of Main Generating Unit.
Siphon condensers without air punp	$2.00 to $3.00
Jet condensers	3.00 to 4.50
Barometric condensers with dry-air pump	4.00 to 6.00
Surface condensers for 26-inch vacuum	3.50 to 5.00
High-vacuum surface condensers	3.50 to 10.00
Leblanc jet condensers and pumps	2.00 to 6.00

249. Choice of Condensers. — The proper selection of condenser and auxiliaries for a proposed installation depends upon the conditions under which the plant is to be operated. These conditions vary so widely in practice that only a few of the more important factors will be considered. The principal advantages and disadvantages of the three types of water-cooled condensers are as follows:*

Advantages	Disadvantages.
Surface Condenser.	
Re-use of condensate for boiler feed.	First cost high.
Re-use of condensate for ice production.	Maintenance high.
Readily adapted to the weighing of condensate for tests.	Requires considerable building space to remove tubes.
Slightly better vacuum obtainable.	Acidulated water or water containing foreign matter in large quantities may preclude the use of surface condensers.
Advantage of low pumping head through siphon action.	More head room necessary to obtain sufficient head on hot-well pump.
Less chance of losing vacuum because a drop in vacuum does not affect water supply.	
Barometric Condenser.	
Condenser proper not costly, but piping to it is expensive.	Long exhaust pipe line to condenser which entails high initial cost and greater possibility of air leaks.
No possibility of flooding turbine as in the case of a low jet condenser.	Loss of vacuum between turbine and condenser, which may amount to ½ inch or even more.
Maintenance low.	As condenser cone generally extends above roof, it does not lend itself to economical station design when boiler room and turbine room are parallel and contiguous.
The use of acidulated water possible.	Waste of condensate.
Requires less circulating water than surface condenser.	
Requires little building space.	
Equipment simple. No hot-well pump necessary and in some forms no vacuum pump is required.	

* A. R. Smith, General Electric Review.

Jet Condenser.

Least expensive type of condenser.	Failure of removal pump would flood turbine. Protection is provided by a vacuum-breaking float valve.
Requires less building space.	Waste of condensate.
Equipment simpler because hot-well pump is not necessary.	
Requires less circulating water than surface condenser.	
Maintenance low.	
The use of acidulated water possible.	

Steam-driven condenser auxiliaries have been universally recommended in preference to motor drives because any disturbances on the electrical end will not affect the auxiliaries. For example, suppose a short circuit occurs on some outside feeder and the speed and voltage is reduced sufficiently to let the condenser auxiliaries drop out. First, the loss of vacuum on the turbine will necessitate the immediate generation of double the amount of steam, but the boiler room is not prepared for this emergency, and the only alternative is to reduce the load. Second, the vacuum pump has to be started, and, third, the circulating pump started and primed. The operations consume considerable time, especially with chaotic periods of interruption. Should there be two turbines on the line, the duration of interruption is doubled.

The dry vacuum pump and hot well pump can conveniently be made motor driven because the motors are small and can be self-starting. An interruption of 30 minutes of the vacuum pump or one minute of the hot-well pump ought to show but little effect on the vacuum.

Motor-driven auxiliaries are very desirable, in that they are cheaper in first cost and maintenance; they obviate the use of considerable steam and exhaust piping and the expense of maintenance and radiation incident thereto; the motor speeds are conducive to high pump efficiencies, and they are easily started and require little attention when running. To enjoy these advantages without sacrificing continuity of service is possible by feeding the auxiliaries for each turbine off a separate auxiliary turbine driving an exciter and a-c. generator. This may seem like an additional complication, but investigation will show that this auxiliary turbine can be operated at a speed of highest economy, and each pump can be operated at the most efficient speed. The auxiliary turbine can be exhausted into its own feed-water heater. See Fig. 357.

Unless an auxiliary turbine is employed, or steam-driven auxiliaries used, there is usually a shortage of exhaust steam for heating the feed water. Take, for example, a case where the turbine is running at half rated load: the steam-driven exciter and boiler feed pump would be taking about 5 per cent of the total steam, which would heat the feed water from 75 deg. fahr. (29 in. vacuum) to 125 deg. fahr. If the main turbine were carrying full rated load, the condition would be worse, as the auxiliary steam would represent only about $3\frac{1}{2}$ per cent and the increase in feed water temperature would be only 35 deg. fahr. Now, if the condenser auxiliaries are steam driven, the total exhaust steam would be about 15 per cent at half load and 9 per cent at full load, and the feed temperature in the former case would be 210 degrees with a waste of $1\frac{1}{2}$ per cent of the steam, and at full load it should be 165 deg. fahr.

The Selection of Steam Turbine Condenser: A. R. Smith, The National Engr., June, 1914, p. 351.

250. Water-Cooling Systems. — When an ample supply of cooling water is unobtainable, for natural or economic reasons, the circulating water may be used over and over again by employing suitable cooling devices. The three most common in practice are

1. The simple cooling pond or tank.
2. The spray fountain.
3. The cooling tower.

251. Cooling Pond. — The water is cooled partly by radiation and conduction but principally by evaporation. The air is seldom saturated normally, and its capacity for absorbing moisture is increased on account of its temperature being raised by contact with the warm water and by radiation. The cooling action is independent of the depth of water and varies directly as the surface, the amount of heat dissipated for each square foot depending upon the temperature of the water, the relative humidity, and the velocity of the air currents. Results of tests are very discordant.

Box in his Treatise on Heat states that the pond surface should approximate 210 square feet per nominal horsepower for an engine working twenty-four hours a day. (Treatise on Heat, Box, p. 152.)

If the engine works only twelve hours per day, the area may be reduced to 105 square feet per horsepower, because the water will cool during the night, but in that case the depth should be such as to give a capacity of 300 cubic feet per horsepower. These figures are based on a reduction in temperature of 122 to 82 deg. fahr., with air at 52 deg. fahr., and humidity 85 per cent, the steam consumption per nominal horsepower being taken at 62.5 pounds. It appears from tests that under ordinary conditions, in the northern part of the United States, with engines using 15 pounds of water per horsepower-hour and a vacuum of 26 inches, a reservoir having a surface of 120 square feet per horsepower would be ample for cooling and condensing water. (W. R. Ruggles, Proc. A.S.M.E., April, 1912, p. 607.)

Box gives the following formula for the rate of evaporation in perfectly calm air:

$$E = (243 + 3.7\,t)\,(V - v), \tag{230}$$

in which

E = evaporation in grains per square foot per hour,
t = temperature of the water, deg. fahr.,
V = maximum vapor tension in inches of mercury at temperature t,
v = actual vapor tension.

Evaporation is greatly affected by the force of the wind and varies from 2 to 12 times the amount determined from equation (230).

Example 50. How many pounds of water will be evaporated per square foot per hour from a pond with the temperature of the water and air 80 deg. fahr.; air perfectly calm; barometric pressure 29.5 inches and relative humidity 70 per cent?

The maximum vapor tension at temperature of 80 degrees is 1.03 inches of mercury. The actual vapor tension will be

$$1.03 \times 0.70\ (= \text{relative humidity}) = 0.721.$$

Substitute these values in equation (230).

$$\begin{aligned} E &= (243 + 3.7 \times 80)\,(1.03 - 0.721) \\ &= 167 \text{ grains per square foot per hour} \\ &= 0.024 \text{ pound per square foot per hour.} \end{aligned}$$

A rough rule is to allow a heat transmission of 3.5 B.t.u. per hr. per sq. ft of pond surface per degree fahr. difference in temperature between that of the air and water.

252. Spray Fountain. — From equation (230) we see that even under the most favorable circumstances an enormous pond surface is necessary. To facilitate evaporation with a view toward reducing the size of the pond, the hot circulating water is sometimes distributed through pipes and discharged through nozzles, falling to the surface of the pond in a spray.

The water issuing from the nozzles creates a draft which aided by the natural breeze, effects the necessary evaporation. The loss of water due to evaporation seldom exceeds 4 per cent of the weight of water circulated. The pressure required at the nozzles is approximately 6 pounds per sq. in. and in many cases the condenser pump is able to furnish the necessary pressure. Under ordinary conditions the power necessary to operate the sprays will average less than $1\frac{1}{2}$ per cent of the power generated by the prime mover. Should the temperature of the condenser discharge water exceed the limit of reduction by single spraying the desired reduction in temperature may be effected by double spraying. In this arrangement the condenser discharge is mixed in the hot well with an equal amount of cooler water flowing through an equalizing valve from the spray pond. The resulting mixture is pumped to the nozzles and resprayed. Some idea of the performance of a spray cooling system may be gained from the data in Tables 95 and 96.

TABLE 96.

SINGLE-SPRAY SYSTEM — 6000-KW. STEAM TURBINE PLANT.

Month.	Relative Humidity, Per Cent.	Temperatures, Degrees Fahrenheit.				
			8 A.M.	12 M.	4 P.M.	Remarks.
Jan.	62	Discharge water	68	73	73	Clear
		After spraying	48	53	53	
		Surrounding air	8	14	20	
Mar.	50	Discharge water	79	86	90	Clear
		After spraying	58	66	70	
		Surrounding air	30	50	43	
May	72	Discharge water	89	94	97	Clear
		After spraying	70	75	78	
		Surrounding air	65	72	70	
July	70	Discharge water	108	118	118	Clear
		After spraying	90	93	93	
		Surrounding air	90	98	102	
Aug.	84	Discharge water	112	114	116	Cloudy
		After spraying	88	89	90	
		Surrounding air	72	74	79	
Nov.	70	Discharge water	89	90	88	Cloudy
		After spraying	62	64	63	
		Surrounding air	27	33	34	

TABLE 97.

DOUBLE-SPRAY SYSTEM.

	First Spraying.	Second Spraying.
Temperature air, deg. fahr.	87.0	88.0
Relative humidity, per cent	48.5	46.0
Temperature hot water, deg. fahr.	122.5	88.7
Temperature, cooled water, deg. fahr.	88.3	78.8
Total degrees cooled, fahr.	....	44.1

Natural ponds without sprays require about 50 times more area than spray cooling systems. A rough rule is to allow 130 B.t.u. per sq. ft. per hr. per degree difference in temperature.

253. Cooling Towers. — A cooling tower consists of a wooden or sheet iron housing open at the top and bottom and so arranged that the hot water may be elevated to the top and distributed in such a manner that it falls in thin sheets or sprays into a reservoir at the bottom, air at the same time being drawn in at the bottom by natural draft or forced in by a fan. The water gives up its heat to the ascending current of air by evaporation, convection and radiation, the latter, however, being a relatively small factor. Of these, evaporation absorbs from 75 to 85 per cent of the heat, convection or direct transfer of heat to the air comes next, while radiation partly in the tower and partly through the piping accounts for the balance. If the air supply is dependent entirely upon the chimney action of the device the system is known as a natural draft or flue cooling tower; if the air is forced into the device by fans the system is called a forced draft cooling tower. Water cooling towers may be classified as (1) forced draft, (2) natural draft — open type or atmospheric, (3) natural draft — closed or flue type, and (4) combined forced and natural draft.

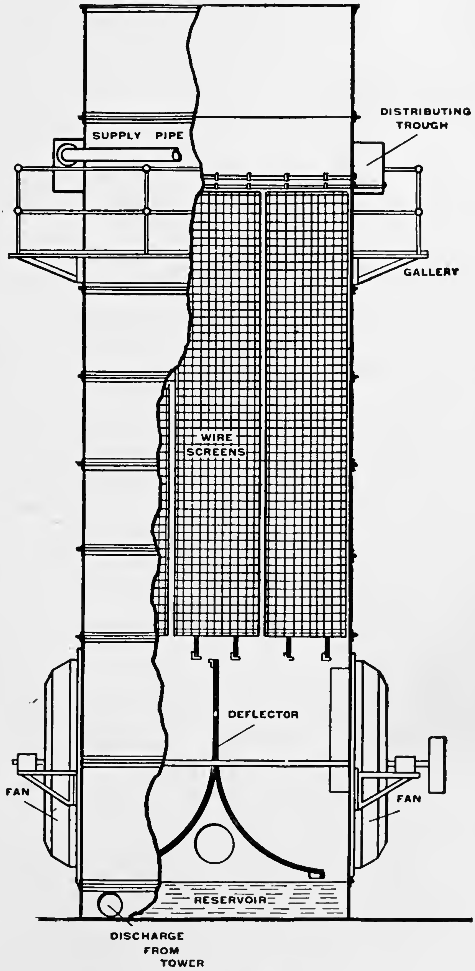


Fig. 332. Barnard-Wheeler Cooling Tower.

Forced draft towers are completely enclosed, except at the top and at the base where provision is made for the fan openings. In the atmospheric type of natural draft tower the sides are louvered and the necessary air is supplied through the open base and through the louvered sides by natural air currents. The flue type of natural draft tower receives its air supply through the chimney action of the flue. The combined forced and natural draft tower may be used with natural draft only for light loads and forced draft for heavy loads.

The different designs vary principally in the method of water distribution. Fig. 332 illustrates the Barnard-Wheeler cooling tower in which the falling water is broken up by vertically suspended galvanized iron wire cloth mats, causing it to trickle in thin sheets to the bottom. A similar result is brought about in the Worthington tower by pieces of terra cotta pipe 6 inches in diameter and two feet long

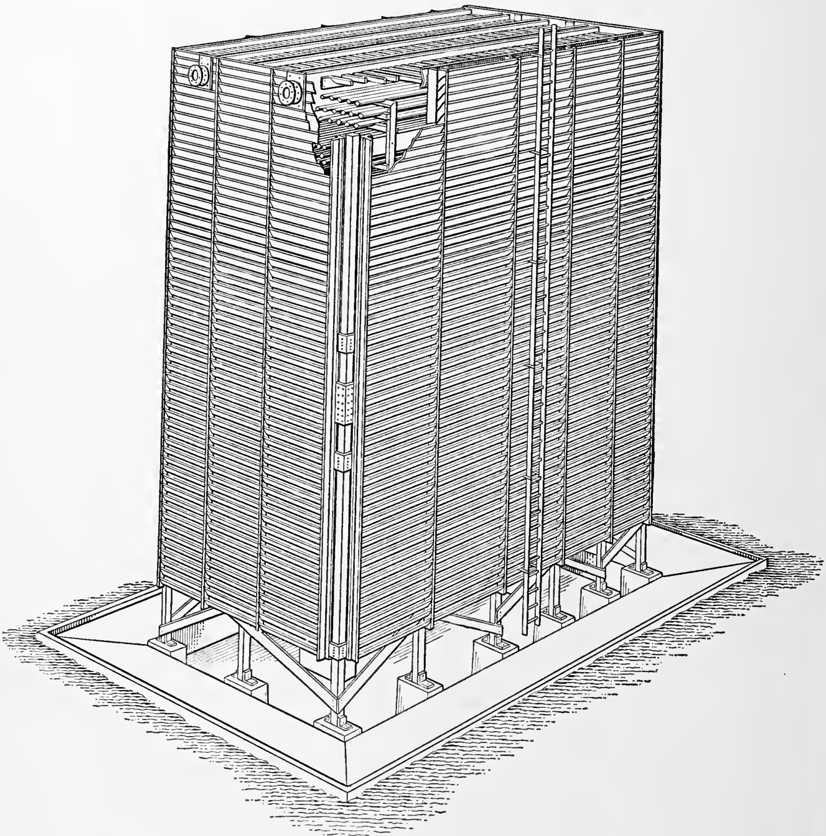


FIG. 333. C. H. Wheeler Atmospheric Cooling Tower.

placed on ends in rows. In the *standard type* of Alberger cooling tower the water trickles down the sides of swamp-cypress boards arranged in honeycomb fashion. In the Alberger *improved type* the fan is placed at the top of the tower with its shaft in a vertical position. The fan is operated by a Pelton water wheel which receives its power from a turbine pump. No oil lubrication is employed, and the operating mechanism is controlled entirely from the engine room. In the Jennison cooling tower the water is divided into a rain of drops, constantly retarded in their fall by a series of perforated 4 × 4-inch galvanized-iron trays arranged in horizontal rows and staggered vertically.

With the best forms of cooling towers, under average conditions, the temperature of the circulating water may readily be reduced from 40 to 50 degrees with a loss not exceeding 3 or 4 per cent of the total quantity of water passing through the tower. The power consumed by the fan in a forced-draft apparatus averages 2 per cent of that developed by the main engines, for the maximum requirements during summer months, and $1\frac{1}{4}$ per cent during the winter.

The location of the tower may be on the engine-rooom floor, on top of the building, or in the yard, the latter being the most adaptable. It may be any reasonable distance from the engine and condenser.

254. Test of Cooling Towers.

RESULTS OF TEST OF NATURAL-DRAFT TOWER, DETROIT.

COMPLETE FIVE-FIFTHS SURFACE INSTALLED.

Proc. A.S.M.E. Mid-Nov., 1909, p. 1205.

Engines:	Two 400-i.hp., 300-kw. MacIntosh & Seymour tandem-compound engines, overhung generators.				
Condensers:	Worthington surface (admiralty type) 1600-sq. ft. reciprocating wet-air pump and circulating pump.				
Tower:	Wood-mat construction, 24,500 sq. ft. evaporating surface, exclusive of shell.				
Test:	March 15 to 16, 1901, 4 p.m. to 4 p.m., 24 hr.				
		A.M.		P.M.	AVERAGE.
Weather:	Barometer (abs.), min.	30.22	30.07;	30.14	30.27
	Temperature air, deg.	18.5	25;	30	25
	Relative humidity, per cent.	76	82;	58	72
Load:	600 kw. max. to 50 kw. min. Average				244.9 kw.
	Engine efficiency = 92.5 = 875 i.hp. max. Average				354.8 i.hp.
Steam:	Weight of condensed steam per hr., lb.				5910.6
	Temperature exhaust steam, deg. fahr.				134.38
	Temperature condensed steam, deg. fahr.				108.78
	Weight of steam per hour, max. load, lb.				13,500
	Vacuum (abs.) 25 to 19, average about				22
	Vacuum corresponding to temperature exhaust steam				25
	Vacuum possible with good condenser (10 deg. difference)				28
Water:	Circulated per hr., lb.				293,536
	Temperature hot well, average, deg. fahr.				87.50
	Temperature cold well, average, deg. fahr.				71.27
	Vaporization loss per hr., lb.				5970
Results:	Condenser surface per kw., sq. ft.				2.66
	Steam per kw-hr., lb.				24.3
	Steam per i.hp-hr., lb.				16.66
	Circulating water per lb. of steam, lb.				49.6
	Steam per sq. ft. condenser surface per hr., lb.				3.7
	Circulating water per sq. ft. tower surface, lb.				12
	Difference in temperature between exhaust steam and discharge, deg. fahr.				47

Cooling:	Max. 20 deg., min. 3 deg.-5 deg. Average	16.23
	Heat dissipated per hr., B.t.u.	4,769,000
	Heat per sq. ft. tower surface, B.t.u.	195
	Heat per sq. ft. per 1000 lb. water, B.t.u.	0.665
Evaporation:	Circulating water, per cent	2.03
	Engine steam, per cent	101
Tower:	Surface per kw. (average load 245 kw.), sq. ft.	100
	Surface per kw. (max. load 600 kw.), sq. ft	40.8
	Surface per 1000 lb. steam max. load, sq. ft	1820
	Surface per 1000 lb. steam average load, sq. ft	4140
	Surface per 1000 lb. circulating water per deg. max. cooling, sq. ft	4.17

Time.	Temperature, Deg. Fahr.					Quantities.				
	Air.	Hot Well.*	Cold Well.	Water Cooling.	Total Heat Head.†	Tower Water, Lb. per Hr.	Heat Dissipated, B.t.u. Lb. per Hr.	Heat per Sq. Ft. Cooling Surface, B.t.u. per Hr.	Circulating Water per Sq. Ft., Lb. per Hr.	Load, Kw.
1	2	3	4	5	6	7	8	9	10	11
12 noon	34	102	89	13	68	375,000	4,880,000	332	25	270
1.30	35	106.5	90	16.5	71.5	‡ 375,000 370,200	6,108,000	415	24.8	315 290
2.30	35	106.5	87.5	19	71.5	375,000	7,120,000	484	25	315
3.30	35	113	88.5	24.5	78	375,000	9,000,000	613	25	350
4.30	32.5	100	84	16	67.5	399,000	6,384,000	434	26.6	365
5.00	28.5	103.5	88	15.5	75	445,500	6,900,000	470	29.7	485
6.00	26	125	94	31	99	417,000	12,930,000	880	27.8	655
7.00	24	121	94	27	97	427,000	11,532,000	785	27.4	570
8.00	24	123	94.5	28.5	99	427,000	12,174,000	827	27.4	600

* Assuming a more efficient condenser, say 10 deg. difference, the probable vacuum would be 26 deg. to 27.5 deg. This condenser actually operated at 40 deg. to 50 deg. difference.

† Total heat head = air heating + lost head. ‡ Difference due to rapid change in load.

For cooling tower calculations and problems in hygrometry see Chapter XXV.

PROBLEMS.

1. Reading of vacuum gauge 26.5, temperature of room 80 deg. fahr., barometer 29.5, temperature of mercury in the barometer 40 deg. fahr. Determine the vacuum referred to a 30-inch barometer.

2. If the absolute temperature in a condenser is 5 inches of mercury and the temperature of the air-vapor mixture in the chamber is 90 deg. fahr., required the percentage of air (by weight) in the mixture.

3. If the temperature within a condenser is 100 deg. fahr. and there is entrained 0.1 lb. of air per lb. of steam, required the maximum degree of vacuum obtainable.

4. Required the volume of aqueous vapor to be withdrawn in order to cool 10,000 lb. of water from 120 to 80 deg. fahr.

5. A 30,000-kw. turbine uses 12 lb. steam per kw-hr., initial pressure 290 lb. abs., superheat 250 deg. fahr., vacuum 28.5 in. referred to a 30-in. barometer; initial temperature of the cooling water 70 deg. fahr., water velocity through tubes 8 ft. per sec. Required:

a. Weight of cooling water.

b. Sq. ft. condenser tube surface.

c. Number of 18 B.W.G. tubes in each pass of the condenser.
d. Length of water travel.

6. A 200-kw. turbine uses 20 lb. steam per kw-hr., initial pressure 150 lb. absolute, superheat 100 deg. fahr., vacuum 27 in. referred to a 30-in. barometer. If an evaporative surface condenser of the forced draft type is used to create the vacuum, required the amount of atmospheric air and water spray which must be forced through the condenser. The temperature of the atmospheric air is 80 deg. fahr., wet bulb thermometer 65 deg. fahr., air issuing from the condenser is completely saturated and its temperature is 15 degrees below that of the vapor in the condenser, fan pressure 4 in. of water.

7. How much "make up" water is necessary for the cooling tower system of a steam engine plant operating under the following conditions: Engines 1000 hp., water rate 20 lb. per i.hp-hr. initial pressure 120 lb. abs., vacuum 26 in., barometer 30 in.; temperature of injection water, discharge water and atmospheric air, 90, 110 and 70 deg. fahr., respectively; relative humidity of air entering and leaving tower 65 and 95 per cent respectively.

CHAPTER XII

FEED WATER PURIFIERS AND HEATERS

255. General. — All natural waters contain more or less foreign matter either in suspension or solution. The organic constituents of this foreign matter are of vegetable and animal origin taken up by water flowing over the ground or by direct contamination with sewage and industrial refuse. Feed water containing organic matter may cause foaming due to the fact that the suspended particles collect on the surface of the water in the boiler and impede the liberation of the steam bubbles arising to the surface.

The suspended inorganic impurities consist of clay, silica, iron, alumina, and the like, in the form of mud and silt. The more common soluble inorganic impurities are lime, magnesia, iron and sodium in the form of carbonates, sulphates and chlorides, oxides of silica, iron and alumina, some free carbonic acid and occasionally free sulphuric acid and hydrogen sulphide.

When raw water is fed into a boiler all of the solids remain in the boiler and are constantly increased in amount by the evaporation taking place. Some of the accumulated impurities deposit on the heating surface as scale, some are present as suspended matter and others remain in solution. The most widely known evidence of the presence of scale-forming ingredients in feed water is known as hardness. If the water contains only such ingredients as carbonates of lime, magnesia and iron which may be precipitated by boiling at 212 deg. fahr. it is said to have *temporary hardness*. *Permanent hardness* is due to the presence of sulphates, chlorides and nitrates of lime, magnesia and iron which are not completely precipitated at a temperature of 212 deg. fahr. Hardness is conveniently determined by means of a standard soap solution as follows:

A 100-cc. (cubic centimeter) sample of water to be tested is put in a 250-cc. bottle and a standard soap solution (this may be obtained from chemical dealers) run in 0.2 cc. at a time, the bottle being shaken vigorously after each addition of the soap solution. Finally a lather is produced that will persist for at least five minutes, and then the volume of soap solution used in cc. gives the degrees "U. S." hardness. One degree "U. S." hardness is equivalent to 1 grain of calcium carbonate per U. S. gallon (1 part in 58,349).

TABLE 98.

WATER AND BOILER SCALE ANALYSES.

Water Analysis. Grains Per U.S. Gallon.	Lake Michigan, Chicago.	Well 115 Feet, St. Bernard, Ohio	San Francisco, Cal.	Schuylkill River, Philadelphia.	Camaguey, Cuba.	Park City, Utah.	Toledo, Ohio.	Kewanee, Ill.	Arkansas River, Florence, Colo.	Surface Water, Auburn Park, Ill.
	1	2	3	4	5	6	7	8	9	10
Silica	0.438	0.677	0.759	0.338	2.873	1.354	0.759	0.373	0.630	0.508
Oxide of iron and aluminum	0.099	0.116	0.116	0.093	0.140	0.350	0.163	0.081	0.075	0.175
Carbonate of lime	3.731	2.271	4.207	0.068	10.270	1.476	5.519	1.721	2.158	2.382
Sulphate of lime	0.962	4.083	0.680	2.257	3.220	1.360	3.950	1.360	18.540	3.154
Carbonate of magnesia	2.092	4.424	2.866	0.884	4.900	0.318	2.592	2.212	4.848	2.875
Sodium and potassium sulphates	Trace	Trace	1.681	Trace	Trace	0.867	Trace	12.928	11.319	Trace
Sodium and potassium chlorides	0.670	0.990	2.970	0.990	5.708	1.980	2.740	26.070	2.028	1.650
Organic matter	0.066	0.584	....	0.700	33.000	2.569	1.052	0.584	0.701	0.584
Total mineral matter	8.058	12.614	13.665	4.672	32.288	7.826	15.885	45.318	40.062	11.096
Chloride of magnesia	....	....	....	....	5.6	....	....	....	....	....

SCALE ANALYSIS — PER CENT.

Character of sample	Hard, brittle.	Medium hardness	Hard, brittle.	Hard, impervious	Very hard.	Very hard.	Hard.	Soft, brittle.	Hard, crystalline.	Medium hardness
Silica	20.60	8.44	11.18	12.30	24.42	19.00	4.96	2.52	6.20	5.7
Oxide of iron and aluminum	10.30	1.30	10.44	6.18	1.02	6.26	11.80	4.92	2.36	2.04
Carbonate of lime	33.86	37.22	40.96	21.26	29.10	29.02	3.74	18.18	18.78	29.86
Sulphate of lime	None	33.82	Trace	34.62	0.96	5.48	55.38	54.76	59.84	39.64
Carbonate of magnesia	6.04	Trace	22.60	Trace	Trace	Trace	8.19	Trace	0.84	Trace
Magnesia (MgO)	15.48	12.01	....	8.20	25.94	1.45	6.83	9.08	4.75	13.8
Moisture and organic matter	12.89	6.22	13.58	11.70	16.66	13.69	8.69	7.40	5.73	7.64
Oil	Trace	....	....	0.27	....	....	....	2.92	....	....
Loss and undetermined	0.83	0.99	1.24	0.23	1.90	1.55	0.38	0.22	1.50	1.32
Lime (CaO)	....	....	....	5.24	....	23.55	....	....	....	....

1. This water will cause the deposit of a moderate amount of scale which will be hard and persistent.

2. This water will cause a large amount of scale to deposit.

3. This water will cause a moderate amount of scale with a decided tendency to galvanic action on account of the large proportion of sodium and potassium salts present.

4. This water will cause the formation of a moderate amount of very hard scale.

5. This water will cause the deposition of a moderate amount of hard scale. The sodium and potassium salts together with the chloride of magnesia will induce galvanic action with consequent corrosion, pitting, etc.

6. This water will cause the formation of some scale. There is also a decided tendency to corrosive action.

7. Will cause a hard and impervious scale to form.

8. Will cause formation of some incrustation of medium hardness. It will also cause considerable trouble due to galvanic action, foaming and priming.

9. This is not a desirable feed water. It will cause the formation of considerable scale and will cause corrosion, pitting, and possibly foaming.

10. Will cause the formation of a moderate amount of very hard scale.

The following factors may be used for specifying hardness of water in terms of calcium carbonate per U.S. gallon.

$$\left.\begin{array}{ll} \text{Magnesium carbonate} & \times\ 1.19 \\ \text{Magnesium sulphate} & \times\ 0.833 \\ \text{Calcium sulphate} & \times\ 0.735 \\ \text{Magnesium chloride} & \times\ 1.05 \\ \text{Calcium chloride} & \times\ 0.901 \end{array}\right\} = \begin{array}{l}\text{hardness as calcium carbonate,} \\ \text{grains per U. S. gallon or U. S.} \\ \text{degrees.}\end{array}$$

It is impossible to judge the quality of feed water merely by the grains of solids per gallon since a large amount of soluble salt such as sodium chloride will not be as deleterious as a very small amount of calcium sulphate.

The scale of hardness usually accepted (grains of dissolved salts per U. S. gallon) is as follows: Soft water, 1 to 10; moderately hard 10 to to 20; very hard water, above 25.

The following is a rough rating according to the number of grains of incrusting solids per United States gallon:

Less than	
8 grains	very good.
12 to 15 grains	good.
15 to 20 grains	fair.
20 to 30 grains	bad.
Over 30 grains	very bad.

This applies to calcium carbonate, magnesium carbonate, and magnesium chloride. For water containing sulphate of calcium and magnesium, divide the first column by 4 for the same rating.

The limiting factor in deciding whether a water carrying a large amount of soluble salts may be used for boiler feed purposes is the amount of blowing down necessary to keep the degree of concentration within the limits found by experience.

256. Scale. — Scale is formed on boiler heating surfaces by the depositing of impurities in the feed water and varies from a porous, friable crust to a dense, very hard coating. The amount of scale formed does not bear a direct relation to the amount of impurities present but depends on the type of boiler, rate of driving and the nature of the scale-forming ingredients in the water. Scale tends to lower the efficiency and capacity of the boiler and may cause overheating of the plates and tubes.

Table 99 gives the results of a number of tests made on locomotive boiler tubes with different thicknesses and characters of scale. The diversity of the results indicates the futility of basing the decrease in conductivity on the thickness of the scale. For example, test No. 1 shows a decrease in conductivity of 9.1 per cent for a scale 0.02 inch thick, while No. 16 shows a decrease of only 6.75 per cent for a scale over 6.5 times as thick. The scale in each case was even, hard, and dense. Again, No. 8 with a very soft scale 0.042 inch thick gives a decrease in conductivity of 9.54 per cent, whereas No. 14, also very soft but twice as thick, gives a decrease of only 4.95 per cent. No doubt the heat transmission is a function of the chemical as well as the physical properties, but further experiments are necessary before any specific conclusion can be drawn.

TABLE 99.

INFUENCE OF SCALE ON HEAT TRANSMISSION.

(Locomotive Boiler Tubes.)

No.	Thickness of Scale, Inches.	Character of Scale.	Decrease in Conductivity due to Scale. Per cent.
1	.02	Hard, dense	9.1
2	.02	Hard	2.02
3	.033	Soft	4.3
4	.033	Very hard	3.5
5	.038	Medium	4.03
6	.04	Soft, porous	6.82
7	.04	Hard, dense	3.07
8	.042	Very soft	9.54
9	.047	Hard	2.75
10	.065	Medium	2.39
11	.07	Soft	2.38
12	.07	Hard	4.43
13	.085	Soft, porous	19.0
14	.089	Very soft	4.95
15	.11	Hard, porous	16.73
16	.13	Hard, dense	6.75

From tests conducted at the University of Illinois, *Railroad Gazette*, Jan. 27, 1899, June 14, 1901. See also *Engineering Record*, Jan. 14, 1905, p. 53; *Power*, February, 1903, p. 70; *Street Railway Review*, July 15, 1901, p. 415.

A moderate amount of scale has little influence on the efficiency and capacity of boilers operating at or below normal rating but for high driving rates scale must not be permitted to accumulate. In the modern central station with its heavy peak loads pure feed water is of vital importance to economy and continuity of operation. For scale prevention see paragraphs 260 to 266.

257. Foaming and Priming. — Boiler troubles due to foaming or priming are often caused by concentration of alkali salts in the water within the boiler, although silt, organic matter, loosened scale, lubricating oil, rate of driving and the design of the boiler all have bearing upon this phenomenon. Where this is caused by excessive concentration it may be largely overcome by frequent blowing down. Surface blowing is, of course, a remedy where it can be applied. Foaming caused by organic matter in suspension may be minimized by filtration.

258. Internal Corrosion. — Corrosion is evidenced by small pits or depressions and by large cup-shaped hollows on the metal surface, and occasionally by a considerable destruction of a large portion of the surface. Carbonic acid gas, occluded oxygen, sodium, calcium and magnesium chlorides are common causes of corrosion. Magnesium and calcium chlorides are very pernicious in that they produce free hydrochloric acid on hydrolysis. Corrosion is also found in boilers using a high percentage of condensate or distilled water. A theory* accepted by physicists embraces the fact that in the presence of a solvent the iron goes into solution as a hydrate before oxidizing. Considering that water is a universal solvent every metal has an inherent tendency to dissolve in water or water solutions. This tendency is called the solution tension of the metal. Opposing this tendency to dissolve is a pressure in the solution tending to resist the entrance into the solution of any more of the metal. This opposing pressure is known as the osmotic pressure of the solution. Accepting this theory, it is only necessary, in order to prevent corrosion, to raise the osmotic pressure of the solution or electrolyte, above the solution tension of the metal. In

TABLE 100.

SUMMARY OF INSPECTOR'S REPORTS FOR THE YEAR 1916.

(Hartford Steam Boiler Inspection and Insurance Company.)

Nature of Defects.	Whole Number.	Dangerous.
Cases of sediment or loose scale	28,212	1,593
Cases of adhering scale	42,877	1,612
Cases of grooving	2,568	315
Cases of internal corrosion	19,008	793
Cases of external corrosion	10,968	814
Cases of defective bracing	984	266
Cases of defective staybolting	2,049	504
Settings defective	9,401	814
Fractured plates and heads	3,711	563
Burned plates	5,361	498
Laminated plates	278	27
Cases of defective riveting	1,448	201
Cases of leakage around tubes	12,554	1,581
Cases of defective tubes or flues	15,080	4,989
Cases of leakage at seams	5,537	373
Water gages defective	4,192	714
Blows-offs defective	4,262	1,337
Cases of low water	420	123
Safety valves overloaded	1,386	235
Safety valves defective	1,695	358
Pressure gages defective	8,351	815
Boilers without pressure gages	32	32
Miscellaneous defects	4,261	662
Total	184,635	19,219
Condemned	25,901	....

* A. H. Babcock, Trans. A.S.M.E., Vol. 37, p. 1119.

the case of boilers, the osmotic pressure of the electrolyte or boiler water, is raised by the addition of alkaline salts. The osmotic pressure being dependent on the concentration of the salts, the corrosive condition of the electrolyte is indicated by its alkaline strength. The alkaline strength to be carried in the boiler depends on the salts used.

Table 100, compiled by the Hartford Steam Boiler Inspection and Insurance Company, shows the number of boilers inspected by that company during the year 1916 and the number found defective from various causes.

259. General Feed Water Treatment. — Table 101 ("Boiler Waters," W. W. Christie) outlines some of the troubles arising from feed water, their cause and means for preventing them.

TABLE 101.

BOILER TROUBLES ARISING FROM USE OF IMPURE FEED WATER.

Trouble.	Cause.	Remedy or Palliation.
Incrustation.	Sediment, mud, clay, etc.	Filtration. Blowing off.
	Readily soluble salts	Blowing off.
	Bicarbonate of magnesia, lime, iron	Heating feed and precipitate. Caustic soda. Lime. Magnesia.
	Organic matter	See below.
	Sulphate of lime	Sodium carbonate. Barium chloride.
Corrosion.	Organic matter	Precipitate with alum Precipitate with ferric chloride } and filter
	Grease	Slaked lime Carbonate of soda } and filter
	Chloride or sulphate of magnesium	Carbonate of soda.
	Sugar Acid	Alkali.
	Dissolved carbonic acid and oxygen	Slaked lime. Caustic soda. Heating.
	Electrolytic action	Zinc plates.
Priming.	Sewage	Precipitation with alum or ferric chloride and filter.
	Alkalies	Heating feed and precipitate.
	Carbonate of soda in large quantities	Barium chloride.

The neutralization or elimination of the impurities may be effected by one or more of the following methods:

1. Chemically.
    Water-softening plants.
    Boiler compounds.

2. Mechanically.
   - Filters.
   - Blow-off.
   - Tube cleaners.
3. Thermally.
   - Feed-water heater.
   - Distillation.

260. Boiler Compounds. — The object of treatment with boiler compounds is to neutralize the evil effects of the impurities in the feed water or to change them into others which are less objectionable and which are easily removed. When properly compounded and introduced into the boiler such preparations are of great benefit, but when improperly used they may produce even greater troubles than the impurities which they are expected to eliminate.

Boiler compounds may be divided into three classes:

1. Those converting the scale-forming elements into new substances which will not form a hard, resisting scale and which are readily removed by skimming, blowing off, or by tube cleaners. For example, feed water containing sulphates of lime and magnesia will form a dense, tenacious scale. If carbonate of soda be added in correct amount the sulphates are converted into insoluble carbonates which are precipitated and form scale varying from a more or less porous, friable crust to a soft "mush" or mud. The resulting sulphate of soda remains in solution and does not form scale unless allowed to concentrate and this is prevented by blowing off. An excess of soda is apt to cause foaming and at high temperatures is liable to attack the inside of gauge glasses. Bisodium and trisodium phosphate, sodium tannate, fluoride of sodium, sugar, etc., have all proved satisfactory, but as each case requires special treatment no detailed discussion is possible within the scope of this work and the reader is referred to the accompanying bibliography.

2. Those enveloping the newly precipitated scale-forming crystals with a surface which prevents them from cementing together. The ingredients used to bring about this result are starches, woody fibers, dextrine, slippery elm, and the like.

3. Those preventing the formation of hard scale by a solvent or "rotting" action, as kerosene and petroleum oils.

Under favorable conditions all that the most effective boiler compound can do is to change the nature of the precipitate from one which adheres to the boiler to one which will be carried in suspension. The accumulation of sludge in the boiler resulting from the use of a compound cannot be entirely removed by blowing off and consequently frequent washing out becomes a necessity.

Compounds for minimizing the formation of scale are recommended for use only in small plants where the cost of treating the water before it enters the boiler is prohibitive or in plants where space limitations prevent the installation of a purifying plant.

Patented Boiler Compounds: Prac. Engr., Aug., 1911, p. 523.

261. Use of Kerosene and Petroleum Oils in Boiler Feed Water. — Kerosene oil and other refined petroleum oils are sometimes used with good effect in boilers to soften scale. These oils are said to change the deposit of lime from a hard scale to a friable material which may be easily removed. To be reasonably effective the kerosene should be introduced after the boiler is emptied and washed and the refilling should be effected from the bottom. Kerosene should not be fed into the boiler with the feed water since it may form a non-conducting film over the heating surfaces.

Use of Kerosene in Boilers: Engr. U. S., Sept. 15, 1905, p. 634; Eng. News, May 24, 1890, p. 497; Power, Nov. 8, 1910, p. 1993; Trans. A.S.M.E., 9-247, 11-937; Locomotive, July, 1890, p. 97.

262. Use of Zinc in Boilers. — Zinc is often introduced into boilers to prevent corrosion. The theory is that a feeble but continuous current of hydrogen is generated over the whole extent of the iron by electrolytic action. The bubbles of hydrogen formed isolate the metallic surface from scale-forming substances. If there is but a little of the scale-forming element it is precipitated and reduced to mud; if there is considerable, coherent scale is produced which takes the form of the iron surface but does not adhere to it, being prevented from doing so by the intervening bubbles of hydrogen. Zinc is ordinarily suspended in the water space of the boiler in the shape of blocks, slabs, or as shavings in a perforated vessel. Electrical connection between the metallic surfaces is essential. Rolled zinc slabs $12 \times 6 \times \frac{1}{2}$ inches have found much favor in marine practice. Generally speaking one square inch of zinc surface is sufficient for every 50 pounds of water in the boiler, though the quantity placed in the boiler should vary with the hardness. The British Admiralty recommends the renewing of the zinc slabs whenever the decay has penetrated to a depth of $\frac{1}{4}$ inch below the surface. Zinc does not prevent corrosion or scale formation in all cases and may even aggravate the trouble.

Use of Zinc in Boilers: Prac. Engr., Dec., 1911, p. 835; Power, Oct. 18, 1910, p. 1874; Sept. 27, 1910, p. 1734.

263. Methods of Introducing Compounds. — Boiler compounds may be introduced into the boiler continuously or intermittently. Small quantities introduced continuously or at short intervals are more effective than large quantities at long intervals. Continuous feeding is ordinarily brought about by connecting the suction side of the feed pump with a reservoir containing the compound in solution, arranged similarly to an ordinary cylinder oil lubricator. In large plants an independent pump is often used to force the solution into the feed line. Intermittent feeding is brought about by temporarily connecting the suction of the feed pump with the reservoir containing the compound. The use of boiler compounds does not necessarily prevent scale from forming in time, though it will reduce the evil to a minimum. In some instances where compounds are used it is found necessary to run a tube cleaner through the tubes at certain intervals, in others such a course has not been found necessary.

264. Mechanical Purification. — Waters containing sand, mud, organic matter, and in fact all matter which is not in solution or in chemical combination with the water may be purified by mechanical filtration. Mud and sand may be eliminated by simply permitting the water to stand for some time in settling tanks. Suspended matter which will not gravitate to the bottom may be removed by filtering the water through coke, cloth, excelsior, or the like. Filters should be in duplicate for continuity of operation.

Vegetable and other organic impurities commonly float on the surface of the water when the boiler is making steam, and may be blown out through a "surface blow-out." (See paragraph 88.)

Precipitated matter may be ejected from the boiler by frequent blowing off before it has time to adhere and bake to a crust. This procedure is particularly essential when boiler compounds are used.

For description and use of mechanically operated tube cleaner see paragraph 92.

265. Thermal Purification. — (See also Live Steam Purifiers, paragraph 298.) The carbonates of lime and magnesia are held in solution in fresh water by an excess of carbon dioxide and are completely precipitated by boiling. At ordinary temperatures carbonate of lime is soluble in approximately 20,000 times its volume of water, at 212 deg. fahr. it is slightly soluble, and at 290 degrees it is insoluble. Sulphate of lime is much more soluble in cold than in hot water, and is completely precipitated at 290 degrees. (Revue de Mécanique, November, 1901, pp. 508, 743.)

Thus it will be seen that the application of heat will completely precipitate these scale-forming elements provided the temperature is high enough and sufficient time is allowed for action. In the commercial type of exhaust and live steam heaters complete precipitation cannot be effected on account of the short time the water is held in them, and because of the limited space for retaining the scale. There is no question but that some of the scale-forming elements are removed from the feed water by exhaust and live steam heaters but the amount precipitated is but a small fraction of the total except in cases of unusually pure water.

Efficiency of Live Steam Feed Heater: Power, Feb. 21, 1911, p. 295.

266. Water Softening. — When feed water contains a large amount of scale-forming material it is usually advisable to "soften" it before allowing it to enter the boiler rather than to introduce the chemical reagent into the boiler. The complete softening of water requires the removal of both its temporary and its permanent hardness. When water is softened outside the boiler and the sludge removed by sedimentation and filtration before delivering it to the heater the chemicals used are almost invariably lime, $Ca(OH)_2$, and soda ash, Na_2CO_3, alone or in combination with each other. Other chemicals may effect the desired result more efficiently but their cost is prohibitive. The chemical changes which take place when these reagents are added to water containing calcium sulphate, $CaSO_4$, magnesium sulphate, $MgSO_4$, calcium bicarbonate, $Ca(HCO_3)_2$ or magnesium bicarbonate, $Mg(HCO_3)_2$, are as follows:

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4. \tag{230}$$

$$MgSO_4 + Ca(OH)_2 + Na_2CO_3 = Mg(OH)_2 + CaCO_3 + Na_2SO_4. \tag{231}$$

$$Ca(HCO_3)_2 + Ca(OH)_2 = 2CaCO_3 + 2H_2O. \tag{232}$$

$$Mg(HCO_3)_2 + 2Ca(OH)_2 = Mg(OH)_2 + 2CaCO_3 + 2H_2O. \tag{233}$$

From these reactions the amount of reagent to be added to raw water may be calculated by considering the combining weights as follows:

For soda ash and calcium sulphate

$$CaSO_4 : Na_2CO_3 = 1 : x, \tag{234}$$

$$40 + 32 + 4\,(16) : 2(23) + 12 + 3(16) = 1 : x, \tag{235}$$

$$x = 0.779,$$

in which

x = soda-ash factor or the weight of soda ash required per lb. of calcium sulphate.

By similar calculations the factors for salts which require soda ash are found to be as follows:

Salt.	Soda-ash factor.
Calcium chloride, $CaCl_2$	0.955
Magnesium chloride, $MgCl_2$	1.113
Magnesium sulphate, $MgSO_4$	0.881
Calcium sulphate, $CaSO_4$	0.779

For salts which require lime:

Salt.	Factor.	
	Lump-lime, CaO.	Hydrated-lime, $Ca(OH)_2$.
Sodium carbonate, Na_2CO_3	0.529	0.699
Magnesium chloride, $MgCl_2$	0.589	0.778
Magnesium sulphate, $MgSO_4$	0.466	0.616
Magnesium bicarbonate, $Mg(HCO_3)_2$	0.767	1.014
Magnesium carbonate, $MgCO_3$	1.330	1.757
Calcium bicarbonate, $Ca(HCO_3)_2$	0.346	0.457
Calcium carbonate, $CaCO_3$	0.560	0.740

If any of the salts tabulated above occur in a water analysis multiply the amount of each by the corresponding factor and the product will represent the weight of reagent to be used.

The sulphates and chlorides of sodium and potassium in raw water need not be considered since they do not add to the hardness.

If a water has been properly softened there will be little if any scale since the small amount of lime and magnesia salts left in the water are of such a character that when precipitated as a result of concentration in the boiler only a slight sludge is formed. This sludge can be kept at a minimum by proper blowing off.

The lime-soda process does not eliminate all the scale-forming salts but removes a large part of them. The precipitates formed are them-

TABLE 102.

EFFECT OF SODA-LIME TREATMENT AND FILTRATION.

Niagara River — Buffalo, N. Y.

Raw.	Gr. per U. S. Gallon.	Treated.	Gr. per U. S. Gallon.
Volatile and organic matter	trace	Volatile and organic matter	trace
Silica	1.85	Silica	0.15
Oxides of iron and alumina	trace	Oxides of iron and alumina	trace
Calcium carbonate	2.20	Calcium carbonate	1.25
Calcium sulphate	2.11	Magnesium hydrate	0.25
Magnesium carbonate	0.48	Sodium sulphate	2.21
Magnesium chloride	0.05	Sodium chloride	0.80
Magnesium nitrate	1.16	Sodium nitrate	1.31
Sodium chloride	0.76		
Total solids	8.61	Total solids	5.97
Suspended matter	0.10		
Free carbonic acid	1.43		
Incrusting substances	7.85	Incrusting substances	1.65

Cost of treatment, 0.8 cent per 1000 gallons.

selves partly soluble in water and it is therefore impossible to reduce the hardness below, say, 4 grains per gallon.

Table 102 shows the influence of soda-lime treatment in a specific case.

Causticity, as used in water treatment, is a term to indicate the presence of an excess of lime added during treatment. *Alkalinity* is a general term used for the presence of compounds having the power to neutralize acids. For an excellent discussion of this subject consult "Water for Steam Boilers – Its Significance and Treatment" by Scott & Bailey, Jour. A.S.M.E., Nov., 1916, p. 867.

Caustic Soda and Boiler Corrosion: Prac. Engr., Feb. 15, 1916, p. 211.

267. Water-softening and Purifying Plants. — The term "water-softening" is ordinarily applied to systems in which the temporary and permanent hardness of the water are eliminated or reduced to a minimum, whereas the term "purifying" refers to systems in which some particular impurity or impurities are neutralized or completely removed. In boiler practice these terms are used synonymously and are applied to all systems of water treatment outside the boiler. Water-softening plants include two types of cold processes, the *intermittent* and the *continuous;* and the *hot process.* The cold-process plant is used chiefly in softening waters for locomotives and in large plants where water is used in considerable quantities. The hot process is commonly used in plants where exhaust steam is available for heating the water.

A typical continuous system is illustrated in Fig. 334. The hard water enters the softener through the inlet pipe, is discharged into the raw water box, whence it passes over the water wheel, and thus generates the power necessary to maintain the reagents in constant agitation. From the water wheel the hard water passes into the top of the cone, where it meets the reagents delivered by the lift pipe and is thoroughly mixed with them. The reagents are dissolved in the mixing tank, located at the ground level, and by means of a steam, electric, or power pump are then elevated into the chemical tank above. One charge is sufficient to last ten hours or more. The reagents are apportioned to the amount of incoming raw water to the dividing box. (Inasmuch as the "head" over this stream varies directly with any fluctuation of the main hard water stream, the two streams are constantly maintained in the same proportion to each other.) In the dividing box this small stream is again divided by a slide which throws one part of the water back into the hard water stream and another part — which determines the rate of flow of the chemicals — into the regulating tank. As the level of water in the regulating tank rises, the float rises likewise and

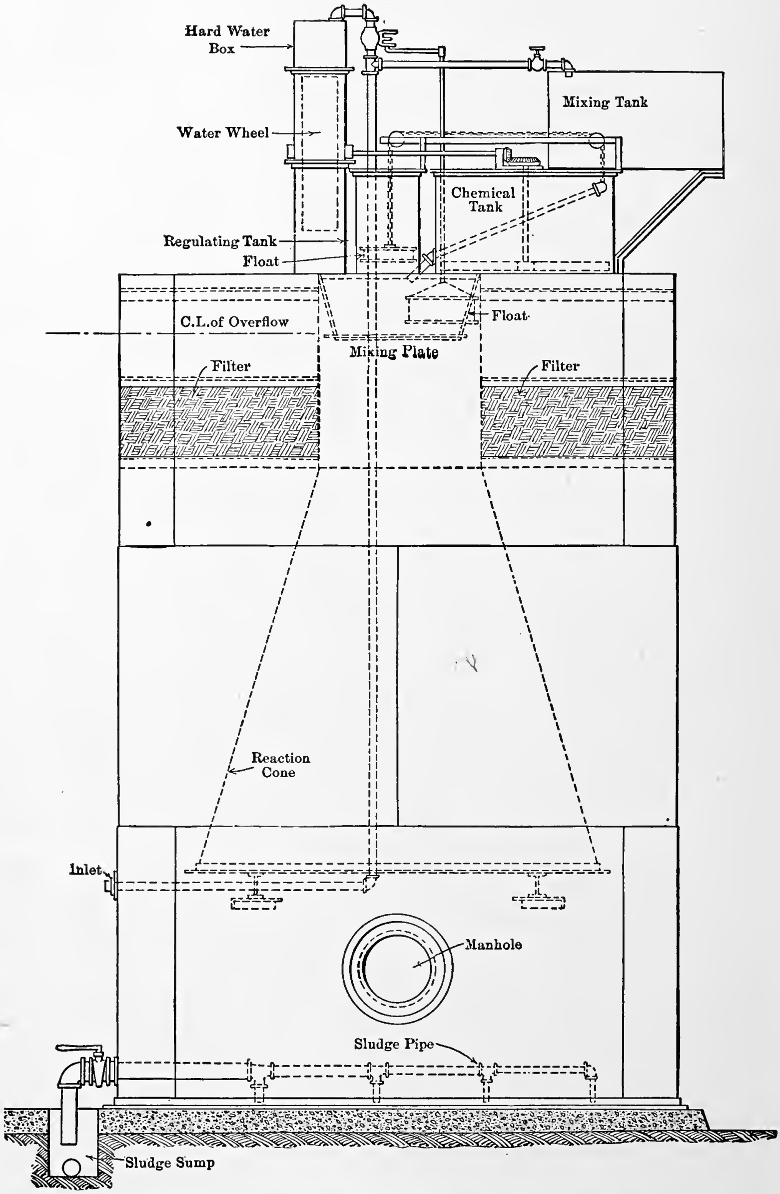


FIG. 334. Kennicott Type K Feed-water Purifier.

by means of a connecting chain lowers the mouth of the lift pipe in the reagent tank. Through this lift pipe the reagents flow into the top of the cone and intimately mix with the raw water. The reaction between the raw water and the reagents starts as soon as they meet, and as the mixture flows from the mixing plate into the reaction cone or downtake, the precipitation of the scale-forming and soap-destroying material commences to take place. Flowing at a constantly decreasing rate, owing to the constantly increasing diameter of the channel, the water passes to the bottom of the cone, turns and flows upward still at a constantly decreasing rate, the precipitate falling away from it as it moves. Finally the water passes through a filter which removes any slight trace of precipitate that remains; and it then is discharged from the top of the softener. The precipitate, which consists of the impurities of the raw water and the softening chemicals in chemical union, falls to the bottom of the main tank and is from time to time discharged therefrom through a sludge valve. An electric indicator is provided which rings a bell half an hour before a new supply of reagents is needed and thus notifies the attendant of the fact. The lift pipe is a tube, flexible for a portion of its length, through which the chemicals leave the chemical tank. By means of the regulating device the mouth of this tube is maintained at a constant depth of immersion in the surface of the dissolved reagents.

In the Scaife system for water purification feed water first enters the heater, where it attains a temperature of from 200 to 210 deg. fahr. As a portion of the free CO_2 is driven off by the heat the carbonates of lime and magnesia are precipitated and are deposited in removable pans

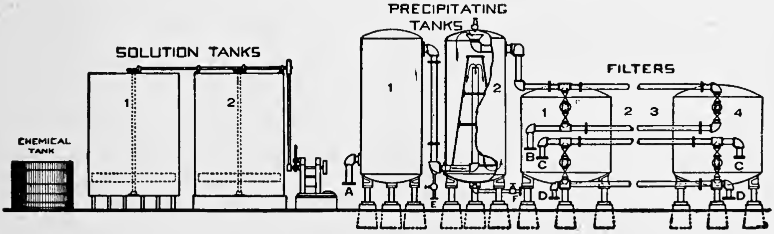


FIG. 335. General Arrangement of Scaife System of Feed-water Purification.

inside the heater. On its way the heated water is forced by the boiler feed pump into a large precipitating tank, where the necessary chemicals are introduced by two small pumps. These pumps take the solution of chemicals from the solution tanks which hold a sufficient quantity to operate the plant from eight to twelve hours. The precipitating tank is so constructed as to cause intimate and thorough mixing of the chemicals with the water. Thus the acids are neutralized, and the scale-forming substances are precipitated by being changed to insoluble substances which sink to the bottom of the precipitating tank, whence they are readily removed. Some of the lighter substances remaining in suspension are carried along with the water as it passes into the filters, which effectively remove all suspended matter. This system is continuous in operation, and purification is accomplished without appreciably retarding the onward flow of feed water. Fig. 335 shows

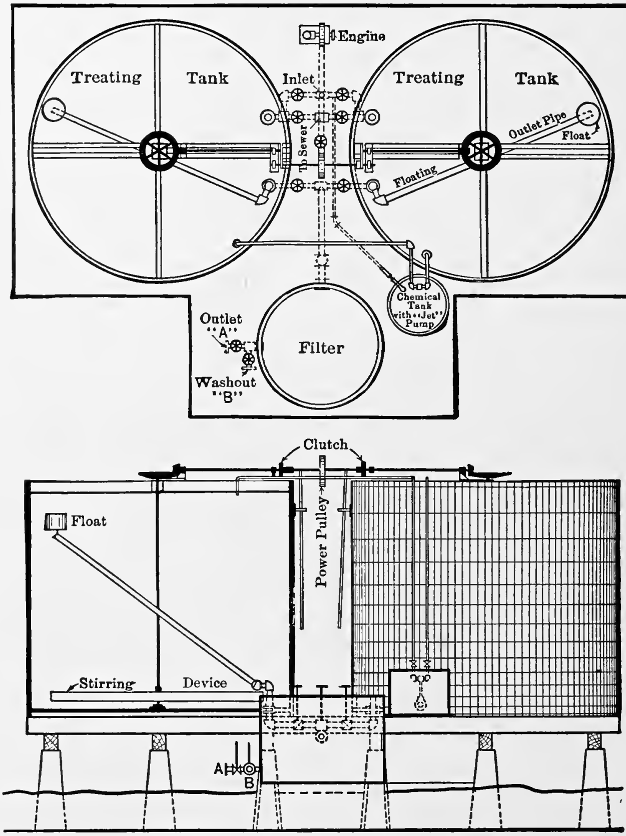


FIG. 336. General Arrangement of We-Fu-Go System of Feed-water Purification.

a modification of the system. The chemicals are pumped from the "chemical tank" into the "solution tanks," where the feed water and chemical solution are thoroughly mixed. The treated water is taken from these tanks and pumped into the "precipitating tanks" where a large portion of the scale-forming element is precipitated. From the precipitating tanks the water is forced through a series of filters to the boiler.

Fig. 336 illustrates the We-Fu-Go system of water purification. In this installation the water supply first enters the settling or treating tanks into which the chemicals are fed. A thorough mixture is effected by the use of the two armed paddles located near the bottom of the tanks. From the treating tanks the water flows by gravity into the filters, which remove all remaining impure solid matter which does not settle to the bottom of the treating tank. The pipes conducting the water from the settling tanks to the filter are fitted with a flexible joint and float so that the outlets are near the surface at all times, rising and falling with the water level. From the filters the purified water gravitates into the clear water storage reservoir, from which it is pumped into an open heater and thence to the boiler. This system is intermittent in operation, and in order to provide sufficient time for thorough

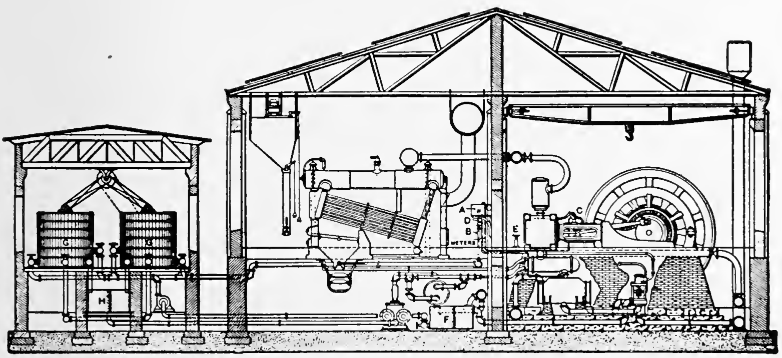


Fig. 337. Anderson System for Preventing Corrosion in Condensers.

chemical treatment of large quantities, two or more settling tanks are employed. Both the We-Fu-Go and Scaife systems are modified in a number of ways to meet different conditions.

Fig. 337 shows the general arrangement of the Anderson system for preventing corrosion in condensers and removing oil from condensed steam. The method consists in injecting into the exhaust steam as it passes from the preheater to the condenser a solution containing a coagulant which changes the emulsion of the cylinder oil to a flaky condition so that it may be separated by settling, flotation, or filtering. The air pump delivers the water to the settling tank *F*, whence it is taken to the open gravity filters *G*, *G*, of a superficial area proportional to the amount of water to be passed and containing a filter bed of four feet of crushed quartz. This will run about four days without any marked difference in efficiency, after which time the bed is stirred to a depth of two feet by mechanical agitators and flushed with clean water, by which all impurities are carried to the sewer. The solution is prepared in tank A, in which the water level is preserved by a ball float and into which filtered water is admitted through pipe B, while the substance with which the water is treated is pumped in through the pipe D by a small pump operated from the main engine. The flow to the "rose head" above the condenser is controlled by the valve E, and a meter in this pipe records the amount being fed. The water ordinarily required for "make up" is sufficient to carry in the solution. There is very little loss of water, and the rapid corrosion of the condenser tubes, which has been so great an obstacle to the successful use of surface condensers, is much reduced. The chemicals used perform a twofold duty, viz., to neutralize the water and make it chemically inactive and to coagulate the oily matter contained in the steam so that mechanical filtration is possible. (Power, June, 1903, p. 304.)

Fig. 338 shows a side elevation and a sectional end elevation of a Permutit water-softening plant. Referring to the sectional end elevation it will be seen that the raw water is delivered to the top of a closed tank and is caused to percolate successively through a layer of crushed marble, Permutit and gravel. This filtration effects the necessary purification. Permutit is the trade name of artificially manufactured hydrous silicates produced from clay, feldspar, soda ash and pearl ash. Permutit has the property of automatically eliminating all hardness from the water passing through it. It is a process of exchange, the calcium and magnesium in the water being replaced by the sodium in the Permutit. The softening continues until the sodium is used up. After the latter is exhausted the Permutit may be restored to its original efficiency by soaking it in common brine. The calcium and magnesium are thrown off and the sodium from the brine takes their place. Permutit is insoluble and a large excess of the reagent may be used without producing causticity, since it automatically gives up only enough soda to effect the required softening. See Power, Feb. 8, 1916, p. 198 and "Chemistry of Permutit," pamphlet published by the Permutit Company, N. Y.

Water-softening plants cost from \$4 to \$5 per horsepower for plants of 1000 horsepower and less, from \$3 to \$4 for plants of 1000 to 2000 horsepower, and as low as \$1.50 for plants of 5000 horsepower or more. The depreciation of wooden tanks is as high as 15 per cent a year, while that of steel tanks should not be greater than 5 per cent. Unless wooden tanks are considerably cheaper than steel tanks they are not a good investment. The cost of water purification varies from a fraction of a cent to 2 cents per 1000 gallons, depending upon the size of the plant and the quantity and character of the impurities. (American Electrician, March, 1905, p. 125.)

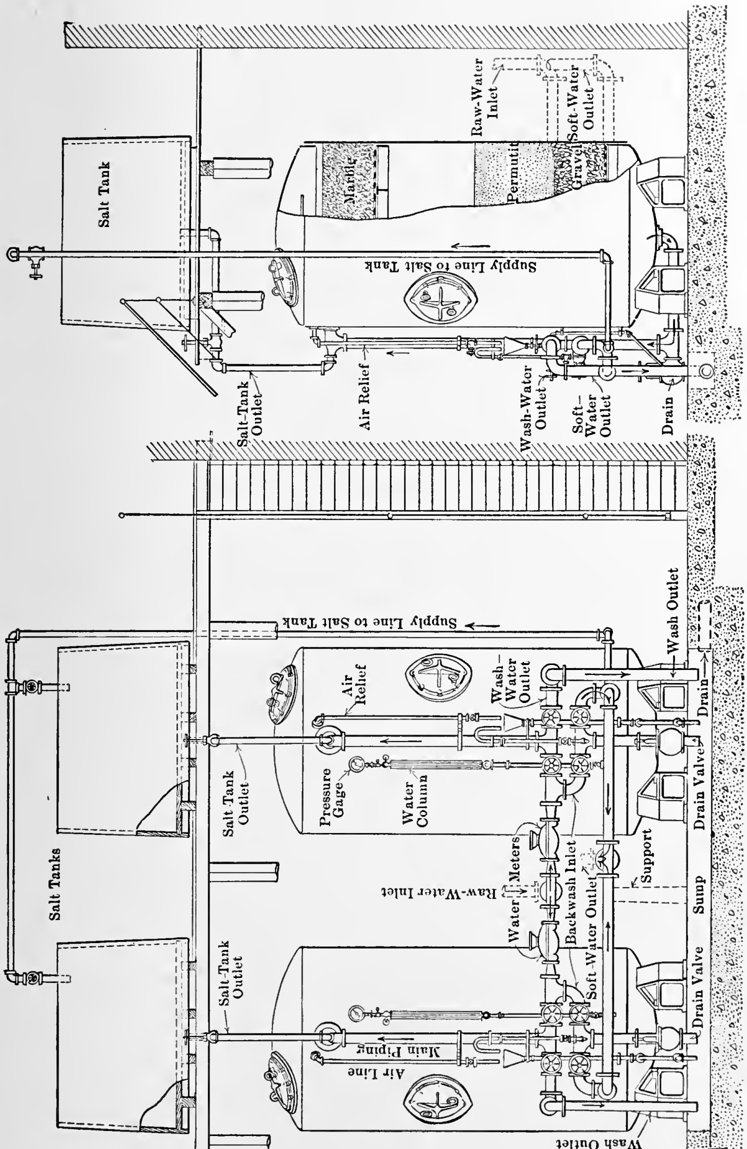


FIG. 338. Side Elevation and End-sectional Elevation of a Permutit Water-softening Plant.

Water Softening and Treatment for Power Plant Purposes: Chem. Engr., Jan., 1910, p. 5; Eng. News, June 6, 1912, p. 1087; Ry. Age Gazette, Aug. 16, 1912, p. 288; Ry. Master Mechanic, May, 1910, p. 153; Power, May 28, 1912, p. 780; Apr. 18, 1911, p. 598; Prac. Engr., U. S., Mar., 1910; see (Serial) 1915.

268. Economy of Preheating Feed Water. — Although a feed-water heater acts to some extent as a purifier its primary function is that of heating the feed water. Generally speaking, for every 10 degrees that the feed water is heated there is a gain in heat of 1 per cent and a corresponding saving of coal, if the heat which warms the feed water would otherwise be wasted. Again, the smaller the difference in temperature between the steam and the feed water the less will be the strain on the boiler shell due to unequal expansion and contraction, an item of no small consequence.

If H represents the heat content of the steam above 32 deg. fahr., t_0 the temperature of the cold water, and t the temperature of the water leaving the heater, then S, the per cent gain in heat due to heating the feed water, may be expressed

$$S = 100 \frac{(t - t_0)}{H - (t_0 - 32)}. \tag{236}$$

The expression is not theoretically correct, since it assumes a constant value of unity for the specific heat, whereas the specific heat varies with the temperature. The variation is so slight, however, that it may be neglected for all practical purposes.

Example 51. Steam pressure 100 pounds gauge; temperature of water entering heater 80 deg. fahr.; temperature of water leaving heater 210 deg. fahr. Required, saving due to heating the feed water.

Here H (from steam tables) is 1188, $t_0 = 80$, $t = 210$.

$$S = 100 \frac{(210 - 80)}{1188 - (80 - 32)}$$
$$= 11.4 \text{ per cent.}$$

This equation gives the *thermal* saving only, and the first cost of the heater, interest, depreciation, attendance, and repairs must be taken into consideration before the *net* saving measured in dollars and cents is ascertained. In the average installation the net saving is a substantial one.

Table 103 based upon equation (236) may be used in determining the percentages of saving due to the increase in feed-water temperature.

Feed-water Heating. — Power, June 25, 1912; Eng. News, Sept. 9, 1909, p. 284; Elec. Wld., March 2, 1911, p. 551; Mech. Engr., Nov. 5, 1909, p. 588; Engr. U. S., Jan. 1, 1906, p. 8, Aug. 15, 1904, p. 15; St. Ry. Jour., July 22, 1905, p. 145.

TABLE 103.

PERCENTAGE OF SAVING FOR EACH DEGREE OF INCREASE IN TEMPERATURE OF FEED WATER.

(Based on Marks & Davis Steam Tables.)

Initial Temp. of Feed.	Boiler Pressure Above Atmosphere.										
	0	20	40	60	80	100	120	140	160	180	200
32	.0869	.0857	.0851	.0846	.0843	.0841	.0839	.0837	.0835	.0834	.0834
40	.0875	.0863	.0856	.0853	.0849	.0846	.0845	.0843	.0841	.0840	.0839
50	.0883	.0871	.0864	.0859	.0856	.0853	.0852	.0850	.0848	.0847	.0846
60	.0891	.0878	.0871	.0867	.0864	.0861	.0859	.0857	.0855	.0854	.0853
70	.0899	.0886	.0879	.0874	.0871	.0868	.0867	.0865	.0863	.0862	.0861
80	.0907	.0894	.0887	.0882	.0878	.0876	.0874	.0872	.0871	.0870	.0869
90	.0915	.0902	.0895	.0890	.0887	.0884	.0882	.0880	.0878	.0877	.0876
100	.0924	.0910	.0903	.0898	.0895	.0892	.0890	.0888	.0886	.0885	.0884
110	.0932	.0919	.0911	.0906	.0903	.0900	.0898	.0896	.0894	.0893	.0892
120	.0941	.0927	.0919	.0915	.0911	.0908	.0906	.0904	.0902	.0901	.0900
130	.0950	.0936	.0928	.0923	.0919	.0916	.0915	.0912	.0911	.0910	.0909
140	.0959	.0945	.0937	.0931	.0928	.0925	.0923	.0921	.0919	.0918	.0917
150	.0969	.0954	.0946	.0940	.0987	.0933	.0931	0930	.0928	.0927	.0926
160	.0978	.0963	.0955	.0948	.0946	.0942	.0940	.0938	.0936	.0935	.0934
170	.0988	.0972	.0964	.0958	.0955	.0951	.0948	.0947	.0945	.0944	.0943
180	.0998	.0982	.0973	.0968	.0964	.0960	.0958	.0956	.0954	.0953	.0952
190	.1008	.0992	.0983	.0977	.0973	.0969	.0968	.0965	.0964	.0963	.0962
200	.1018	.1002	.0993	.0987	.0983	.0978	.0977	.0974	.0973	.0972	.0971
210	.1029	.1012	.1003	.0997	.0993	.0989	.0987	.0984	.0983	.0982	.0981
220	......	.1022	.1013	.1007	.1003	.0999	.0997	.0994	.0992	.0991	.0990
230	......	.1032	.1023	.1017	.1013	.1009	.1007	.1004	.1002	.1001	.1000
240	......	.1043	.1034	.1027	.1023	.1019	.1017	.1014	.1012	.1011	.1010
250	......	.1054	.1044	.1008	.1034	.1029	.1027	.1024	.1022	.1021	.1020

Multiply the factor in the table corresponding to any given initial temperature of feed water and boiler pressure by the total rise in feed-water temperature; the product will be the percentage of saving.

269. Classification of Feed-water Heaters. — Feed-water heaters may be classified according to the *source* of heat, as

1. *Exhaust steam,* in which the heat is received from the exhaust of engines, pumps, etc.

2. *Flue gas,* in which the waste chimney gases are the source of the heat.

3. *Live steam purifiers,* or those using steam at boiler pressures; or according to the method of heat *transmission,* as

1. *Open heaters,* in which the steam and feed water mingle and the steam in condensing gives up its heat directly to the water.

2. *Closed heaters,* in which the steam and water are in separate chambers and the steam gives up its heat to the water by conduction.

Heaters may also be classified according to the pressure of the heating steam, as

1. *Vacuum* or *primary,* in which the pressure is less than atmospheric and applies particularly to heaters utilizing the exhaust of condensing engines. These are always of the closed type. Open heaters in which the pressure is less than atmospheric are not usually classed as vacuum

heaters. 2. *Atmospheric* or *secondary,* in which the pressure is atmospheric or, literally, that corresponding to the back pressure on the engines and pumps.

3. *Pressure,* in which the pressure corresponds to that in the boiler and in which the heat is used primarily for purifying purposes.

CLASSIFICATION OF A FEW TYPICAL HEATERS.

Exhaust steam	Open	Atmospheric	Cochrane Hoppes Stillwell Webster	
	Closed	Atmospheric Vacuum or pressure	Wainwright Wheeler	Water Tube
			Otis Berryman	Steam Tube
Flue Gas			Green American Sturtevant	
Live Steam	Open	Pressure	Hoppes Baragwanath	

Heaters may be still further classified as

1. *Induced,* in which only such steam is admitted as is induced by its condensation. That is, the feed water condenses the steam. This creates a partial vacuum which draws in more steam.

2. *Through,* in which all the steam is forced through the heater irrespective of condensation.

270. Open Heaters. — Fig. 339 gives a sectional view of a Cochrane special feed heater and receiver and is a typical example of an open heater. Exhaust steam enters the heater through a fluted oil separator as indicated, and passes out at the top, while the oily drips are automatically drained to waste by a suitable ventilated float. The feed water enters through an automatic valve and is distributed over a series of copper trays so arranged and constructed that the water is forced to fall in a finely divided stream before reaching the reservoir in the bottom. The steam coming in contact with the water particles gives up latent heat and condenses. Some of the scale-forming element is deposited on the surface of the trays, from which it may be removed. The suspended matter is eliminated by a coke filter in the bottom of the chamber, and the floating impurities are decanted by a skimmer or overflow weir. The particular heater shown in the illustration is especially designed for use in a steam-heating plant; i.e., besides performing all the functions of an open heater, it provides for the reception and heating of the condensation returned to it from the heating system.

Fig. 340 shows a section through a Hoppes open heater, illustrating the "pan" type. Exhaust steam enters at H, passes through oil filter O, and completely surround pans T, T. The feed water enters at B,

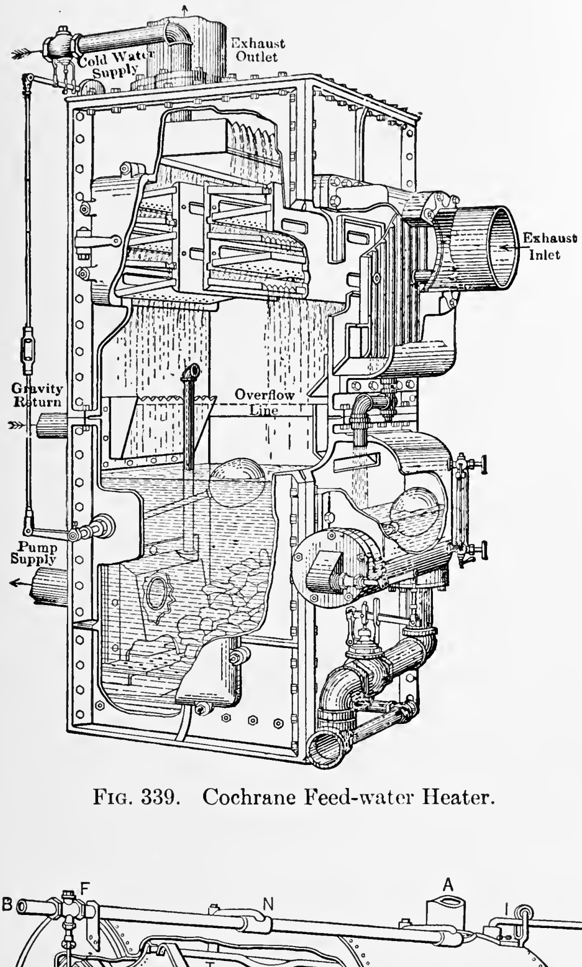


FIG. 339. Cochrane Feed-water Heater.

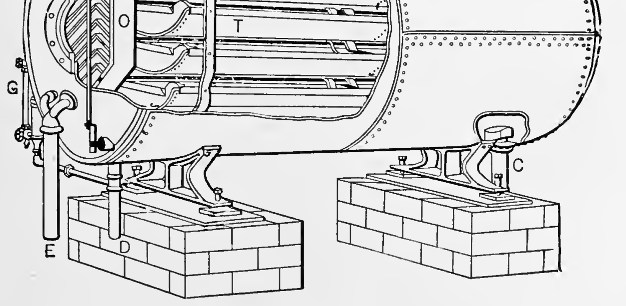


FIG. 340. Hoppes Horizontal Feed-water Heater.

and the rate of flow is regulated by valve F, which is controlled by a suitable float in the lower part of the chamber. The water in flowing over the sides and bottoms of the pans comes in direct contact with the steam.

271. Combined Open Heater and Chemical Purifier. — Combined feed-water heaters and chemical purifiers are finding increased favor with some engineers in many districts where the feed water is particularly bad and when space limitations preclude the use of water-softening plants. Although better than the plain open heater the purification is not thorough because of the short time that the water is in the heater.

272. Temperatures in Open Heaters. — The temperature to which feed water is raised in an open heater may be determined as follows:

Let H represent the heat content of the steam entering the heater,
t_0 the temperature of the water entering heater,
t the temperature of the water leaving heater, and
S the ratio of exhaust steam to the feed water, by weight.

Then, allowing a loss of 10 per cent due to radiation, etc., $0.9\,S\,(H - t + 32)$ will be the B.t.u. given up by the exhaust steam to each pound of feed water, and $(t - t_0)$ will be the B.t.u. absorbed by each pound of water.

Therefore $0.9\,S\,(H - t + 32) = t - t_0$, from which

$$t = \frac{t_0 + 0.9\,S\,(H + 32)}{1 + 0.9\,S}. \qquad (237)$$

TABLE 104.

FINAL FEED-WATER TEMPERATURES. OPEN HEATER.

(Temperature of steam, 212 degrees F.)

Per Cent of Total Steam Used by Auxiliaries.	Initial Temperature of Feed Water, Degrees F.									
	40	50	60	70	80	90	100	110	120	130
2	60.1	69.9	79.7	89.5	94.4	109.2	119.0	128.8	138.7	148.5
3	69.9	79.6	89.3	90.1	108.8	118.6	128.3	138.0	147.8	157.5
4	79.5	89.1	98.8	108.5	118.1	127.8	137.4	147.1	156.7	166.4
5	89.0	98.5	108.1	117.7	127.2	136.8	146.4	155.9	165.5	175.1
6	98.3	107.7	117.2	126.7	136.2	145.7	155.2	164.7	174.2	183.6
7	107.4	116.8	126.2	135.6	145.0	154.4	163.8	173.2	182.5	192.1
8	116.4	125.7	135.0	144.4	153.7	163.0	172.4	181.8	191.0	200.3
9	125.2	134.5	143.7	153.0	162.2	171.5	180.7	190.0	199.2	208.5
10	133.3	143.1	152.3	161.4	170.6	179.8	189.0	198.1	207.3	212.0
11	142.5	151.6	160.7	169.7	178.9	188.2	197.0	206.2	212.0*	212.0*
12	150.9	159.9	168.9	177.9	187.0	196.0	205.0	212.0*	212.0*	212.0*

* All of the steam not condensed.

If more steam passes through the heater than can be condensed by the feed water, then this equation gives t a fictitious value; in other words, t can never be greater than the temperature of the exhaust steam.

Substituting $t = 212$, the maximum obtainable temperature with exhaust steam at atmospheric pressure, and solving for S, we find that only 17 per cent of the main engine exhaust is necessary to heat the feed water to a maximum. t_0 is assumed to be 60 deg. fahr.

Table 104 has been determined from this equation and gives the final temperatures obtainable in open heaters for various conditions of operation.

Example 52. A power plant has 1200 i.hp. of engines using 20 pounds of steam per i.hp-hour. Auxiliaries use 2400 lb. steam per hr. Pressure in heater 0 pounds gauge, temperature of hot-well supply 110 deg. fahr. Required temperature of feed water leaving heater.

Here $H = 1150$ (from steam tables), $t_0 = 110$, $S = 0.10$.

Substituting these values in (237),

$$0.9 \times 0.10\,(1150 - t + 32) = t - 110.$$
$$t = 198 \text{ deg. fahr.}$$

273. Pan Surface Required in Open Feed-water Heaters. — Pan or tray surface required varies according to the quality of the water with regard to both scale-making material and mud, and may be approximated by the formula

$$\text{Pan surface, sq. ft.} = \frac{\text{Pounds of water heated per hour}}{c}. \tag{238}$$

	Vertical Type.	Horizontal Type.
For very muddy water, c	118	110
Slightly muddy water, c	166	155
For clean water, c	500	400

274. Size of Shell, Open Heaters. — General proportions of open heaters vary considerably on account of the different arrangements of pans or trays, filter and oil-extracting devices. A fair idea of the size of shell required may be obtained by the formulas

$$\text{Area of shell} = \frac{\text{Horsepower}}{a \times \text{length in feet}}, \tag{239}$$

$$\text{Length of shell} = \frac{\text{Horsepower}}{a \times \text{area in square feet}}, \tag{240}$$

$a = 2.15$ for very muddy water,
$a = 6$ for slightly muddy water,
$a = 8$ for clean water.

The horsepower in this case is obtained by dividing the weight of water heated per hour by the steam consumption of the engine per horsepower per hour.

Pans containing 2.5 square feet and less are usually made round, and larger sizes rectangular in plan. When circumstances will permit it is better to have not more than six pans in any one tier, since it is advisable to proportion the pans so as to obtain as low a velocity over each as practicable.

Distance between trays or pans is seldom less than one-tenth the width for rectangular and one-fourth the diameter for round pans. Volume of storage and settling chamber in horizontal heaters varies from 0.25 for good quality of water to 0.4 of the volume of the shell for muddy water, 0.33 being about the average. In the vertical type the settling chamber represents respectively 0.4 and 0.6 the volume of the shell with clear and muddy water. Filters occupy from 10 to 15 per cent of the volume of the shell in the horizontal type and from 15 to 20 per cent in the vertical type, the smaller percentage corresponding to clear water and the larger to muddy water or water containing a considerable quantity of impurities.

Open Heaters: Cassier's Mag., Aug., 1903, p. 33; Engr. U. S., Jan. 1, 1906, pp. 17, 78; St. Ry. Jour., Feb. 4, 1905, p. 227; Elec. Wld., Apr. 27, 1911, p. 1051.

275. Types of Closed Heaters. — Closed heaters may be grouped into two classes:

1. Water tube, Fig. 341, and
2. Steam tube, Fig. 345.

Closed heaters, both water tube and steam tube may operate with

1. *Parallel currents,* where the water and steam flow in the same direction, Fig. 344, or with

2. *Counter currents,* where the water and steam flow in opposite directions, Fig. 343.

Water-tube heaters may be still further classified as

1. *Single-flow,* in which the water flows through the heaters in one direction only, Fig. 341.

2. *Multi-flow,* in which the water flows back and forth a number of times, as in Fig. 343.

3. *Coil heater,* in which the water flows through one or more coils, as in Fig. 344.

4. *Film,* in which the water is forced across the heating surface in a thin sheet or film.

276. Water-tube Closed Heaters. — Fig. 341 shows a section through a feed-water heater of the single-flow straight-tube type. The tubes are of plain brass and the shell of cast iron. The tubes are expanded into the tube sheets by a roller expander. To provide for expansion the upper tube sheet and water chamber are secured to the main shell by means of a special expansion joint the details of which are shown in Fig. 342. *R* is a ring or gasket of soft annealed copper and *G*, *G* two gaskets of special packing with brass wire cloth insertion. These gaskets form a flexible expansion joint between *C* and tube sheet *D*, so that the whole upper chamber, which is carried solely by the tubes, is free to move up and down as the tubes expand or contract under varying temperatures.

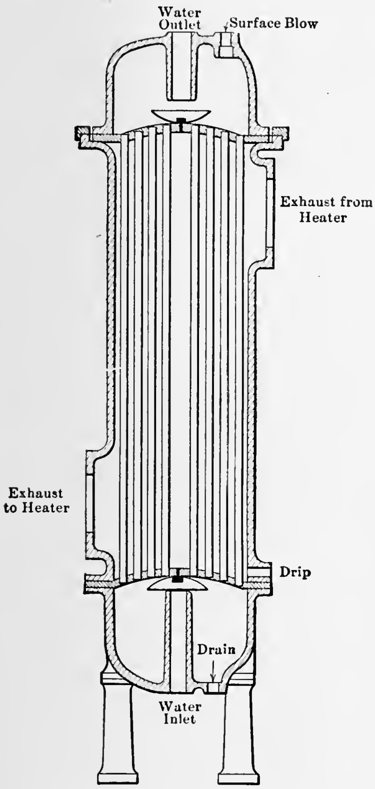


FIG. 341. Goubert Single-flow Closed Heater.

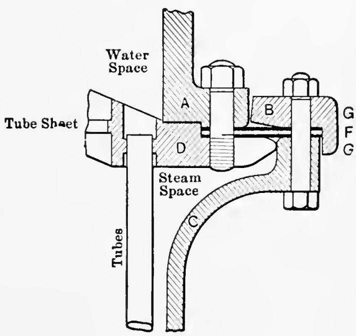


FIG. 342. Details of Expansion Joint, Goubert Heater.

Fig. 343 shows a section through a Wainwright heater, illustrating the multi-flow water-tube type. The body of the heater is of cast iron, the tubes of corrugated copper. The water passes through the tubes and the steam surrounds them. The feed water and exhaust steam do not mingle, and hence the oil in the exhaust does not contaminate the water. The water chambers are divided into several compartments, as shown in the illustration, and the partitions are so arranged that the flow of feed water is directed back and forth through the various groups of tubes in succession. This arrangement gives a higher velocity of flow than the non-return type of heater, and therefore increases the rate of heat absorption. The mud and impurities settle at the bottom and are discharged through the mud blow-off. Such impurities as rise to the surface are removed by the surface blow-off. The tubes are corrugated to allow for expansion and at the same time to increase the transmission of heat. Referring to Fig. 343: Exhaust steam enters at *A* and leaves at *E*, and the portion which is condensed

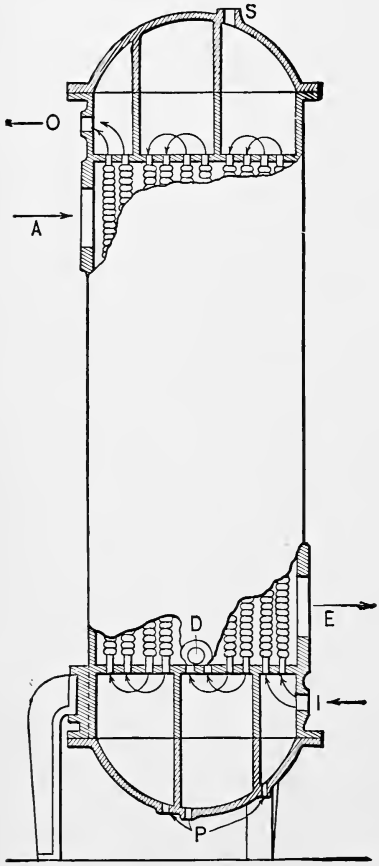


Fig. 343. Wainwright Multiflow Closed Heater.

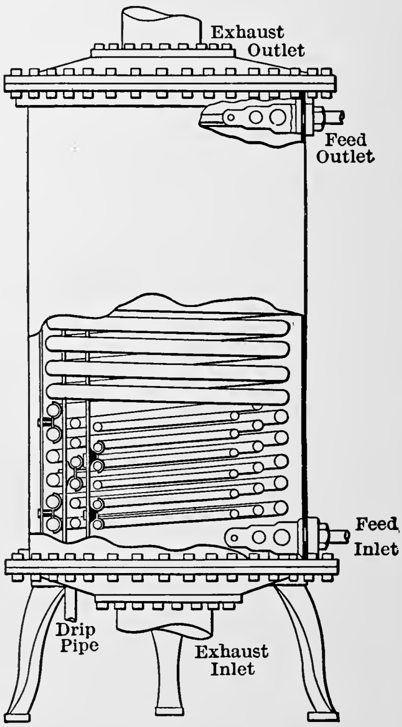


Fig. 344. Typical Coil Heater.

is drawn off at *D*. Feed water enters at *I* and is discharged at *O*. *P*, *P* are mud blow-offs and *S* is an opening for a safety valve. Fig. 356 gives results of tests showing the relative efficiencies of plain and corrugated tubes for various velocities.

Fig. 344 shows a partial section through a Harrisburg feed-water heater. This apparatus is a typical example of the coiled-tube heater. Three sets of concentric copper coils are brazed to gun-metal manifolds and supported by clamp stays as indicated in the illustration. Feed water enters the heater at the bottom manifold and passes through the coils to the feed outlet. The exhaust steam enters the heater at the bottom and surrounds the coils in its passage to the outlet at the top. The coils are designed to withstand a pressure of 600 pounds per square inch.

277. Steam-tube Closed Heaters. — Fig. 345 shows a section through an Otis heater, illustrating the steam-tube type. Here the exhaust

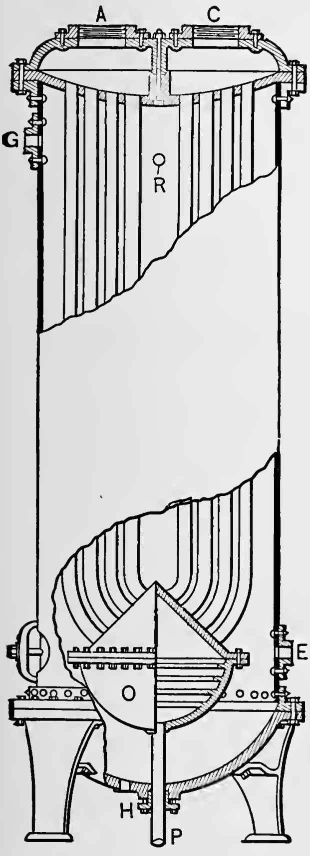


Fig. 345. Otis Steam-tube Feed-water Heater.

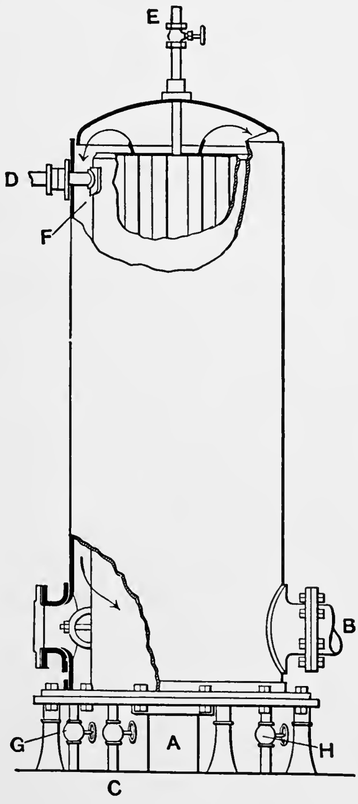


Fig. 346. Baragwanath Steam-jacketed Feed-water Heater.

steam passes *through* the tubes which are surrounded by the feed water. The exhaust steam enters at A, and passes down one section of tubes into the enlarged space of the water and oil separator O, in which the condensation and oil are deposited. From this chamber the steam passes up through the other section of tubes to outlet C, thus passing twice through the entire length of the heater. The water enters at E and is discharged at G. R is the blow-off opening. The tubes are of seamless brass and are curved to allow for expansion. Condensed steam is withdrawn at P.

Fig. 346 shows a partial section through a Baragwanath steam-jacketed steam-tube heater. Exhaust steam enters at A, passes up through the tubes, returns down annular space E between the inner shell and jacket, and passes out at B. Feed water enters at C and leaves at D. E is the scum blow-off, G the heater drain, and H the jacket drain.

278. Film Heaters. — The heating element in a film heater consists usually of two spirally corrugated tubes, one within the other, the water path being the small annular clearances between the two. Thus the water is directed in a spiral path due to the corrugations, and for a given velocity the particles of water come more often in contact with the heating surface than in plain tubes because they are contained within an annular space whose perimeter is large in comparison with its area. This type of heater though highly efficient in heat transmission necessitates the use of comparatively pure water and is not commonly used for feed water heating.

279. Heat Transmission in Closed Heaters. — Since the closed heater is practically the same in principle as a surface condenser the laws of heat transmission are practically identical in both cases. The temperature of the steam and water are higher in the atmospheric heater but otherwise the heat exchange is the same in all heaters and condensers of the water-tube type. Increasing the velocity of the water passing through the heater increases the rate of heat transmission and thereby renders the heating surface more effective. In order to employ moderately high velocities and at the same time allow sufficient time in which to raise the temperature to a maximum, the tubes should be as long as practicable and of small diameters. Other things being equal, a heater containing a large number of tubes of small diameter is more efficient than one containing a small number of large tubes. It is important to proportion the heater according to the amount of water to be heated and the maximum temperature to which the water must be raised. In designing a heater, then, the maximum temperature to which the water is to be raised and the coefficient of heat transfer are assumed and the amount of heating surface is calculated from equations 241 or 242.

Although recent experiment* shows that the amount of heat transmitted through the heating surface is proportional to some power of the mean temperature difference the value of the exponent is not far from unity (0.8 to 0.9) and it may be safely taken as such, particularly

* Jour. A.S.M.E., Aug., 1915, p. 433.

in view of the liberal factor allowed in the assumed value of the coefficient of heat transfer, U. With this assumption the extent of heating surface may be calculated from the following adaptation of equation (210)

$$S = \frac{cw\,(t_2 - t_0)}{Ud}. \qquad (241)$$

in which

S = total tube heating surface, sq. ft.,
c = mean specific heat of water; this may be taken as 1.0,
w = weight of water heated per hr.,
t_2 = final temperature of the feed water, deg. fahr.,
t_0 = initial temperature of the feed water, deg. fahr.,
U = mean coefficient of heat transfer for the entire surface, B.t.u. per sq. ft. per deg. difference in temperature per hour.,
d = mean temperature difference between the steam and that of the water.

$$d = \frac{t_2 - t_0}{\log_e \frac{t_s - t_0}{t_s - t_2}}. \quad \text{(See Equation (219))}$$

Substituting this value of d in equation (241) (taking $c = 1$) and reducing we have

$$S = \frac{W}{U} \log_e \frac{t_s - t_0}{t_s - t_2}. \qquad (242)$$

For a given extent of heating surface S, the temperature difference between that of the steam and the feed water leaving the heater may be calculated by solving equation (242) for $t_s - t_2$, thus

$$\frac{t_s - t_0}{t_s - t_2} = e^n. \qquad (243)$$

in which e = base of the Naperian logarithm = 2.718

$$n = \frac{SU}{w}.$$

By taking different extents of area S and solving for the corresponding values of $t_s - t_2$ the temperature gradient for a given heater may be obtained as illustrated in Fig. 347.

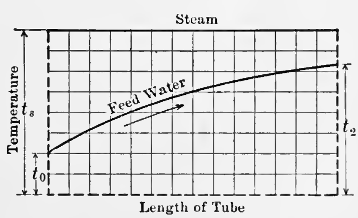


FIG. 347. Temperature Gradient in Feed-water Heater Tube.

From equation (241) it will be seen that extent of heating surface depends upon the weight of water to be heated, the temperature of the steam, the desired temperature of the feed-water heater and the value of U.

Since the extent of heating surfaces increases rapidly as t_2 approaches t_s, and becomes infinity for $t_2 = t_s$, it is desirable to limit t_2 to some practical figure. An average maximum for $t_2 = t_s - 4$.

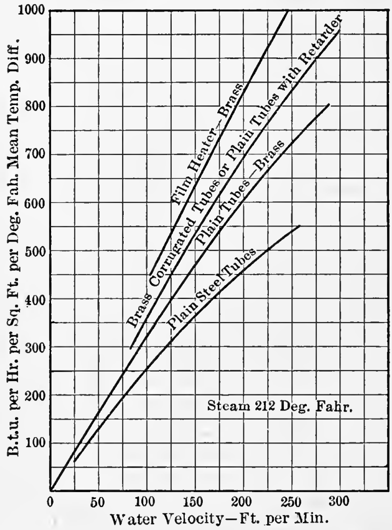


FIG. 348. Coefficient of Heat Transfer. (For General Design.)

The coefficient of heat transfer varies within wide limits depending upon type of heater and the conditions of operation, and ranges from $U = 150$ in steel tube heaters with low water velocities to 1000 or more in the film type of corrugated brass tube heaters with water velocity of 7 ft. per second. In practice a liberal factor is allowed for possible heat reduction due to the presence of air and the accumulation of oil scale or other deposit on the tube surfaces.

For steam coils submerged in water and from which the condensation is withdrawn as rapidly as it is formed the value of U in Table 105a appears to give satisfactory results.

Example 53. Determine the length of $\frac{3}{4}$ in. (O. D.), $\frac{1}{16}$ in. thick brass tubes in a closed heater designed to heat water from 60 to 196 deg. fahr., steam temperature 212 deg. fahr., water velocity 2 ft. per sec., $U = 400$.

$$S = \frac{\pi d l}{12} = \frac{3.14}{12} dl = 0.197\, l.$$

l = length of tube, ft.

$$w = \frac{2 \times 3600 \times \pi d^2 \delta}{144 \times 4} = \frac{7200 \times 3.14 \times (\frac{5}{8})^2 \times 62.4}{144 \times 4} = 957 \text{ lb. per hr.}$$

Substituting these values in equation (241),

$$0.197\, l = \frac{957}{400} \log_e \frac{212 - 60}{212 - 196}.$$

From which $l = 27.3$ ft. approx.

Example 54. A 200-sq. ft. closed heater is rated at 40,000 lb. of water per hour, initial temperature, 60 deg. fahr., temperature steam 212 deg. fahr., $U = 300$. Required the final temperature of the water.

From equation (243),

$$\frac{t_s - t_0}{t_s - t_2} = e^n.$$

$$e = 2.718$$

$$n = \frac{SU}{w} = \frac{200 \times 300}{40{,}000} = 1.5,$$

$$\text{whence} \qquad \frac{212 - 60}{212 - t_2} = 2.718^{1.5},$$

$$\text{or} \qquad t_2 = 172.4 \text{ deg. fahr.}$$

TABLE 105.

HEAT TRANSMISSION IN CLOSED FEED-WATER HEATERS.

(Based on Commercial Designs.)

Type of Heater.	Coefficient of Heat Transfer, U.	
	Range.	Average.
Single-flow plain brass tubes	150– 300	200
Single-flow, corrugated brass tubes	250– 400	300
Single-flow, steel tubes	125– 175	150
*Spiral coils, plain brass tubes	250– 500	350
Multi-flow plain brass tubes	250– 500	350
Multi-flow, corrugated brass tubes	350– 700	400
Plain brass tubes with retarders	350– 900	450
Film heater with corrugated tubes	500–1100	600

* For small coils and high water velocities these values may be increased 100 per cent.

TABLE 105a.

HEAT TRANSFER—SUBMERGED STEAM COILS.

Mean Temperature Difference.	Coefficient of Heat Transfer, U.		
	Iron.	Brass.	Copper.
50	100	200	220
100	175	275	300
150	200	375	400
200	225	450	475

Example 55. Determine the size of vacuum and atmospheric heaters for a condensing plant of 1200 i.hp. Engines use 20 pounds of steam per i.hp-hr.; auxiliaries use the equivalent of 10 per cent of the main engine steam; vacuum 25 incnes referred to 30-inch barometer; feed water, $t_0 = 50$ degrees; temperature of hot well, $t_2 = 110$ degrees; coefficient of heat transmission, $U = 300$ B.t.u.

Vacuum or Primary Heater.

Feed water for main engines,

$$20 \times 1200 = 24{,}000 \text{ pounds per hour.}$$

Feed water used by auxiliaries,

$$10 \text{ per cent of } 24{,}000 = 2400 \text{ pounds per hour.}$$

Total feed,

$$W = 24{,}000 + 2400 = 26{,}400 \text{ pounds per hour.}$$

TABLE 106.

DEGREES OF DIFFERENCE BETWEEN STEAM TEMPERATURE AND ACTUAL AVERAGE TEMPERATURE OF FEED WATER.

Vacuum Heaters between Engine and Condenser.

Initial Temperature of Water, t_0.	24″ Vacuum. Temperature 141° F. Final Temperature of Water.						25″ Vacuum. Temp. 134° F. Final Temperature of Water.			
	105	110	115	120	125	130	110	115	120	125
40	62.9	59.3	55.3	50.9	46.2	40.6	50.1	45.7	40.6	34.6
50	59.2	55.6	51.9	47.7	43.1	37.8	46.7	42.4	37.7	32.1
60	55.5	52.1	48.4	44.4	40.1	35.1	43.2	39.3	34.7	29.4
70	51.6	48.2	44.8	41.1	36.9	32.1	39.7	35.9	31.2	26.6
80	47.6	44.2	41.0	37.5	33.5	29.2	35.9	32.4	28.4	23.8

Initial Temperature of Water, t_0.	26″ Vac. Temp. 125° F. Final Temp. of Water.				27″ Vac. Temp. 114°. Final Temp. of Water.			28″ Vac. Temp. 100°. Final Temp. of Water.		
	105	110	115	120	90	100	105	70	80	90
40	42.0	40.4	35.0	28.3	44.4	36.0	30.8	43.2	36.3	27.9
50	39.9	37.3	32.3	25.9	40.8	32.8	28.0	39.1	32.7	24.8
60	38.2	34.1	29.4	23.4	36.9	29.6	25.1	34.7	28.8	21.6
70	34.3	30.8	26.9	20.7	33.0	26.2	22.1	......	24.7	18.1
80	30.8	27.3	23.3	18.1	28.7	22.5	18.7	......	......	14.4

Atmospheric Heaters.

Atmospheric Pressure. Temp. 212° F. Final Temperature of Water.						Initial Temperature of Water, t_0.
192	196	200	204	208	210	
70.6	65.7	60.1	53.5	44.8	38.2	40
67.9	63.1	57.6	51.2	42.8	36.4	50
65.1	60.4	55.2	48.9	40.7	34.7	60
62.2	57.7	52.6	46.6	38.7	32.9	70
59.4	54.9	50.0	44.2	36.6	31.0	80
56.4	52.3	47.4	41.8	34.4	29.1	90
53.1	49.3	44.7	39.3	32.4	27.3	100
51.9	47.9	43.4	38.2	31.4	26.4	105
50.3	46.4	42.1	36.9	30.2	25.5	110
48.8	45.0	40.6	35.7	29.2	24.5	115
47.2	43.5	39.2	34.4	28.0	23.5	120
45.6	41.9	37.8	33.1	26.9	22.5	125
43.9	40.3	36.4	31.7	25.8	21.5	130

TABLE 107.

SQUARE FEET OF HEATING SURFACE REQUIRED TO HEAT 1000 POUNDS OF WATER PER HOUR.
$U = 350.$

Vacuum Heaters between Engine and Condenser.

Initial Temperature of Feed Water, t_0.	24″ Vacuum. Temperature 141° F.						25″ Vacuum. $t_s = 134°$ F.			
	Final Temperature of Feed Water.									
	105	110	115	120	125	130	110	115	120	125
40	2.93	3.36	3.36	4.50	5.22	6.29	3.93	4.65	5.58	7.01
50	2.64	3.29	3.57	4.15	4.93	6.01	3.65	4.36	5.28	6.65
60	2.29	2.93	3.22	3.86	4.58	5.65	3.29	4.01	4.93	6.29
70	1.93	2.50	2.86	3.43	4.22	5.29	3.07	3.58	4.57	5.86
80	1.50	2.07	2.43	3.01	3.72	4.86	2.36	3.07	4.01	5.36

t_0.	26″ Vac. $t_s = 125°$.				27″ Vac. $t_s = 114°$.			28″ Vac. $t_s = 100°$.		
	Final Temperature of the Feed Water.									
	105	110	115	120	90	100	105	70	80	90
40	4.43	4.93	6.07	8.18	3.22	4.72	6.01	1.93	3.14	5.08
50	3.92	4.57	5.72	7.73	2.79	4.36	5.58	1.43	2.57	4.57
60	3.36	4.15	5.36	7.51	2.29	3.86	5.08	.78	1.93	3.93
70	2.86	3.65	4.79	6.86	1.71	3.22	4.50	......	1.14	3.14
80	2.29	3.07	4.28	6.28	.86	2.21	3.79	......	......	1.93

tmospheric Heaters.

Atmospheric Pressure. Temp. 212° F.						Initial Temperature of Feed Water, t_0.
Final Temperature of Feed Water.						
192	196	200	204	208	210	
6.01	6.65	7.58	8.73	10.72	12.74	40
5.94	6.58	7.44	8.58	10.51	12.51	50
5.79	6.44	7.15	8.36	10.38	12.30	60
5.58	6.22	7.01	8.23	10.15	12.15	70
5.37	6.01	6.87	8.00	9.94	11.85	80
5.15	5.79	6.57	7.80	9.80	11.79	90
4 93	5.51	6.36	7.51	9.52	11.50	100
4.79	5.46	6.22	7.37	9.37	11.37	105
4.65	5.28	6.15	7.22	9.23	11.15	110
4.50	5.15	5.93	7.08	9.08	11.07	115
4.36	4.93	5.79	6.93	8.95	10.94	120
4.15	4.79	5.64	6.79	8.80	10.80	125
3.86	4.65	5.44	6.57	8.65	10.58	130

From equation (242),

$$S = \frac{W}{U} \log_e \frac{t_s - t_0}{t_s - t_2}$$
$$= \frac{26{,}400}{300} \log_e \frac{134 - 50}{134 - 110}$$
$$= 110 \text{ square feet.}$$

On the basis of $\frac{1}{3}$ square foot of surface per horsepower the rating of this heater will be

$$110 \times 3 = 330 \text{ horsepower.}$$

Atmospheric or Secondary Heater.

The temperature of the feed water leaving the atmospheric heater, equation (237), will be

$$t = \frac{t_0 + 0.9\, S\,(H + 32)}{1 + 0.9\, S},$$

where $S = 0.10, \quad t_0 = 110 \text{ degrees}, \quad H = 1150 \text{ B.t.u.},$

whence
$$t = \frac{110 + 0.9 \times 0.10\,(1150 + 32)}{1 + 0.9 \times 0.10}$$
$$= 198 \text{ degrees.}$$

The required surface is

$$A = \frac{W}{U} \log_e \frac{t_s - t_0}{t_s - t_2},$$

where $t_s = 212, \quad t_0 = 110, \quad t_2 = 198,$

whence
$$A = \frac{26{,}400}{300} \log_e \frac{212 - 110}{212 - 198}$$
$$= 175 \text{ square feet.}$$

The horsepower rating will be

$$175 \times 3 = 525.$$

280. Open vs. Closed Heaters. — Open and closed heaters have their respective advantages and a careful study of the various influencing conditions is necessary for an intelligent choice. The following parallel comparison brings out a few of the distinguishing features:

OPEN HEATER.	CLOSED HEATER.
Efficiency.	
With sufficient exhaust steam for heating, the feed water may reach the same temperature as the steam.	The maximum temperature of the feed water will always be 2 degrees or more lower than the temperature of the steam.
Scale and oil do not affect the heat transmission.	Scale and oil deposit on the tubes and the heat transmission is lowered.
Pressures.	
It is not ordinarily subjected to much more than atmospheric pressure.	The water pressure is slightly greater than that in the boiler when placed on the pressure side of the pump as is customary.

Safety.

Sticking of the back pressure valve may cause it to "blow up" if provision is not made for such an emergency.	It will safely withstand any pressure likely to occur.

Purification.

Since the exhaust steam and feed water mingle, provision must be made for removing the oil from the steam.	Oil does not come in contact with the feed water.
Scale and other impurities precipitated in the heater are readily removed.	Scale is removed with difficulty.

Location.

Must always be placed above the pump suction and on the suction side.	May be placed anywhere on the pressure side of the pump.

Pumps.

With supply under suction two pumps are necessary and one must handle hot water.	One cold-water pump is necessary.

Adaptability.

Particularly adaptable for heating systems where it is desired to pipe the "returns" direct to heater.	All vacuum or primary heaters are necessarily of this type.

281. "Through" Heaters. — Fig. 349 shows a typical installation of a through heater in a non-condensing plant.

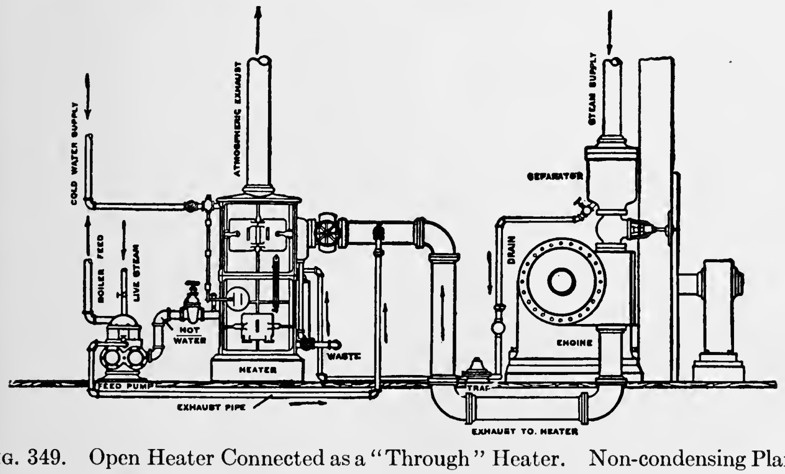


Fig. 349. Open Heater Connected as a "Through" Heater. Non-condensing Plant.

It is evident that *all* the steam must pass *through* the heater. Now, one pound of exhaust steam in condensing gives up approximately 1000 B.t.u. Hence, if the initial temperature of the feed water is 50 degrees and the final temperature 210, the engine furnishes $\frac{1000}{210 - 50}$

= 6.26, say, six times the quantity necessary for heating the feed water to a maximum. Therefore the area of the pipe supplying the heater with steam need be but one sixth that of the main exhaust. With the heater connected as in Fig. 349 the connections must necessarily be the same size as the exhaust pipe.

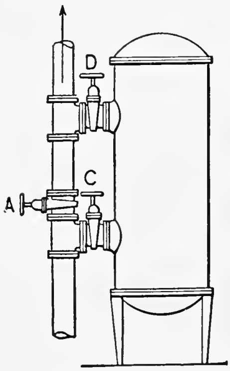


Fig. 350.

With this arrangement the heater cannot be "cut out" while the engine is in operation and hence it is not adapted for plants working continuously. For the purpose of cutting out a heater while the plant is in operation a through heater may be by-passed as in Fig. 350. Advantage may be taken here of the permissible reduction in the size of pipes and fittings, i.e., valves, etc., at *C* and *D* need be but one half the size of those at *A*. This reduction in size may prove to be a considerable item in large installations.

282. Induced Heaters. — Fig. 351 shows a typical installation of an induced heater in a non-condensing plant and Fig. 352 an induced primary heater in a condensing plant.

In the arrangement in Fig. 351 the number of fittings is reduced to a minimum and the heater may be readily cut out. Since induced heaters

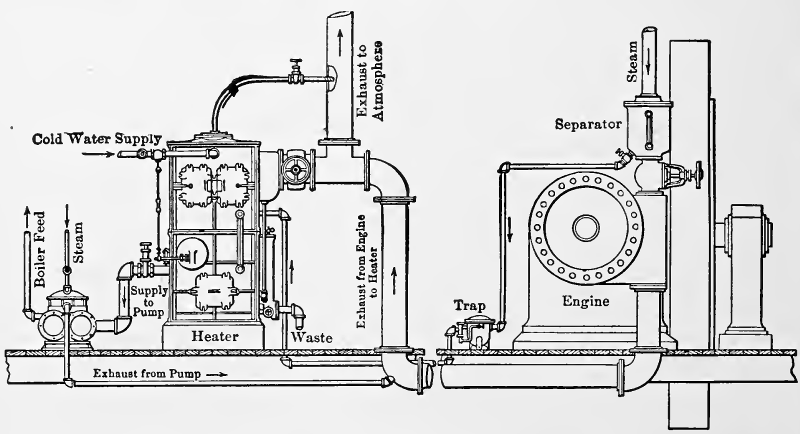


Fig. 351. Open Heater Connected as an "Induced" Heater. Non-condensing Plant.

are apt to become air-bound, a vapor pipe or vent is inserted in the top of the heater as shown. This pipe varies from $\frac{1}{2}$ to $1\frac{1}{2}$ inches in diameter, depending upon the size of heater.

Closed Heaters: Am. Elecn., May, 1900, p. 236, July, 1900, p. 354, Oct., 1905, p. 530; Cassier's Mag., Aug., 1903, p. 330; Eng. U. S., Jan. 1, 1906, p. 13; Power, April, 1902, p. 11.

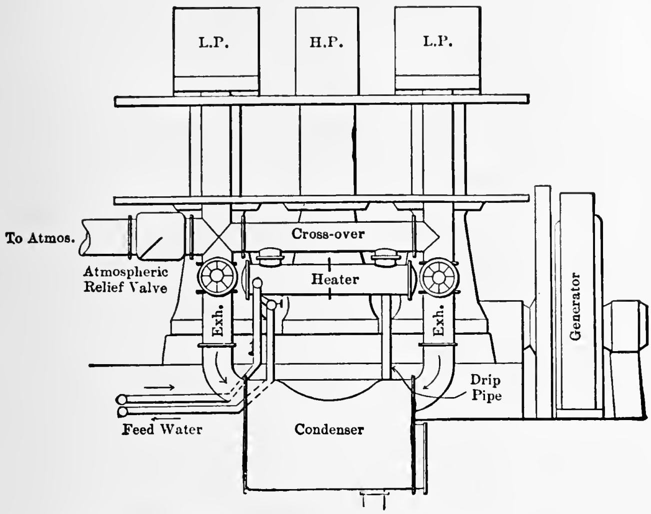


Fig. 352. Closed Heater Connected as an "Induced" Heater. Condensing Plant.

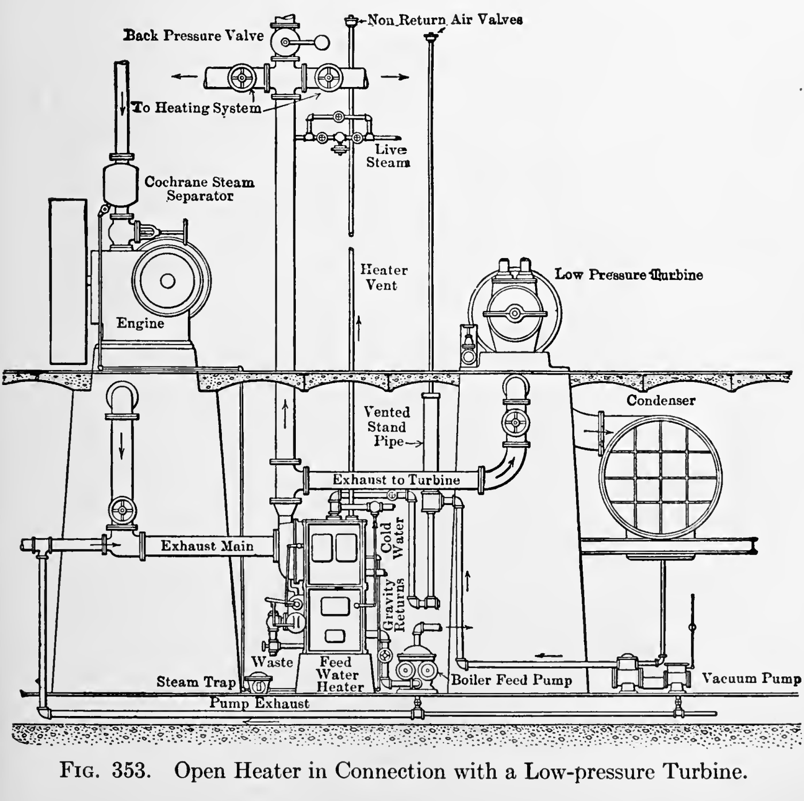


Fig. 353. Open Heater in Connection with a Low-pressure Turbine.

283. Live-steam Heaters and Purifiers. — The function of a live-steam heater and purifier is primarily that of purification and hence it is not ordinarily installed unless the feed water contains scale-forming elements such as sulphates of lime and magnesia. These, as previously

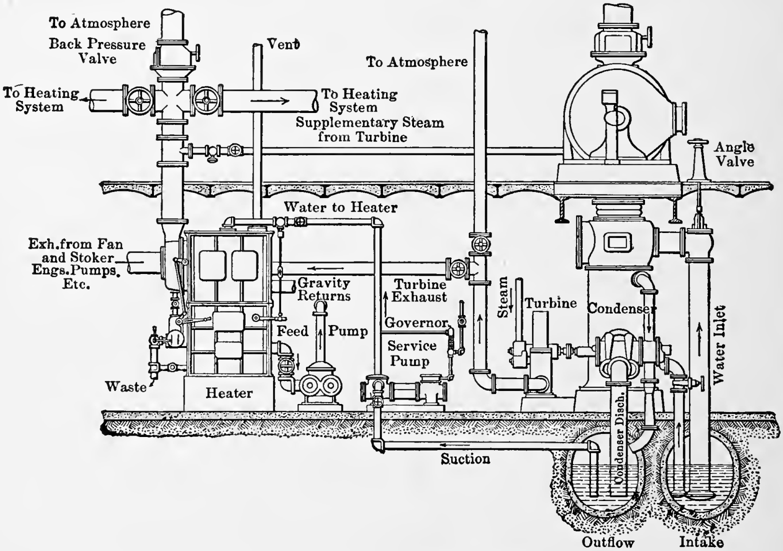


FIG. 354. Open Heater in Connection with a Jet Condenser.

stated, are not entirely precipitated until a temperature of approximately 300 deg. fahr. is reached; hence no amount of heating with exhaust steam atmospheric pressure will thoroughly purify feed water containing these elements.

Fig. 355 shows a section through a Hoppes live-steam purifier. Since the purifier is subjected to full boiler pressure, the shell and heads are constructed of steel. Within the shell are a number of trough-shaped pans or trays placed one above another and supported on steel angle ways. Steam from the boiler enters the chamber at *A* and comes in contact with feed water and condenses. The water on entering the heater at *B* is fed into the top pan and, overflowing the edges, follows the under side of the pan to the center and drops into the pan below. It flows over each successive pan in the same manner until it reaches the chamber at the bottom, whence it gravitates to the boiler through pipe *C*. As the steam inclosed in the shell comes in contact with the thin film of water, the solids held in solution are separated and adhere to the bottom of the pans in the same manner that stalactites form on the roofs of natural caves. Authentic tests show that live-steam heaters may increase the boiler efficiency. (See Power, Feb. 21, 1911, p. 295.)

The purifier should be set in such a position as will bring the bottom of the shell two feet or more above the water level of the boilers, as in Fig. 356. *N* is the feed pipe from pump to purifier and should be provided

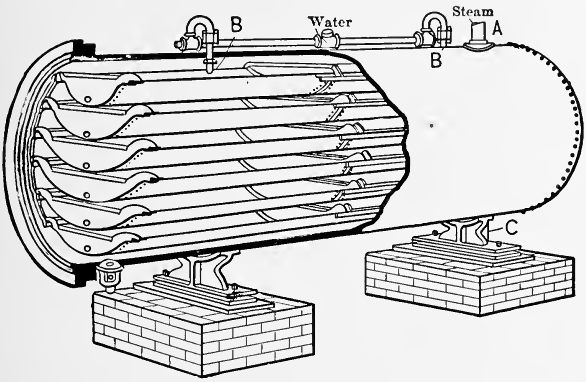


FIG. 355. Hoppes Live-steam Purifier.

with a check valve. *D* is the gravity pipe through which the purified water flows to the boiler. This pipe should be carried below the water level of the boilers and all branch pipes should be taken off below the

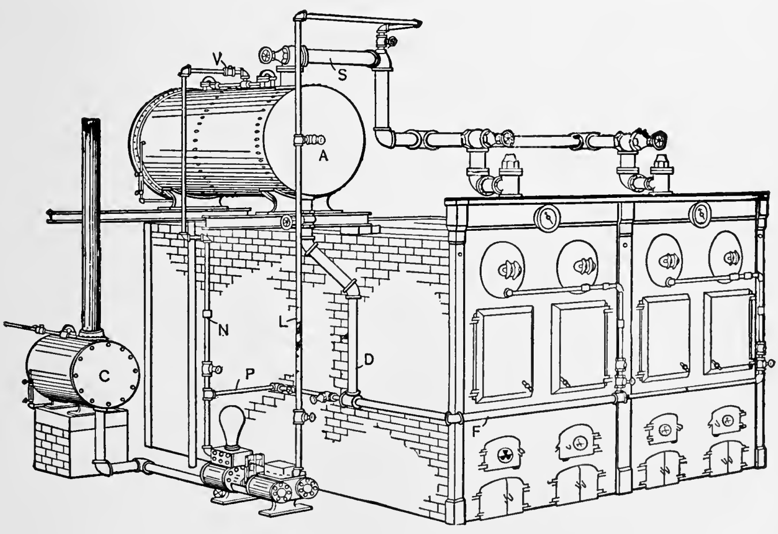


FIG. 356. Typical Installation of a "Live-steam" Purifier.

water line. Pipe *L* leads from top of pipe *S* to pump or other steam-using device. This is necessary in order that air and other non-condensable gases liberated from the water may be removed from the purifier, which would otherwise become air-bound. In the illustration the feed pump takes its supply from an exhaust steam heater *C*. The purifier is provided with a suitable by-pass so that the water may be fed directly to the boiler when necessary.

Live Steam Heated Feed Water: Elec. Engr., Lond., June 29, 1906; Cassier's Mag., Oct., 1911, p. 543; Elec. Rev., Lond., May 20, 1898, p. 667; Eng. Rec., Aug. 30, 1898, p. 467; Power, March 31, 1908, p. 498, Feb. 21, 1911, p. 295.

284. Distillation of Make-up Water. — In large central stations equipped with turbines and surface condensers the condensate furnishes a supply of distilled water for boiler feed purposes. To provide for leakage losses an additional supply of water must be had from some other source.

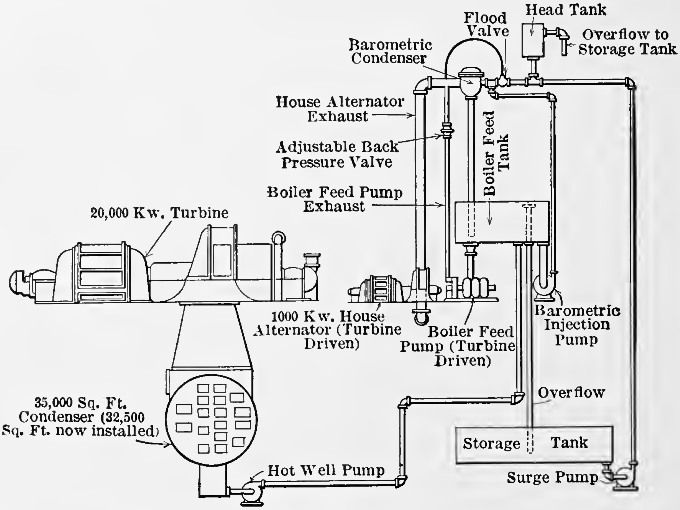


FIG. 357. Feed-water Heating System at the Connors Creek Station of the Detroit Edison Co.

In some situations raw make-up water is sufficiently pure to warrant its introduction into the system without treatment but in most cases it is too hard for direct use even though the quantity required is a relatively small percentage of the total weight of water fed to the boilers. In a number of recent installations all make-up water is distilled, thus insuring a continuous supply of pure water. Fig. 357 illustrates the principles of the feed-water system as installed in the Connors Creek Station of the Detroit Edison Company. The condensate from the main surface condensers is discharged into one end of a large tank shown as the boiler feed tank. A centrifugal pump draws its water from the same end of this tank and discharges it into the head of a barometric condenser. The relatively cold condensate is picked up by the second pump before it has time to mix with the mass of water in the tank and serves as injection water for the barometric condenser. The house-service alternator turbine and the boiler feed pump turbine exhaust into this barometric condenser so that the condensate from the main unit takes up all the heat of the auxiliary steam. The foot of the barometric condenser is immersed in the hot end of the boiler feed tank. The mixture is then picked up by the boiler feed pump and delivered to the boilers. The barometric condenser is therefore the equivalent of an open feed-water heater in which exhaust steam from auxiliaries mixes with and heats the condensate from the main units.

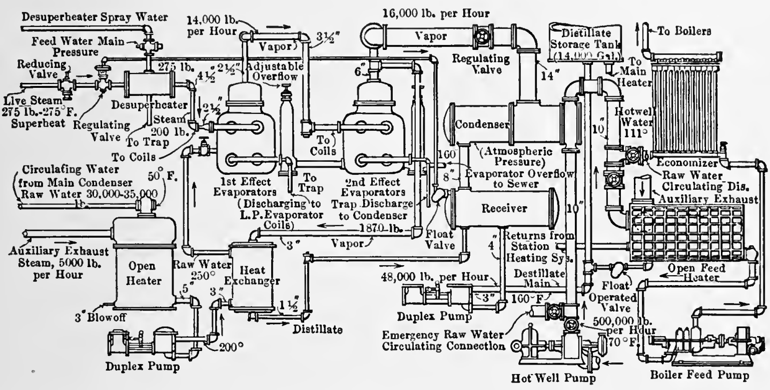


FIG. 358. Make-up Water Evaporator System, Buffalo General Electric Co.

The make-up water is boiled in an evaporator heated by high pressure steam and the resulting vapor passes directly to the barometric condenser in which it mixes with the auxiliary exhaust and thus becomes part of the feed water. For a full description of this interesting installation, consult "The Connors Creek Plant of the Detroit Edison Company," C. F. Hirshfeld, Trans. A.S.M.E., Vol. 37, 1915.

Fig. 358 gives a diagrammatic arrangement of the make-up water evaporator system of the River Station of the Buffalo General Electric Company, Black Rock, Buffalo, which is representative of the latest practice. Raw water is taken from the circulating water outlet of the main unit condensers and follows the course of the arrow heads from the open heater at left of the diagram, through the various appliances, to the economizer and thence to the boiler. Most of the impurities are precipitated in the evaporators from which they are discharged to waste. For complete details consult Power, Feb. 13, 1917, p. 202.

285. Fuel Economizer. — Although any device which effects a saving in fuel is a fuel economizer the term "fuel economizer" without qualification refers to a closed heater which receives its heat supply from the flue gases. Two types of economizers are found in practice, (1) those which are independent of the boiler and (2) those which are integral with the boiler and form a part of the heating surface. The independent type is the more common and is usually constructed of cast iron to obviate danger of corrosion. The integral type is usually constructed of wrought iron or steel tubes and is to all intents and purpose a part of the boiler proper. The present tendency toward higher boiler pressures makes the use of an economizer almost a necessity because of the otherwise high temperatures of the escaping flue gases; in fact, practically all modern large central stations are equipped with economizers.

Fig. 359 gives a general view of a Green economizer, illustrating a typical flue gas heater. It consists of a series of cast-iron tubes 9 to

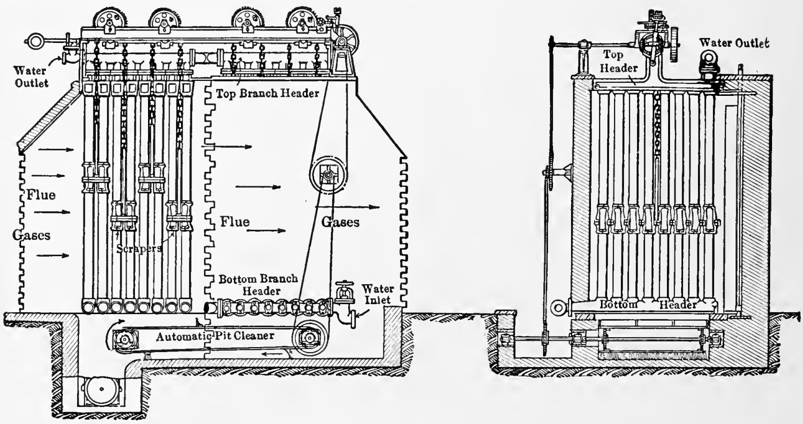


Fig. 359. Green Economizer.

10 feet in length and $4\frac{5}{8}$ inches in diameter, which are arranged vertically in sections of various widths across the main flue between boiler and chimney. When in position the sections are connected by top and bottom headers, and the headers are connected to branch pipes running lengthwise, one at the top and the other at the bottom. Both of the branch pipes are outside the brickwork which incloses the apparatus. The waste gases are led to the economizer by the ordinary flue from the boiler to the chimney, but a by-pass must be provided for use when the economizer is out of service for cleaning or for repairs. The feed water is forced into the economizer through the lower branch pipe nearest the point of exit of gases, and emerges through the upper branch pipe nearest the point where the gases enter. Each tube is encircled with a set of triple overlapping scrapers which travel continuously up and down the tubes at a slow rate of speed, the object being to keep the external

surfaces free from soot. The mechanism for working the scrapers is placed on top of the economizer, outside the chamber, and the motive power is supplied either by a belt from some convenient shaft or small independent engine or motor. The power for operating the gearing varies from 1 to ½ horsepower per 1000 square feet of economizer surface, depending upon the number and length of tubes. The apparatus is fitted with blow-off and safety valves, and a space is provided at the bottom of the chamber for the collection of soot. For continuous plant operation the soot is automatically cleaned as shown in the illustration.

This type of economizer is also used as an *air heater* for drying and heating purposes. The air heater is similar in design to the water heater with the exception of the direction of flow and size of tubes. The tubes in the air economizer are $3\frac{7}{8}$ inches internal diameter by 9 feet

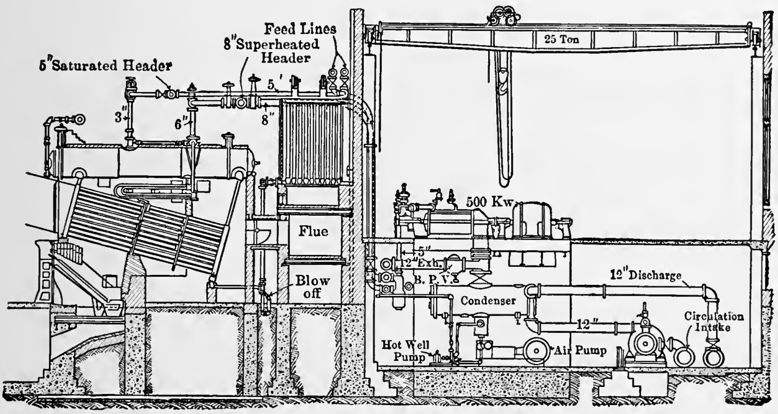


FIG. 360. Typical Economizer Installation.

in length, as against $4\frac{5}{8}$ inches internal diameter for the water economizer. In the latter the water enters at the bottom header and passes out from the top header; in the former the air is forced by a fan first through one set of tubes and up through another set, and then down again, and so on until it leaves the heater.

Fig. 361 shows a section through a 25,000-sq. ft. Badenhausen boiler as installed in the Highland Park plant of the Ford Motor Company and illustrates an economizer element integral with the boiler. Feed water enters drum 6, flows down the rear bank and enters the forward bank of tubes connecting drums 5 and 6. The economizer element is baffled so that the gases are forced to travel down the front bank and up the rear bank of tubes. The resulting difference in temperature creates a positive circulation of the water in the economizer element. The integral type of economizer is not commonly used in this country but a modification of this arrangement, which appears to be the tendency in large central stations, is the subdivision of the

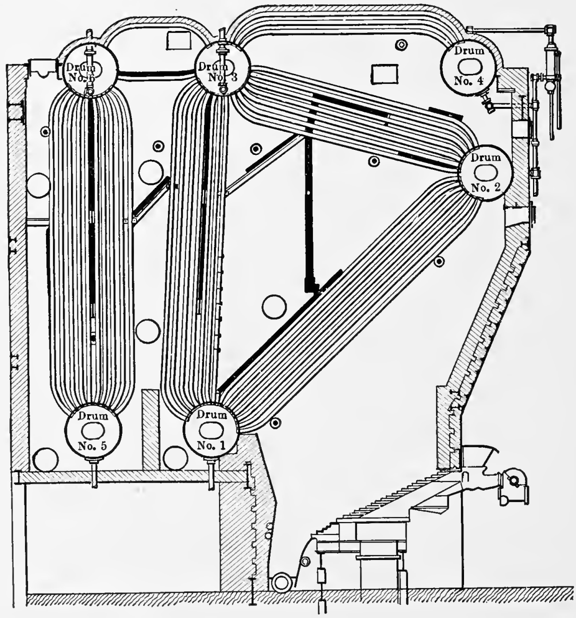


Fig. 361. 25,000-sq. ft. Badenhausen Boiler with Economizer Element Integral with Heating Surface.

heating surface so that each boiler has its own economizer. With large units the heating surface is often arranged in three or four sections. To maintain a constant velocity of the gases through the economizer passages each section is made narrower.

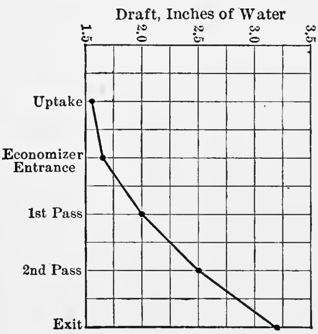


Fig. 362. Pressure Drop through 8500-sq. ft., 3-section Economizer — Fan Draft.

Economizers have been installed in connection with chimney draft but the extra height of stack necessary to compensate for the reduction in draft caused by the lower temperatures of the gases and by the resistance of the tubes usually offsets the gain. In order to obtain an overall economy it is necessary to have some form of mechanical draft to force the gases through the economizer at proper speed. The loss in draft due to the reduction in temperature of the flue gases may be calculated as shown in paragraph 127. The loss in draft due to the resistance of the tubes varies directly with the length of the economizer and as the square of the velocity. The pressure drop through an economizer 40 sections long with mean gas velocity of 1500 ft. per min. is approximately 0.25 in. water. This loss naturally varies with the design of the economizer. See curves in Fig. 362 for a specific example.

286. Temperature Rise in Economizers. — The heat transfer in an economizer follows the same basic law as the heat transmission through any heating surface, viz.:

$$SUd = w_1c_1\,(t - t_0), \tag{245}$$

$$= w_2c_2\,(t_2 - t_1), \tag{246}$$

in which

S = total heating surface, sq. ft.,
U = mean coefficient of heat transmission, B.t.u. per hr. per sq. ft. per deg. mean temperature difference,
d = mean temperature difference between the two fluids, deg. fahr.,
w_1 and w_2 = weights, respectively, of the fluid to be heated and the flue gas,
c_1 and c_2 = mean specific heats respectively of the fluid to be heated and the flue gas,
t_0 and t = initial and final temperature of the fluids to be heated, deg. fahr.,
t_2 and t_1 = initial and final temperature of the flue gas, deg. fahr.

By an analysis similar to that developed in paragraph 242 it may be shown that for either parallel or counter flow

$$d = \frac{t_i - t_f}{\log_e \dfrac{t_i}{t_f}}, \tag{247}$$

in which

t_i, t_f = initial and final temperature difference between the two fluids.

By combining equations (245) to (247) and reducing (see Sibley Journal, Jan., 1916, p. 129) we have as an expression for the temperature rise in the feed water

$$x = \frac{t_2 - t_0}{\dfrac{N-1}{10^n - 1} + N}, \tag{248}$$

in which

x = temperature rise in the feed water, deg. fahr.,

$$N = \frac{w_1c_1}{w_2c_2},$$

$$n = \frac{SU\,(N-1)}{2.3\,w_1}.$$

Other notations as previously designated.

TABLE 108.

AVERAGE MEAN COEFFICIENT OF HEAT TRANSFER IN ECONOMIZERS.

(Clean Cast-iron Tubes.)

B.t.u. Per Sq. Ft. Per Deg. Fahr. Difference in Temperature.

Velocity of the Gases, Ft. per Min.	Mean Temperature Difference Between Flue Gas and Feed Water, Deg. Fahr.				
	250	275	300	350	400
500	2.2	2.3	2.5	2.7	2.8
1000	3.0	3.2	3.3	3.4	3.6
1500	3.6	3.8	4.0	4.2	4.5
2000	4.0	4.3	4.5	4.7	5.0

Equation (248) applied strictly to counterflow which is the usual economizer practice. See Fig. (364).

Example 56. Calculate the final feed-water and flue-gas temperature for an economizer installation operating under the following conditions. Boiler heating surface 12,000 sq. ft.; economizer surface 7500 sq. ft.; initial feed-water temperature 100 deg. fahr. and initial flue-gas temperature 650 deg. fahr. when the boiler is operating at 100 per cent above standard rating; coal used, Illinois screenings, 11,400 B.t.u. per lb.

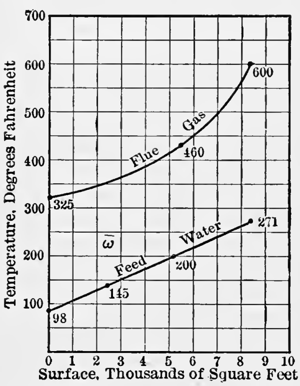


Fig. 363. Temperature of Flue Gas and Feed Water in an 8000-sq. ft. Economizer—Fan Draft.

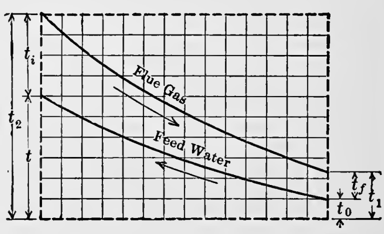


Fig. 364. Counter Current Flow.

It has been shown (paragraph 21) that the theoretical weight of air per lb. of any coal is approximately 7.5 lb. per 10,000 B.t.u. Therefore for the coal specified,

$1.14 \times 7.5 = 8.65$ lb. = theoretical air requirements per lb. of coal.

Assuming an air excess of 50 per cent at maximum load and allowing 15 per cent for ash the probably actual weight of flue gas per lb. of coal $= 1.5 \times 8.65 + 0.85 = 13.8$ lb., or in round numbers 14 lb.

Since the evaporation at rating is equivalent to 3.45 lb. from and at 212 deg. per sq. ft. heating surface per hr., at 100 per cent overload the total weight of water, w, fed to the boiler is

$$w = 2 \times 12{,}000 \times 3.45 = 82{,}800 \text{ lb. per hr.}$$

Assuming an overall efficiency of 75 per cent the weight of coal required is

$$\frac{970.4 \times 82{,}800}{11{,}400 \times 0.75} = 9400 \text{ lb. per hr.}$$

The total weight of flue gas, w_2, is

$$w_2 = 9400 \times 13 - 131{,}600 \text{ lb. per hr.}$$

Assume the mean specific heat of the water to be unity and that of the flue gas to be 0.25.

Assume $U = 4.25$, which is an average value for a modern economizer with initial flue gas temperature of 650 deg. fahr. Substituting these values in equation (248),

TABLE 109.

ECONOMIZER PROPORTIONS IN MODERN CENTRAL STATIONS.

Name of Plant.	Size of Boiler Unit Nominal Horse-power.	Boiler Heating Surface Per Unit, Sq. Ft.	Economizer Surface Per Boiler Unit, Sq. Ft.	Ratio Economizer to Boiler Surface.
Buffalo General Electric	1140	11,400	9435	0.825
*Cleveland Municipal Plant, 53rd Street Station	1013	10,134	5400	0.525
Commonwealth Edison Co., Fisk Street Station	1225	12,250	8500	0.692
Northwest No. 3	1220	12,200	6566	0.540
†Delray, No. 1	483	4,830	4896	0.490
Public Service, Joliet, Ill.	992	9,919	6730	0.679
Public Service, New Jersey, Essex Station	1373	13,730	7750	0.564
Regina, Sask., Can.	500	5,000	2320	0.464

* One economizer for 5 boilers. † One economizer for 2 boilers.

$$N = \frac{w_1c_1}{w_2c_2} = \frac{82{,}800 \times 1}{131{,}600 \times 0.25} = 2.52.$$

$$n = \frac{SU(N-1)}{2.3\,w_1} = \frac{7500 \times 4.25\,(2.52 - 1)}{2.3 \times 82{,}800} = 0.254.$$

$$x = \frac{t_2 - t_0}{\dfrac{N-1}{10^n - 1} + N} = \frac{650 - 100}{\dfrac{2.52 - 1}{10^{0.254} - 1} + 2.52} = 126 \text{ deg. fahr.}$$

Since $x = t - t_0$, the final temperature of the feed water is

$$t = 126 + 100 = 226 \text{ deg. fahr.}$$

The heat absorbed by the feed water must be equal to that given up by the flue gas, or

$$w_1c_1\,(t - t_0) = w_2c_2\,(t_2 - t), \tag{249}$$

from which

$$\frac{t_2 - t_1}{t - t_0} = \frac{w_1c_1}{w_2c_2} = N. \tag{250}$$

Substituting the known quantities in equation (250)

$$\frac{650 - t_1}{226 - 100} = 2.52,$$

or

$$t_1 = 337.0 \text{ deg. fahr.} = \text{final temperature of the flue gas.}$$

For parallel flow as in Fig. 365 the final flue gas temperature may be

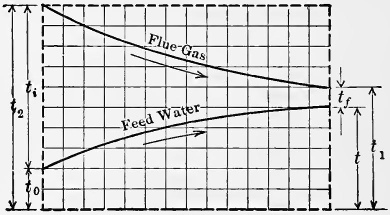


FIG. 365. Parallel Current Flow.

calculated from the following formula which has deauced from equations (245) to (247).

$$t_1 = \frac{t_2 - \frac{a}{b}}{e^m} + \frac{a}{b}, \tag{251}$$

in which

$$a = \frac{w_1c_1}{w_2c_2} t_2 + t_0,$$

$$b = \frac{w_1c_1}{w_2c_2} + 1.$$

$$m = \frac{(1 + B)\, SU}{w_2c_2}.$$

Other notations as previously designated.

287. Value of Economizers. — The general conclusions drawn from current practice is that an economizer installation results in:

(1) A saving in fuel ranging from 7 to 20 per cent.

(2) A very small gain and often an actual loss in overall economy when installed in connection with feeble chimney draft and underloaded boilers.

(3) A substantial overall gain in economy where the boilers are forced and mechanical draft is employed.

(4) Maximum overall economy when the boilers are forced far above their rating and the auxiliaries are electrically driven and pure feed water is available.

(5) Decreased wear and tear on the boilers due to the high feed-water temperature.

(6) A large storage of hot water for sudden peak demands.

TABLE 110.

ECONOMIZER PERFORMANCES.

Number of Plant.	Number of Economizer Tubes Installed.	Temperatures, Deg. Fah.					
		Gases Entering Economizer	Gases Leaving Economizer	Fluid Entering Economizer	Fluid Leaving Economizer	Rise in Temperature of Fluid.	Actual Saving in Fuel, Per Cent.
Water Heater.							
1	160	435	279	84.2	196.2	112.0	12.5
2	....	416	254	40.0	185.4	125.4	13.8
3	960	620	293	101.0	237.0	136.0	18.3
4	520	548	295	96.0	200.0	104.0	9.2
5	520	603	325	93.5	203.8	110.3	9.7
6	384	368	245	103.0	202.6	99.6	12.4
7	448	537	326	71.2	203.4	132.2	17.5
Air Heater.							
1	72	301	257	70.0	152.0	82.0	....
2	240	512	319	54.0	201.6	147.6	9.0
3	96	557	376	41.0	200.0	159.0	14.0
4	192	417	369	74.0	210.0	136.0	....

Compiled from "The Book of the Economizer," 1912, published by the Green Engineering Co.

288. Factors Determining Installation of Economizers. — Some of the more important factors to be considered before installing an economizer are:

(1) Temperature of the flue gas. The higher the temperature of the flue gas the greater will be the thermal saving. With the standard type of boiler operating with high pressures, 300 lb. per sq. in. or more, economizers are practically indispensable. See Table 36 for flue gas temperatures incident to boiler overloads.

(2) Initial temperature of the feed water. With electrically driven auxiliaries exhaust steam is not available for heating the feed water and an economizer is desirable. Even with initial temperature as high as 200 deg. fahr. overall economy may result from the use of an economizer.

(3) Purity of the feed water. With impure feed water the formation of scale within the tubes may seriously affect the efficiency of heat transmission and the cost of cleaning may prove excessive. Internal corrosion may also be caused by impure feed water.

(4) Minimum temperature of the flue gas. The flue gas temperature should not be lowered below the dew point since the condensation of the vapor content may cause the soot to adhere to the tubes and render its removal a costly problem. An average minimum is 240 deg. fahr. With coals high in sulphur content the moisture forms sulphuric acid which corrodes the tubes.

(5) Increased capacity due to the additional heating surface.

(6) Cost of additional building space. With the independent type of economizer this is of secondary importance.

(7) Cost of producing the draft. For chimney draft this means cost of the extra height of stack necessary to overcome the loss in draft. This may range from 20 to 40 per cent of the total cost of the chimney. In the modern mechanical draft installation the power required to operate the fan ranges from one per cent to four per cent of the main generator output.

(8) First cost. Economizers cost approximately $1.25 per sq. ft. of surface for pressures under 250 lb. per sq. in., though the cost naturally varies with the cost of raw material.

(9) Boiler pressure. Cast-iron superheaters are used for working pressures as high as 400 lb. per sq. in. but the cost increases rapidly with increase in pressure above 250 lb. per sq. in. It is doubtful if cast iron will be used in projected new plants where pressures of 500 lb. or more are being seriously considered.

289. Choice of Feed-water Heating System. — The heating of feed water and its delivery to the boiler in the most economical manner is a problem involving such a large number of combinations that a general analysis is impracticable. The following discussion of a specific case will give some idea of the manner in which this problem may be attacked.

Example 57. Determine the most economical manner of heating the feed water for a power plant of 1000 horsepower operating under the following conditions: Schedule 10 hours per day and 310 days per year; load factor on the ten-hour basis 0.8; cost of coal $2.50 per ton of 2000 pounds; heat value of the coal 13,500 B.t.u. per pound; average boiler efficiency 65 per cent; engines use 20 pounds of steam per i.hp-hour; steam pressure 150 pounds absolute; temperature of cold water 60 degrees; vacuum 26 inches referred to 30-inch barometer; interest 5 per cent; depreciation $8\frac{1}{2}$ per cent; maintenance 1 per cent; insurance $\frac{1}{2}$ per cent; taxes 1 per cent; total charges 16 per cent; charges for attendance and maintenance assumed to be the same in each case and credit for the chimney assumed to offset debit for economizer space. Many of the influencing conditions are left out for the sake of simplicity.

The most likely combinations are

(1) Atmospheric, all auxiliaries steam driven, water taken from cold well.
(2) Same as (1) except that water is taken from hot well.
(3) Economizers, auxiliaries electrically driven, chimney draft, water from cold well.
(4) Vacuum heater, economizer, and electrically driven auxiliaries, fan draft.

(5) Vacuum heater, atmospheric heater, and steam auxiliaries.
(6) Atmospheric heater, economizer, steam auxiliaries, fan draft.
(7) Vacuum and atmospheric heaters, economizers, steam auxiliaries, and electrical fan.
(8) Vacuum, atmospheric heater, economizer, and chimney draft, auxiliaries operating condensing except feed pumps and stoker engines which exhaust into the atmospheric heater.

The difference between the total heat furnished by the boiler and the heat returned in the feed water is the net heat put into the steam by the boiler. Evidently the system which shows the least net heat required to produce one horsepower will be the most economical as far as coal consumption is concerned, although not necessarily the cheapest when both operating and fixed charges are considered.

Prices vary so much that it is practically impossible to give costs of installations which will bear criticism and the prices taken in this problem are approximate only.

Case I.

Atmospheric heater, auxiliaries steam driven, feed from cold well.

This arrangement and that of Case II are the most common in power plants of this size.

The power consumption of the auxiliaries operating non-condensing varies from 8 to 12 per cent of the total power developed. Assume it to be 10 per cent.

The temperature of the feed water leaving the heater may be determined by equation (237).

$$t = \frac{t_0 + 0.9\,S\,(\lambda + 32)}{1 + 0.9\,S}.$$

Substituting $S = 0.10, \quad \lambda = 1146, \quad t_0 = 60,$

$$t = \frac{60 + 0.9 \times 0.10\,(1146 + 32)}{1 + 0.9 \times 0.10}$$
$$= 152.$$

The *net heat* furnished by the boiler to produce one indicated horsepower-hour in the engine is evidently the heat necessary to raise 20 + 10 per cent of 20 = 22 pounds of water from 152 deg. fahr. to steam at 150 pounds pressure; i.e., the net heat furnished is

$$22 \times 1071.2 = 23{,}564 \text{ B.t.u.}$$

Now, 1 i.hp. = 2546 B.t.u.

Therefore the heat efficiency of this arrangement is

$$\frac{2546}{23{,}564} = 10.8 \text{ per cent.}$$

Probable First Cost.

Steam pumps	$400.00
Condenser with steam-driven air and circulating pumps	3000.00
1000-horsepower open heater	480.00
Piping	1200.00
	$5080.00

Fuel Consumption.

Average horsepower-hours per year = 1000 (rated horsepower) × 0.8 (curve load factor) × 310 (days per year) × 10 (hours per day) = 2,480,000.

Pounds of coal per i.hp-hour = net heat furnished per i.hp-hour ÷ net heat absorbed by the boiler per pound of coal = 23,564 ÷ (13,500 × 0.65) = 2.68.

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.68}{2000} = 3323.$$

Fuel and Fixed Charges.

Fuel, 3323 tons at $2.50	$8308.00
Fixed charges, 16 per cent of $5080	812.00
	$9120.00

Case II.

Same as Case I, except that feed is taken from the hot well. This arrangement is possible only when the condensing water is suitable for feed purposes.

Assume the temperature of the water from the hot well as it enters the heater to be 110 degrees.

The temperature of the feed water leaving the heater will then be 198 degrees (from equation (237)).

$$\text{Net heat furnished} = 22 \times 1025.2 = 22{,}554 \text{ B.t.u.}$$

$$\text{Efficiency} = \frac{2546}{22{,}554} = 11.3 \text{ per cent.}$$

$$\text{Pounds of coal per i.hp-hr.} = \frac{22{,}554}{13{,}500 \times 0.65} = 2.62.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.62}{2000} = 3248.$$

Fuel and Fixed Charges.

Fuel, 3248 tons at $2.50	$8120.00
Fixed charges (same as Case I)	812.00
	$8932.00

Case III.

Economizers, auxiliaries electrically driven, chimney draft, water from the cold well.

Practice gives an average of 3 per cent of the main engine output as the power required to operate the electrical auxiliaries in a plant of this size.

The temperature rise of the feed water leaving the economizer is found to be 119 deg. fahr. (equation 248).

Temperature of feed water entering boiler = 119 + 60 = 179 degrees.

Net heat furnished = (20 + 3 per cent of 20) × 1044.2 = 21,510 B.t.u.

$$\text{Efficiency} = \frac{2545}{21{,}510} = 11.8 \text{ per cent.}$$

Probable First Cost.

Economizers	$3500.00
Motor feed pump	600.00
Condenser with electrically driven air and circulating pump	6000.00
Piping and wiring	1000.00
	$11,100.00

Fuel Consumption.

$$\text{Pounds of coal per i.hp-hr.} = \frac{21{,}510}{13{,}500 \times 0.65} = 2.45.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.45}{2000} = 3038.$$

Fuel and Fixed Charges.

Fuel, 3038 tons at $2.50	$7595.00
Fixed charges, 16 per cent on $11,100	1776.00
	$9371.00

Case IV.

Vacuum heater, economizer, electrically driven auxiliaries, fan draft.

The vacuum heater may be relied upon to raise the temperature of the feed water to 110 degrees.

The economizer will increase this 107 degrees (from equation (248)), giving the feed water a temperature of 217 degrees as it enters the boiler.

The electrical fan for the mechanical-draft system will require approximately 2 per cent of the main system engine power, making a total of $3 + 2 = 5$ per cent for all auxiliaries.

$$\begin{aligned}\text{Net heat furnished} &= (20 + 5 \text{ per cent of } 20) \times 1006.2\\ &= 21{,}130 \text{ B.t.u.}\end{aligned}$$

$$\text{Efficiency} = \frac{2545}{21{,}130} = 12.05 \text{ per cent.}$$

Probable First Cost.

For the sake of simplicity it is assumed that the high first cost of the chimney plus its low depreciation and maintenance will offset the low first cost of the mechanical-draft system plus its higher maintenance and depreciation charges:

Economizers	$3500.00
Motor feed pump	600.00
Motor-driven pumps and condenser	6000.00
Motor-driven fan	750.00
Piping and wiring	1200.00
Vacuum heater	200.00
	$12,250.00

Fuel Consumption.

$$\text{Pounds of coal per i.hp-hr.} = \frac{21{,}130}{13{,}500 \times 0.65} = 2.41.$$

$$\text{Tons per year} = \frac{2{,}480{,}000 \times 2.41}{2000} = 2988.$$

Fuel and Fixed Charges.

Fuel, 2988 tons at $2.50	$7470.00
Fixed charges, 16 per cent of $12,250	1960.00
	$9430.00

In like manner Cases V, VI, VII, and VIII have been treated and are tabulated in the summaries.

SUMMARY (1).

Case.	Temperature of Feed Water.	Power Consumed by Auxiliaries.	Efficiency.	First Cost.	Fuel Cost per Year.	Cost of Operation per Year.
	Degrees F.	Per Cent.	Per Cent.			
I..........	152	10	10.8	$5,080	$8,308	$9,120
II..........	198	10	11.3	5,080	8,120	8,932
III.........	179	3	11.8	11,100	7,595	9,371
IV.........	217	5	12.05	12,250	7,470	9,430
V..........	208	10	11.4	5,280	7,900	8,744
VI.........	294	14	12	9,000	7,750	9,190
VII........	290	10	12.2	9,300	7,380	9,570
VIII.......	270	8	12.3	8,250	7,075	8,395

SUMMARY (2).

Case.	Efficiency.	First Cost.	Fuel.	Cost per Year.
I..............	8	1	8	4
II..............	7	1	7	2
III.............	6	6	4	6
IV.............	3	7	3	7
V..............	5	2	6	3
VI.............	4	4	5	5
VII............	2	5	2	8
VIII...........	1	3	1	1

Summary (2) gives the ranking; thus: Case I is eighth in point of efficiency first in cheapness of installation; eighth in yearly cost of fuel; and fourth in yearly cost of operation. Case VIII is apparently the best arrangement for the *given conditions.*

Bleeding Turbines to Heat Feed Water: Power, May 15, 1917, p. 652.

PROBLEMS.

1. Determine the amount of soda ash and lime necessary to soften 10,000 gallons of water as per analysis, Col. 2, Table 98.

2. In a certain plant it costs 30 cents per 1000 lb. to evaporate water from feed temperature of 60 degrees to steam at 115 lb. abs. and 50 deg. superheat; required the saving in per cent if the feed water is heated by exhaust steam to 210 deg. fahr.

3. A 2000-kw. turbo-generator plant uses 18 lb. steam per kw-hr., initial pressure 140 lb. abs., back pressure 3 in. abs., superheat 100 deg. fahr., temperature of the

condensate 100 deg. fahr.; auxiliaries develop 100 hp. and use 30 lb. steam per hp-hr. (non-condensing), initial pressure 115 lb. abs., steam dry at admission; required the temperature of the feed water if the auxiliary exhaust is discharged into an open heater.

4. Required the tube surface necessary for a closed heater suitable for the conditions in Problem 3. Assume $U = 350$.

5. If the tubes are $\frac{1}{2}$ inch inside diameter, required the total length of water travel for the conditions in Problem 4, assuming a water velocity through the tubes of 120 ft. per min.

6. Calculate the final feed-water and flue-gas temperatures for an economizer installation operating under the following conditions: Boiler heating surface 10,000 sq. ft., economizer surface 6500 sq. ft., initial feed-water temperature 120 deg. fahr., initial flue-gas temperature 700 deg. fahr. when the boiler is operating at 150 above standard rating; coal used, Illinois washed nut, 13,500 B.t.u. per lb.

CHAPTER XIII

PUMPS

290. Classification. — Pumps used in connection with steam power plants may be conveniently classified under five groups according to the principles of action.

1. *Piston pumps,* in which motion and pressure are imparted to the fluid by a reciprocating piston, plunger, or bucket. The action is positive and a certain definite amount of fluid is handled per stroke under predetermined conditions of pressure and velocity.

2. *Centrifugal pumps,* in which the fluid is given initial velocity and pressure by a rotating impeller. The action is not positive, as the amount of fluid discharged is not necessarily proportional to the impeller displacement.

3. *Positive displacement rotary pumps,* in which motion and pressure are imparted to the fluid by a rotating impeller or screw. The volume discharged is practically equal to the impeller displacement regardless of pressure.

4. *Jet pumps,* in which velocity and pressure are imparted to the fluid by the momentum of a jet of similar or other fluid. The ordinary steam injector is the best known of this group.

5. *Direct-pressure pumps,* in which the pressure of one fluid acts directly on the surface of another fluid, thereby imparting all or part of its energy to the latter. The pulsometer is an example of this type.

These groups may be variously subdivided as follows:

Piston	Direct-acting	Simplex; Duplex	Air. Vacuum. Forcing. Lifting.
	Fly-wheel; Power driven	Simplex; Duplex; Triplex	
Centrifugal	Volute; Turbine	Single stage; Multi-stage	Vacuum. Forcing. Lifting.
Rotary	Power driven	Forcing; Lifting	
Jet	Injector; Ejector	Positive; Automatic	
Direct pressure	Pulsometer	Lifting	
	Air-lift	Lifting	

Piston or *plunger pumps* are the most common in use. Small boiler-feed pumps, city waterworks pumps and force pumps are ordinarily of this type. In the direct-acting type, Fig. 367, the water plunger and steam piston are secured to a single piston rod and the steam pressure is transmitted directly to the water. There is no flywheel, connecting rod, or crank. The velocity of the delivery is proportional to the resistance offered by the water; when the resistance equals the forward effort of the steam pressure the pump stops. This class of pump is well adapted for boiler-feeding purposes, since it may be operated as slowly as suits the requirements of feeding by simply throttling the discharge. The steam consumption is very large in proportion to the work performed, since the steam is not used expansively.

Flywheel pumps, Figs. 380, 428, are ordinarily classified as pumping engines. In this class steam may be used expansively, as sufficient energy is stored in a flywheel to permit the drop in steam pressure during expansion. These pumps find wide application in city waterworks, elevator plants, and the like, where high duty is required. They are little used as stationary boiler feeders, but are used to some extent in river-boat practice and in plants operating continuously for long periods at comparatively steady loads. Practically all sizes of dry-air pumps and a number of large jet condenser pumps are of this type.

Piston pumps, Fig. 387, driven by gearing or belting are ordinarily classified as power-driven pumps. The driving power may be steam engine, electric motor, or gas engine. The single-cylinder machine is often designated as a "simplex" power-driven pump, the two-cylinder as a "duplex," the three-cylinder as a "triplex," and so on.

Centrifugal pumps, Fig. 415, are supplanting to a considerable extent the present type of piston pump for many uses. Though particularly adapted for low heads and large volumes, they are used in many situations requiring extremely high heads. They are not as efficient as high-grade pumping engines, but the extremely low first cost frequently offsets this disadvantage, and they are much used in connection with dry docks, irrigating plants, sewage systems, and as circulating and vacuum pumps in condensing plants.

Rotary pumps, Fig. 424, are employed to a limited extent in the same field as the centrifugal pump. Being positive in action, they permit of a much lower rotative speed for the same delivery pressure.

Jet pumps, Fig. 391, are seldom used as pumps in the ordinary sense of the word, on account of their extremely low efficiency, but are frequently employed for discharging water from sumps. Their greatest field of application lies in boiler feeding and in this respect their efficiency is comparable with that of the average piston pump. A recently developed multi-jet air pump gives great promise of superseding the present type of dry-air pump for vacuum purpose. See paragraph 309.

Direct-pressure pumps operated by steam, such as the "pulsometer," Fig. 430a, are used principally for pumping out sumps, surface drains, and the like, where the operation is intermittent. Direct-pressure pumps of the air-lift type, Fig. 431, are quite common and are used a great deal in situations where water is to be pumped from a number of scattered wells.

291. Boiler-feed Pumps, Direct-acting Duplex. — Figs. 366 and 367 illustrate a typical duplex boiler-feed pump, which consists virtually of two direct-acting pumps mounted side by side, the water ends and the

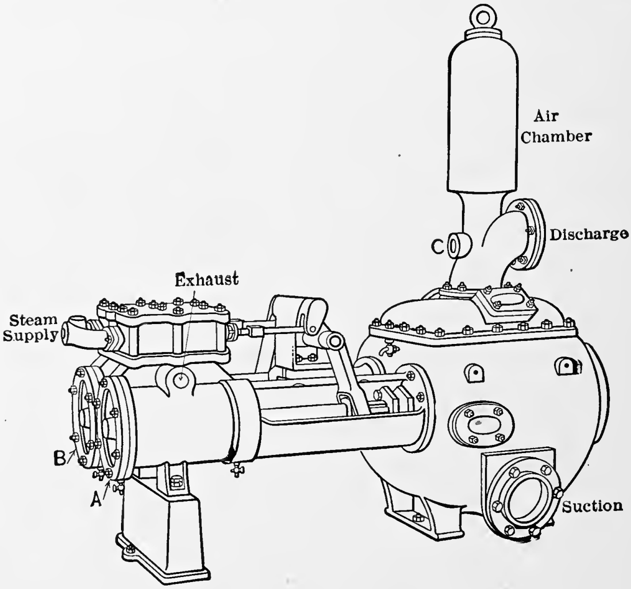


FIG. 366. Typical Duplex Pump.

steam ends working in parallel between inlet and exhaust pipe. The piston rod of one pump operates the steam valve of the other through the medium of bell cranks and rocker arms. The pistons move alternately, and one or the other is always in motion, the flow of water being practically continuous.

In general construction the steam pistons and valves are similar to those of steam engines. The valves in duplex pumps, however, have no lap. In order to reduce the valve travel to a minimum, and still have sufficient bearing surface between the steam ports and the main exhaust ports to prevent the leakage of steam from one to the other, separate exhaust ports are provided which enter the cylinder at nearly the same point as the steam ports. This arrangement offers a simple means of cushioning the piston by exhaust steam, thus preventing it from striking the cylinder heads at the ends of the stroke. The valves of the duplex pump having no lap would, if connected rigidly to the valve stem, open one port as soon as the other had been

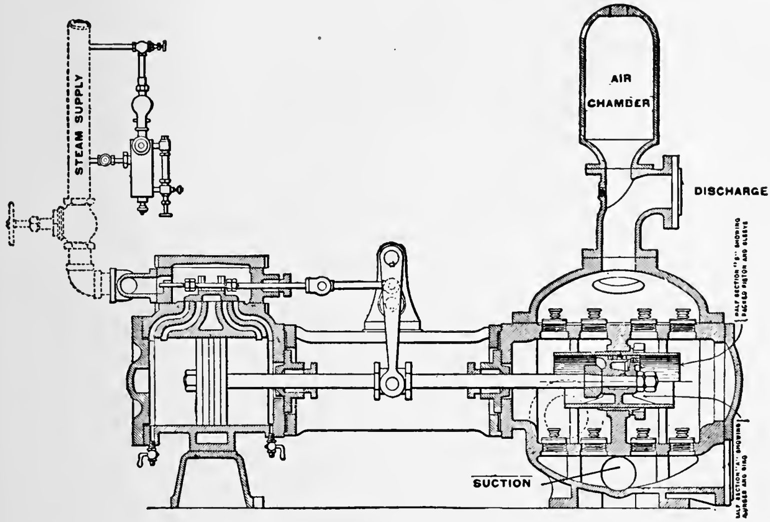


Fig. 367. Section through a Typical Duplex Boiler-feed Pump.

closed, at about mid-stroke of the piston, thus cutting down the stroke to about one fourth the usual length. To obviate this difficulty the valves are given considerable lost motion by allowing sufficient clearance between the lock nuts on the valve stem; the latter, therefore, imparts no motion to the valve until the piston operating it has nearly completed the stroke. The lost motion between valves and lock nuts renders it impossible to stop the pump in any position from which it cannot be started by simply admitting steam, and therefore the pump has no dead centers. When one piston moves to the end of the stroke it pulls or pushes the opposite valve to the end of its travel; then when the piston starts back to the other end of its stroke the valve remains stationary, owing to the lost motion, until the piston has completed about one half the stroke. During this time the opposite piston has completed a full stroke and the valve operated by it will have opened the steam port wide, so that while one valve covers both steam ports the other is at the end of its travel. In some makes of pumps the stem is rigidly attached to the valves, the lost motion being adjusted outside the steam chest as shown in Figs. 368 and 369, which represent two common constructions of duplex valve gear.

Fig. 370 shows the valve and piston in the position occupied at the commencement of the stroke. At one end of the valve the steam port P is open wide and at the opposite end the exhaust port E is open wide.

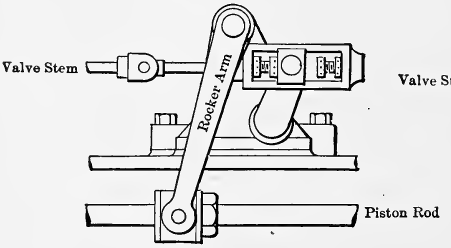


FIG. 368.

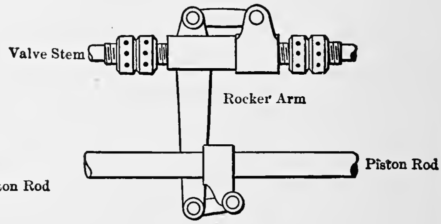


FIG. 369.

When the piston nears the opposite end of the stroke and reaches the position shown in Fig. 371 the steam escape through the exhaust port E is cut off by the piston, and since the steam port is closed, the remaining steam is compressed between the piston and cylinder head, thus arresting the motion of the piston gradually without shock or jar.

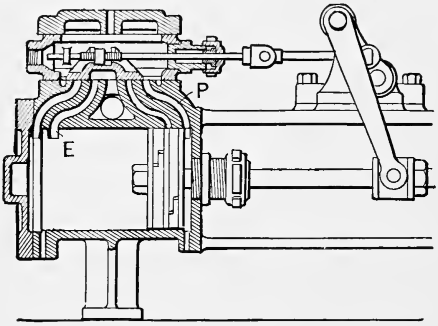


FIG. 370.

The construction of the water end of single-cylinder and duplex pumps is practically the same; any slight differences which may be found are confined to minor details which in no way affect the general design or operation of the pump. The piston is double acting, the single-acting cylinder being confined to power pumps or to steam pumps intended for very high pressures. In the old-style pumps it was the custom to use one large valve with a lift sufficient to give the required passage, but in modern practice the required area is divided among several small valves, so that each one is easily and cheaply removed in case of accident or wear, and slip is lessened.*

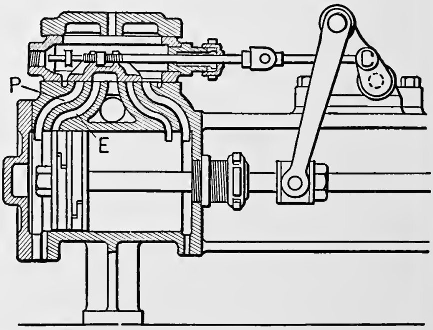


FIG. 371.

* The modern Riedler pump is an exception. See Engineer, U. S., Nov. 15, 1907, p. 1040.

The valves are carried by two plates or decks, the suction valves being attached to the lower plate and the delivery valves to the upper one, as shown in Fig. 368.

The valves in practically all boiler-feed pumps are of the flat disk type, Fig. 372, held firmly to the seat by conical springs and guided by a bolt through the center.

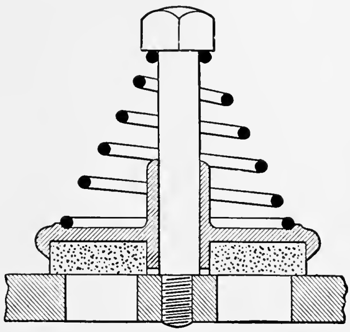


Fig. 372. A Typical Pump Disk Valve.

All pumps are provided with an air chamber on the discharge side, which acts as a cushion for the water, prevents excessive pounding, and insures a uniform flow. Fig. 373 shows a section through the steam end of a compound duplex pump.

292. Feed Pumps with Steam-actuated Valves. — Single-cylinder direct-acting pumps, Fig. 374, are ordinarily operated by steam-actuated valves. The steam enters the chest *C* and passes to the left through the annular opening *A* formed between the reduced neck of the valve and the bore of the steam chest. It is thus projected against the inside surface of the valve head *H* before escaping through the port *P* and passing to the cylinder. Both the pressure and impulse

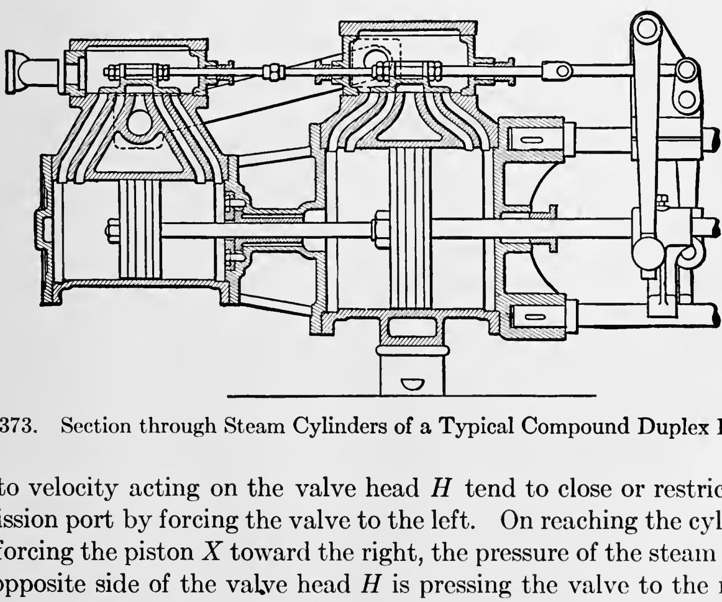


Fig. 373. Section through Steam Cylinders of a Typical Compound Duplex Pump.

due to velocity acting on the valve head *H* tend to close or restrict the admission port by forcing the valve to the left. On reaching the cylinder and forcing the piston *X* toward the right, the pressure of the steam upon the opposite side of the valve head *H* is pressing the valve to the right, a movement which would give the admission more port opening at *A* and deliver more steam to the cylinder. The valve then holds a position depending upon the relative intensity of the two pressures, which tend to move it in opposite directions, the admission steam, tending to close the valve, and cylinder steam, tending to open the valve wider.

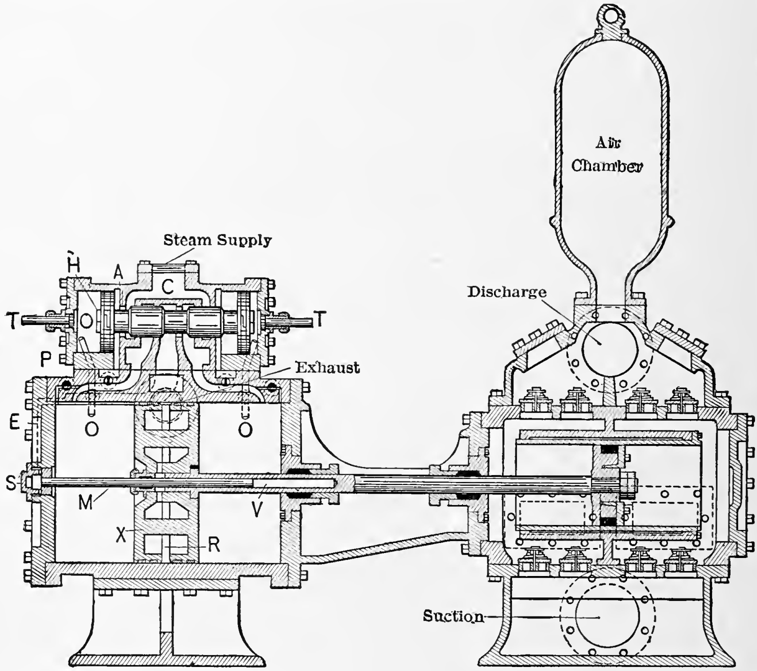


FIG. 374. Marsh Boiler-feed Pump. A Typical Steam-actuated Valve Gear.

The steam valve, therefore, is always in a balanced position. The steam piston is grooved at the center, forming a reservoir for live steam R which is supplied from the upper chamber of the steam chest by passage E to the cylinder cap S, and thence by tube M and the hollow piston rod V. The steam in this annular piston space reverses the steam valve by pressing alternately against the outer surfaces of the valve heads H through the connecting passages O, O near each end of the cylinder. The tappets T are for the purpose of moving the valve by hand in case it fails to move automatically. Steam-actuated valves are not as positive in action as mechanically operated valves, and hence are little used in situations where positive action is essential, as in fire-pump service.

293. Air and Vacuum Chambers. — Air chambers in piston pumps are for the purpose of causing a steady discharge of water and of reducing excessive pounding at high speeds by providing a cushion for the water. The water discharged under pressure compresses the air in the air chamber somewhat above the normal pressure of discharge during each stroke of the water piston, and when the piston stops momentarily at the end of the stroke the air expands to a certain extent and tends to produce a uniform rate of flow.

The volume of the air chamber varies from 2 to $3\frac{1}{2}$ times the volume of the water piston displacement in single-cylinder pumps, and from 1 to $2\frac{1}{2}$ times in the duplex type. High-speed pumps are provided with air chambers of from 5 to 6 times the piston displacement. The water level in the air chamber should be kept down to one fourth the height of the chamber. In slow-running pumps sufficient air may be carried into the pump chamber along with the water, but with high speeds a large part of the air will be discharged, and air must be forced into the chamber by mechanical means. The larger the chamber the more uniform will be the discharge pressure.

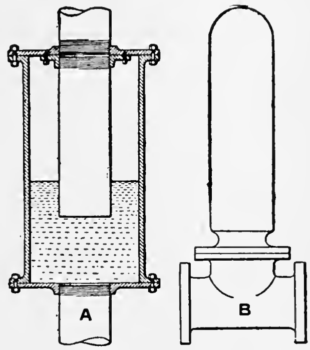


Fig. 375. Forms of Vacuum Chambers.

Vacuum chambers are frequently provided for the purpose of maintaining a uniform flow of water in the suction pipe and assisting in the reduction of slip. Such chambers should be of slightly greater volume than the suction pipe and of considerable length rather than diameter.

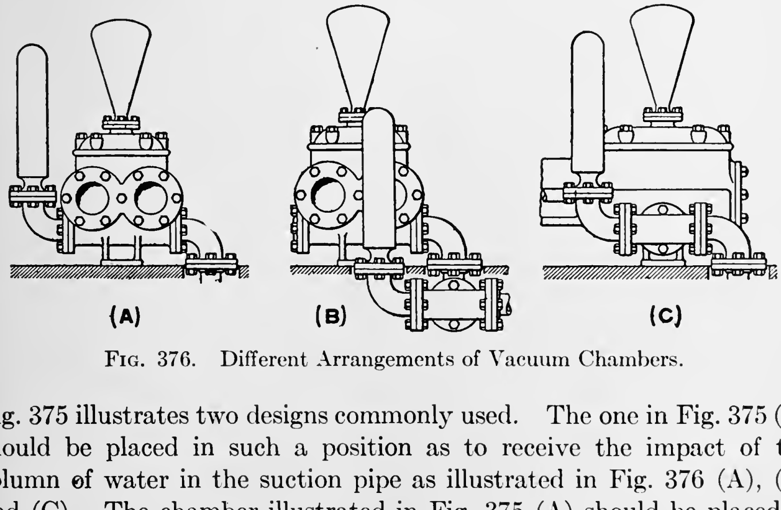


Fig. 376. Different Arrangements of Vacuum Chambers.

Fig. 375 illustrates two designs commonly used. The one in Fig. 375 (B) should be placed in such a position as to receive the impact of the column of water in the suction pipe as illustrated in Fig. 376 (A), (B) and (C). The chamber illustrated in Fig. 375 (A) should be placed in the suction pipe below but close to the pump.

294. Water Pistons and Plungers. — In cold-water pumps the water pistons are usually packed with some kind of soft packing. Fig. 377 (A) shows the details of a piston with square *hydraulic packing*. The body E is fastened to the piston rod by nut C; packing is placed at D, and

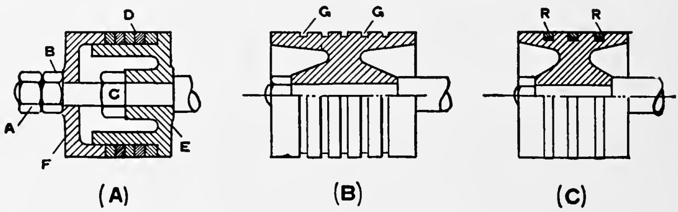


Fig. 377. Types of Water Pistons.

follower F is forced up by the nut B and locked by nut A. For large sizes the design is the same except that the follower is set up by a number of nuts near the edge. In hot-water pumps the pistons are often packed by means of *metallic piston rings* R, R, Fig. 377 (C), similar to those in steam pistons, or merely by *water grooves* G, G, Fig. 377 (B). The water end is often fitted with a *plunger* instead of a piston, as in

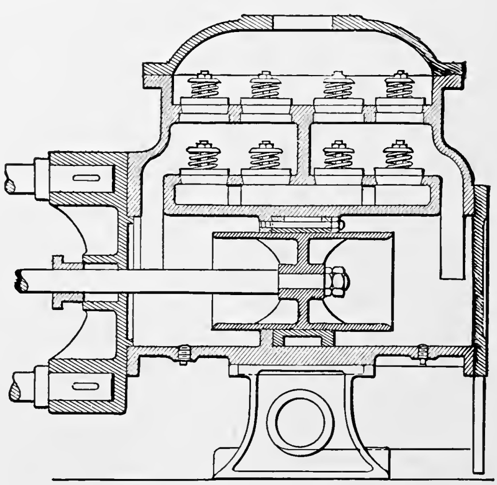


Fig. 378. Plunger with Metal Packing Ring.

Figs. 378 to 380. The piston is more compact, but the plungers do not require a bored cylinder, so that the first cost is not materially different.

Fig. 378 shows a plunger with metal packing ring. When leakage becomes excessive it is necessary to renew the ring, which is readily removed.

In Fig. 379 the plunger is packed with hydraulic packing as in the follower type of pump piston. The great difficulty with the above types of piston and plunger is in keeping the packing tight or in know-

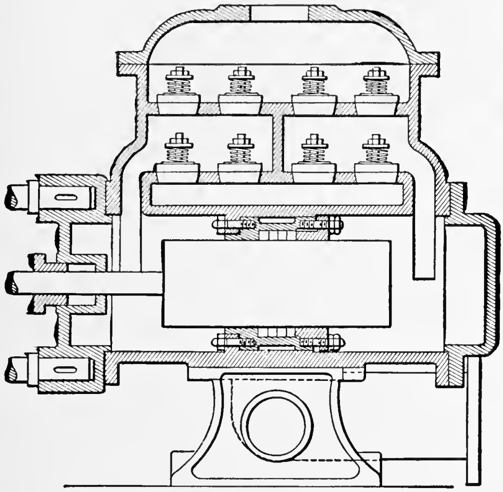


Fig. 379. Plunger with Hydraulic Packing.

ing when it is leaking, and the trouble necessary to replace the packing. The *outside packed plunger*, Fig. 380, obviates these disadvantages to a great extent, since leakage is readily detected and repacking is performed

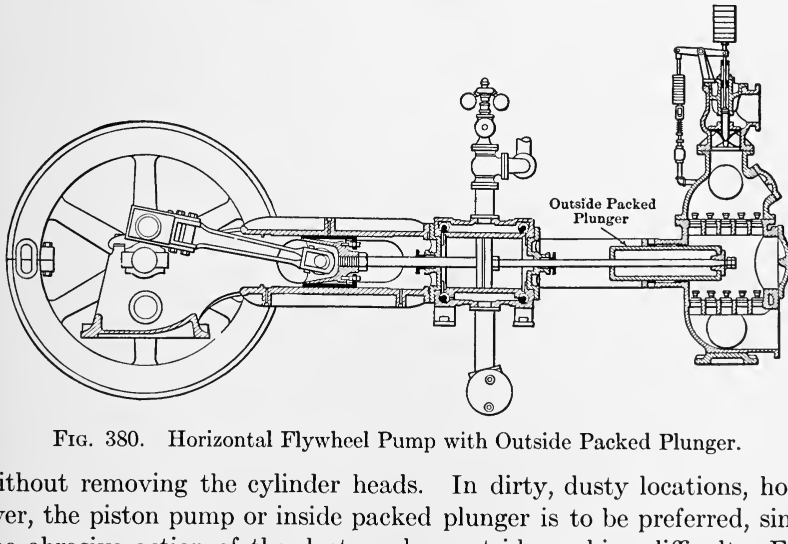


Fig. 380. Horizontal Flywheel Pump with Outside Packed Plunger.

without removing the cylinder heads. In dirty, dusty locations, however, the piston pump or inside packed plunger is to be preferred, since the abrasive action of the dust renders outside packing difficult. Fig. 380 illustrates a high-duty elevator pump with outside packed plunger.

295. Performance of Piston Pumps. — Direct-acting pumps as a class are wasteful of fuel and low in efficiency, due largely to the non-expansive use of steam. The average small duplex boiler-feed pump uses from 100 to 200 pounds of steam per i.hp-hr., depending upon the speed, and the mechanical efficiency varies from 50 per cent to 90 per cent. When new and in proper working condition the mechanical efficiency is seldom less than 85 per cent; but such pumps, as a rule, are given scant attention, and the average efficiency is not far from 65 per cent. The term "mechanical efficiency" in this connection refers to the ratio of the actual water horsepower to the indicated horsepower of the steam cylinder. The loss includes the slip of the piston

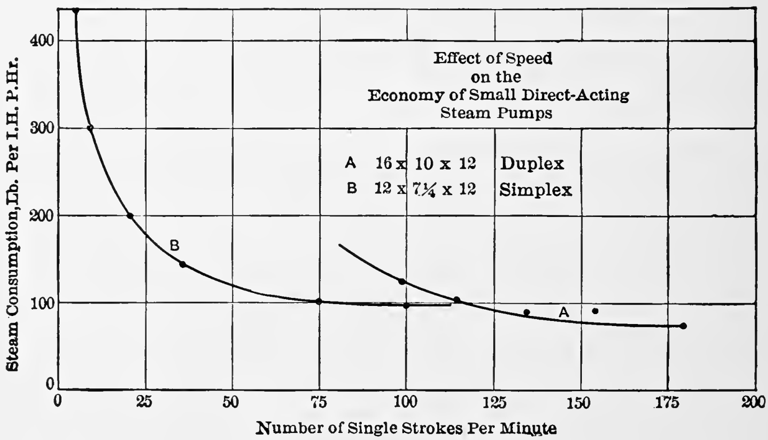


Fig. 381.

and valves. A steam consumption of 150 pounds per i.hp-hour with mechanical efficiency of 65 per cent is equivalent to a power consumption of about 5 per cent of the rated boiler capacity, although if the exhaust steam is used for feed-water heating the actual heat consumption may be but 1 to 1.5 per cent. Compound direct-acting pumps running non-condensing use from 50 to 100 pounds of steam per i.hp-hour. Single-cylinder flywheel pumps of the slow-speed type, running non-condensing, use about 50 pounds of steam per i.hp-hour. Multi-cylinder flywheel pumps of the high-duty type use about 25 pounds per i.hp-hour when running non-condensing, and as low as 10 pounds when operating condensing. High-grade *direct-connected* motor-driven power pumps have a mechanical efficiency from line to water load, at normal rating, of about 80 per cent. The efficiency of *geared* pumps at normal rating varies with the character of the gearing and the degree of speed reduction, and may range anywhere from 40 to 70 per cent.

The steam consumption of all direct-acting boiler pumps decreases with the increase in speed. This is illustrated by curve *B*, Fig. 381, plotted from the tests of a 12-in. by $7\frac{1}{4}$-in. by 12-in. direct-acting single-cylinder pump at Armour Institute of Technology, and curve *A* based on experiments with a 16-in. by 12-in. duplex fire pump at Massachusetts Institute of Technology.

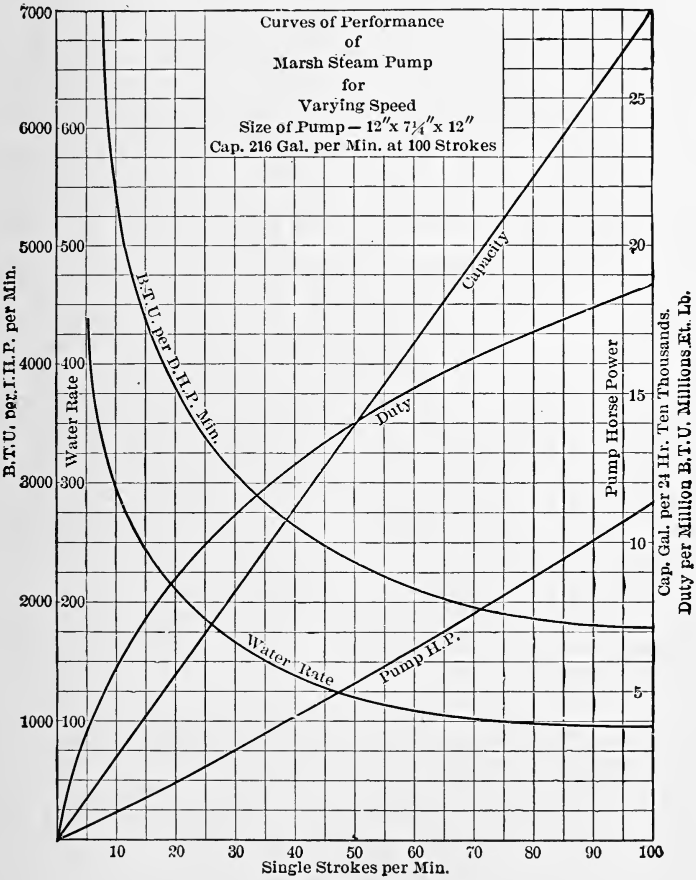


FIG. 382.

Fig. 382 gives the details of the performance of a 12-in. by $7\frac{1}{4}$-in. by 12-in. Marsh boiler-feed pump at the Armour Institute of Technology.

The determination of the power consumption of a boiler-feed pump is best illustrated by the following example.

Example 58. A small direct-acting duplex pump uses 150 pounds of steam per i.hp-hour. Gauge pressure 150 pounds per square inch; feed-water temperature 64 deg. fahr. Required the per cent of rated boiler capacity necessary to operate the pump.

The head pumped against, 150 pounds per square inch, is equivalent to $150 \times 2.3 = 345$ feet of water.

The friction through the valves, fittings, and pipe, and the vertical distance between suction and feed-water inlet, are assumed to be equivalent to 20 per cent of the boiler pressure, giving a total head of $150 + 30 = 180$ pounds per square inch, or 414 feet of water.

A boiler horsepower, taking into consideration leakage losses and the steam used by the feed pump, will be equivalent to the evaporation of approximately 32 pounds of water per hour from a feed temperature of 64 deg. fahr. to steam at 150 pounds gauge.

The actual work done in pumping 32 pounds of water against a head of 414 feet is

$$414 \times 32 = 13{,}248 \text{ foot-pounds.}$$

This corresponds to

$$\frac{13{,}248}{60 \times 33{,}000} = 0.0067 \text{ horsepower.}$$

The total heat of one pound of steam above 64 deg. fahr. is 1163 B.t.u. The heat delivered to the pump per i.hp-hour is

$$1163 \times 150 = 174{,}450 \text{ B.t.u.}$$

The amount used by the pump for each boiler horsepower, disregarding efficiency, is

$$174{,}450 \times 0.0067 = 1168 \text{ B.t.u. per hour.}$$

The mechanical efficiency of the average feed pump ranges from 50 to 85 per cent, depending upon its condition and the number of strokes per minute. Assuming it to be 65 per cent, the heat used by the pump per hour to deliver 32 pounds of water into the boiler is

$$1168 \div 0.65 = 1796 \text{ B.t.u.}$$

A boiler horsepower is equivalent to 33,479 B.t.u. per hour. Therefore the per cent of boiler output necessary to operate the pump is

$$100 \times \frac{1796}{33{,}479} = 5.36 \text{ per cent.}$$

If the exhaust steam is used for heating the feed water, the steam consumption will be 0.73 per cent of the boiler capacity, thus: The weight of steam consumed per boiler horsepower-hour

$$\frac{1796}{1163} = 1.54 \text{ pounds.}$$

Allowing a 10 per cent loss, the heat in the exhaust available for heating the feed water is

$$[1150 - (64 - 32)]\,0.9 \times 1.54 = 1550 \text{ B.t.u.}$$

$1796 - 1550 = 246$ B.t.u., or the net heat required by the pump per hour to deliver 32 pounds of water to the boiler.

The per cent of boiler output necessary to operate the pump is

$$100 \frac{246}{33{,}479} = 0.73.$$

Pump performances are generally given in terms of the foot-pounds of work done by the water piston per thousand pounds of dry steam or per million B.t.u. consumed by the engine, thus:

$$1.\ \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Weight of dry steam used}} \times 1000. \tag{253}$$

$$2.\ \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat units consumed}} \times 1{,}000{,}000. \tag{254}$$

(See A.S.M.E. code for conducting duty trials of pumping engines, Trans. A.S.M.E., Vol. 37, 1915.)

Example 59. A compound feed pump uses 100 pounds of steam per i.hp-hour; indicated horsepower, 48; capacity, 400 gallons per minute; temperature of water, 200 deg. fahr.; total head pumped against, 175 pounds per square inch; steam pressure, 100 pounds gauge; moisture in the steam, 3 per cent. Required the duty on the dry steam and on the heat-unit basis.

175 pounds per square inch is equivalent to $175 \times 2.4 = 420$ feet of water at 200 deg. fahr.

Weight of 400 gallons of water at 200 deg. fahr. $= 400 \times 8.03 = 3212$ pounds.

Work done per minute $= 3212 \times 420 = 1{,}329{,}040$ foot-pounds.

Weight of dry steam supplied per minute

$$= \frac{100 \times 48}{60} \times 0.97 = 77.6 \text{ pounds.}$$

B.t.u. supplied per minute

$$= \frac{100 \times 48}{60} (0.97 \times 879.8 + 309 - 200 + 32) = 79{,}552.$$

Duty per thousand pounds of dry steam

$$= \frac{1{,}349{,}040}{77.6} \times 1000 = 17{,}384{,}150 \text{ foot-pounds.}$$

Duty per million B.t.u.

$$= \frac{1{,}349{,}040}{79{,}552} \times 1{,}000{,}000 = 16{,}958{,}000 \text{ foot-pounds.}$$

Table 111 may be used in approximating the duty, thus:

The mechanical efficiency of the pump in the preceding problem is

$$\text{Efficiency} = \frac{\text{p.hp.}}{\text{i.hp.}} = \frac{1{,}349{,}040}{33{,}000 \times 48} = 85 \text{ per cent.}$$

At the intersection of vertical column "85" and horizontal column "100" of Table 111, we find 16.82 millions. See, also, Table 79.

TABLE 111.

PERFORMANCE OF STEAM PUMPS. DUTY IN MILLIONS OF FOOT-POUNDS PER MILLION B.T.U.

Mechanical efficiency = $\frac{\text{Pounds discharged per min.} \times \text{head in feet}}{\text{I.hp. of steam cylinder} \times 33{,}000}$.

Steam Consumption, Lb. per I.hp-hour.		0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55	0.50
Initial Press. 100 Lb. Gauge, Non-Cond.	200	9.40	8.91	8.42	7.92	7.42	6.93	6.44	5.94	5.45	4.95
	190	9.90	9.39	8.86	8.36	7.83	7.31	6.79	6.27	5.74	5.22
	180	10.45	9.90	9.38	8.80	8.25	7.70	7.15	6.60	6.05	5.50
	170	10.90	10.50	9.90	9.32	8.74	8.15	7.57	6.99	6.41	5.83
	160	11.75	11.13	10.51	9.90	9.28	8.66	8.04	7.42	6.81	6.19
	150	12.55	11.90	11.22	10.59	9.90	9.29	8.49	7.92	7.26	6.60
	140	13.45	12.75	12.02	11.32	10.61	9.90	9.20	8.49	7.76	7.07
	130	14.49	13.71	12.96	12.20	11.42	10.69	9.90	9.15	8.38	7.62
	120	15.67	14.85	14.03	13.20	12.38	11.55	10.71	9.90	9.08	8.25
	110	17.10	16.21	15.31	14.40	13.50	12.60	11.70	10.80	9.90	9.00
	100	18.81	17.82	16.82	15.82	14.84	13.86	12.88	11.88	10.89	9.90
	90	20.90	19.80	17.76	16.72	15.67	14.63	13.58	12.54	11.49	10.45
	80	23.51	22.27	21.03	19.80	18.56	17.32	16.08	14.85	13.61	12.37
	70	26.90	24.50	24.04	22.64	21.22	19.80	18.40	16.98	15.58	14.14
	60	31.35	29.70	28.05	26.40	24.75	23.10	21.43	19.80	18.15	16.50
	50	37.62	35.64	33.64	31.68	29.68	27.72	25.76	23.76	21.98	19.80
	40	47.02	44.55	42.07	39.60	37.12	34.65	32.17	29.70	27.22	24.75
	30	62.70	59.40	56.10	52.80	49.50	46.20	42.86	39.60	36.30	33.00
Initial Press. 150 Lb. G. Vacuum 2 Lb. Abs.	25	68.40	64.80	61.20	57.60	54.00	50.04	46.80	43.20	39.60	36.00
	20	85.50	81.00	76.50	72.00	67.50	63.00	58.50	54.00	49.50	45.00
	18	95.00	90.00	85.00	80.00	75.00	70.00	65.00	60.00	55.00	50.00
	16	106.87	101.25	95.62	90.00	84.37	78.75	73.12	67.50	61.87	56.25
	15	114.00	108.00	102.00	96.00	90.00	84.00	78.00	72.00	66.00	60.00
	14	122.14	115.71	109.29	102.86	96.43	90.00	83.57	77.14	70.71	64.28
	13	131.53	124.61	117.61	110.77	103.84	96.92	90.00	83.07	76.15	69.22
	12	142.50	135.00	127.50	120.00	112.50	105.00	97.50	90.00	82.50	75.00
	11	155.45	147.30	139.00	130.90	122.70	114.50	106.30	98.16	90.00	81.80
	10	171.00	162.00	153.00	144.00	135.00	126.00	117.00	108.00	99.00	90.00

Tables 112 and 113 give the maximum theoretical height to which pumps may lift water by suction at different temperatures. In practice these figures cannot be realized. It is customary to have the water gravitate to the pump for all temperatures over 120 deg. fahr.

TABLE 112.

MAXIMUM HEIGHTS TO WHICH PUMPS CAN RAISE WATER BY SUCTION.

(Temperature of Water 40 Deg. Fahr.; Barometer 29.92.)

Vacuum in Suction Pipe, Inches of Mercury.	Theoretical Lift.	Probable Actual Lift.	Vacuum in Suction Pipe, Inches of Mercury.	Theoretical Lift.	Probable Actual Lift.
	Feet.	Feet.		Feet.	Feet.
1	1.1	0.9	16	18.0	14.4
2	2.2	1.8	17	19.1	15.3
3	3.3	2.7	18	20.2	16.1
4	4.5	3.6	19	21.4	17.1
5	5.6	4.5	20	22.5	18.0
6	6.7	5.4	21	23.7	18.9
7	7.9	6.3	22	24.8	19.8
8	9.0	7.2	23	25.9	20.7
9	10.1	8.1	24	27.0	21.6
10	11.3	9.0	25	28.2	22.7
11	12.4	9.9	26	29.3	23.9
12	13.5	10.8	* 27	30.4	24.3
13	14.6	11.7	28	31.6	25.2
14	15.8	12.6	29	32.7	26.1
15	16.9	13.5	† 29.68	33.6	....

* Vacua greater than 27 inches are practically unobtainable in pumping practice except in connection with condensers.

† Maximum theoretical vacuum obtainable with water at 40 degrees F. and barometer of 29.92 inches.

TABLE 113.

MAXIMUM THEORETICAL HEIGHT TO WHICH A PUMP CAN LIFT WATER BY SUCTION AT DIFFERENT TEMPERATURES.

(Barometer 29.92.)

Temperature of Feed Water.	Maximum Theoretical Lift.	Temperature of Feed Water.	Maximum Theoretical Lift.
Deg. fahr.	Feet.	Deg. fahr.	Feet.
40	33.6	130	29.2
50	33.5	140	27.8
60	33.4	150	25.4
70	33.1	160	23.5
80	32.8	170	20.3
90	32.4	180	16.7
100	31.9	190	12.8
110	31.3	200	7.6
120	30.3	210	1.3

296. Size of Boiler-feed Pump. — Reciprocating Piston Type. — Let

D = diameter of water cylinder, inches.
d = diameter of the steam cylinder, inches.
L = length of stroke, inches.
N = number of working strokes per minute.
H = head in feet between suction and boiler water level.
R = resistance in pounds per square inch between suction level and boiler water level due to valves, pipes, and fittings.
p = boiler pressure, pounds per square inch.
S = ratio of the water actually delivered to the piston displacement.
W = weight of water delivered, pounds per hour.
I = indicated horsepower of the pump at maximum capacity.
E = mechanical efficiency of the pump, taken as the ratio of the water horsepower at the discharge opening to the indicated horsepower of the pump, steam end.

Then

$$W = \frac{\pi}{4} \cdot \frac{D^2}{144} \cdot \frac{LN}{12} \times 60 \times 62.5 \times S = 1.7\, D^2LNS. \tag{255}$$

$$D = 0.77 \sqrt{\frac{W}{LNS}}. \tag{256}$$

$$d = D \sqrt{\frac{p + R + 0.433\, H}{Ep}}. \tag{257}$$

$$I = \frac{W\,(p + R + 0.433\, H)\, 2.3}{33{,}000 \times 60 \times E}. \tag{258}$$

In average practice the piston or plunger displacement is made about twice the capacity found by calculation from the maximum amount of water required for the engine, to allow for leakage, steam consumption of the auxiliaries and blowing off.

For pumps with strokes of 12 inches or over, the speed of the plunger or piston is usually limited to 100 feet per minute as a maximum to insure smooth running. For shorter strokes a lower limit should be used. The maximum number of strokes ranges from 100 for strokes over 12 inches in length to 200 for strokes under 5 inches. Boiler-feed pumps should be designed to give the desired capacity at about one-half the maximum number of strokes or less.

Pump slip varies from 2 to 40 per cent, depending upon the condition of the piston and valves and the number of strokes. An average value for piston and plunger pumps in first-class condition is 8 per cent when operating at rated capacity, but it is wise to allow a much larger figure, say 20 per cent, for leakage caused by wear.

The area of the steam cylinder is made from 2 to 2.5 times that of the water end to allow for the various friction losses and the drop in pressure between the pump throttle and the boiler. The total head pumped against includes the suction lift, the friction of valves and fittings, the distance between the suction inlet and the boiler level and the boiler pressure. The excess head varies in practice from 15 to 40 per cent of the boiler pressure; an average figure is 25 per cent. In allowing for the drop in steam pressure between boiler and pump a liberal figure is 25 per cent.

The application of equations (255) to (258), including the practical considerations stated above, is best illustrated by a specific example.

Example 60. Determine the size of direct-acting single-cylinder feed pump necessary to supply water to 1000 horsepower of boilers operating at rated capacity. Gauge pressure 100 pounds per square inch; feed-water temperature 150 deg. fahr.

One horsepower is equivalent to the evaporation of 34.5 pounds of water from and at 212 deg. fahr.; but the pump is usually designed to supply about twice the required amount of water.

Thus $W = 62{,}400$ (under the given conditions).
$S = 0.8$ (by assumption).
$LN = 1200$ (on the basis of 100 feet per minute).

Substitute these values in (256):

$$D = 0.77\sqrt{\frac{62{,}400}{1200 \times 0.8}} = 6.2 \text{ inches, — call it 6 inches,}$$

since the assumptions have been very liberal.

Assume $(0.433\,H + R) = 0.25\,p$ and $E = 0.65$.

Substitute these values in (257):

$$d = 6\sqrt{\frac{100 + 25}{0.65 \times 100}}$$
$$= 8.35, \text{— call it 8.5 inches.}$$

Allowing 100 strokes per minute the length of the stroke must be

$$L = 1200 \div 100 = 12 \text{ inches.}$$

The dimensions of the pump are $8\frac{1}{2}$-in. by 6-in. by 12-in.

The indicated horsepower at maximum load may be obtained by substituting the proper values in (258), thus:

$$I = \frac{62{,}400\,(100 + 25)\,2.3}{33{,}000 \times 60 \times 0.65}$$
$$= 13.9 \text{ i.hp.}$$

297. Steam-pump Governors. — Fig. 383 shows a section through a Fisher pump governor, illustrating a device for maintaining a practically constant pressure in the discharge pipe irrespective of the quantity of water flowing. It embodies a pressure-reducing valve in the steam supply pipe of the pump, actuated by the slight variations in water pressure. When the demand for water increases, the pressure in the discharge pipe tends to decrease, and this drop in pressure (transmitted to the pump governor by suitable piping) causes more steam to be admitted, which increases the speed of the pump. The governor is connected to the steam inlet of the pump at *B* and the steam enters at *A*. Double-balanced valve *C* regulates the supply of steam to the cylinder by the amount it is raised from the seat. The valve is held open by spring *G*, the compression of which may be regulated by hand wheel *K*. The water pressure from the discharge pipe acts on piston *F* and tends to overcome the resistance of the spring. The difference in pressure between the water and the spring determines the position of valve *C*.

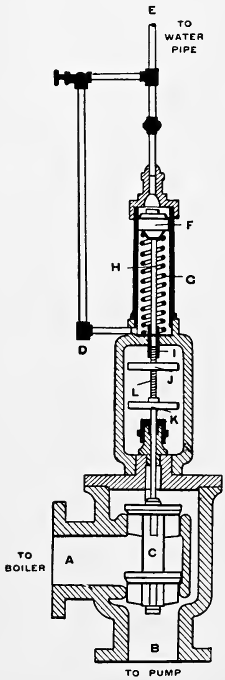


Fig. 383. Fisher Pump Governor.

Piston rod *H* is pinned to sleeve *I* and valve stem *L* screwed into this sleeve by means of hand wheel *K*. Hence, during ordinary operation, the piston, piston-rod sleeve, valve stem, and valve act as a single unit. By turning the hand wheel *K*, valve stem *L* will screw into sleeve *I* and the tension on the spring will be increased. Hand wheel *J* serves as a lock nut and prevents *K* from turning during normal operation.

298. Feed-water Regulators. — The water level in the boiler should be kept as nearly constant as possible, and this necessitates considerable attention on the part of the fireman, especially with fluctuating loads. There are a number of devices on the market which are designed to automatically maintain a constant level, and in many small plants where the duties of the fireman are numerous such devices in connection with high and low water alarms are of considerable assistance. Their action, however, is not always positive on account of wear or sticking of parts, and engineers as a rule prefer to rely upon hand regulation.

Fig. 384 shows a section through a Kitts feed-water regulator, consisting of two parts, the chamber *F* and the regulating valve *V*. The float chamber is connected to the boiler or water column at *O* and *E*, and the regulating valve to the feed main at *R* and to the boiler feed pipe at *W*. When the water in the boiler falls below the mean level, the weight *B* overcomes the counterweight *G* and closes needle valve *L* by means of compound levers. At the same time an extension on valve *L* lifts spring *A* and opens exhaust valve *D*. This removes the steam pressure from the top of diaphragm *C*, in the regulating valve, through the agency of pipe *K*. The pressure from the pump raises the disk *T* and water flows into the boiler until the water rises to the mean level.

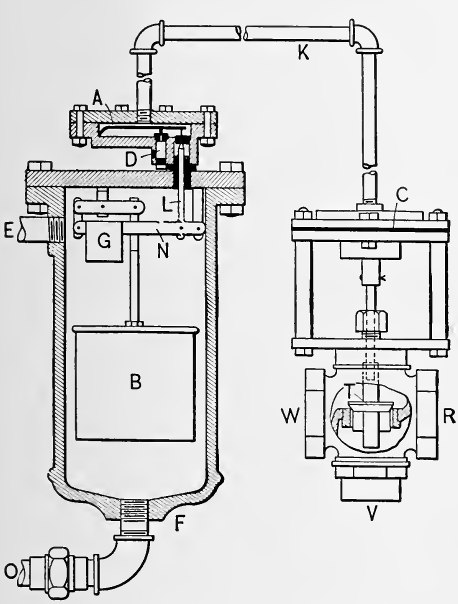


FIG. 384. Kitts Feed-water Regulator. (Counterbalanced-weight Type.)

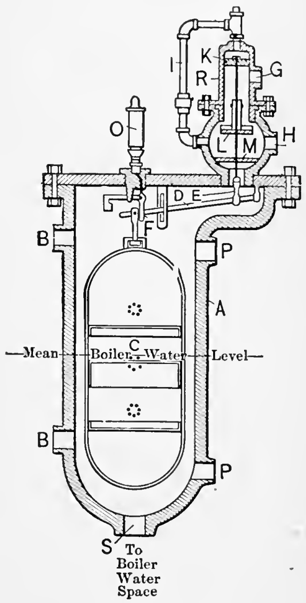


FIG. 385. Rowe Feed-water Regulator. (Float Type.)

When weight *B* becomes submerged its weight is overcome by counterweight *G*, valve *L* is opened and exhaust valve *D* is closed. This admits steam pressure to the diaphragm *C* and forces disk *T* to its seat, cutting off the supply of water to the boiler.

The Rowe feed-water regulator, Fig. 385, depends for its operation on a familiar float-controlled valve mechanism. The vessel *A* is connected to the boiler above and below the water line, and the float *C*, following the water level up and down, actuates a balanced valve in accordance with the boiler-feed requirements. When this apparatus is used to regulate the feed of a single boiler the opening *G* in the valve chamber is connected to the steam space of the boiler and the outlet *H* is carried to the steam inlet of the feed-water pump. When the water level is normal the float closes the valve *L* and thereby cuts off the supply of steam to the pump cylinders. Communication between chambers *A* and *R* is prevented by means of a diaphragm *M*. When the water level falls below normal the float pulls the valve down, opening the way for steam to pass from the inlet *G* to the outlet *H* and thence to the pump. When the regulator is used to control a battery of boilers the pump discharge delivers into the inlet *G* and the water passes through *H* to the boiler-feed main. Should the water level fall beyond a predetermined limit by reason of any accidental discontinuance of the water supply which the apparatus cannot correct, the float would open the valve *F* of the alarm whistle *O* mounted on the top of the main vessel.

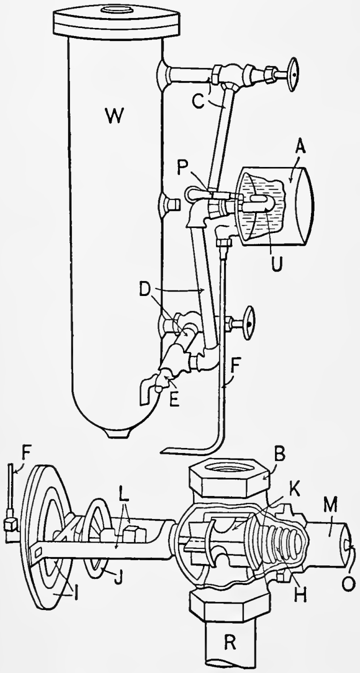


Fig. 386. "S–C" Feed-water Regulator. (Thermo-pressure Type.)

Fig. 386 gives the general details of the "S–C" feed-water regulator which differs from the types just described in the manner of actuating the water regulating valve. A small copper vessel, *A*, partly filled with water, is in communication with diaphragm *I* through the medium of tube *F*. The water in vessel *A* is independent of the boiler supply. A small copper U-tube, *U*, projects into chamber

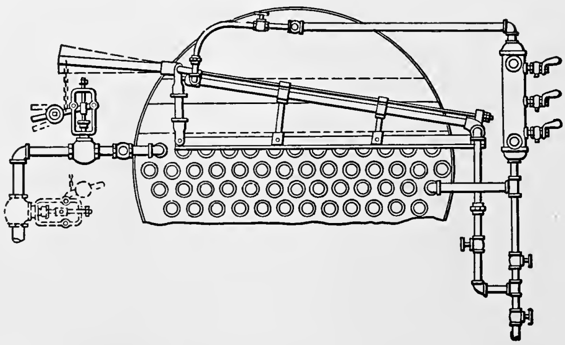


Fig. 387. Copes Feed-water Regulator. (Thermo-expansion Type.)

A, as indicated. When the water in the boiler is at its highest level the U-tube is filled with water and the pump regulator valve *V* is not

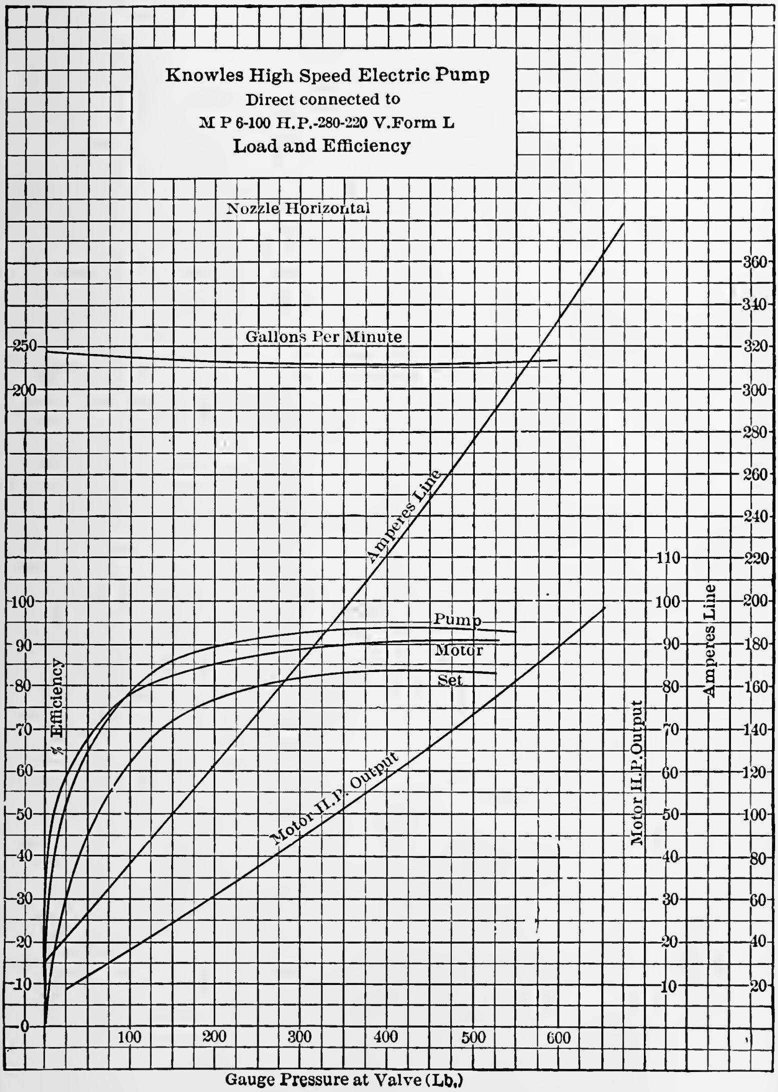

Knowles High Speed Electric Pump
Direct connected to
M P 6-100 H.P.-280-220 V.Form L
Load and Efficiency
Nozzle Horizontal
Gallons Per Minute
Amperes Line
Pump
Motor
Set
Motor H.P. Output
% Efficiency
Motor H.P. Output
Amperes Line
250
200
100
90
80
70
60
50
40
30
20
10
0
110
100
90
80
70
60
50
40
30
20
10
360
340
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
100
200
300
400
500
600
Gauge Pressure at Valve (Lb.)

Fig. 388.

feeding. As the level of the water in the boiler drops, the water recedes from the outer surface of the U-tube, and the upper branch of the tube is surrounded with steam. The steam causes the water in the vessel *A* to boil, and the pressure generated is transmitted through pipe *F* to diaphragm *I*, thereby opening controlling valve *K*. Wheel *J* permits of hand control. Regulators of this type installed in the power plant of the Armour Institute of Technology are giving excellent service.

The Copes feed-water regulator, Fig. 387, depends for its operation upon the expansion and contraction of an inclined tube. As illustrated, this inclined tube is so placed that it contains steam when the water in the boiler is at its lowest level. As the water gradually rises in the boiler it rises in the tube also. When the level of the water is as shown in the illustration the part of the tube filled with steam is at boiler pressure and temperature and that part containing water is at a lower temperature. When an increased load comes on there is a slight drop in steam pressure, accompanied by a more rapid liberation of steam from the entire body of water within the boiler, causing a corresponding increase in its volume and a rise in the boiler water level. This at once raises the level of the water in the expansion tube slightly, increases the amount of the tube submerged in the water and decreases the tube temperature, causing it to shorten. Since this tube is connected by a simple system of levers to a balanced valve in the feed line, shortening of the tube causes the valve to close, so that increase in water level results in a decrease in the rate of feed.

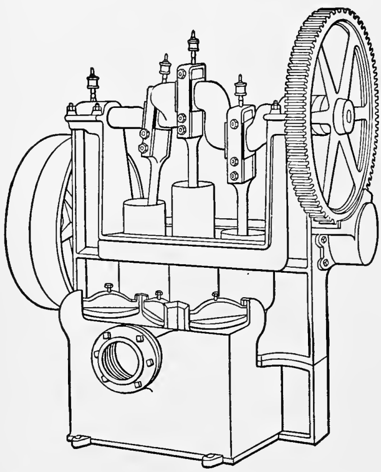


Fig. 389. A Typical Geared Triplex Pump.

299. Power Pumps. — Piston Type. — Piston pumps, geared, belted, or direct connected to electric motors, gas engines, and water motors, are used chiefly where steam power is not available. Their general utility is evidenced by the rapidly increasing number installed in situations formerly occupied by the direct-acting steam pump. The efficiency of this type of pump depends in a large measure upon the character of the driving motor and the efficiency of the transmitting mechanism. High-speed power pumps direct connected to electric motors give efficiencies from line to water horsepower as high as 83 per cent, while the low-speed geared type seldom exceed 70 per cent. The curves in Fig. 388 give the performance of a direct-connected triplex pump, and those in Fig. 390 the performance of a triplex pump geared to an electric motor. Both of these performances are exceptionally good and are considerably above the average.

For a General Treatise on the Design and Operation of Pumping Machinery consult "Pumping Machinery," by A. M. Greene; John Wiley & Sons, 1911.

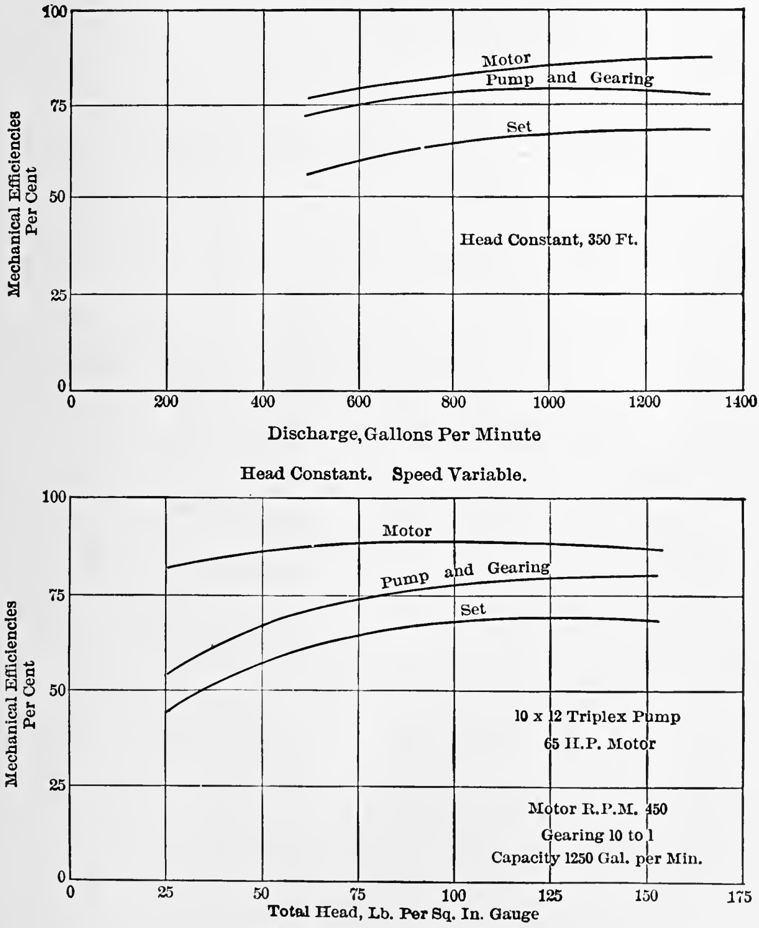


FIG. 390. Performance of a 65-horsepower, Motor-driven Triplex Pump. Geared Type.

300. Injectors. — As a boiler feeder the injector is an efficient and convenient device, cheap and compact, with no moving parts, delivers hot water to the boiler without preheating, and has no exhaust steam to be disposed of. Its adoption in locomotives is practically universal, but in stationary practice it is limited to small boilers or single boilers or as a reserve feeder in connection with pumps. The objections to an injector are its inability to handle hot water, the difficulty of maintaining a continuous flow under extreme variation of load, and the uncertainty of operation under certain conditions. Fig. 391 illustrates the simplest form of single-tube injector. Boiler steam is admitted at *A* and, flowing through nozzle and combining tube to the atmosphere through *G*, partially exhausts the air from pipe *B*, thereby causing the water to rise until it comes in contact with the steam. The steam emerging from nozzle *C* at high velocity condenses on meeting the water

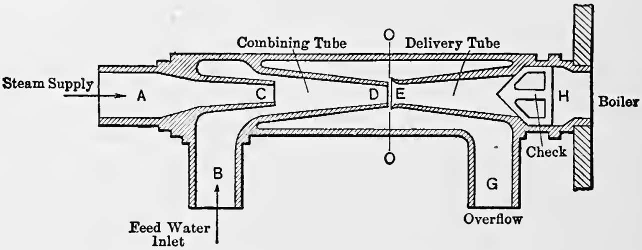


Fig. 391. Elementary Steam Injector.

and imparts considerable momentum to it. The energy in the rapidly moving mass is sufficient to carry it across opening *O*, lift check *H* from its seat and force it into the boiler. The steam then ceases to escape at *G*.

301. Positive Injectors. — Fig. 392 shows a section through a Hancock injector, illustrating the principles of the double-tube positive type. Its operation is as follows: Overflow valves *D* and *F* are opened and steam is admitted, which at first passes freely through the overflow to the atmosphere and in so doing exhausts the air from the suction pipe. This causes the feed water to rise until it meets the jet of steam and the two are forced through the overflow. As soon as water appears at the overflow, valve *D* is closed, valve *C* partially opened, and valve *F* closed. This admits steam through the forcing jet *W* and, the overflow valves being closed, the water is fed into the boiler. In case the action is interrupted for any reason it is necessary to restart it by hand.

The chief advantage of the double-tube positive type lies in its ability to lift water to a greater height and to handle hotter water than the single-tube. Its range in pressure is also greater, that is, it will start with a lower steam pressure and discharge against a higher back pressure. Double-tube injectors are used almost exclusively in locomotive work.

302. Automatic Injectors. — Fig. 393 shows a section through the Penberthy injector. Its operation is as follows: Steam enters at the top connection and blows through suction tube *c* into the combining tube *d* and into chamber *g*, from which it passes through overflow valve *n* to the overflow *m*. When water is drawn in from the suction intake and begins to discharge at the overflow, the resulting condensation of the steam creates a partial vacuum above the movable ring *h* and the latter is forced against the end of tube *c*, cutting off the direct flow of water to the overflow. The water then passes into the boiler. Spill

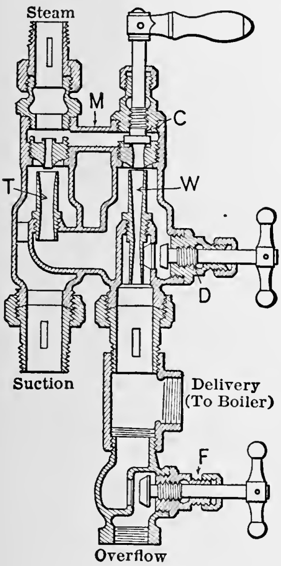


Fig. 392. Hancock Double-tube Injector.

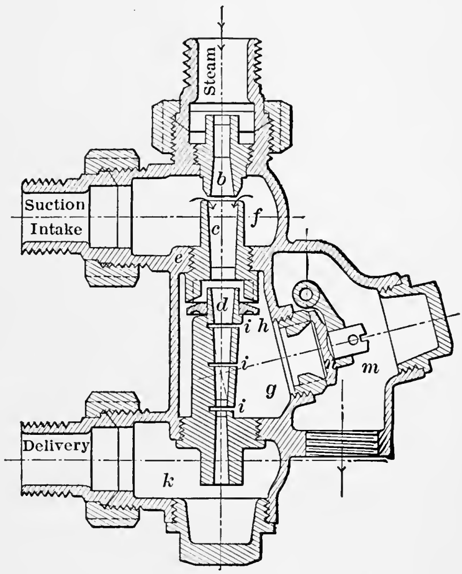


Fig. 393. Penberthy Automatic Injector.

holes *i*, *i*, *i* are for the purpose of relieving the excess of water until communication with the boiler has been established. The action of opening and closing the overflow is entirely automatic. Where the conditions are not too extreme the automatic injector is to be preferred for stationary work because of its restarting features. It is also used on traction, logging, and road engines, where its certainty of action and special adaptability render it invaluable for the rough work to which such machines are subjected.

Injectors, Theory of: Trans. A.S.M.E., 10–339; Sibley Jour., Dec., 1897, p. 101; Power, May, 1901, p. 23; Thermodynamics of the Steam Engine, Peabody, Chap. IX; Theory of the Steam Injector, Kneass.

Injectors, General Description: Engr. U. S., Oct. 1, 1907, Nov. 15, 1907, July 15, 1904, p. 501, Feb. 2, 1903, p. 151; Power, Aug. 1906, p. 478; Engr., Lond., March 10, 1905, p. 244; Engineering, Aug. 30, 1895, p. 281.

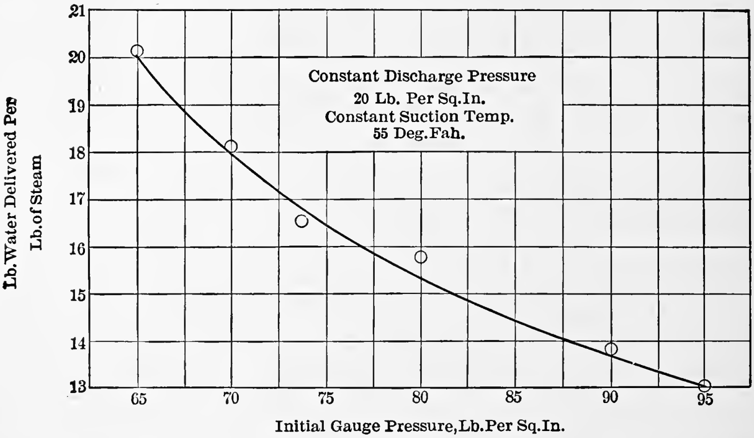


FIG. 394. Performance of an Automatic Injector with Varying Initial Pressure.

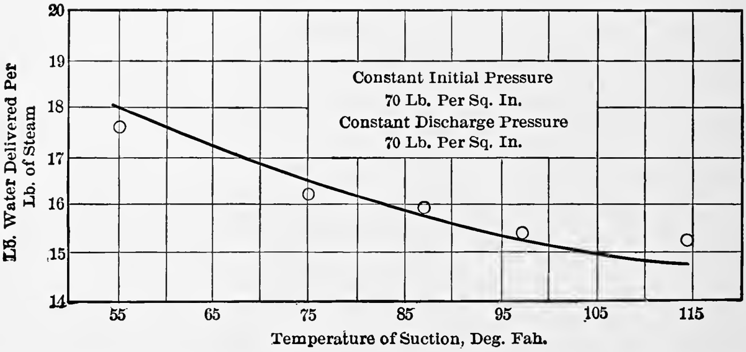


FIG. 395. Performance of an Automatic Injector with Varying Suction Temperature.

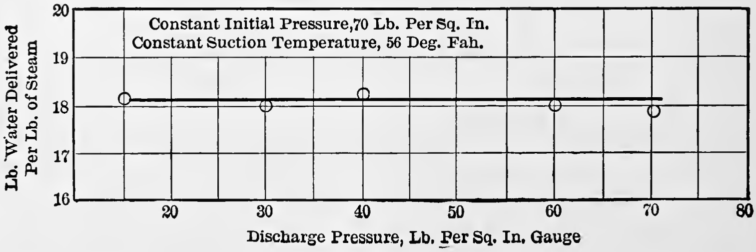


FIG. 396. Performance of an Automatic Injector with Varying Discharge Pressure.

303. Performance of Injectors. — The performance of an injector may be very closely determined from the equation

$$w = \frac{xr + q - t + 32}{t - t_0} \quad \text{(Kneass, "Theory of the Injector," p. 83),} \tag{259}$$

in which

w = pounds of water delivered per pound of steam supplied,
x = quality of the steam supplied,
r = heat of vaporization,
q = heat of the liquid,
t = temperature of the discharge water,
t_0 = temperature of the suction water.

Figs. 394, 395, and 396 give the performance of a Desmond automatic injector as tested at the Armour Institute of Technology. The results check very closely with those calculated from above equation. Referring to Fig. 394 it will be seen that the weight of water delivered per pound of steam decreases as the initial pressure is increased, all other factors remaining the same. From Fig. 395 it will be noted that the weight of water delivered per pound of steam decreases as the temperature of suction supply is increased up to a point where the injector "breaks" or becomes inoperative. This critical temperature varies with the different types of injectors, being highest for the double-tube

TABLE 114.

RANGE IN WORKING PRESSURES.

Standard " Metropolitan " Steam Injectors.

Suction Temperature, Deg. Fahr.	Automatic.				
	Suction Head, Feet.				
	2	8	14	20	Under Pressure.
Under 60	25 to 150	30 to 130	42 to 110	55 to 85	20 to 160
100	26 to 120	33 to 100	55 to 80	....	25 to 125
120	....	....	....	....	26 to 85
140	....	....	....	....	....

Suction Temperature, Deg. Fahr.	Double Tube.				
	Suction Head, Feet.				
	2	8	14	20	Under Pressure.
Under 60	14 to 250	23 to 220	27 to 175	42 to 135	14 to 250
100	15 to 210	26 to 160	37 to 120	46 to 70	15 to 210
120	20 to 185	30 to 120	42 to 75	....	20 to 185
140	20 to 120	35 to 70	....	....	20 to 120

type, but seldom exceeds 160 deg. fahr. Fig. 396 shows that the weight of water delivered per pound of steam is practically constant for all discharge pressures within the limits of the apparatus.

Table 114 gives the range of working steam pressures for standard "Metropolitan" injectors with varying suction heads and temperatures, and, though strictly applicable to this particular type only, is characteristic of all makes.

In selecting an injector the following information is desirable for best results:

1. The lowest and highest steam pressure carried.
2. The temperature of the water supply.
3. The source of water supply, whether the injector is used as a lifter or non-lifter.
4. The general service, such as character of the water used, whether the injector is subject to severe jars, etc.

304. Injector vs. Steam Pump as a Boiler Feeder. — From a purely thermodynamic standpoint the efficiency of an injector is nearly perfect, since the heat drawn from the boiler is returned to the boiler again, less a slight radiation loss. As a pump, however, the injector is very inefficient and requires more fuel for its operation than very wasteful feed pumps. This is best illustrated by an example:

Example 61. Compare the heat consumption of a high-grade injector with that of an ordinary duplex boiler feed pump when feeding water to a boiler. Make all necessary assumptions. An injector of modern construction will deliver say 15 pounds of water to the boiler per pound of steam supplied, with delivery temperature of 150 deg. fahr. This corresponds to a heat consumption of 71.3 B.t.u. per pound of water delivered, thus:

With initial pressure of 115 pounds absolute,

$$H = 1188.8.$$

Heat of the water delivered to the boiler,

$$150 - 32 = 118 \text{ B.t.u. above 32 deg. fahr.}$$

Heat of 1 pound of steam above a feed temperature of 150 deg. fahr.,

$$1188.8 = 118 = 1070.8 \text{ B.t.u.}$$

Heat required to deliver 1 pound of water to the boiler,

$$\frac{1070.8}{15} = 71.3 \text{ B.t.u.}$$

A simple direct-acting duplex pump consumes say 200 pounds steam per i.hp-hour. Assume the extreme case where the exhaust steam will not be used for heating the feed water and the latter is fed into the boiler at 60 deg. fahr.

The heat supplied to the pump per i.hp-hour,

$$200 \{1188.8 - (60 - 32)\} = 232{,}160 \text{ B.t.u.}$$

Assuming the low mechanical efficiency of 50 per cent, the heat required to develop one horsepower at the water end will be

$$232{,}160 \div 0.50 = 464{,}320 \text{ B.t.u. per hour.}$$

Since the steam pressure is 100 pounds gauge, the equivalent head of water at 60 deg. fahr. is

$$2.3 \times 100 = 230 \text{ feet.}$$

Assume the friction in the feed pipe, the resistance of valves, etc., to be 30 per cent of the boiler pressure; the total head pumped against will be

$$230 + 69 = 299, \text{ say } 300 \text{ feet,}$$

$$1 \text{ horsepower-hour} = 1{,}980{,}000 \text{ foot-pounds per hour,}$$

$$\frac{1{,}980{,}000}{300} = 6600 \text{ pounds;}$$

that is, 1 horsepower at the pump will deliver 6600 pounds of water per hour to the boiler against a head of 300 feet.

The heat consumption per pound of water delivered,

$$\frac{464{,}320}{6600} = 70.3 \text{ B.t.u.}$$

If the feed water is heated to say 210 deg. fahr. by the exhaust steam from the pump, the heat consumption will be 63.7 B.t.u. as against 70.3 without the heater.

Thus even in this extreme case of poor steam-pump performance the heat consumption lies in favor of the pump. With the better grades of pumps this disparity is considerably greater, and decidedly so if the exhaust steam is used to preheat the feed water. For intermittent operation the condensation losses in the pump may more than offset this gain. Other conditions, however, such as compactness, low first cost, and ease of operation are oftentimes considerations and the heat consumption is of minor importance.

305. Vacuum Pumps. — The different types of vacuum pumps employed in steam power plant practice may be divided into four general classes:

1. Wet-air pumps.
2. Tail pumps.
3. Dry-air pumps.
4. Condensate pumps.

(1) *Wet-air pumps* are for the purpose of withdrawing water and noncondensable gases from apparatus under less than atmospheric pressure. Standard low level jet-condenser wet-air pumps handle simultaneously

the circulating water, condensate, and all entrained air and are, in fact, a combination of circulating pump and vacuum pump. Surface condenser wet-air pumps deal with the condensate and its air entrainment. Wet-air pumps may be of the reciprocating, centrifugal, rotary jet, rotary positive displacement and steam jet type.

(2) The terms "wet-vacuum pump," "wet-air pump," and "tail pump" are often used synonymously but in order to differentiate between pumps handling injection water, condensate and air from those dealing only with the injection water and condensate the term "wet-air pump" has been applied to the former and "tail pump" to the latter.

(3) *Dry-air pumps* are for the purpose of withdrawing the non-condensable gas from apparatus under a vacuum and discharging it against atmospheric or greater pressure. They are to all intents and purposes, air compressors. The term "dry air" is a misnomer since the gases exhausted are almost invariably saturated with water vapor. These pumps may be of the reciprocating, rotary, positive displacement, hydro-centrifugal and steam jet types.

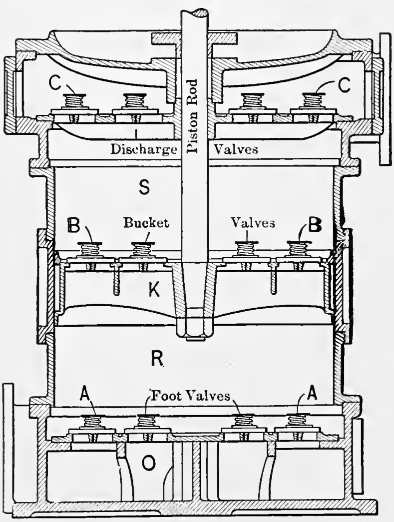


Fig. 397. Dean Air Pump.

(4) Condensate pumps are for the purpose of withdrawing condensed steam from surface condensers and are usually of the reciprocating, rotative or centrifugal types.

306. Wet-air Pumps for Jet Condensers. — Fig. 397 shows a section of the cylinder of a Dean twin-cylinder wet-air pump as applied to a standard low-level jet condenser and illustrative of the reciprocating type. There are three sets of valves, the suction or foot valves *A*, *A*, the lifting or bucket valves *B*, *B*, and the head or discharge valves *C*, *C*. On the upward stroke of the piston or bucket a partial vacuum is formed in the chamber between the bucket and the lower head, causing the water and air in the bottom of the barrel to lift the foot valves *A*, *A* from their seats and flow into the cylinder. On the downward stroke the foot valves *A*, *A* close and water and air are entrapped in chamber *R* between the lower head and the bucket. As the bucket descends, the pressure of air in the cylinder lifts the bucket valves *B*, *B* from their seats and permits the air and water to escape to the upper portion *S* of the cylinder between the head plate and the bucket. On the next upward stroke the water and air are forced through the discharge valves

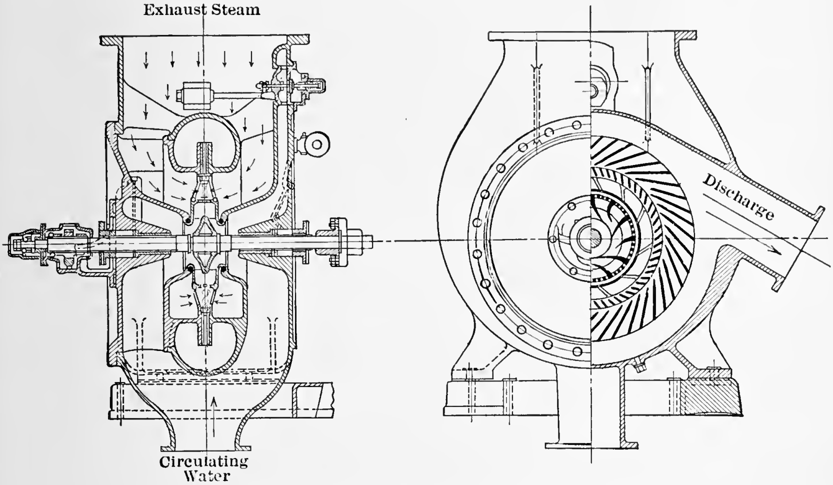


FIG. 398. Rees "Roturbo" Jet Condenser.

C, C into the hot well. This discharge of water and air from the top compartment is simultaneous with influx of water and air in the lower chamber.

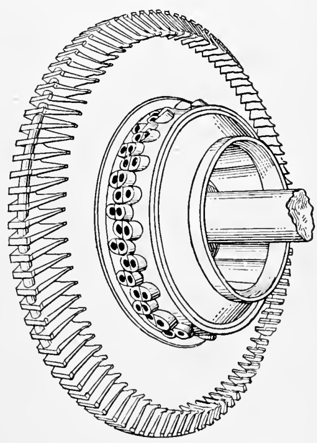


FIG. 399. Impeller for Rees "Roturbo" Jet-condenser Pump.

Fig. 398 shows a vertical section and sectional end elevation of a Rees Roturbo rotary jet condenser illustrating an adaptation of the rotary-jet pump as a jet condenser. This pump is a development of a special type of centrifugal pump the unique feature of which is the employment of a revolving pressure chamber. The hollow impeller, Fig. 399, lifts the circulating water in much the same manner as in any centrifugal pump. The space between the periphery of the impeller and the inner circumference of the fan wheel forms the mixing chamber in which the exhaust steam is brought into contact with radial jets of water. The fan wheel itself acts as an ejector and exhausts the mixture of circulating water and vapor. The operation is as follows: circulating water is drawn through the suction pipe into the revolving pressure chamber, on the periphery of which nozzles are arranged as shown in Fig. 399, and is forced through the nozzles in radiating jets which are arranged to impinge in pairs. The water jets, which are made fan shaped and subdivided into a fine spray, are projected in lines radiating from the shaft (but still rotating as a whole with the impeller) across a space into which the exhaust steam blows. The circulating water leaving the nozzles, condensate, and air entrainment are picked up by the blades of the fan and discharged through a volute guide chamber to the hot well.

The Connersville jet condenser is a typical example of an application of a rotary positive-displacement wet-air pump. In this device the circulating water, condensate, and air entrainment are handled by a Connersville cycloidal 3-lobe type rotary pump. (A cross section through a typical 2-lobe cycloidal pump is shown in Fig. 424.)

The steam-jet type of wet-air pump is exemplified in the ejector condenser. See paragraph 237.

306a. Wet-air Pumps for Surface Condensers. — These pumps exhaust the condensate and air entrainment from surface condensers. The vacuum pumps of a steam heating system come also under this head.

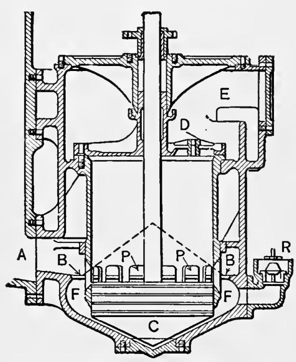


FIG. 400. Edwards Air Pump.

The Edwards air pump, Fig. 400, is a typical example of a wet-air pump of the reciprocating type. Referring to Fig. 400, the condensed steam flows continuously by gravity from the condenser into the base of the pump through passage *A* and annular space *B*. As the piston *C* descends it forces the water from the lower part of the casing *F* into the cylinder proper through the ports *P*, *P*. On the upward stroke the ports in the piston are closed and the air and water discharged through head valves *D* and exhaust port *E* to the hot well. The seats of valves *D* are constructed with a rib between each valve and a lip around the outer edge, so that each valve is water-sealed independently of the others. In ordinary air pumps the clearance between the bucket and head valve seat is necessarily large, due to the space occupied by the bucket valves and the ribs on the under side of the valve seating. This clearance space reduces the capacity of the pump, since the air above the bucket must be compressed above atmospheric pressure before it can be discharged, and on the return stroke will expand and occupy a space which should be available for a fresh supply of air from the condenser. In the Edwards air pump the clearance space is reduced to a minimum, since there are no bucket valves to limit it. The absence of suction or foot valves still further increases the capacity of the pump for similar reasons. These pumps are arranged either single, double, or triplex; steam, electric, or belt driven; slow or high speed.

Fig. 401 shows a partial axial and an end section through a C. H. Wheeler & Co.'s high-vacuum "Rotrex" pump. This pump is of the wet-vacuum type and handles both air and water of condensation but it is also adapted for dry air purposes. The apparatus consists of a cylindrical casing and a rotor mounted eccentrically on the shaft. This shaft is carried in outboard ring oil bearings which are entirely independent of the stuffing boxes. The division between the suction and discharge space in the pump cylinder is maintained by a radius cam carried on a shaft independent of the stuffing boxes. This cam is operated from the rotor shaft by a lever and crank on the outside of the casing. The clearance spaces are water sealed. The discharge

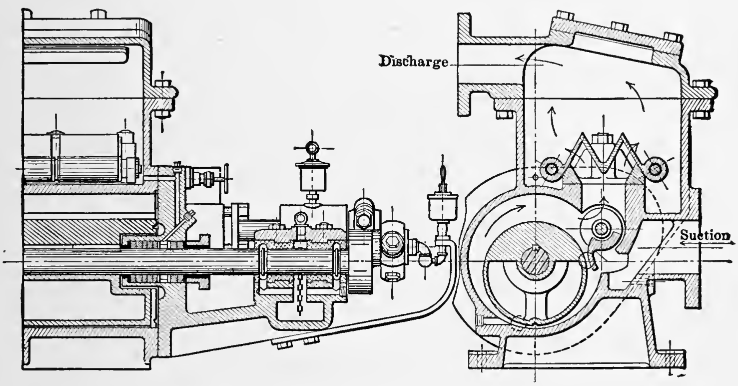


Fig. 401. High-vacuum "Rotrex" Pump.

valves are of the Gutermuth type. Pump speed 200 to 300 r.p.m. The manufacturers guarantee that on dead-end test a vacuum may be obtained within one half inch of the barometer, and within one inch of the barometer under operating conditions.

307. Size of Wet-air Pumps. — Since the wet-air pump for jet condenser must deal with the mixture of injection water, condensate, and all air entrainment, the problem of design is essentially that of determining the volume of mixture to be withdrawn under condenser pressures and temperatures. The volume of injection water and condensate for a given set of conditions may be readily calculated, but the volume of air entrained with the injection water and condensate and that introduced by leakage is an unknown quantity and can only be estimated. The amount of air mechanically mixed with the injection may vary from 1 to 5 per cent by volume at atmospheric pressure and temperature. The amount of air in feed water varies from less than 1 per cent by volume, if the heater is of the open type, to 5 per cent or more if the heater is of the closed type and raw water is fed directly into the heater. Air leakage is an unknown quantity varying within wide limits and is dependent upon the tightness of joints, stuffing boxes and the like. A very liberal factor is usually allowed in estimating total air entrainment, an average figure being about 10 per cent by volume of the circulating water for the combined air and wet-vacuum pump for jet condensers and 10 per cent by volume of the feed water for surface condensers.

Let Q = total volume of air and water in cubic feet per hour to be handled by the pump,

V = volume of cooling water in cubic feet per hour,

v = volume of condensed steam in cubic feet per hour,

v_a = volume of air at pressure p_a and temperature t_a,

t_a = temperature of the air entering the condenser, deg. fahr.,

t_2 = temperature of the discharge water, deg. fahr.,

t_0 = initial temperature of the cooling water, deg. fahr.,

p_a = atmospheric pressure, pounds per square inch,

p_c = total pressure in the condenser, pounds per square inch,

p_v = pressure of aqueous vapor at temperature t_2,

then $(V + v)$ = volume of water to be pumped from the condenser per hour.

The air entering the condenser will be increased in volume on account of the reduction in pressure and the increase in temperature. If v_a is the original volume under pressure p_a and temperature t_a the final volume on entering the condenser is

$$\text{Final volume} = v_a \frac{p_a}{p_c - p_v} \times \frac{t_2 + 460}{t_a + 460}, \tag{260}$$

and the total volume to be exhausted per hour by the pump is

$$Q = V + v + v_a \frac{p_a}{p_c - p_v} \times \frac{t_2 + 460}{t_a + 460}. \tag{261}$$

Example 62. Estimate the piston displacement of a wet-air pump suitable for average reciprocating engine practice.

Under average conditions of reciprocating-engine practice the hot-well temperature is about 110 deg. fahr. and the absolute back pressure 4 inches of mercury. Assuming 70 deg. fahr. as the initial temperature of the circulating water and allowing 10 per cent as the air entrainment,

$$\begin{array}{lll} p_a = 29.92 & t_0 = 70 & v = 0.04\,V \\ p_c = 4 & t_2 = 110 & v_a = 0.1\,V. \\ p_v = 2.59 & t_a = t_0 = 70 & \end{array}$$

Substitute these values in (261)

$$Q = V + 0.04\,V + 0.1\,V \frac{29.92}{4.0 - 2.59} \times \frac{110 + 460}{70 + 460}$$
$$= 3.3\,V.$$

Average practice gives 3 V as the pump displacement per hour for a single-acting pump and 3.5 V for a double-acting pump, the cylinders being ordinarily proportioned on a piston velocity of 50 feet per minute at rated capacity.

Wet-air pumps are usually independently driven, making it possible to vary the speed of the pump irrespective of the engine speed and to create a vacuum before starting the engine. Occasionally, however, when the load is constant, as in pumping-engine practice, the pump may be driven by the main engine.

The combined air, condensate and circulating pump (with the exception of pumps of the Rees "roturbo jet" type) is not adapted for high-vacuum work on account of the enormous increase in air volume at very low pressures. With cold injection water and a good air-tight condensing system vacua as high as 2 inches absolute are possible with the standard type of jet condenser air pumps but practice recommends the use of separate air and wet-vacuum pumps for vacua higher than 26 inches.

Since the wet-air pump for surface condenser handles only the condensed steam and air, its theoretical capacity, neglecting clearance, may be determined by eliminating V from equation (261) which then becomes

$$Q = v + v_a \frac{p_a}{p_c - p_v} \times \frac{t_2 + 460}{t_a + 460}. \qquad (262)$$

The volume of air entering the condenser varies so much with the character of the power-plant equipment and the conditions of operation that any assumed average value of v_a may lead to serious error.

Average steam turbine practice gives

$Q = 20\ v$ for 26-inch vacuum,
$Q = 30\ v$ for 27-inch vacuum,
$Q = 40\ v$ for 28-inch vacuum,
$Q = 50\ v$ for 29-inch vacuum.

Average reciprocating engine practice gives

$Q = 85$ per cent of above for vacua up to 27 inches.

308. Tail Pumps.—As previously stated the term "tail" pump has been applied to pumps which deal with the combined circulating water and condensate merely to distinguish between this type and that dealing with the entire condenser water supply including the air entrainment. In practice the terms tail pump and wet-air pump are used synonymously. Almost any type of water pump may be used for the purpose of withdrawing the combined circulating water and condensates but the centrifugal pump appears to be the more common in use. Quite recently the "screw-pump" has been developed to a high point of efficiency and it is not unlikely that this type may supplant to a certain extent the present type of centrifugal pump. A typical tail pump installation is shown in Fig. 295. The Leblanc jet condenser,

Fig. 294, and the C. H. Wheeler low-head high-vacuum jet condenser, Fig. 296, involve the use of centrifugal tail pumps. The power required to drive this style of pump may be calculated from equation (263). In this connection the total head pumped against must include the suction head due to the vacuum in the condenser.

309. Dry-air or Dry-vacuum Pumps. — Dry-air or dry-vacuum pumps are used in connection with jet or surface condensers where a high degree of vacuum is essential as in steam turbine practice. Such pumps are intended to exhaust the saturated non-condensable vapors only. Air pumps for jet condensers must deal with much larger volumes of air than those for surface condensers, other things being equal, because of the air entrained with the circulating water. Dry-air pumps may be divided into four general groups: (1) the reciprocating piston, (2) positive rotary displacement, (3) hydro-centrifugal and (4) steam-jet.

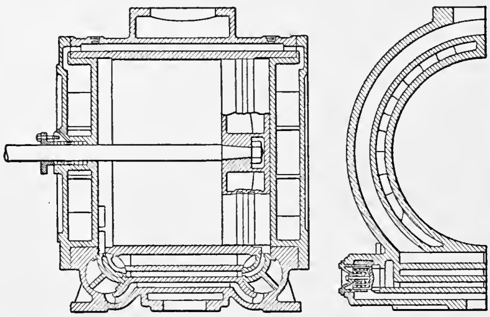


Fig. 402. Air Cylinder Construction of Wheeler Dry-vacuum Pump.

Fig. 402 shows a section through the cylinder of a Wheeler dry-vacuum pump illustrating the single-cylinder, single-stage reciprocating piston group. The admission valves A and A are mechanically controlled and the discharge valves are of the usual spring loaded type. The rotary admission valves are adjusted so that for a short instant at dead center communication is established between both ends of the cylinder so as to reduce the air pressure in the clearance space down to the suction pressure on the other side of the piston.

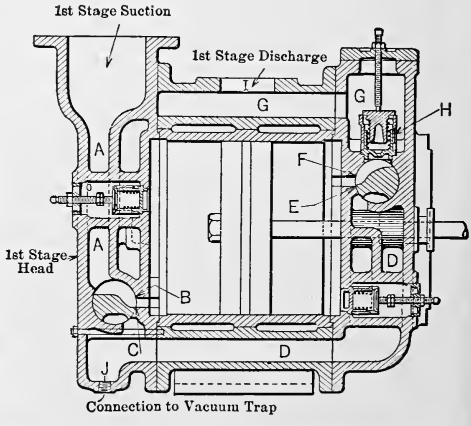


Fig. 403. Air Cylinder Construction of Worthington Two-stage Single-cylinder Dry-vacuum Pump.

Fig. 403 shows a section through the cylinder of a Worthington single-cylinder two-stage dry-vacuum pump and which possesses many advantages over the single-cylinder mechanism. The cycle of operation is as follows: With piston moving as indicated air is drawn into the head-end of the cylinder until the piston reaches the end of its stroke. On the return stroke the air drawn in the head end of the cylinder is

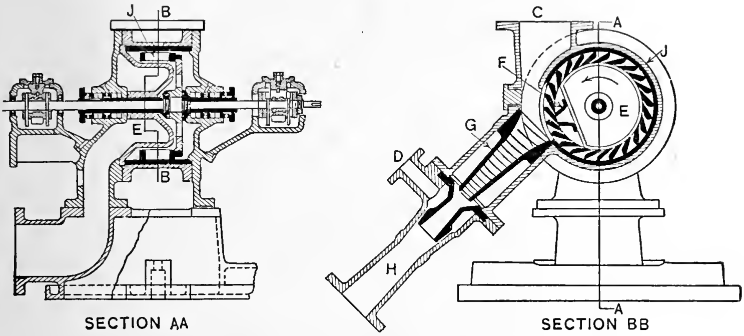


FIG. 404. Leblanc Air Pump.

transferred (at condenser pressure) through passage D and valve E to the crank end of the cylinder. On the next stroke the air charge is compressed through spring loaded valve H to somewhat more than atmospheric pressure.

The Leblanc, Thyssen, Wheeler Turbo-air Pump and the Worthington Hydraulic Vacuum Pumps are well-known examples of the hydro-centrifugal or hurling-water dry-air pumps. They differ very little

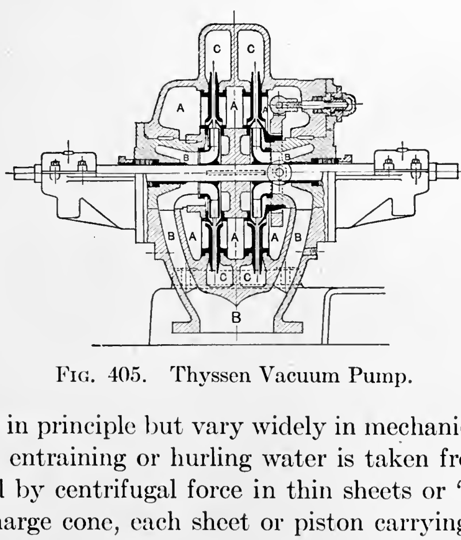


FIG. 405. Thyssen Vacuum Pump.

from each other in principle but vary widely in mechanical construction. In these pumps entraining or hurling water is taken from a circulating tank and hurled by centrifugal force in thin sheets or "pistons" into a diffuser or discharge cone, each sheet or piston carrying with it a layer of saturated air drawn in from the condenser. The water is used over and over again since very little heat is abstracted from the air. This style of pump is in common use and has superseded the reciprocating pump to a great extent. It may be driven by motor or turbine, is very compact, and owing to the absence of valves and reciprocating parts requires very little attention. The power requirements, however, are from two to three times that for a reciprocating pump.

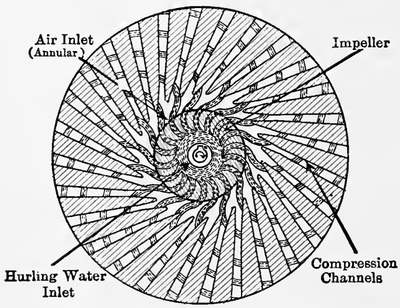


Fig. 406. Diagrammatic Arrangement of Elements in Wheeler Turbo-air Pump.

Fig. 408 shows an application of the Parsons augmenter which is one of the earliest applications of a steam jet for withdrawing the non-condensable vapors from a condenser. Referring to the illustration, a pipe is led from the bottom of the main condenser to an auxiliary or augmenter having about one-twentieth of the cooling surface of the main condenser. At the point indicated a small jet is provided which acts as an ejector and draws out the air and vapor from the condenser and delivers it to the air pump. The water seal prevents the air and vapor from returning to the condenser. With this arrangement if there is a vacuum of $27\frac{1}{2}$ or 28 inches in the condenser there need be only 26 at the air pump, which therefore may be of smaller size, the jet compressing the air and vapor and the augmenter condenser cooling them so that the volume is reduced about one half. The steam jet uses about $1\frac{1}{2}$ per cent of the steam used by the prime mover at full load. The net saving on the average condenser due to the use of the augmenter averages 5 per cent; with light condensers the saving is negligible. The kinetic ejector is a development of the Parsons vacuum augmenter but since it is little used in this country no attempt will be made to describe it. A notable installation of the kinetic ejector is in the Fisk Street Station of the Commonwealth Edison Company, Chicago, Illinois.

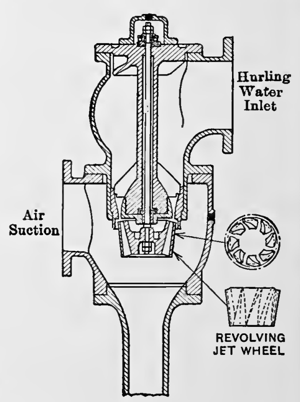


Fig. 407. Worthington Hydraulic Vacuum Pump.

Fig. 409 shows a general assembly of the C. H. Wheeler "Radojet" pump which is the latest development of the steam jet for vacuum purposes and which promises to supersede the hydro-centrifugal pump for general condenser practice. This device consists essentially of a compound live-steam jet; a primary jet which withdraws the saturated air from the condenser and compresses it to four or five inches above condenser pressure and a secondary jet which picks up the discharge from the primary and forces it out against atmospheric pressure. By forcing the discharge into an open feed-water heater the latent heat of steam used by the jets may be reclaimed. The primary jet is effected by a number of small expanding nozzles discharging into a conical diffusing chamber. The secondary jet is radial in form and discharges into an annular volute chamber. There are no moving parts and the apparatus is very compact and simple. The same degree of vacuum may be developed for identical operating conditions as with the hydro-centrifugal air-pump and at a lower power cost.

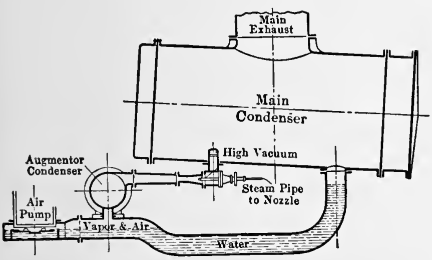


Fig. 408. Parsons Vacuum Augmenter.

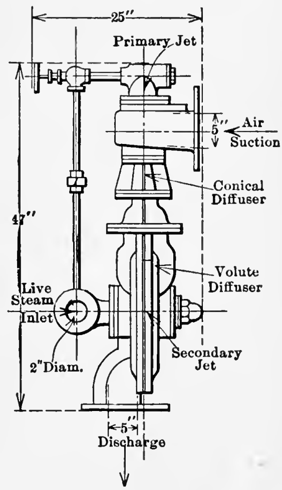


Fig. 409. C. H. Wheeler "Radojet" Dry-vacuum Pump.

310. Size of Dry-air Pumps. — The volumetric capacity of a dry-air pump for condenser service is based upon experience rather than theory because the amount of air in the steam and the air infiltration are very uncertain quantities. Since the air to be dealt with is saturated with water vapor the pump displacement or its equivalent will be much larger than if dry air only were supplied. The volume of mixture which must be exhausted for a given weight of dry air for different vacua and air-pump suction temperatures is shown in Fig. 410. The curves are based on equation (260) and give the volume of mixture containing one pound of dry air at various condenser pressures and corresponding saturated vapor temperatures. The great reduction in volume effected by cooling the air-pump suction is clearly shown. The marked superiority of counter current over parallel current flow for high vacua is chiefly due to the greater reduction in temperature of the air and its vapor content.

The following capacities for dry-air pumps appear to conform with current practice:

Q = 20 v to 30 v for vacua under 27 inches.

Q = 35 v to 50 v for vacua of 28 inches or over, both referred to a 30-inch barometer.

Q = air-pump displacement, cu. ft. per hr.

v = volume of condensate, cu. ft. per hr.

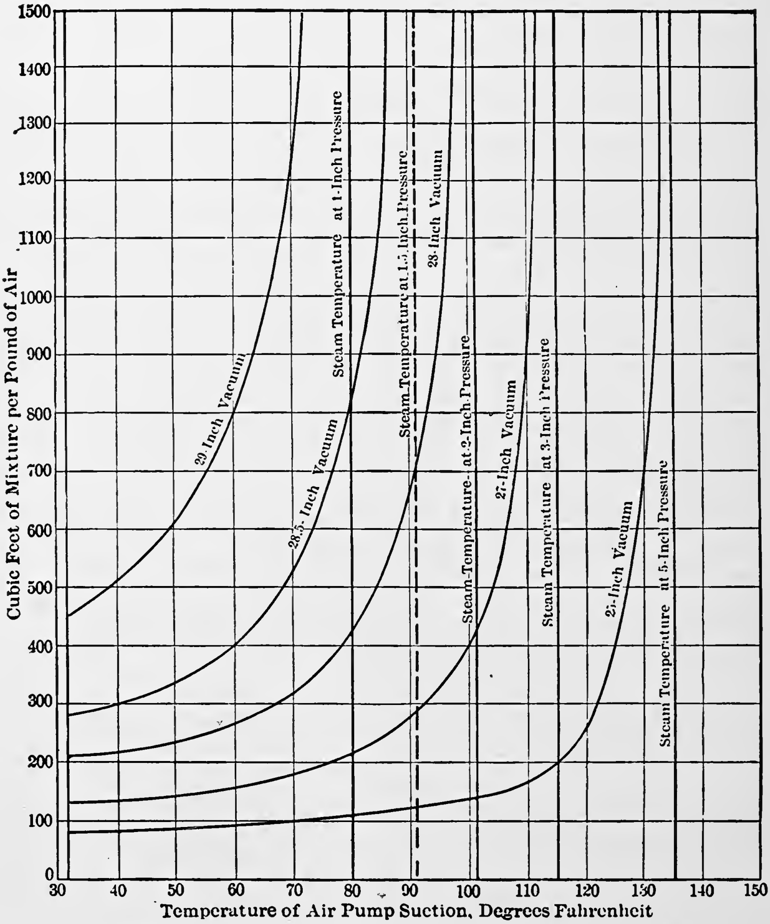


FIG. 410. Cubic Feet of Saturated Air Containing One Pound of Dry Air for Various Vacua and Air Temperatures.

In a number of recent large condenser installations the air pumps are proportioned on a basis of Q = 50 v.

The curves in Fig. 411, though strictly applicable to a specific case, represent the general characteristics of an ordinary reciprocating vs.

a hydro-centrifugal air-pump. Referring to the curves, it will be seen that the reciprocating pump is superior to the hydro-centrifugal for vacua below the line BB and for vacua above BB the latter is the more

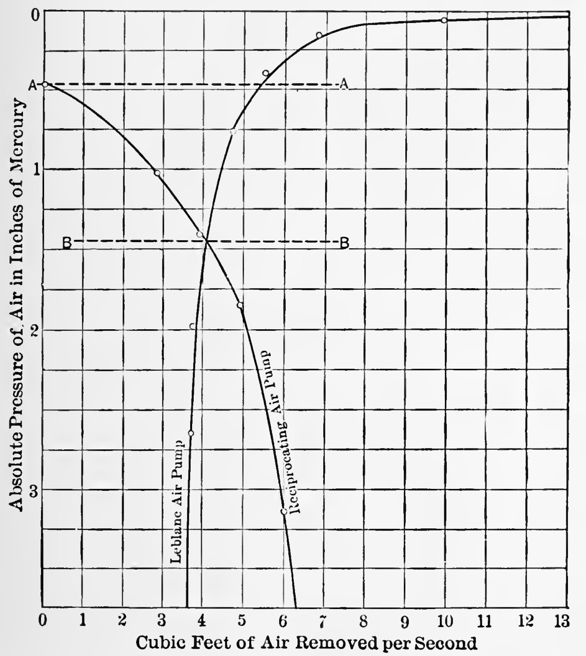


FIG. 411. Comparative Tests — Reciprocating Air Pumps vs. Leblanc Air Pumps.

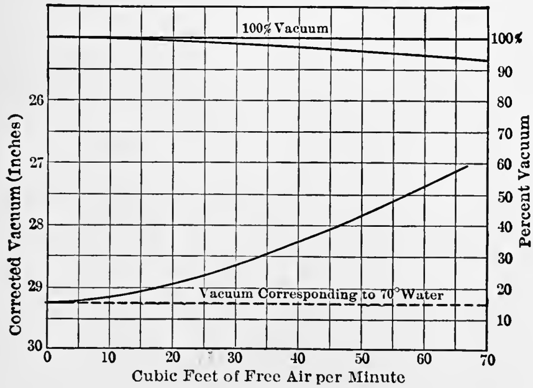


FIG. 412. Test of Wheeler Turbo-air Pump.

effective. For vacua above AA the hydro-centrifugal pump is in a class of its own. With tight condensers in which air leakage is kept to a minimum a reciprocating air-pump of the Worthington two-stage single-cylinder type (Fig. 403) may maintain a higher vacuum than the hydro-centrifugal type for the same temperature range.

311. Centrifugal Pumps. — Centrifugal pumps consist of two essential elements, (1) a rotary impeller which draws in the water at its center and (2) a stationary casing which guides the water thrown from the ends of the impeller to the discharge outlet. Increase of peripheral speed increases the energy in the impeller. This increase in energy may take the form of increase in pressure or potential energy, or it may be in the form of increase in rate of flow or kinetic energy. In general there is an increase in both kinetic and potential energy. The impeller may be of the open type, Fig. 414 (B), or closed, Fig. 414 (A). The casing may be cylindrical and concentric with the impeller, Fig. 418, or of spiral form, Fig. 413. It may be plain or fitted with diffusion vanes and any number of impellers may be employed. The shape of the impeller and casing and the number of impellers or stages determine the efficiency of the pump and its adaptability to certain conditions of service.

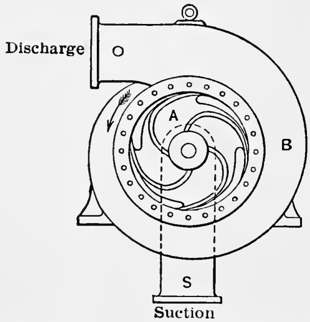


FIG. 413. A Typical Centrifugal Pump.

Centrifugal pumps are generally classified as

1. Volute.
2. Turbine.

Fig. 413 gives an end view of a typical single-stage volute pump with end plate removed so as to expose the impeller, and Fig. 415 shows a

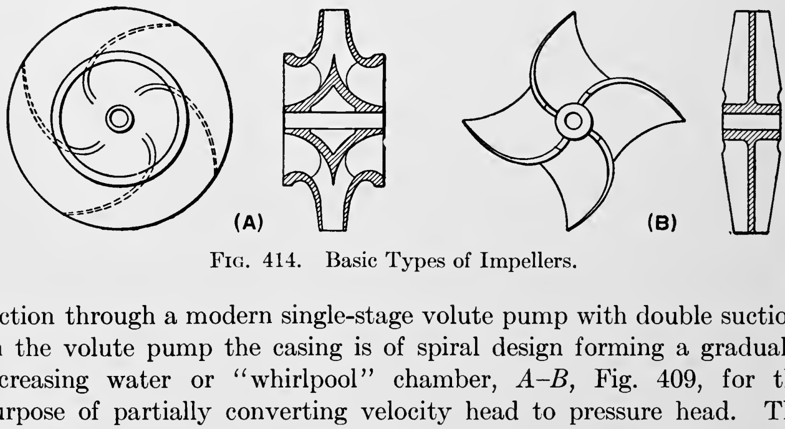


FIG. 414. Basic Types of Impellers.

section through a modern single-stage volute pump with double suction. In the volute pump the casing is of spiral design forming a gradually increasing water or "whirlpool" chamber, *A–B*, Fig. 409, for the purpose of partially converting velocity head to pressure head. The older forms of volute pumps were very inefficient, seldom delivering more than 40 per cent of the energy supplied and usually not adapted to lifts greater than 50 feet. The modern pumps give efficiencies as high as 80 per cent, and the lift is limited only by the speed of the impeller. As a general rule the volute pump is of single-stage construction

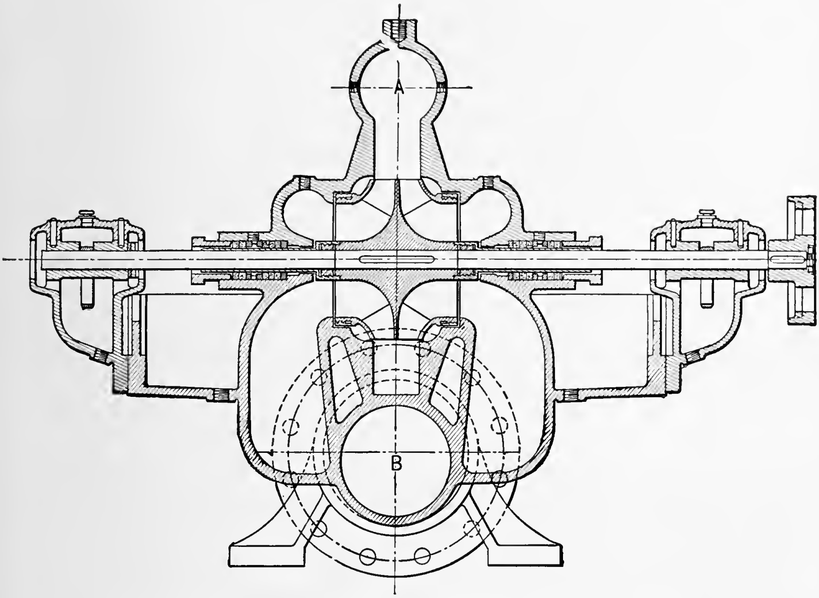


Fig. 415. Typical Single-stage Double-suction Volute Pump.

and limited to comparatively low lifts, 120 feet and under, though two-stage pumps of this type are on the market designed for heads as high as 1000 feet.

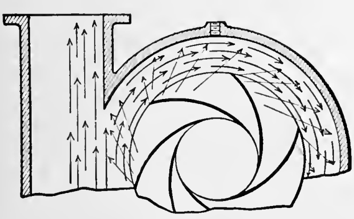


Fig. 416. Direction of Water from the Impellers of a Centrifugal Pump without Diffusion Vanes.

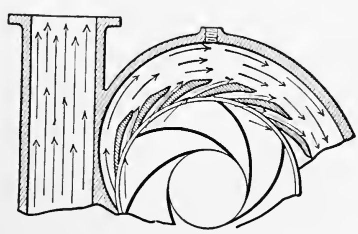


Fig. 417. Effect of Diffusion Vanes on the Direction of Water.

In the usual design of volute pumps the stream of water in the casing is at cross current with that thrown out from the impeller as shown in Fig. 416. The turbine pump is provided with a system of diffusion vanes or expanding ducts, disposed between the periphery of the impeller and the annular casing, somewhat like the guide vanes in a reaction turbine water wheel, so that the fluid emerges tangentially at about the velocity in the casing (see Fig. 417). The casing is usually concentric with the impeller and of uniform cross section though the volute casing is sometimes used in this connection. For high lifts these pumps are compounded, thereby reducing the peripheral velocity and decreasing the friction losses. Fig. 418 shows a section through a three-stage Worthington turbine pump as installed in the testing laboratories of the Armour Institute of Technology and designed to deliver 200 gallons per minute against a 750-foot head at 2500 r.p.m.

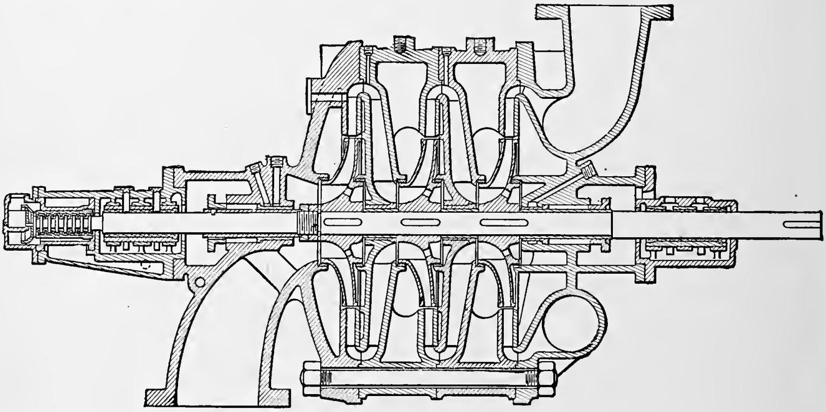


FIG. 418. Worthington Three-stage Turbine Pump.

In view of past developments it is probable that the centrifugal pump will supplant the piston type of pump for practically all purposes, except perhaps for deep-well service and for very heavy pressures. Centrifugal pumps are now used for boiler feeding, circulating condensing water, hot-well and wet-vacuum purposes and for various applications of industrial service. Efficiencies above 70 per cent are not unusual and the head against which the pump may operate is limited only by the peripheral speed at which the impeller may be safely run. Although the equivalent heat efficiency of the high-grade piston pump is superior to that of the centrifugal pump, other items, such as low first cost, decreased cost of repairs and the like, frequently offset this advantage. Some of the advantages of the centrifugal pump as compared with the piston type are:

1. Low first cost,
2. Compactness,
3. Absence of valves and pistons,
4. Low rate of depreciation,

5. Uniform pressure and flow of water,
6. Simplicity of design and ease of operation,
7. Freedom from shock,
8. High rotative speed, permitting direct connection to electric motors and steam turbines,
9. Ability to handle dirty water, sewage and the like,
10. In case of stoppage of delivery, the pressure cannot increase beyond the predetermined working pressure, and
11. Ease of repair.

Some of the disadvantages are:

1. Efficiency not as high as the best grade of piston pumps,
2. Cannot be direct connected to low-speed engines when high lifts are desired, and
3. The rate of flow cannot be efficiently regulated for wide ranges in duty.

312. Performance of Centrifugal Pumps. — For best efficiency a centrifugal pump must be properly designed for the intended service as to curvature of vanes, diameter and speed of impeller, and number of stages. Figs. 419 to 421 are based upon experiments with De Laval centrifugal pumps. When a practically uniform head is required at constant speed with varying water supply as in city water works, hydraulic elevator systems or boiler feeding, the impeller vanes are designed to give the characteristic curve illustrated in Fig. 419 which protects the motor from possible overload. See also Fig. 430.

In dry-dock and other variable head work, in order not to overload the motor, the power should be practically constant through wide variations of head and at the same time the efficiency should not vary seriously. A desirable characteristic for such a pump is illustrated in Fig. 420.

In water-supply systems in which the friction of the piping is a large part of the total head at full delivery, the characteristic shown in Fig. 421 is especially useful. Thus, when the system reduces its demand for water and the frictional head is consequently considerably reduced, the pump would automatically adjust itself to the reduced head without change of speed. Figs. 419 and 422 are based upon experiment and show the relationship between speed, head, capacity, efficiency and power consumption of various types of pumps.

The theory involved in the operation of centrifugal pumps and rules for design are beyond the scope of this book and the reader is referred to the accompanying bibliography.

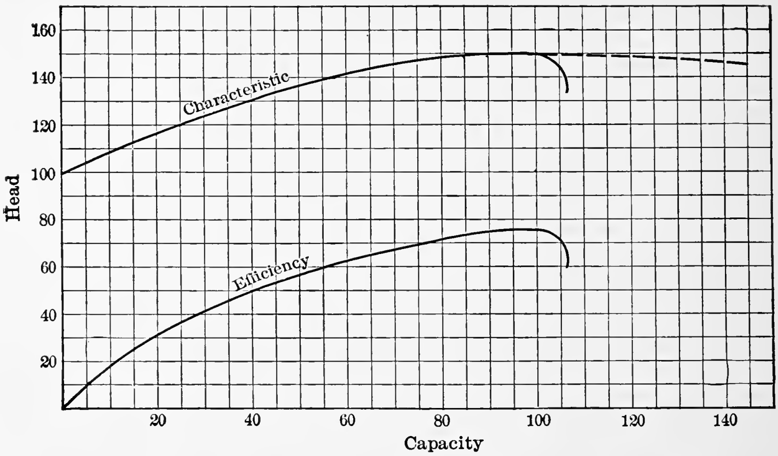


FIG. 419. Centrifugal Pump Characteristic for Hydraulic Elevator Service, Boiler Feeding, etc.

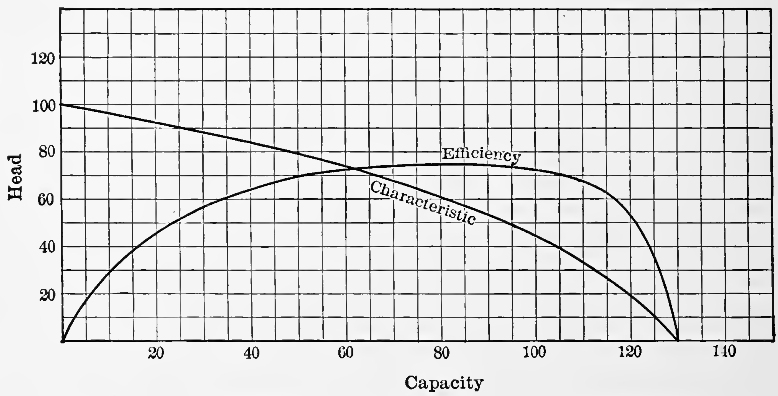


FIG. 420. Centrifugal Pump Characteristic for Dry-dock Service.

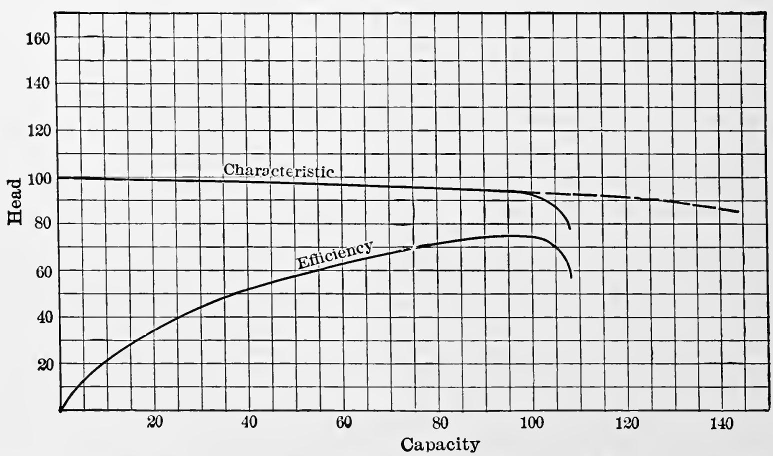


FIG. 421. Centrifugal Pump Characteristic for Water Works with Large Friction Head.

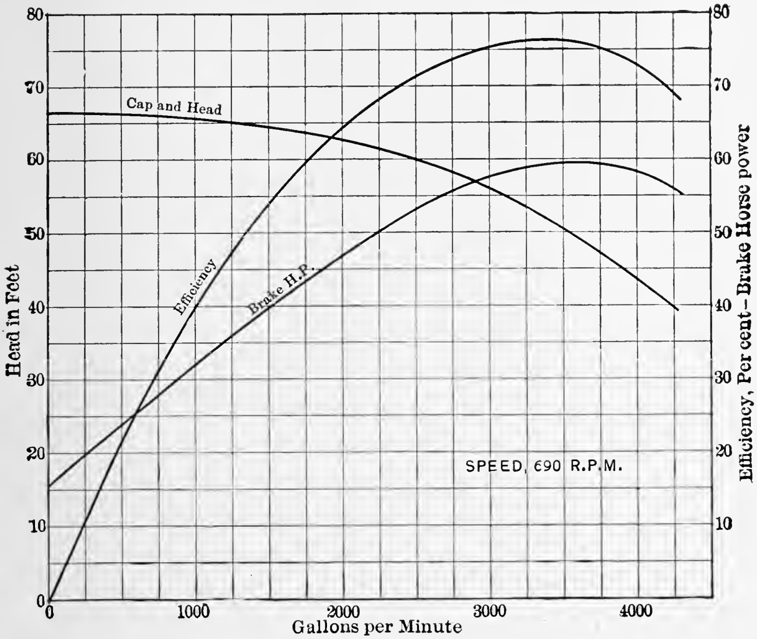


FIG. 422. Performance of Worthington 10-inch Volute Pump.

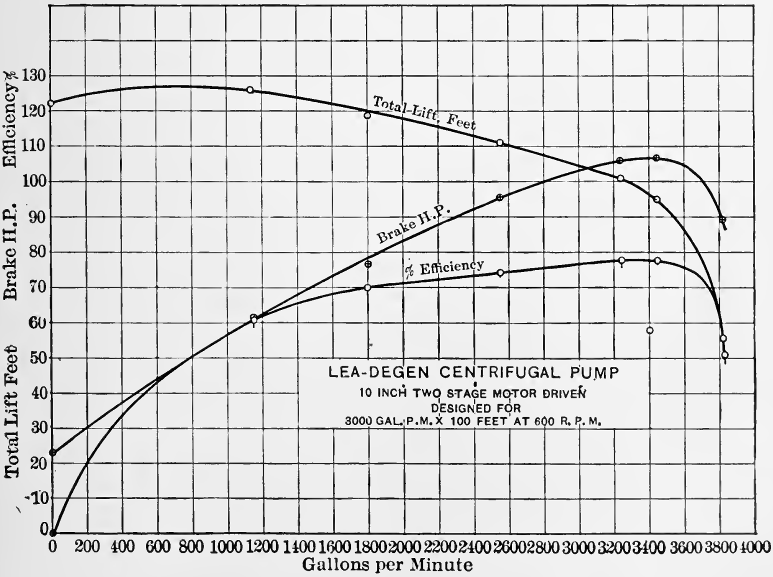


FIG. 423. Performance of Two-stage Lea-Degen Centrifugal Pump.

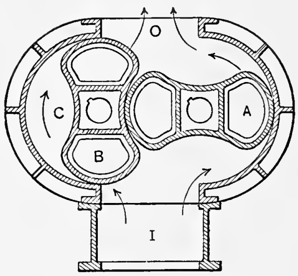


Fig. 424. Two-lobe Cycloidal Pump.

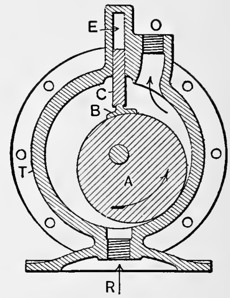
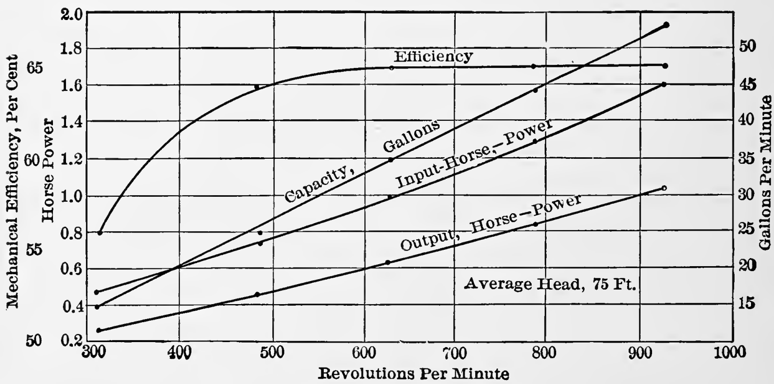
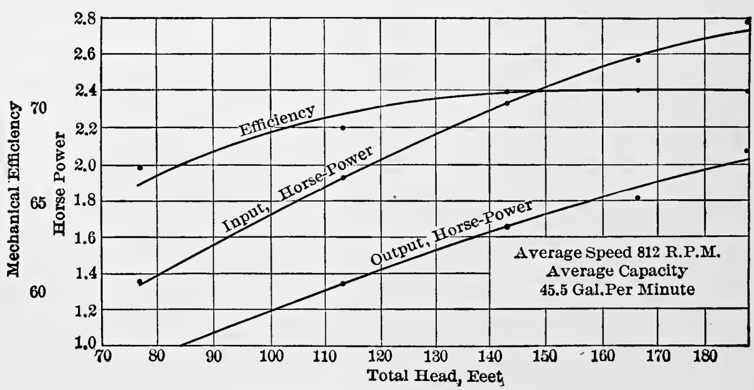


Fig. 425. Rotary Pump with Movable Butment.



Head Constant, Speed Variable.



Speed Constant, Head Variable.

Fig. 426. Performance of a Small Rotary Pump.

313. Rotary Pumps. — Rotary pumps are often used for circulating cooling water in condenser installations, and give about the same efficiency as centrifugal pumps under similar conditions of operation. For moderate pressure and large volumes they offer the advantage of low rotative speed, thus permitting direct connection to slow-speed steam engines. At high speeds they are noisy, due chiefly to the gearing. They occupy considerably less space than piston pumps of the same capacity, but require more room than the centrifugal type.

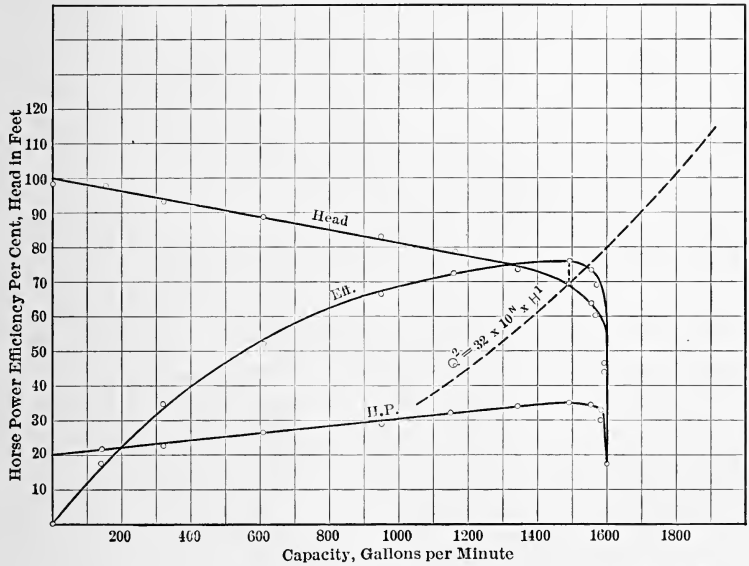


FIG. 427. Test Curves for 8-inch "Screw" Pump, American Well Works.

Fig. 424 shows a section through a two-lobe cycloidal pump. The shafts are connected by wheel gearing, the power being applied to one of the shafts. The water is drawn in at *I* and forced out at *O*, the displacement per revolution being equal to four times the volume of chamber *A*. There is no rubbing between impellers and casing. In this type of pump the pressure is independent of the speed of rotation, and the capacity varies almost directly with the speed. The slip varies from 5 to 20 per cent according to the discharge pressure.

Fig. 425 shows a section through a rotary pump with movable butment. Fig. 426 illustrates the performance of a 45-mm. Siemens-Schuckert rotary pump at different speeds and discharge pressures. (Zeit. d. Ver. Deut. Ing., June 24, 1905, p. 1040.) Large rotary pumps give much higher efficiencies, but the general characteristics are about the same. A combined efficiency of pump and engine as high as 84 per cent has been recorded. (Trans. A.S.M.E., Vol. 24, p. 385.)

Screw pumps may be grouped with the rotary positive-displacement class. The Quimby screw pump is one of the best-known examples of this type of pump and consists essentially of two right and left square thread screws revolving in a double casing. The liquid to be pumped is drawn in at the outer ends of the cylinder and forced toward the center by the action of the two pairs of intermeshing threads. The discharge is from the center of the casing. Power is applied to one of the screws and the second is driven by means of a pair of gears. The screws run in close fit with the casing but without actual contact. Quimby pumps operate at speeds varying from 600 to 1500 r.p.m., depending upon the size and service for which they are intended. Fig. 427 shows the performance of an 8-inch "screw" pump built by the American Well Works.

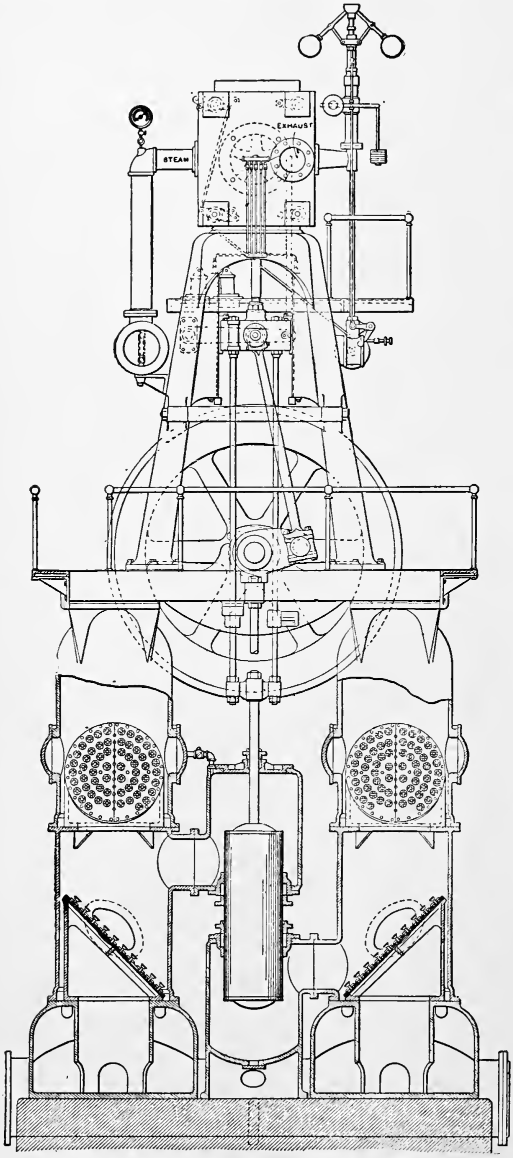


FIG. 428. 10,000,000-gallon Circulating Pump.

314. Circulating Pumps. — This term is ordinarily applied to the pumps which supply cooling water to surface condensers. The three types found in condenser practice are (1) the centrifugal, (2) the rotary positive-displacement, and (3) the reciprocating-piston pump. The centrifugal pump is by far the more common in use. For high lifts and in connection with very large units the high-duty reciprocating piston pump has been used because of its high overall efficiency but such installations are exceptional. The rotary pump is occasionally used where the driving unit is a slow-speed reciprocating engine. In small and medium-sized installations the screw pump has also been used but in the majority of plants the centrifugal pump appears to be the best selection.

The power required by the circulating pumps is the largest item of the condenser auxiliaries, and therefore every effort should be made to reduce the pumping head to a minimum. Where it is possible to seal the circulating water discharge pipe the system operates as a siphon and the static head is the difference in level of intake and discharge canals. Where the discharge head cannot be sealed the static head is the difference in level of intake water and the top pass in the condenser. The total head pumped against in any case is the sum of the static head (suction plus discharge) and the friction head lost in the condenser and piping. The brake horsepower necessary to deliver the circulating water is

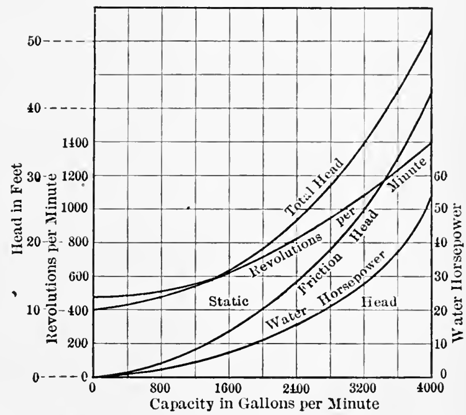


FIG. 429. Typical Performance Curves of a Typical Centrifugal Pump.

$$\text{Br.hp.} = \frac{WH}{33{,}000\,E}, \tag{263}$$

in which

W = weight of circulating water, lb. per min.,
H = total head, ft.,
E = mechanical efficiency of the pump.

The static head of course remains constant, other conditions being the same, for all rates of flow, but the friction head increases with the square of the quantity pumped. This is illustrated in Fig. 429.

Example 63. Calculate the power required to drive the circulating pump for a surface condenser installation when operating under the following conditions: Maximum capacity of main turbine 10,000 kw., water rate 15 lb. per kw-hr., ratio of cooling water to condensate 60, suction head 5 ft., friction heat 20 ft., static discharge head 15 ft., pump efficiency 70 per cent.

From equation (263),

$$\text{Br.hp.} = \frac{15 \times 10{,}000 \times 60\ (5 + 20 + 15)}{60 \times 33{,}000 \times 0.7} = 261\ \text{(approx.)}.$$

If the pump is motor driven allowing an overall motor efficiency of 85 per cent the pump will require

$$\frac{261}{10{,}000 \times 1.34 \times 0.85} = \begin{cases} 0.023 \text{ or } 2.3 \text{ per cent of the main} \\ \quad \text{generator output.} \end{cases}$$

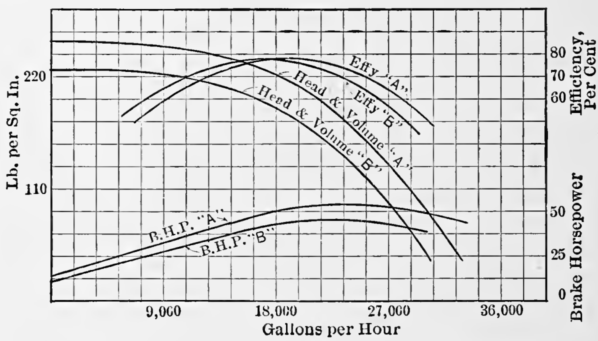


Fig. 430. Typical Performance Curves of a Rees "Roturbo" Boiler-feed Pump.

315. Centrifugal Boiler-Feed Pumps. — In power plants having capacities over 1000 boiler horsepower direct-acting and power-driven triplex boiler-feed pumps have been largely superseded by turbine- or motor-driven centrifugal pumps. For plants under 1000 horsepower the direct-acting pump offers the advantage of low first cost and ease of operation. The most economical drive for a centrifugal boiler-feed pump is a steam turbine using the exhaust steam for heating the feed water, though motor drives are sometimes used to advantage in large central stations. One great advantage of a steam turbine-driven centrifugal pump is that its delivery may be throttled down to zero when the pump is operating at its normal speed. A further advantage is that it delivers a uniform and even supply without pulsations or the need of air chambers or relief valves, thus avoiding vibration and water hammer. The turbine-driven pump is occasionally equipped with a water-pressure governor which regulates the speed of the turbine and adjusts it automatically to any load. The power required to deliver water to the boiler may be calculated with the aid of equation (263). In practice the centrifugal boiler-feed pump requires from less than

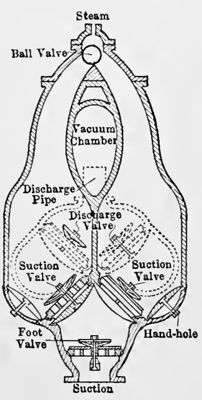


Fig. 430a. The Pulsometer.

one per cent to five per cent of the boiler steam depending upon the size, load, type of drive and disposition of the exhaust.

Example 64. Calculate the power required to drive the centrifugal feed pump for a turbine installation when operating under the following conditions: Maximum output of main turbine 10,000 kw., water rate (including auxiliary steam) 16 lb. per kw-hr., boiler pressure 200 lb. gauge.

When specific figures are not available it is customary to assume 25 per cent of the boiler pressure as the friction head, whence $H = (200 + 50)\ 2.6 = 650$ ft. (2.6 = ft. of water at boiler temperature corresponding to 1 lb. per sq. in.). Assume a pump efficiency of 65 per cent.

From equation (263),

$$\text{Br.hp.} = \frac{16 \times 10{,}000 \times 650}{60 \times 33{,}000 \times 0.65} = 81.$$

If the pump is turbine driven and the latter used 40 lb. of steam per b.hp-hr. the pump will require

$$\frac{81 \times 40}{160{,}000} = 0.02 \text{ or 2 per cent of the total weight of steam generated.}$$

If the pump turbine exhaust is used for feed water heating the pump will require only 0.3 per cent of the total steam generated. (See example 58.)

316. Condensate or Hot-well Pumps. — The centrifugal pump is now quite universally used for pumping the condensate from surface condensers. Condensate pumps must deliver water against the head corresponding to the vacuum, plus the friction head and the static head. The pump cannot create a vacuum sufficiently greater than the vacuum in the condenser to draw water into the impeller by suction, therefore the condensate should be supplied under a head of three or four feet or more. If the head on the suction side is less than this the pump "cavitates" or becomes vapor bound and is unable to remove the water. Condensate pumps are built in single-stage and two-stage types. These pumps are ordinarily operated without automatic control and are permitted to operate at constant speed. The power required to operate the pump may be calculated with the aid of equation (263).

Example 65. Calculate the power required to drive the condensate pump for a turbine installation when operating under the following conditions: Maximum output of main turbine 10,000 kw., water rate 15 lb. per kw-hr., vacuum 28 inches referred to a 30-inch barometer.

Suction head corresponding to 28 in. of mercury = 31 ft.

Assume a friction and discharge head of 29 ft.; efficiency 50 per cent.

Substituting these values in equation (263),

$$\text{Br.hp.} = \frac{10{,}000 \times 15 \times (31 + 29)}{60 \times 33{,}000 \times 0.5} = 9.4 \text{ (approx.).}$$

317. Air Lift. — The air lift is a simple arrangement of piping whereby water may be raised by means of compressed air. There are no working parts, and no valves are employed except to regulate the supply of air. Its particular field of application lies in pumping water from a number of scattered wells, and on account of the total absence of working parts it is peculiarly adapted to handling water containing sand, grit and the like. The device consists of a partially submerged water pipe and air supply variously arranged as in Fig. 431 (A) to (D). Compressed air forced into the water pipe at or near the bottom decreases the density of the column and the difference in weight between the

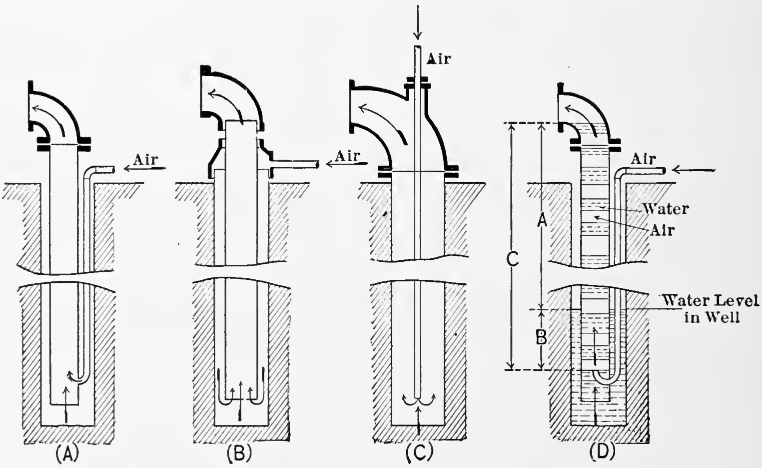


FIG. 431. Various Arrangements of the "Air Lift."

solid column of water B and the air-water column A causes the flow. The successful operation of this device depends upon the ratio of the depth of submersion B to the total head C.

The quantity of air necessary to operate an air lift may be closely approximated from the equation (see Prac. Engr. U. S., April 1, 1912, p. 354)

$$V = \frac{L}{\log \frac{S + 34}{34} \times C}, \tag{264}$$

in which

V = cubic feet of free air per gallon,
S = actual submergence in feet,
C = coefficient determined from experiment.

The actual submergence S may be determined from the relationship

$$S = \frac{LS_p}{l_p}, \tag{265}$$

in which

L = actual lift in feet (A, Fig. 431),

S_p = submergence percentage $\left(100\,\frac{B}{C}, \text{ Fig. } 431\right)$,

l_p = lift percentage $\left(100\,\frac{A}{C}, \text{ Fig. } 431\right)$.

The coefficient C may be approximated as follows:

$$C = 255 - 0.1\,L. \tag{266}$$

For the air pressure required for any lift and any percentage of submergence it is convenient to divide the actual submergence in feet by 2 to get the gauge pressure in pounds. This gives enough pressure in excess of that due to water head to allow for the pipe friction and other losses.

The efficiency ("water" horsepower divided by "air" horsepower) varies from 30 to 50 per cent, increasing as the ratio $\frac{B}{C}$ increases from 0.55 to 0.85. (Engineer, U. S., Aug. 15, 1904, p. 564.) A number of tests gives efficiencies ("water" horsepower divided by i.hp. of steam cylinder) varying from 20 to 40 per cent. The horsepower required to compress one cubic foot of free air to different pressures per square inch, as determined from actual practice, is approximately as given in Table 115.

TABLE 115.

Pressure in Pounds.	Horse Power Required to Compress 1 Cubic Foot.	Pressure in Pounds.	Horse Power Required to Compress 1 Cubic Foot.
176	0.434	60	0.159
140	0.376	45	0.145
100	0.201	30	0.121
80	0.189		

(Engr., Lond., Aug. 14, 1903, p. 174; Dec. 11, 1903, p. 568; Feb. 12, 1904, p. 172.)

When it becomes necessary to raise water to a height exceeding say 175 feet above the level in the well, it is customary to use two or more pumps, the total lift being divided between them.

Air Lift: Power, June 22, 1915, p. 843; Eng. and Contr., Aug. 9, 1916, p. 137; Bulletin No. 450, Univ. of Wis.; Prac. Engr., April 1, 1912.

Pulsometer: Tech. Quar., Sept., 1901; Public Works, Aug. 15, 1904; Engr. U. S., July 15, 1904; Experimental Eng., Carpenter, p. 621.

Turbine Pump Design: Jour. A.S.M.E., Sept., 1915., p. 538.

New Centrifugal Pump Duty Record: Iron Age, Apr. 20, 1916, p. 940.

Centrifugal Boiler Feed Pumps: Power, Oct. 31, 1916, p. 609; Aug. 24, 1915, p. 276; Nov. 16, 1915, p. 693; Dec. 29, 1914, p. 934.

Characteristic Curves of Centrifugal Pump: Jour. W. Soc. Eng., Oct., 1914, p. 776.

Pumping Units of Various Types for Small Water Supply Systems: Munic. Jour., June 22, 1916, 879.

PROBLEMS.

1. A direct-acting duplex boiler-feed pump uses 125 lb. steam per i.hp-hr. Initial steam pressure 115 lb. absolute, feed-water temperature 180 deg. fahr. What per cent of the total steam generated by the boiler is necessary to operate the pump?

2. A triple-expansion pumping engine delivers 30,310,000 gallons of water in 24 hours against a head of 61 lb. per sq. in., initial steam pressure 200 lb. abs., developed hp. 800, water rate 10.33 lb. per br.hp-hr., steam initially dry. Required the duty per 1000 lb. of dry steam and per million, B.t.u.

3. Determine the cylinder dimensions of a direct-acting single-cylinder feed pump suitable for a 500-hp. boiler, maximum overload 100 per cent, boiler pressure 115 lb. abs., feed-water temperature 70 deg. fahr.

4. Required the probable i.hp. when operating at maximum capacity.

5. Which is the more economical in heat consumption as a boiler feeder, an injector or a motor-driven triplex power pump? Boiler pressure 100 lb. abs., feed water supply 60 deg. fahr., injector delivers 16 lb. of water per lb. of steam, overall efficiency of pump and motor 60 per cent.

6. Approximate the cylinder dimensions of a wet-air pump for a 750-hp. engine using 16 lb. steam per i.hp-hr., initial pressure 150 lb. abs., vacuum 26 in. (barometer 30 in.), dry steam at admission, initial temperature of injection water 70 deg. fahr.

7. Required the horsepower necessary to operate a centrifugal circulating pump for a surface condenser installation using 1000 gallons of water per minute, total head pumped against 50 ft., initial temperature of circulating water 70 deg. fahr.

8. If the pump in Problem 7 is installed in connection with a 1000-hp. engine and the ratio of cooling water to condensed steam is 30 to 1, required the per cent of main engine power necessary to operate the pump.

9. If the pump in Problem 8 is driven by a steam engine using 50 lb. steam per hp-hr. and the exhaust is used for heating the feed water, required the per cent of main engine heat supply necessary to operate the pump. Main engine initial pressure 150 lb. abs., vacuum 26 in. (barometer 30 in.), circulating-pump engine initial pressure 100 lb. abs., back pressure 16 lb. abs. Assume dry steam at admission in both cases.

CHAPTER XIV

SEPARATORS, TRAPS, DRAINS

318. Live-steam Separators. General. — The function of a steam separator is the removal of entrained water from steam.

Unless a boiler is liberally provided with superheating surface, the steam may contain an amount of moisture varying from 0.3 to 5 per cent. If the boiler is poorly proportioned or forced far above its rating, this percentage may be greatly increased. The quality of the steam is still further reduced by condensation in the steam pipe, which may vary from 1 to 10 per cent, depending upon the length of pipe and efficiency of covering.

One of the effects of moisture in steam is to increase its density and reduce its elastic force. It also increases its conductivity, so that during the work of expansion more heat is absorbed from the walls of the cylinder and discharged into the atmosphere or into the condenser without doing useful work. (Ewing, "The Steam Engine," p. 151.) Although the heat loss from this cause is small, the danger arising from the introduction of a considerable amount of water in the cylinder renders the removal of the moisture necessary. See par. 193 for influence of moisture on steam consumption.

The essentials of a good separator are high efficiency as a water eliminator, ample storage capacity for any sudden influx of water, simplicity and durability in construction, and small resistance to the current of steam passing through. A good separator may be relied upon to remove practically all of the moisture from steam containing under ten per cent entrainment and all but two per cent from steam containing as much as twenty per cent. (Engineer, U. S., Jan. 15, 1904.)

Table 116 gives the results of a series of tests made by Professor R. C. Carpenter in 1891 of six steam separators. (Power, July, 1891, p. 9.) Conclusions from these tests were:

1. That no relation existed between the volume of the several separators and their efficiency.

2. No marked decrease in pressure was shown by any of the separators, the most being 1.7 pounds by separator E.

3. Although changed direction, reduced velocity, and perhaps cen-

trifugal force are necessary for good separation, still some means must be provided to lead the water out of the current of the steam.

A series of tests made at Armour Institute of Technology in 1905 on a number of separators showed that the *efficiency of separation decreased as the velocity of the steam increased.** At the low velocity of 500 feet per minute all separators were equally efficient, at a velocity of 5000 feet per minute several had little effect on eliminating the moisture present, and at a velocity of 8000 feet per minute only one gave efficient results.

TABLE 116.

TESTS OF STEAM SEPARATORS.

(R. C. Carpenter.)

Make of Separator.	Test with Steam of about 10 Per Cent of Moisture.			Tests with Varying Moisture.		
	Quality of Steam Before.	Quality of Steam After.	Efficiency.	Quality of Steam Before.	Quality of Steam After.	Average Efficiency
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
B............	87.0	98.8	90.8	66.1–97.5	97.8–99	87.6
A............	90.1	98.0	80.0	51.9–98	97.9–99.1	76.4
D............	89.6	95.8	59.6	72.2–96.1	95.5–98.2	71.7
C............	90.6	93.7	33.0	67.1–96.8	93.7–98.4	63.4
E............	88.4	90.2	15.5	68.6–98.1	79.3–98.5	36.9
F............	88.9	92.1	28.8	70.4–97.7	84.1–97.9	28.4

319. Classification of Separators. — Separators are based on one or more of the following principles of action:

1. *Reverse current.* The direction of the flow is abruptly changed, usually through 180 degrees. This causes the water in the steam, on account of its greater specific gravity, to be thrown into a receiving vessel, while the steam passes on in a reverse direction.

2. *Centrifugal force.* A rotary motion is imparted to the steam whereby entrained water particles are eliminated by centrifugal force.

3. *Baffle plates.* The flow is interrupted by corrugated or fluted plates to the surfaces of which the water particles adhere and from which they fall by gravity tc the well below.

4. *Mesh.* The separation is brought about by mechanical filtration through screens or meshes.

The following outline shows the classification of typical separators, in accordance with the above principles:

* See Power, May 11, 1909, p. 834.

Live-steam separators	Reverse current	Hoppes. Stratton.
	Centrifugal	Keystone. Mosher. Robertson.
	Baffle plate	Bundy. Austin. Detroit.
	Mesh	Direct. Potter.
Exhaust-steam separators	Jacketed baffle	Baum.
	Absorption	Loew.

320. Types of Separators. *Reverse-current Steam Separators.*—Fig. 432 shows a section through a Hoppes steam separator and illustrates the principle of reverse-current separation. Steam may flow through in either direction. Both the inlet and outlet ports are surrounded by

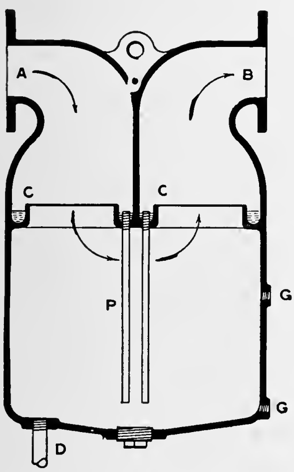


FIG. 432. Hoppes Steam Separator.

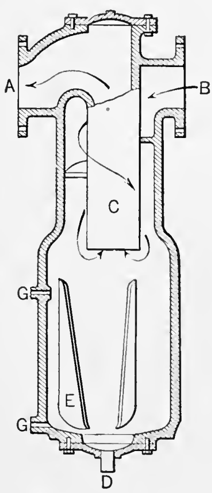


FIG. 433. Stratton Steam Separator.

gutters C, C, partly filled with water, which intercept the moisture following the surface of the pipe, while the downward plunge of the steam throws the entrained water to the bottom of the separator. The condensation is carried from the troughs by pipe P to the well below, from which it is trapped at D in the usual way. The velocity of the steam in passing through this separator is greatly reduced to prevent the steam from taking up the water in the bottom of the well. This is brought about by increasing the area of the passage through the separator.

Fig. 433 gives a sectional view of a Stratton separator, which, though primarily of the reverse-current type, embodies also the principle of centrifugal force. The separator consists of a vertical cast-iron cylinder with an internal central pipe C extending from the top downward for about half the height of the apparatus, leaving an annular space between the two. The current of team on entering is deflected by a curved partition and thrown tangentially to the annular space at the side, near the top of the apparatus. It is thus whirled around with all the velocity of influx, producing the centrifugal action which throws the particles of water against the outer cylinder. These adhere to the surface, so that the water runs down continuously in a thin sheet around the outer shell into the receptacle below. The steam, following in a spiral course to the bottom of the internal pipe, abruptly enters it, and passes upward and out of the separator without having once crossed the stream of separated water. The rapid rotation of the current of steam imparts a whirling motion to the separated water which tends to

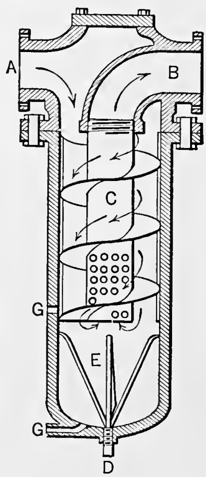


Fig. 434. Keystone Steam Separator.

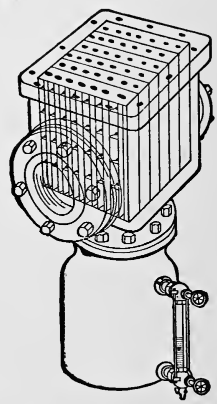


Fig. 435. Bundy Steam Separator.

interfere with its proper discharge from the apparatus. The separator has therefore been provided with wings or ribs *E* projecting at an acute angle to the course of the current, which have the effect of breaking up this whirling motion and allowing the water to settle quietly at the bottom, whence it passes off through the drain pipe *D*.

Centrifugal Steam Separators. Fig. 434 shows a section through a Keystone or Simpson's centrifugal separator. The separator consists of a cast-iron cylinder with vertical pipe *C* extending downward about two-thirds of the whole length; this pipe has a thread or screw wound spirally around it, the space between the threads being somewhat greater than the area of the steam pipe. The steam passing around the spiral course causes the water to be thrown against the outer walls by centrifugal force, while the dry steam passes through the small holes in the central pipe. The water passes down the outer walls, where its motion is arrested by obstructing ribs *E*, and is thence carried away by a drip pipe *D* to a suitable drain.

Baffle-plate Steam Separators. — Fig. 435 gives an interior view of a Bundy separator and illustrates the application of baffle plates for live-steam separation. This separator consists of a rectangular cast-iron casing with a cylindrical receiver beneath it. Directly across the steam passage are baffle plates corrugated for the reception of entrained water. The plates consist of vertical castings, each containing a main artery or channel which leads directly to the receiver. The fronts of the plates are flat, with a series of recesses sloping inwards and downwards, terminating in an opening of capillary size leading to the main artery. The plates are staggered, so that the steam must impinge against all of them in its passage. The particles of water adhere to the plates, collect, and fall by gravity into the receiver. The flanges at the bottom constrict the opening of the reservoir so as to prevent the steam from picking up any portion of the water.

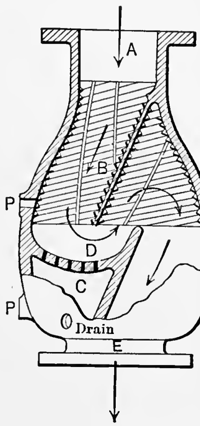


Fig. 436. Austin Steam Separator.

Fig. 436 shows a section through an Austin separator and illustrates another class embodying the fluted baffle-plate principle. The steam in passing through the chamber impinges against the fluted baffle plate *B*. The moisture adheres to the surfaces, collects and trickles along the corrugations to the bottom of the well. These corrugations are formed in such a manner that the steam cannot come in contact with the water particles after they have been once eliminated. A perforated diaphragm *D* prevents the water in the well from coming in contact with the steam. The current of steam is also reversed, thus giving additional separating properties to the apparatus.

Mesh Separators. — Fig. 437 shows a section through a "direct" separator, illustrating the principle of mesh separation. These separators are made with steel bodies and cast-iron heads and bases, in all sizes up to six inches inclusive, the larger sizes being constructed of cast iron or boiler plate. The cone *C*, perforated lining *E*, and diaphragm *S* are made of cold-rolled copper; the cone *O* is a substantial gray-iron casting, resting on three cast-iron supports hooked over the top of inner pipe as indicated. The method of operation is as follows: The accumulated moisture around the walls of the steam pipe is caught by the upper edge of cone *C* and carried down back of lining *E* to the water chamber. The current of steam entering the separator impinges upon the conical surface, which is composed of solid plate *O* covered with sieve *S*, through which water may freely pass but from which it cannot readily escape. Passing through the sieve and depositing on the solid surface of the cone *O*, this water is carried by conductors *P* to the water chamber. Perforated lining *E* permits the moisture content of the steam to pass through the opening to the water below and prevents it from coming in contact again with the current of steam. A trough is provided at the lower edge of the inverted cup which leads all the water that may adhere to it to the water chamber. The steam flows through the passages indicated by arrows and is subjected to a whipsnapping action which tends to throw off any remaining moisture. The perforated plate *D* prevents the steam from picking water out of the water chamber.

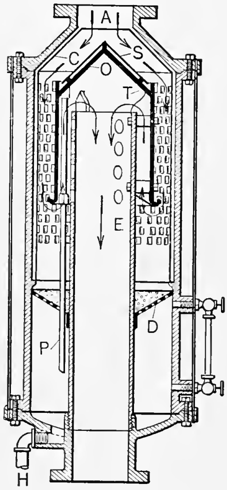


Fig. 437. "Direct" Steam Separator.

321. Location of Separators. — Live-steam separators may be located

1. Inside the boiler,
2. Between boiler and engine,
3. At the steam chest.

Where the steam pipe is very short, and particularly in marine and locomotive work where the tossing of the boiler induces excessive priming, the separator may be placed inside the boiler and its function becomes that of a dry pipe. In this location it prevents the water due to foaming and priming from passing to the engine, and reduces condensation in the pipe by supplying dry steam. The "Potter mesh" and the "De Rycke centrifugal" are types of separators designed for this service.

The arrangement of separator between engine and boiler, other than at the throttle or inside the boiler, is sometimes necessary for economy of space. Where possible, however, the separator should be placed close to the steam chest.

Current practice recommends that a receiver separator, which is an ordinary separator with a volume of two to four times that of the high-pressure cylinder, be placed close to the engine if the load is intermittent or sharply fluctuating. This forms a cushion for absorbing the force of the blows caused by cut-off, delivers steam at a practically uniform pressure, and reduces the vibration of the piping to a minimum. It also provides a reservoir for sudden demands made by the engine. Smaller pipes and higher velocities may be used with this arrangement.

322. Exhaust-steam Separators and Oil Eliminators. — The function of an exhaust-steam separator is the removal of cylinder oil from the steam exhausted by engines and pumps. In plants where exhaust steam is used for heating it is quite essential to remove the oil from the steam before it enters the heating system, for the oil not only reduces the efficiency of the radiators by coating them with an excellent non-conducting film but is an element of danger to the boiler itself. In condensing plants the separator will prevent the oil from fouling the condenser tubes and those of the vacuum heater if one is installed; this is an important factor, since the oil or grease lowers the efficiency of the heat transmission.

In a general sense a live-steam separator is also an oil eliminator, and all the separators previously described perform this function to a certain extent, since the underlying principles governing the elimination of oil from exhaust steam are similar to those employed in removing water from steam. Most of the separators described above are also designed in lighter form, as oil eliminators, but by far the greater number are based on the fluted baffle-plate principle, of which the Hine, Bundy, Cochrane, Utility, Peerless, and Keily are well-known examples. This type of oil separator will eliminate a considerable portion of the oil in the steam, provided the baffle plates or corrugated surfaces are frequently cleaned.

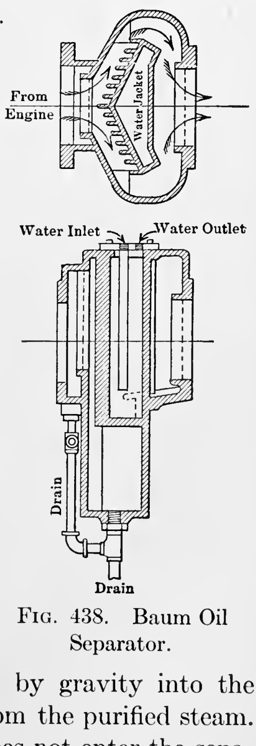


Fig. 438. Baum Oil Separator.

It is a well-established fact that oil can be more effectually removed from wet than from dry steam, and some makers, notably the Austin Separator Company, inject a cold-water spray into the separator chamber. A similar result is brought about in the Baum separator, Fig. 438, in which the corrugated baffle plate is hollow and cold water is forced through the chamber thus formed. Referring to Fig. 438: The diverged baffle plate forms the wall of a chamber in which cold water is continually circulated. This circulation causes moisture to appear on the baffle-plate surface. The particles of oil, coming in contact with this moist surface as the steam current is diverged, adhere to it and fall by gravity into the well below, where they are completely isolated from the purified steam. A large portion of the oil and water, however, does not enter the separator at all but is caught by the inside ledge near the junction of the exhaust pipe and the separator. The oil and condensation which are carried along the bottom of the pipe come in contact with this ledge and are carried directly to the outlet pipe.

A very successful method of removing oil from steam is to project the steam on to the surface of a body of water. The water may be hot or cold and will hold the oil if it once reaches the surface. It is essential, however, to reduce the velocity of the steam as it passes on its way to the outlet. Baldwin's grease separator is based upon this principle. (Baldwin on Heating, p. 234.)

The most efficient method of removing oil is by combined filtration and absorption. (Engineering News, May 22, 1902, p. 406.) A large chamber filled with coke, brick, broken tile, or other absorption material is placed in series with the exhaust pipe. The steam passing through this chamber is entirely freed from oil and moisture, provided the absorbing material is sufficient in quantity and is replenished as soon as it becomes saturated with oil. The annoyance attending the removal and replenishing of the absorbing material at frequent intervals and the great size of the apparatus are serious drawbacks. An example of this system of purification in which many of the objectionable features are reduced to a minimum is the Loew grease and oil extractor, Fig. 439. The exhaust steam enters the chamber at the top, strikes a large deflecting plate shaped like an inverted V, and permits part of the condensation and oil to be drawn off by the drain pipe. The steam then rises and is deflected, as indicated, against a series of shelves filled with fibrous material covered with coarse wire screens. The grease is removed from each shelf by suitable drains. This apparatus is sectional and any number of sections may be added without affecting the rest.

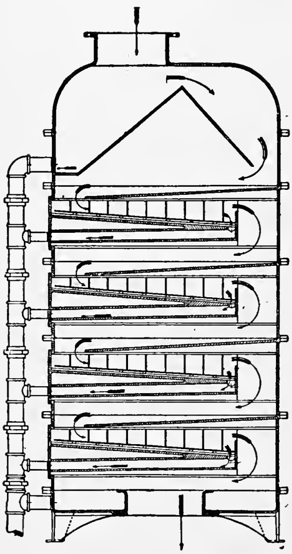


Fig. 439. Loew Grease Extractor.

In a non-condensing plant where the exhaust steam is used for heating purposes the oil separator is ordinarily placed in the main exhaust pipe just before it enters the heating system. Where several branches enter one main it is not customary to place a separator in each branch, one large separator located as above being sufficient.

In condensing plants oil separators are seldom installed except where surface condensers are used, in which case the separator may be placed anywhere between the engine and condenser. In case a vacuum heater is used the separator may be placed on either side of the heater, depending upon the type of separator. If the separator is of the "jacket-cooling" or "spray" type, it may be placed between the engine and the vacuum heater; if, however, it is of the "baffle-plate" type, the oil will be more efficiently removed if the separator is placed between the heater and condenser so that it will get the benefit of the moisture formed in the heater. In the latter location, however, the separator will not prevent the oil from fouling the heater tubes.

Where a jet condenser is used and water is taken from the hot well, the hot well itself acts as an oil separator. (Trans. A.S.M.E., 24–1144.)

All separators, steam and oil, should be provided with gauge glasses and should be thoroughly drained and the drainage should be automatic.

323. Exhaust Heads. — The function of the exhaust head is the elimination of oil and water from steam exhaust before permitting it to be discharged into the atmosphere. Unless removed, the water and oil rot the roofs and walls in summer and pollute the atmosphere surrounding the plant. The exhaust head also acts as a muffler, reducing the noise of the escaping steam. Exhaust heads are built on the same principle as steam and oil separators and most separator builders manufacture them. Fig. 440 shows a section through a typical exhaust head.

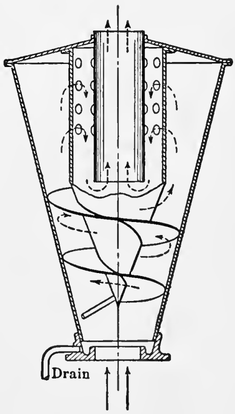


Fig. 440. A Typical Exhaust Head.

The condensation is ordinarily drained to waste, though with proper purification it may be returned to the boiler. With an efficient oil separator in the exhaust line the condensation in the exhaust head may be returned directly to the boiler without further purification.

Live-steam separators are proportioned so that it is only necessary, in the average installation, to specify the size of pipe, the type of engine, the steam pressure, and the style, whether horizontal or vertical. Gauge glasses, gauge cocks, and companion flanges are usually provided by the maker. In some cases the capacity of the reservoir is also specified. In specifying oil extractors the following additional data are necessary for an intelligent choice: the number of engines and pumps exhausting into the line, the location of the separator, the steam pressure, *velocity*, and the quality and quantity of cylinder oil used.

A guarantee of efficiency and of material and workmanship is often demanded.

Oil Separation from Water of Condensation: Jour. A.S.M.E., June, 1915, p. 345.
Electrostatic Separation of Oil and Water: Met. and Chem. Engr., Mar. 15, 1916, p. 343.

324. Drips. — No matter how thoroughly a steam pipe or reservoir may be covered with insulating material considerable condensation takes place. With the best covering this loss approximates one sixth of a pound of steam per square foot of pipe surface per hour for steam pressures of one hundred pounds, and runs as high as one pound of steam for bare pipes. See Fig. 467 for results of experiments on the loss of heat from bare pipes, and Fig. 468 for data on the efficiency of pipe coverings. In addition to this water of condensation, from $\frac{1}{2}$ to 2 per cent of moisture is carried over by the steam from the boiler. This water, unless thoroughly removed, is a constant source of danger to the engines and causes water hammer and leaky joints in the piping.

A joint on a steam pipe may safely withstand a steam pressure of 100 pounds without leaking and still leak badly under a water pressure of half that amount. This is due to the fact that the steam with its high temperature causes the pipe to expand, thus insuring a tight joint, while the entrained water (which cools as it collects) causes the pipe to contract and allows a leak.

The entrained water and water of condensation are usually spoken of as "drips." Drips may be divided into two classes, low pressure and high pressure.

325. Low-pressure Drips. — Low-pressure drips include the steam condensed in heating systems, exhaust steam feed heaters of the close type, exhaust steam piping, receiver barrels, steam chests, and exhaust heads. As these drips are impregnated with oil and are useless for boiler feed without purification, they are usually discharged to waste. Most city ordinances require the drips to be cooled to 100 deg. fahr. before being discharged into the sewer. In this case they must be first discharged into a tank and permitted to cool. This tank must be vented to the atmosphere to prevent back pressure. Fig. 441 shows an installation in which the heat abstracted from the drips, etc., is used to heat the feed water. The drips from the throttle valve and steam chest in a non-condensing plant are ordinarily discharged into the exhaust pipe as shown in Fig. 442. In a condensing plant the throttle drips are piped to a trap or to the free exhaust pipe. The returns from a steam-heating system are sometimes classified as low-pressure drips. They are invariably returned to the boiler.

In small plants all the low-pressure drips may be connected to one large pipe and this pipe in turn to a single trap, provided there is but little difference in pressure in the various drip pipes. In case of different pressures separate leads should be run to waste or traps.

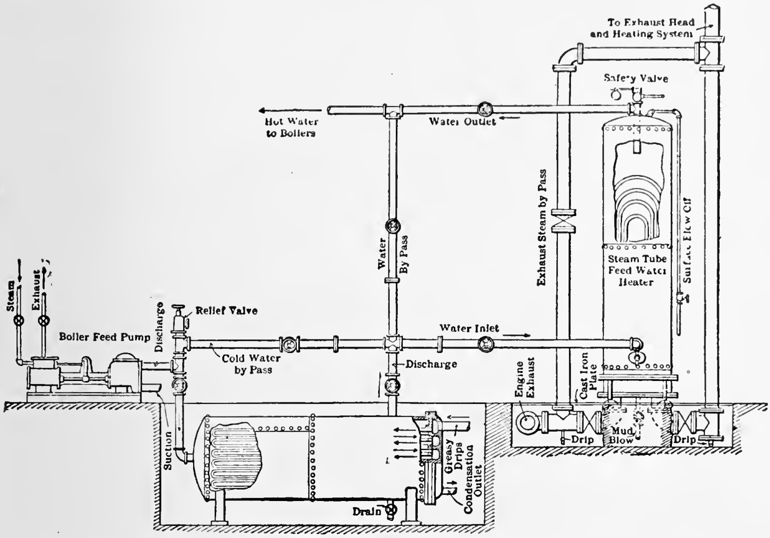


FIG. 441. Closed Heater Installation for Abstracting Heat from Oily Drips.

The drips from the receiver and vacuum heater barrels in a condensing plant are oftentimes under less than atmospheric pressure, and sometimes the pressure varies from a slight vacuum to 10 or 20 pounds gauge, and consequently cannot be disposed of as described above. If possible, the heaters and receivers should be placed so as to drain into the condenser (see Fig. 455). Should this arrangement prove impracticable, the barrels may be drained by a trap especially arranged as shown in Fig. 456.

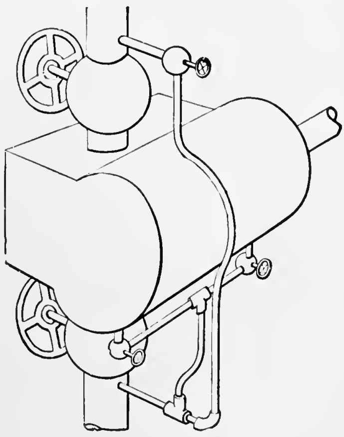


FIG. 442. Simple Method of Draining Drips.

326. Size of Pipe for Low-pressure Drips. — In the average exhaust-steam feed-water heater one pound of steam in condensing gives up approximately 1000 heat units. This will heat about 6 pounds of water from 60 to 200 deg. fahr. Hence the area of the drip which carries the water of condensation from the closed heater need be but one fifth that of the feed pipe.

In no case, however, should a pipe smaller than one half inch in diameter be used. Should the same pipe be used for both exhaust head and heater drips, an area of one fourth area of feed pipe would prove of ample capacity. In practice it is customary to use the size of pipe conforming with the outlet furnished by the manufacturer of the apparatus, and only when several pieces of apparatus are connected to one main are calculations made for the size of this main.

The drip pipe from the throttle valve is ordinarily one half inch in diameter irrespective of the size of steam pipe; this is also true of the steam-chest drip.

327. High-pressure Drips. — High-pressure drips consist of those which are condensed under boiler pressure and include the steam condensed in steam pipes, cylinder jackets of engines, reheating coils of receivers, and separators. Being free from oil and containing considerable heat, they are usually returned to the boiler. Drips may be returned to the boiler automatically by means of

1. Steam traps,
2. Holly steam loop,
3. Pumps.

328. Classification of Steam Traps. — Steam traps may be divided into two classes, depending on their use, — return and non-return. Both of these two classes may be subdivided into five types according to the principle of operation, viz.:

I. Float.
II. Bucket.
III. Bowl.
IV. Expansion.
V. Differential.

CLASSIFICATION OF A FEW WELL-KNOWN STEAM TRAPS.

Steam Traps	Float		McDaniel. Cookson.
	Bucket		Acme. Albany.
	Dump		Bundy. Morehead.
	Expansion	Metal	Columbia. Geipel.
		Volatile-Fluid	Dunham. Heintz.
	Differential		Flinn. Siphon.

Return Traps.

Traps which receive the condensed steam and return it to a boiler having considerably higher pressure than that acting on the returns are known as *return traps.* They are made in a great variety of styles. The general principle of operation is shown in Fig. 452 and described in paragraph 330.

Non-return Traps.

Non-return traps, as the name implies, are used where the water of condensation is not returned to the boiler but is discharged into any receptacle having less than boiler pressure.

329. Types of Traps. *Float Traps.* — Fig. 443 shows a section through a McDaniel improved trap, illustrating the principles of the float type. A hollow sphere C of seamless copper pivoted at E rises and falls with the change of water level in the vessel. The discharge valve M is operated by the float. When the trap is empty the float is in its lowest

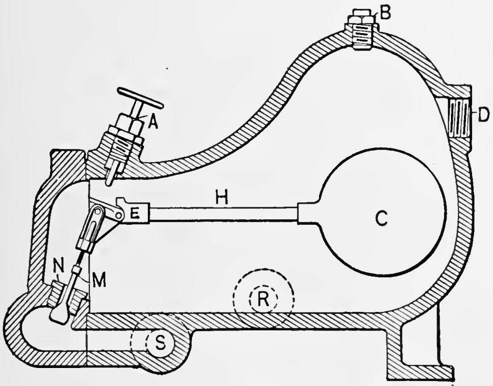


Fig. 443. McDaniel Float Trap.

position and the discharge valve is closed. Water of condensation flows into the trap by gravity through opening D to a certain depth, when the float opens the discharge valve and the steam pressure acting on the surface of the water forces it through outlet S to tank or atmosphere. After the water is discharged the float closes the valve and permits the condensation to collect again. A gauge glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is a danger of considerable steam leakage through the discharge valve, due to unequal expansion of valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous, since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present. When the trap is working lightly, this adjustment is apt to throttle the area and create such a high velocity of discharge as to cause a rapid wear of valve and seat. This defect is more or less evident in all steam traps discharging continuously. For this reason all wearing parts should be accessible and readily replaceable.

Bucket Traps. — Fig. 444 shows a section through an "Improved Acme" steam trap. The water of condensation enters the cast-iron vessel at A, filling the space D between the bucket E and the walls of the trap. This causes the bucket to float and forces valve V against its seat (valve V and its stem being fastened to the bucket as indicated). When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing valve V from its seat. This

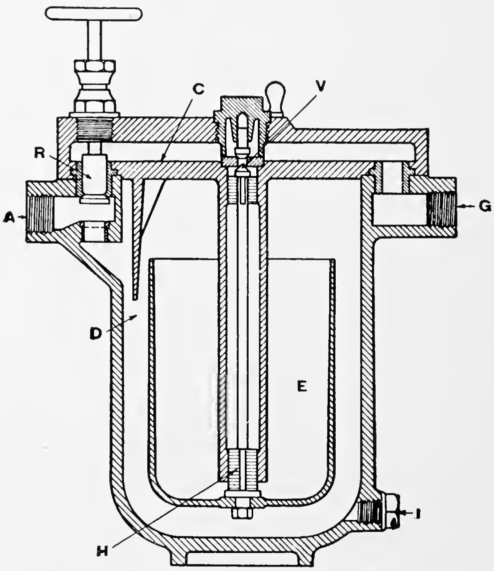


FIG. 444. Acme Bucket Trap.

permits the steam pressure acting on the surface of the water in the bucket to force the water through the annular space H to discharge opening G. When the bucket is emptied it rises and closes valve V and another cycle begins. By closing valve R the trap is by-passed and the condensation blows directly through passage C to discharge G. The discharge from this type of trap is intermittent.

Dump or Bowl Traps. — Fig. 445 shows sections through a Bundy bowl trap of the "return" type. The water enters the bowl through trunnion D and rises until its weight overbalances counterweight E and the bowl sinks to the bottom. As the bowl sinks, arm G, which is a part of the bowl, rises and engages the nuts N on valve stem H and opens valve I, thus admitting live steam pressure on to the surface of the water. The trap then discharges like all others. After the water is discharged weight E sinks and raises bowl A, which in turn closes valve I, and the cycle begins again. Bowl traps are necessarily intermittent in their discharge.

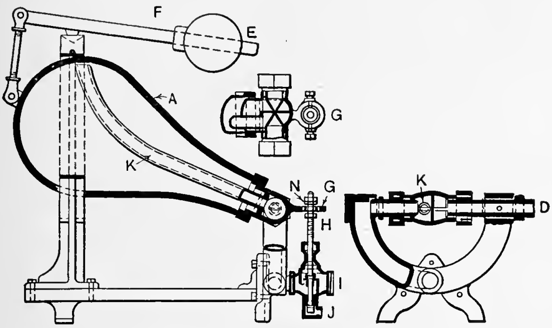


FIG. 445. A Typical Tilting Trap.

Fig. 456 shows the application of a bowl trap to a receiver where the drips are under a vacuum, and Fig. 457 a similar application to an engine receiver where the pressure varies from less than atmospheric pressure to a pressure of 40 or 50 pounds.

Expansion Traps. — Expansion traps may be divided into two groups:

(1) Those in which the discharge valve is operated by the relative expansion of metals and

(2) Those in which the action of a volatile fluid is utilized.

Expansion traps will never freeze, as they are open when cold and all the water drains out before the freezing temperature is reached.

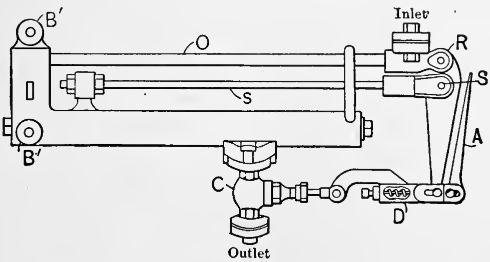


FIG. 446. A Typical Expansion Trap.

Since traps of this type have little capacity for holding water, 5 to 10 feet of pipe should be provided between the trap and the pipe to be drained in order that the condensation may collect and cool.

Fig. 446 shows the general appearance of a Columbia expansion trap in which the valve is operated by the expansion of metallic tubes.

Water gravitates to the trap through opening marked "inlet," passes through *brass* pipe *O*, then downward to the main body of the valves and back to outlet valve *C*. Below pipe *O* and parallel to it is an *iron* rod *S*, at the end of which is the support or fulcrum of lever *R*. The lower end of this lever is connected to the stem of the valve *C*, so that any movement of the lever is communicated to it. When the trap is cold, valve *C* is open and all water of condensation passes out. The moment steam enters the pipe *O* it expands. The amount of expansion is multiplied several times by the action of the lever *R*, so that the movement of the valve is much greater than the expansion of the pipe *O*. The compensating spring *D* prevents the brass tube from damaging itself by excessive expansion. Lever *A* permits the trap to be blown through by hand.

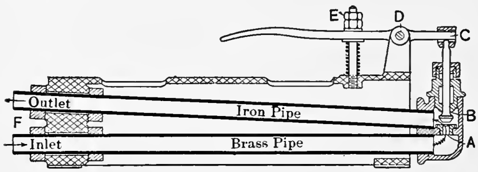


Fig. 447. Geipel Expansion Trap.

Fig. 447 shows a section through a Geipel trap in which the valve is operated directly by the expansion of two metallic tubes and the movement is not multiplied by levers as with the Columbia. The lower or brass pipe constitutes the inlet and is connected to the vessel to be drained; the upper or iron pipe is the outlet for discharge. The two pipes form the sides of an isosceles triangle, the base *F* of which is rigid, while the apex *A* is free to move in a direction at right angles to the linear expansion of the tubes. When cold, the brass pipe is contracted and the apex, in which the valve seat is placed, is moved down so that the valve is open and the water is discharged. As soon as steam enters the brass pipe the latter expands and forces the valve seat against the valve. The trap may be adjusted for any pressure by means of the lock nuts *E*. When it is desired to blow through, the valve may be operated by hand by pressing the lever.

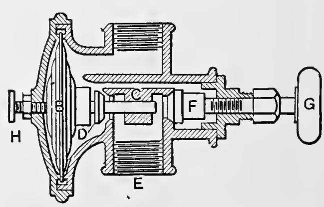


Fig. 448. Dunham Expansion Trap.

Fig. 448 shows a section through a Dunham trap. It operates upon the expansion principle, utilizing a fluid of a volatile character as its motive force. The corrugated bronze disk *B* is filled with a volatile fluid, and expands and contracts according to the pressure exerted by the fluid. The water enters at the top, surrounds disk *B* and passes through valve opening *D* to discharge outlet at *E*. As soon as steam strikes the disk *B* the volatile fluid flashes into a vapor and causes the disk to expand. This expansion forces valve *D* against its seat and the discharge ceases. The valve will remain closed until the condensation collects and cools the disk *B*, which then contracts, opens the valve, and condensation enters as before. The adjustment, however, is such that the discharge may be made continuous instead of intermittent.

The Dunham trap is claimed to be the smallest trap of its capacity on the market. The 1-inch size, having a capacity for draining 10,000 lineal feet of 1-inch pipe under 60 pounds pressure, weighs but 5 pounds and may be connected to the pipe line as if it were a globe valve.

Fig. 449 shows an internal view of a Heintz steam trap. This works on the principle of the volatile-fluid expansion trap but in a different manner from any of those described above. The requisite movement

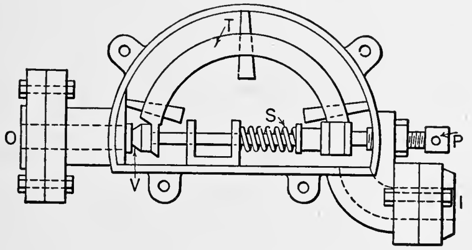


FIG. 449. Heintz Expansion Trap.

is obtained by the elongation and contraction of the extremities of a bent metallic tube *T* filled with a highly volatile fluid. This tube is inclosed in a cast-iron box and presses against the point of regulating screw *P*. The other extremity of the tube carries the valve and is free to move under the action of the variations of temperature. Spring *S* has no connection with the action of the trap. It is used as a simple means of holding one end of the expansion tube on its pivot. The trap operates as follows: Water enters at *I*, surrounds the tube *T* and passes through the valve to the discharge outlet *O*. As soon as steam enters the chamber the volatile fluid in the tube flashes into a vapor and the pressure thus created tends to straighten out the tube; this forces the valve against its seat and the discharge ceases. As the trap cools, the tube returns to its normal position and the discharge valve is opened, thus permitting the condensation to drain out. The adjustment permits of continuous or intermittent discharge and of variable pressures.

Differential Traps. — Fig. 450 shows a cross section through a Flinn differential trap. The column of water *X* acting on diaphragm *D* closes valve *V*. The water entering pipe *E* and the action of the spring equalize column *X* and open the valve. Describing the action in further detail, the water of condensation enters at A, fills lower chamber Y, pipe X, and receiving chamber C up to the level of the top of pipe E. This column of water acting on the under side of the diaphragm D forces the valve to its seat against the counter pressure of the spring S.

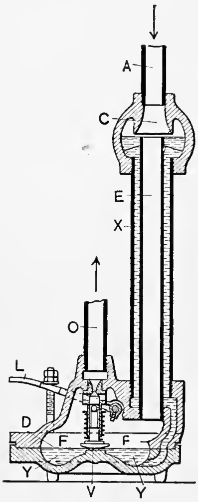


Fig. 450. Flinn Differential Trap.

Any additional water that enters the trap overflows through pipe E, filling chamber F and pipe E to a point about midway of its height, where the effect of the column of water in pipe X is balanced. The pressure on each side of the diaphragm is then equal, the short column in pipe E, aided by the spring, balancing the pressure of the longer column in pipe X. Any further increase in the height of the water in pipe E causes a depression of the valve V, which allows water to escape until the column has fallen to a level a little below the middle of pipe E, when this valve closes again. This action is repeated at intervals according to the quantity of water entering the trap. So long as the water keeps coming in sufficiently large quantities the valve remains wide open.

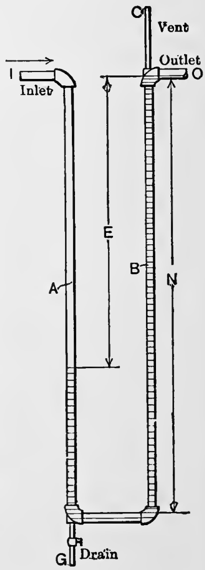


Fig. 451. Simple Siphon Trap.

Fig. 451 gives a general view of a siphon trap which is much used in draining low-pressure systems, as, for example, the separator in an exhaust steam heating system. It consists essentially of two legs A and B, which may be close together or any distance apart but the lengths of which must be sufficiently great to prevent pressure acting through pipe I from forcing the water out of B. C is a vent pipe extending to the air to prevent siphoning; O is the discharge for the condensed steam. In ordinary operation the leg B is filled with water which is constantly overflowing, and A with steam and water, the total pressure in both legs being equal. The siphon trap is applicable for low pressure only, as it requires approximately 2.3 feet of vertical space E for each pound per square inch pressure in the pipe. The maximum allowable head is represented by vertical distance N.

330. Location of Traps. — Wherever possible a trap should be located so that the condensation will flow into it by gravity. This will insure positive drainage. Sometimes, however, the coils, cylinders, or pipes to be drained are located in a pit or trench or lie on a basement floor where it is impossible to set the trap so as to receive the drains by gravity without placing it in an inaccessible position. With very low pressures this is often unavoidable, but with pressures of five pounds or more the trap may be placed above the point to be drained. If a trap is set in an exposed place a drain should be provided at the lowest point to free the pipe of water when steam is shut off. A dirt catcher or strainer should be placed in the pipe leading to the trap to prevent scale, etc., from reaching the valve. All pockets and dead ends should be drained, and no condensation should be allowed to accumulate. High- and low-pressure drips should be kept separate. All tanks should have gauge glasses.

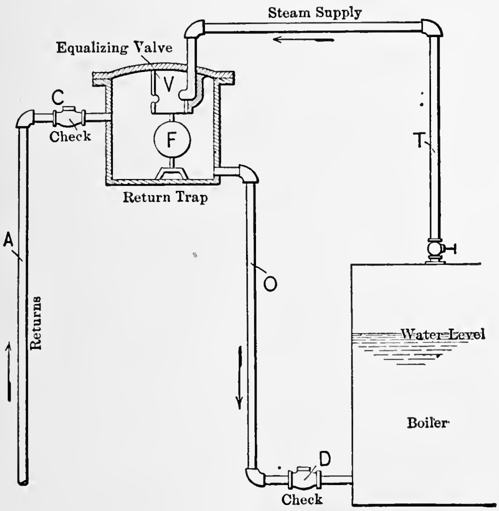


Fig. 452. Return Trap.

Fig. 452 shows the application of a float trap for automatically returning water to the boiler. For this purpose the trap must be placed three feet or more above the water line in the boiler, so that the water may gravitate to the latter. Water is forced into the trap from the returns through pipe *A* until it reaches a level where the float opens the equalizing valve *V* and permits steam from the boiler to enter the trap, thus equalizing the pressures. The water then flows into the boiler by gravity through check valve *D*. At the end of discharge the float closes the equalizing valve and another cycle begins. Check valve *C* prevents the water from being forced back to the return pipe. If the pressure in the return pipe *A* is not sufficient to force the water into the trap, a pump or another trap may be used to effect this result.

Practically any high-pressure trap may be converted into a return trap by the proper installation and an "equalizing" valve.

Figs. 453 and 454 show different applications of steam traps to the receiver coils and jackets of triple-expansion pumping engines. The drawings are self-explanatory.

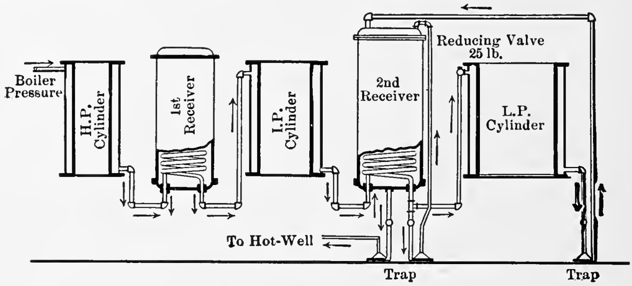


FIG. 453. Drainage System for Jackets and Receivers of Triple-expansion Pumping Engines.

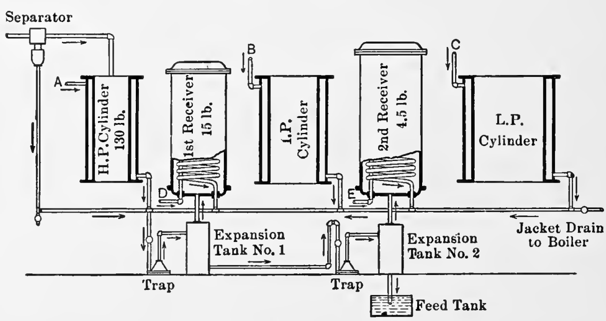


FIG. 454. Drainage System for Jackets and Receivers of Triple-expansion Pumping Engines.

331. Drips under Vacuum. — Conditions frequently make it necessary to remove condensation from apparatus working under a vacuum, as, for example, a primary heater.

The simplest method is to pipe the drips to the condenser and permit the condensation to gravitate to it as in Fig. 455. Where this is impracticable, as in an installation with the condenser above the heater, a steam trap is usually employed. Fig. 456 shows the application of a Bundy trap to a vacuum or primary heater. A close-fitting weighted check valve W, set to open outwards, prevents intake of air through the discharge pipe while the trap is filling. Connection E is made from the vent underneath the valve stem V back to the heater so as to equalize the pressures. The operation is as follows: Condensation gravitates from the heater through check C to the body of the trap, the check W being closed. When the bowl is full enough to overcome the weight of the counterbalance, it sinks and opens up the live-steam valve V. This admits steam to the trap through pipe D, which in turn closes check C and forces the water past the weighted check W to the discharge tank. After the water is discharged the bowl returns to its original position and closes valve V, the weight closes check W, the vent check equalizes the pressure in the bowl and heater, and condensation gravitates to the trap again.

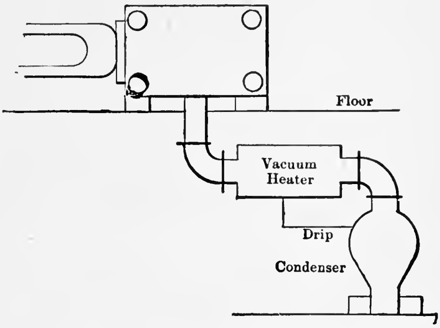


FIG. 455. Gravity Drainage; Vacuum Heater.

332. Drips under Alternate Pressure and Vacuum. — Occasionally the load on an engine is of such a character that the pressure in the receiver alternates from a pressure of 30 or 40 pounds absolute to a vacuum of varying degree. Where the periods of vacuum operation are very few and of short duration, as in the average installation, no attention is paid to the vacuum and the condensation is removed by a trap in the ordinary way. If, however, the periods are of sufficient duration and

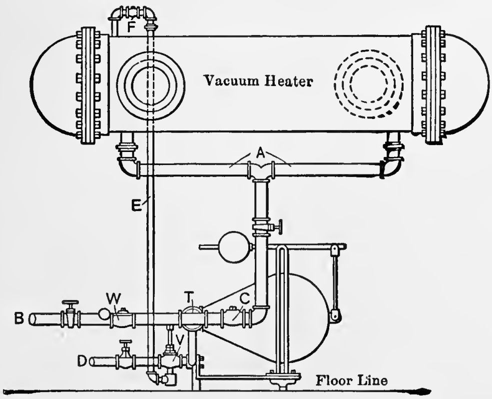


FIG. 456. Method of Draining Heater under Vacuum.

frequency, the ordinary method is not applicable and the arrangement shown in Fig. 457 may be used. The trap is placed below the receiver as indicated. The delivery pipe is provided with a weighted check or resistance valve W set so as to open outwards from the trap, also a spring water relief valve R. Another weighted check P is placed in the line leading from the vent to the atmosphere, and a plain check C in the line leading back into the receiver. This arrangement of valves permits the venting of the trap after discharge and effectually excludes air from the trap when there is less than atmospheric pressure on the receiver. With the relief valve set to open at

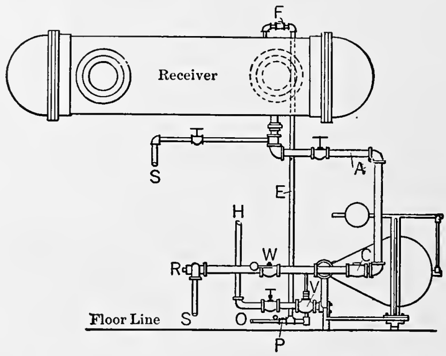


Fig. 457. Method of Draining Receiver under Alternate Vacuum and Pressure.

a pressure in excess of the maximum receiver pressure it acts as a "stop" in the pipe and the water must enter the trap. When the trap discharges, the live steam supplied through the pipe attached to the steam valve forces the water through the weighted check and relief valves into the sewer or receiving tank. When working with a vacuum, the pressures in receiver and trap are equalized through the vent connection and the condensation flows into the trap by gravity. The operation of discharge is the same as in the case of pressure.

333. The Steam Loop. — Fig. 458 illustrates the principles of the "steam loop" for automatically returning high-pressure drips to the boiler. In the figure the loop is returning the condensation from a steam separator to a boiler above the level of the separator. The apparatus is very simple, consisting of a horizontal and two vertical lengths of plain pipe placed as indicated. Pipes R and B may be covered but "horizontal" A is left uncovered, as its function is that of a condenser. The operation is as follows: Circulation is first started by opening stop valve O at the bottom of the drop leg until steam escapes. The valve is then closed and the steam in the horizontal A condenses and gravitates to the drop leg B. On account of the slight reduction in pressure in the horizontal a mixture of spray and steam flows from the separator chamber to the horizontal, and, condensing, gravitates to the drop leg. The column of water in the drop leg rises until its static head balances the difference in pressure in the riser R and the horizontal. In other words, a decrease in pressure in the horizontal produces similar effects on the contents of the riser and drop leg but in a degree inversely proportional to their densities. Any further accumulation causes an equal amount to pass from the bottom of the column to the boiler, since the pressure in the boiler is then less than that at the bottom of the column; that is, the steam pressure on the top of the water column

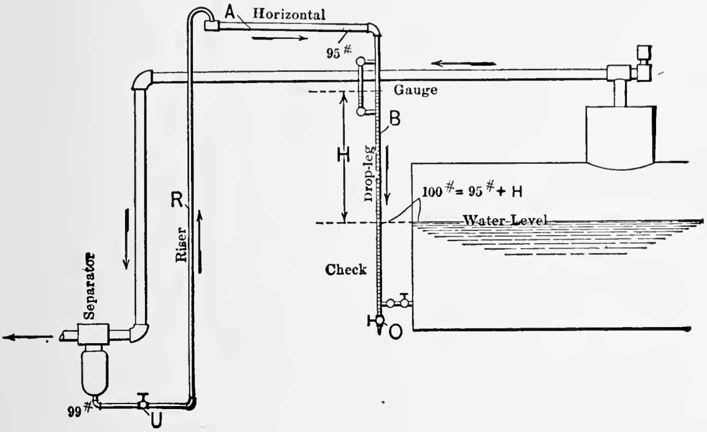


FIG. 458. General Arrangement of the Simple "Steam Loop."

plus the hydrostatic head H is greater than the pressure in the boiler. Once started the process is continuous and requires no further attention.

334. The Holly Loop. — In the application of the steam loop where many points requiring drainage are connected to many boilers and conditions are more complex, some method other than the simple one of radiation may be advisable to secure the necessary lower pressure at the top of the loop. Such a method is illustrated in Fig. 459. This arrangement differs from the simple loop in that all condensation first gravitates to a "Holly" receiver (shown in detail in Fig. 460) before passing into the "riser." The receiver is placed below the lowest point to be drained and serves as a storage for large or unusual quantities of water and enables the riser to act at a constant rate independent of variable discharge into the receiver. Furthermore, the lower pressure in the discharge chamber necessary to secure the lifting of the mingled steam and water through the riser, instead of being created by condensation as in the simple loop, is produced by a reducing valve B discharging into the feed-water heater. The operation of the Holly loop is as follows: Circulation is started by opening valve D until steam appears. Valve D is then closed and the reducing valve is put into commission. Condensation from separators, traps, and pipes gravitates to the "receiver," from which it is forced into the "riser" in the form of a spray. The spraying effect is produced by a series of holes

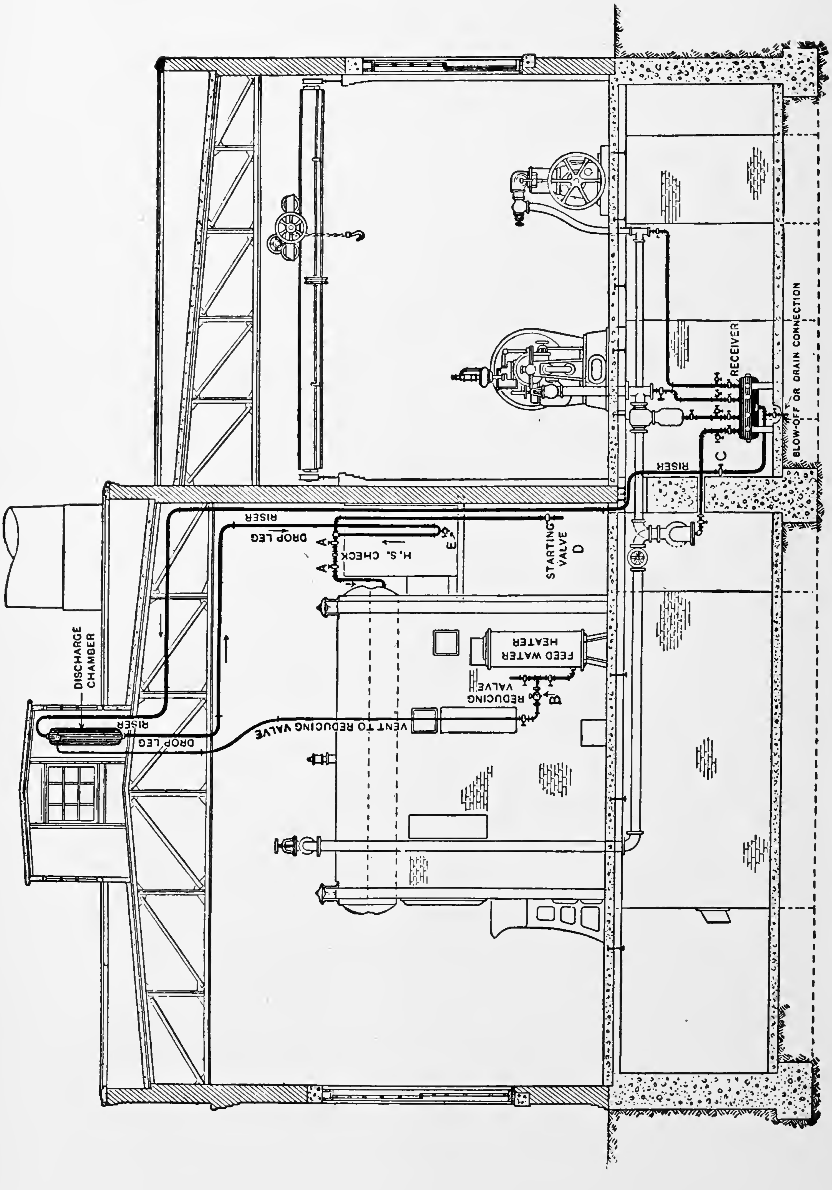


Fig. 459. General Arrangement of the Holly Loop.

drilled in pipe A, Fig. 460. From this receiver the spray and moisture rise to the "discharge chamber," on account of the lower pressure at that point, where the steam and entrained water are separated, the water gravitating to the bottom of the chamber and thence to the drop leg, and the steam discharging through the reducing valve into the heater. The principles of operation are exactly the same as in the simple steam loop.

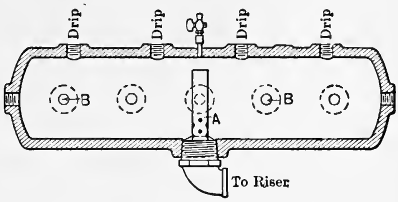


Fig. 460. Holly Receiver.

335. Returns Tank and Pump.—Low-pressure drips in connection with heating systems may be returned to the boiler along with the condensation from the heating system by a combined pump and receiver as shown in Fig. 461. The height of water in the tank controls the operation of the pump through the medium of a float and throttle valve. This combination of float and balanced throttle valve is sometimes called a "pump governor." In the illustration the pump forces the returns through a closed heater before

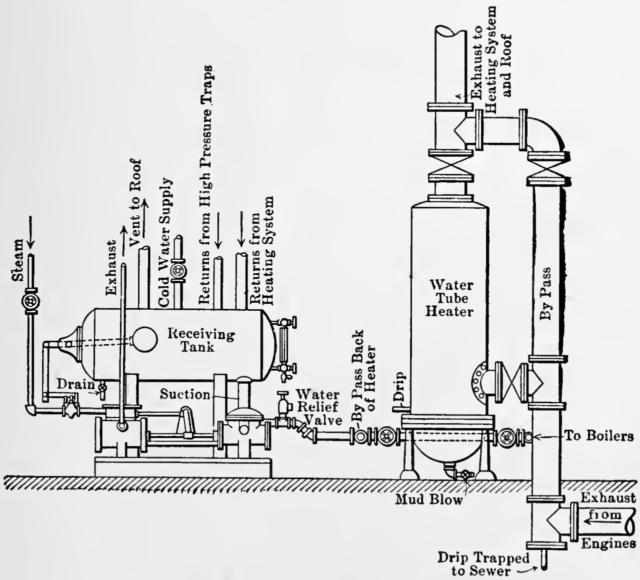


Fig. 461. Returns Tank and Pump.

delivering them to the boiler, though they are oftentimes returned directly. The tank is vented to the atmosphere to prevent it from becoming "air bound." The cold-water supply or make-up water is sometimes discharged into the receiving tank as indicated. With open heaters the cold supply is ordinarily controlled by a float within the heater itself.

336. Office Building Drains. — In the power plants of tall office buildings the public sewers are often above the basement level, and it is necessary to remove all liquid wastes mechanically.

The Shone pneumatic ejector has been found to serve this purpose effectually. This apparatus is placed in a pit in the basement floor into which all sewage, drips from engines, washings from boilers, and ground water gravitate, and are automatically discharged into the street sewer by means of compressed air.

Fig. 462 gives a sectional view of a Shone ejector of ordinary construction. It consists essentially of a closed vessel furnished with inlet and discharge connections fitted with check valves, *A* and *B*, opening in opposite directions with regard to the ejector. Two cast-iron bells, *C* and *D*, are linked to each other, in reverse positions, the rising and falling of which control the supply of compressed air through the agency of automatic valve *E*.

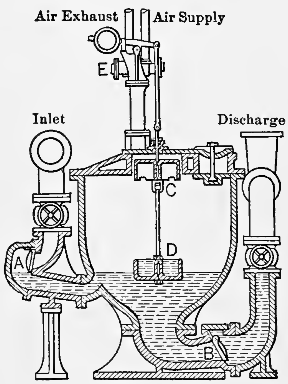


Fig. 462. Shone Ejector.

The bells are shown in their lowest position, the supply of compressed air is cut off from the ejector, and the inside of the vessel is open to the atmosphere. The sewage gravitating into the ejector raises the bell *C*, which in turn actuates the automatic valve *E*, thereby closing the connection between the inside of the ejector and the atmosphere and opening the connection with the compressed air. The air pressure expels the contents through the bell-mouthed opening at the bottom and the discharge valve *B* into the main sewer. Discharge continues until the level falls to such a point that the weight of the sewage retained in the bell *D* is sufficient to pull it down, thereby reversing the automatic valve. This cuts off the supply of compressed air and reduces the pressure to that of the atmosphere.

The positions of the bells are so adjusted that compressed air is not admitted until the ejector is full, and is not allowed to exhaust until emptied down to the discharge level; thus the ejector discharges a fixed quantity each time it operates.

Two ejectors, each of a capacity suitable for handling the average flow of tributary sewage and so arranged that they can work either independently or together, are usually installed at each ejector station.

The main sanitary sewer of the building usually discharges directly into the ejectors, the surface water, drips, etc., being collected in a neighboring sump. The latter is connected to the sanitary sewer through a trap or back-water valve.

CHAPTER XV

PIPING AND PIPE FITTINGS

337. General. — The advent of high pressures and superheat is responsible for the elimination of many of the older systems of piping, the tendency being towards greater uniformity in design, particularly in electric central-station work. In isolated stations the conditions of operation and installation are so variable that each case presents an entirely different problem. In any system of piping the fundamental object is to conduct the fluid in the safest and most economical manner. The material should be the best obtainable and the system so flexible that a break-down in one element will not necessitate the closing down of the entire plant. On the other hand, flexibility increases the number of parts and, unless first cost is of little importance, tends to weaken the system as a whole. It is a safe general proposition to say that the best pipe and fittings, irrespective of first cost, will prove the most economical in the end, but few owners of power plants are willing to take this view.

338. Drawings. — An assembly drawing of the entire installation giving the location of all valves and fittings is necessary in order to avoid interference, and particularly where a number of fittings are to be close together. Detailed drawings should also be provided of each division of the piping to facilitate installation, as, for example, the high-pressure steam, the exhaust steam, the feed water, the condensing water, the oil, the heating, and the sanitary piping. As a rule, lower and more uniform bids will be obtained from an isometric or perspective sketch, as in Fig. 463, than from conventional plan and elevation drawings, due, no doubt, to the greater ease with which the drawing is interpreted. A complete set of specifications for a piping system is given in paragraph 479 and illustrates the usual practice along this line.

339. Materials for Pipes and Fittings. — The following materials are used in the construction of pipes for steam, water, and gases.

	Average Tensile Strength.
Low-carbon or mild steel	65,000 lb. per sq. in.
Wrought iron	50,000 lb. per sq. in.
Cast iron, high grade	20,000 lb. per sq. in.
Cast steel	50,000 lb. per sq. in.
Wrought copper	33,000 lb. per sq. in.
Brass	18,000 lb. per sq. in.
Special alloys and compounds	15,000–85,000 lb. per sq. in.

Mild Steel. — The greater portion of the piping in the average steam power plant is of mild steel, lap or butt welded for high pressures and riveted for very low pressures and large diameters. Steel pipe is considerably cheaper than that manufactured from other material and fulfills practically all requirements for general service.

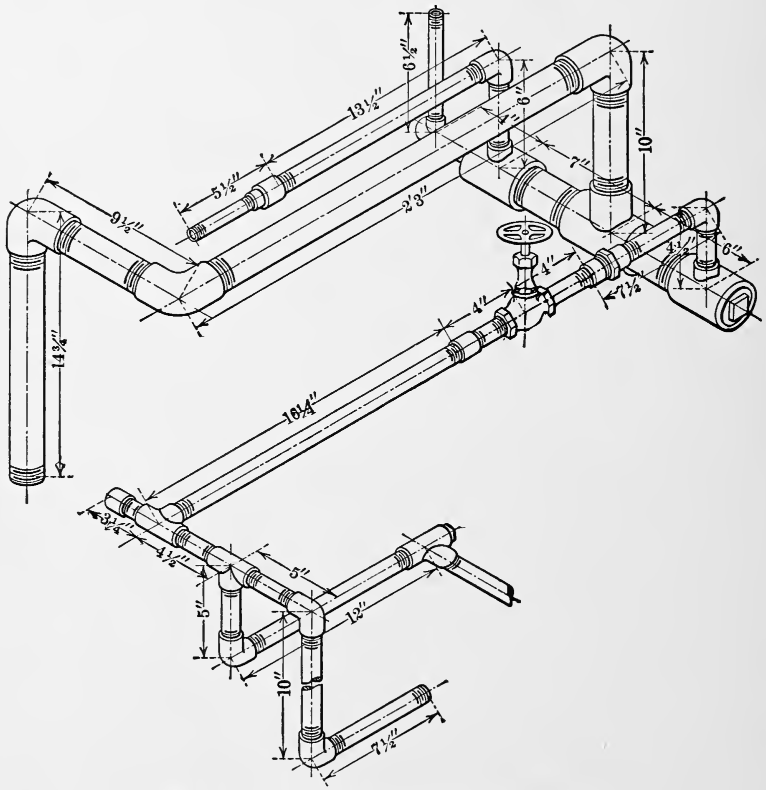


Fig. 463. A Typical Isometric Pipe Drawing.

Wrought Iron. — "Wrought-iron" pipe in a commercial sense refers to mild-steel pipe and unless stress is laid upon the term "puddled iron" mild steel is ordinarily furnished. Puddled-iron pipe is not much in evidence in steam power plant work since mild steel is cheaper and fulfills all requirements. Wrought-iron pipe appears to resist corrosion to a greater extent than mild-steel pipe. Numerous laboratory investigations have been made of late which show that mild steel is equal if not superior to wrought iron in many ways but in actual service the latter appears to have the longer life.

Cast-iron Pipes. — Cast iron is little used for high-pressure steam piping except occasionally in the construction of manifold headers.

The chief objections to cast iron for high-pressure steam are its weight and lack of homogeneity. It is mostly used in connection with water service and sanitation. For manifold headers and the like steel pipes with welded connections have superseded cast iron in the modern plant.

Cast-steel Pipe. — Cast-steel headers are sometimes used in power plants for highly superheated steam, since the material is not affected by temperature variations to the same extent as mild steel. High first cost and the difficulty of securing castings free from blowholes have prevented its more general use.

Copper Pipes. — Copper steam pipes were in common use for many years in marine service on account of their flexibility. To increase the bursting strength, pipes above 6 inches in diameter were generally wound with a close spiral of copper or composition wire. In recent years wrought-iron and steel pipe bends have practically superseded copper for flexible connections. As a rule the use of copper pipes should be avoided on account of the rapid deterioration of the metal under high temperatures and stress variations. The cost is prohibitive for most purposes and this alone prevents it from being seriously considered in the manufacture of pipe. Copper expansion joints are occasionally used in low-pressure work.

Brass Pipes. — Brass is little used in the construction of pipes on account of its high cost. It withstands corrosive action much better than iron or steel and is sometimes used in connecting the feed main with the boiler drum. Special alloys, nickel steel, "ferrosteel," malleable iron, and the like have been used in the manufacture of pipes, and possess points of superiority over wrought iron and steel for some purposes, but the cost is prohibitive for average steam power plant practice.

Materials for Fittings. — Elbows, tees, flanges, and similar fittings are usually made of cast iron, malleable iron, or pressed steel, though cast steel, "ferrosteel," and other steel compounds are used to a limited extent. Standard cast-iron fittings are recommended for saturated steam and for pressures of 100 pounds per square inch or less, and extra heavy cast-iron fittings for higher pressures. Malleable-iron fittings are lighter and neater than cast-iron and are extensively used for small sizes of steam and gas pipe. Cast or pressed steel is recommended for very high pressures and superheat.

340. Size and Strength of Commercial Pipe. — Wrought-iron and mild-steel pipes are marketed in standard sizes. Those most commonly used in steam power plants are designated as

1. Merchant or standard pipe.
2. Full-weight pipe.

3. Large O.D. pipe.
4. Extra heavy.
5. Double extra heavy.

Table 118 gives the dimensions of standard "full-weight" pipe, which is specified by the nominal inside diameter up to and including 12 inches and based on the Briggs' standard. Pipes larger than 12 inches are designated by the actual outside diameter (O.D.), and are made in various weights as determined by the thickness of metal specified. Manufacturers specify that "full-weight" pipe may have a variation of 5 per cent above or 5 per cent below the nominal or table weights, but merchant pipe, which is the standard pipe of commerce, such as manufacturers and jobbers usually carry in stock, is almost invariably under the nominal weight. It varies somewhat among the different mills, but usually lies between 5 and 10 per cent under the table weight. The smaller sizes of merchant pipe, $\frac{1}{8}$ inch to 3 inches, are butt-welded and the larger sizes are lap-welded.

Extra heavy and double extra heavy pipe have the same external diameter as the standard, but are of greater thickness and hence the internal diameter is smaller. Taking the thickness of the standard pipe as 1, that of the extra heavy is approximately 1.4 and of the double extra heavy 2.8.

Wrought-iron and steel pipes are ordinarily designed with factors of safety of from 6 to 15, with an average not far from 10. The standard hydrostatic tests to which the various pipes are subjected at the mills are as follows:

	Hydrostatic Pressure, Lb. per Sq. In.
Standard, butt-welded, $\frac{1}{8}$–3 in.	600 to 1,000
Standard, lap-welded, 3–12 in.	500 to 1,000
Extra heavy, butt-welded, $\frac{1}{8}$–3 in.	600 to 1,500
Extra heavy, lap-welded, $1\frac{1}{2}$–12 in.	600 to 1,500
Double extra heavy, butt-welded, $\frac{1}{8}$–$2\frac{1}{2}$ in.	600 to 1,500
Double extra heavy, lap welded, $1\frac{1}{2}$–8 in.	1,200 to 1,500

The pressure necessary to burst piping is far above anything likely to occur in ordinary practice on account of the thickness of material necessary to permit of threading. (See Table 117.)

Riveted Pipes. — For low pressures and large diameters, pipes are constructed of thin sheets of boiler steel with riveted joints, the seams being either longitudinal and circumferential, or spiral. Such pipes are not necessarily limited to large sizes and low pressures, though this is the usual practice.

Pipe fittings are classed as screwed, flanged or welded.

TABLE 117.

BURSTING PRESSURE OF "STANDARD" MILD-STEEL PIPE.*

No. of Specimen.	Nominal Diameter, Inches.	Actual Bursting Pressure, Lb. per Sq. In.	No. of Specimen.	Nominal Diameter, Inches.	Actual Bursting Pressure, Lb. per Sq. In.
†1	1	7800	‡7	3	3500
†2	1	7700	‡8	3	3500
†3	1	7700	†9	3	3000
		Average 7730			Average 3330
†4	2	4950	§10	4	1800
†5	2	4800	§11	4	1700
‡6	2	5500			Average 1750
		Average 5080	§12	5	2500
			§13	5	2600
					Average 2550
			§14	6	3200

* Tests made at Armour Institute of Technology.

Specimens were taken at random from a lot of new pipe; length of test specimens, 5 ft. Specimens threaded at both ends and capped.

† Failed at weld. ‡ Failed in body of pipe. § Failed at threaded end.

341. Screwed Fittings, Pipe Threads. — For screw connections the ends of pipes and fittings are threaded to conform to the Briggs or United States standard system, as shown in Fig. 464. The end of the pipe is tapered 1 to 32 with the axis, the angle of the thread being

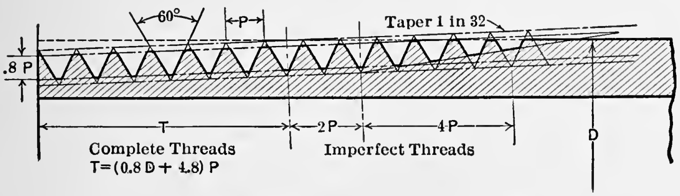


FIG. 464. Standard U. S. Pipe Thread.

60 degrees and slightly rounded at top and bottom. The proper length of perfect threads is given by the formula

$$T = \frac{(0.8\,D + 4.8)}{n}, \tag{267}$$

in which

T = length in inches,
D = actual external diameter of the tube, inches,
n = number of threads per inch.

The imperfect portion of the thread is simply incidental to the process of cutting. The object of the taper is to facilitate "taking hold" in making up the joint. Table 118 gives the number of threads per

TABLE 118.

STANDARD DIMENSIONS OF WROUGHT-IRON AND STEEL STEAM, GAS, AND WATER PIPE.

Diameter.			Nominal Thickness.	Circumference.		Transverse Areas.			Length of Pipe per Square Foot of		Length of Pipe Containing One Cubic Foot.	Nominal Weight per Foot.	Number of Threads per Inch of Screw.
Nominal Internal.	Actual External Diameter.	Approximate Internal Diameter.		External.	Internal.	External.	Internal.	Metal.	External Surface.	Internal Surface.			
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Sq. Inch.	Sq. Inch.	Sq. Inch.	Feet.	Feet.	Feet.	Pounds.	
1/8	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513	.241	27
1/4	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18
3/8	.675	.494	.091	2.121	1.552	.358	.1917	.1663	5.657	7.73	751.2	.559	18
1/2	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14
3/4	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270	1.115	14
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11 1/2
1 1/4	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11 1/2
1 1/2	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11 1/2
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11 1/2
2 1/2	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8
3 1/2	4	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8
4 1/2	5	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.49	8
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	23.271	8
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	28.177	8
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.065	8
11	11.75	11	....	36.914	34.558	108.434	95.033	13.401	.325	.347	1.51	45.028	8
12	12.75	12	....	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8

inch for various sizes of standard pipe. When properly constructed a screwed joint will hold against any pressure consistent with the strength of the pipe. For example, the ultimate bursting strength of a "standard" 2-inch pipe is about 5000 pounds per square inch, while the stripping strength of the joint (with perfect threads) is 225,000 pounds. The threads, however, are often poorly cut and the parts screwed together improperly cleaned and lubricated, thus causing leakage between the threads.

TABLE 119.

STANDARD BOILER TUBES.

Table of Standard Dimensions.

Diameter.		Standard Thickness.		Transverse Areas.		Area of Surface per Foot of Tube.		Nominal Weight per Foot — Lb.				
External.	Internal.	Nearest B.W.G.		External.	Internal.	External.	Internal.	Standard Thickness.	One Extra Wire Gauge.	Two Extra Wire Gauges.	Three Extra Wire Gauges.	Four Extra Wire Gauges.
Ins.	Ins.	No.	Ins.	Sq. In.	Sq. In.	Sq. Ft.	Sq. Ft.					
1	0.810	13	.095	0.785	0.515	.262	.212	0.90	1.04	1.13	1.24	1.35
1¼	1.060	13	.095	1.227	0.882	.327	.277	1.15	1.33	1.45	1.60	1.74
1½	1.310	13	.095	1.767	1.348	.392	.343	1.40	1.62	1.77	1.96	2.14
1¾	1.560	13	.095	2.405	1.911	.458	.408	1.66	1.91	2.09	2.31	2.53
2	1.810	13	.095	3.142	2.573	.523	.474	1.91	2.20	2.41	2.67	2.93
2¼	2.060	13	.095	3.976	3.333	.589	.539	2.16	2.49	2.73	3.03	3.32
2½	2.282	12	.109	4.909	4.090	.654	.597	2.75	3.05	3.39	3.72	4.12
2¾	2.532	12	.109	5.940	5.035	.720	.663	3.04	3.37	3.74	4.11	4.56
3	2.782	12	.109	7.069	6.079	.785	.728	3.33	3.69	4.10	4.51	5.00
3¼	3.010	11	.120	8.296	7.116	.851	.788	3.96	4.46	4.90	5.44	5.90
3½	3.260	11	.120	9.621	8.347	.916	.853	4.28	4.82	5.30	5.88	6.38
3¾	3.510	11	.120	11.045	9.676	.982	.919	4.60	5.18	5.69	6.32	6.86
4	3.732	10	.134	12.566	10.939	1.047	.977	5.47	6.09	6.76	7.34	8.23
4½	4.232	10	.134	15.904	14.066	1.178	1.108	6.17	6.88	7.64	8.31	9.32
5	4.704	9	.148	19.635	17.379	1.309	1.231	7.58	8.52	9.27	10.40	11.23
6	5.670	8	.165	28.274	25.250	1.571	1.484	10.16	11.19	12.57	13.58	14.65

342. Flanged Fittings. — In cast-iron pipes, valves, tees, and other fittings the flange is always a part of the casting, but for joining the two ends of a steel or wrought-iron pipe the flanges may be fastened to the pipe in a number of ways. Fig. 465, *A* to *H*, illustrates methods most commonly used. In *A* to *C* the pipes are screwed into cast-iron or forged-steel flanges and the two faces, with metallic or composition gasket between, are drawn together by bolts. *A* illustrates the most common and inexpensive of flanged joints, which requires no special tools and can be made up at the place of erection. It gives satisfactory results for pressures of 100 pounds or less, but for higher pressures leakage is apt to take place between the threads. The flanges are sometimes made with a long thread and a recess which can be calked with soft metal. A similar joint is made with the pipe screwed beyond the face of the flange and the two faced off together, either plane or as shown in B, which is known as a *male and female* or *hydraulic*

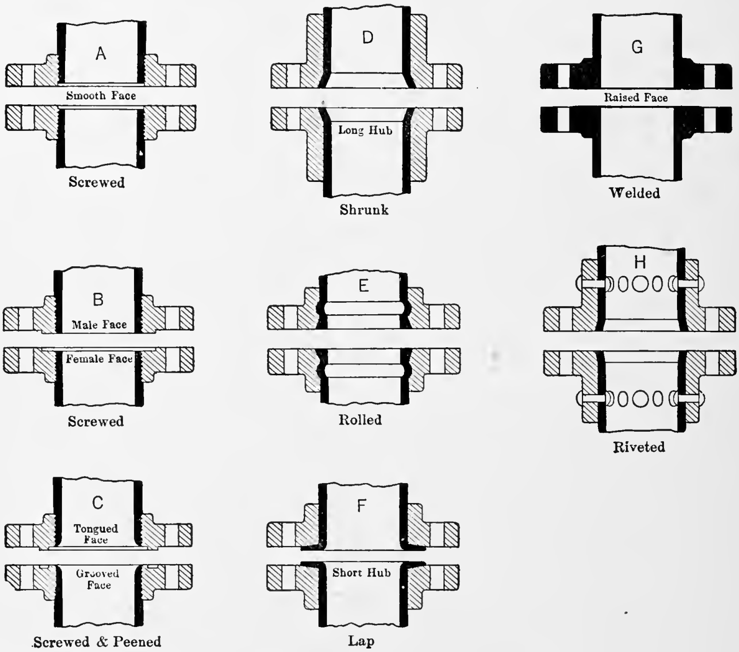


FIG. 465. Types of Pipe Flanges.

joint. This method forms a very reliable joint, since the ends of the pipe bear on the gasket, and the gasket is prevented from being blown out. An objection lies in the difficulty of opening the line to remove the gasket or replace a fitting. C is a modification known as the *tongued and grooved* joint, which uses an extremely narrow gasket. Such flanges may be subjected to severe strains when the bolts are drawn up, owing to the small area of contact. Corrugated copper or steel gaskets are recommended, since soft material is apt to be squeezed out. In C the ends of the pipe are *peened,* which is an improvement over the simple screwed joint. D illustrates a *shrunk* joint. The flanges are bored for a shrink fit and forced over the pipe when at a red heat. After cooling the end is beaded over into a recess on the face of the flange and a light cut taken from both. H shows a modification in which the hub is riveted to the pipe. E illustrates a joint constructed by rolling the pipe into a corrugation in the flange. The end of the pipe is then faced off flush.

TABLE 120.

DIMENSIONS OF CAST-IRON PIPE.*

Nominal Inside Diameter, Inches.	Standard Thickness and Weight.								
	Class A. 100 Feet Head. 43 Pounds Pressure.			Class B. 200 Feet Head. 86 Pounds Pressure.			Class C. 300 Feet Head. 130 Pounds Pressure.		
	Thickness, Inches.	Weight per Foot.	Weight per Length.	Thickness, Inches.	Weight per Foot.	Weight per Length.	Thickness, Inches.	Weight per Foot.	Weight per Length.
4	.42	20.0	240	.45	21.7	260	.48	23.3	280
6	.44	30.8	370	.48	33.3	400	.51	35.8	430
8	.46	42.9	515	.51	47.5	570	.56	52.1	625
10	.50	57.1	685	.57	63.8	765	.62	70.8	850
12	.54	72.5	870	.62	82.1	985	.68	91.7	1,100
14	.57	89.6	1,075	.66	102.5	1,230	.74	116.7	1,400
16	.60	108.3	1,300	.70	125.0	1,500	.80	143.8	1,725
18	.64	129.2	1,550	.75	150.0	1,800	.87	175.0	2,100
20	.67	150.0	1,800	.80	175.0	2,100	.92	208.3	2,500
24	.76	204.2	2,450	.89	233.3	2,800	1.04	279.2	3,350
30	.88	291.7	3,500	1.03	333.3	4,000	1.20	400.0	4,800
36	.99	391.7	4,700	1.15	454.2	5,450	1.36	545.8	6,550
42	1.10	512.5	6,150	1.28	591.7	7,100	1.54	716.7	8,600
48	1.26	666.7	8,000	1.42	750.0	9,000	1.71	908.3	10,900
54	1.35	800.0	9,600	1.55	933.3	11,200	1.90	1141.7	13,700
60	1.39	916.7	11,000	1.67	1104.2	13,250	2.00	1341.7	16,100
72	1.62	1283.4	15,400	1.95	1545.8	18,550	2.39	1904.2	22,850
84	1.72	1633.4	19,600	2.22	2104.2	25,250	......	......	....

* Adopted standards of Am. Water W'ks Ass'n. The above weights are per length to lay 12 feet, including standard sockets; proportionate allowance to be made for any variation. All weights are approximate.

Dimensions of Riveted Steel Pipes: Power, March 7, 1911, p. 377.

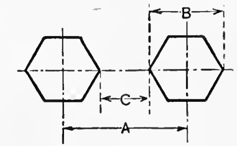
One of the best commercial joints is illustrated by *F* and is known as the *lap joint.* The pipe is expanded as indicated and a light cut is then taken from the flared ends to insure a tight joint. The flanges are loose and permit of considerable flexibility in shifting them through various angles. This is sometimes called the *Van Stone* joint.

Pipes with flanges *welded* on the end as in *G* have proved the most reliable of all and though costly are considered the standard for high-pressure and high-temperature work. The faces are ordinarily raised $\frac{1}{32}$ to $\frac{1}{16}$ inch inside the bolt holes and ground to a steam-tight fit, so that thick gaskets are unnecessary.

For moderately high pressures and temperatures any of the joints when well made will prove satisfactory. For extremely high pressures and temperatures the lap or welded joints are preferable.

TABLE 121.

THE AMERICAN STANDARD OF 1914.

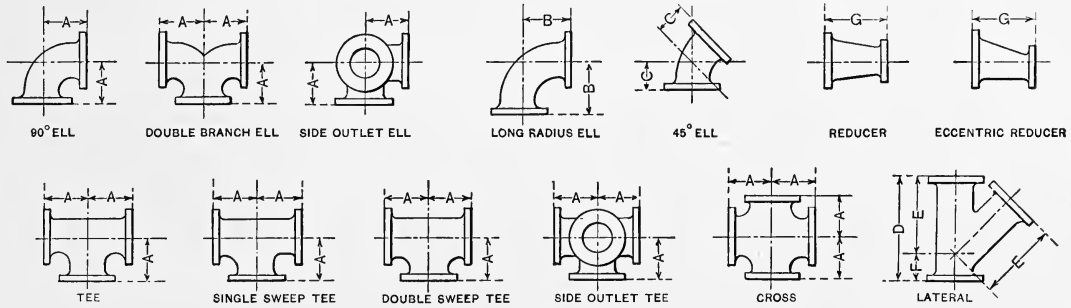
Diameter of Pipe.	Thickness of Pipe.	Minimum Thickness (Fractions of an Inch).	Stress on Pipe per Sq. Inch.	Diameter of Flange.	Thickness of Flange.	Width of Flange Face.	Diameter of Bolt Circle.	No. of Bolts.	Diameter of Bolts.	Effective Area.	Stress per Sq. In. on Bolt Metal.	Diameter of Bolt Holes.	 (On Chord)		
													A.	B.	C.
							Standard Weight Flanges.								
1	0.43	$\frac{7}{16}$	143	4	$\frac{7}{16}$	$1\frac{1}{2}$	3	4	$\frac{7}{16}$	0.093	264	$\frac{9}{16}$	2.12	0.91	1.21
$1\frac{1}{4}$	0.44	$\frac{7}{16}$	178	$4\frac{1}{2}$	$\frac{1}{2}$	$1\frac{5}{8}$	$3\frac{3}{8}$	4	$\frac{7}{16}$	0.093	412	$\frac{9}{16}$	2.38	0.91	1.47
$1\frac{1}{2}$	0.45	$\frac{7}{16}$	214	5	$\frac{9}{16}$	$1\frac{3}{4}$	$3\frac{7}{8}$	4	$\frac{1}{2}$	0.126	438	$\frac{5}{8}$	2.73	1.00	1.73
2	0.46	$\frac{7}{16}$	286	6	$\frac{5}{8}$	2	$4\frac{3}{4}$	4	$\frac{5}{8}$	0.202	486	$\frac{3}{4}$	3.35	1.21	2.14
$2\frac{1}{2}$	0.48	$\frac{7}{16}$	357	7	$\frac{11}{16}$	$2\frac{1}{4}$	$5\frac{1}{2}$	4	$\frac{5}{8}$	0.202	750	$\frac{3}{4}$	3.88	1.21	2.67
3	0.50	$\frac{7}{16}$	428	$7\frac{1}{2}$	$\frac{3}{4}$	$2\frac{1}{4}$	6	4	$\frac{5}{8}$	0.202	1093	$\frac{3}{4}$	4.23	1.21	3.02
$3\frac{1}{2}$	0.52	$\frac{7}{16}$	500	$8\frac{1}{2}$	$\frac{13}{16}$	$2\frac{1}{2}$	7	4	$\frac{5}{8}$	0.202	1488	$\frac{3}{4}$	4.94	1.21	3.73
4	0.53	$\frac{1}{2}$	500	9	$\frac{15}{16}$	$2\frac{1}{2}$	$7\frac{1}{2}$	8	$\frac{5}{8}$	0.202	972	$\frac{3}{4}$	2.87	1.21	1.56
$4\frac{1}{2}$	0.55	$\frac{1}{2}$	562	$9\frac{1}{4}$	$\frac{15}{16}$	$2\frac{3}{8}$	$7\frac{3}{4}$	8	$\frac{3}{4}$	0.302	823	$\frac{7}{8}$	2.96	1.44	1.52
5	0.56	$\frac{1}{2}$	625	10	$\frac{15}{16}$	$2\frac{1}{2}$	$8\frac{1}{2}$	8	$\frac{3}{4}$	0.302	1016	$\frac{7}{8}$	3.25	1.44	1.81
6	0.60	$\frac{9}{16}$	667	11	1	$2\frac{1}{2}$	$9\frac{1}{2}$	8	$\frac{3}{4}$	0.302	1463	$\frac{7}{8}$	3.63	1.44	2.19
7	0.63	$\frac{5}{8}$	700	$12\frac{1}{2}$	$1\frac{1}{16}$	$2\frac{3}{4}$	$10\frac{3}{4}$	8	$\frac{3}{4}$	0.302	1991	$\frac{7}{8}$	4.11	1.44	2.67
8	0.66	$\frac{5}{8}$	800	$13\frac{1}{2}$	$1\frac{1}{8}$	$2\frac{3}{4}$	$11\frac{3}{4}$	8	$\frac{3}{4}$	0.302	2600	$\frac{7}{8}$	4.50	1.44	3.06
9	0.70	$\frac{11}{16}$	818	15	$1\frac{1}{8}$	3	$13\frac{1}{4}$	12	$\frac{3}{4}$	0.302	2194	$\frac{7}{8}$	3.43	1.44	1.99
10	0.73	$\frac{3}{4}$	833	16	$1\frac{3}{16}$	3	$14\frac{1}{4}$	12	$\frac{7}{8}$	0.420	1948	1	3.69	1.66	2.03
12	0.80	$\frac{13}{16}$	923	19	$1\frac{1}{4}$	$3\frac{1}{2}$	17	12	$\frac{7}{8}$	0.420	2805	1	4.40	1.66	2.74
14	0.86	$\frac{7}{8}$	1000	21	$1\frac{3}{8}$	$3\frac{1}{2}$	$18\frac{3}{4}$	12	1	0.550	2915	$1\frac{1}{8}$	4.86	1.88	2.98
15	0.90	$\frac{7}{8}$	1072	$22\frac{1}{4}$	$1\frac{3}{8}$	$3\frac{5}{8}$	20	16	1	0.550	2510	$1\frac{1}{8}$	3.90	1.88	2.02
16	0.93	1	1000	$23\frac{1}{2}$	$1\frac{7}{16}$	$3\frac{3}{4}$	$21\frac{1}{4}$	16	1	0.550	2856	$1\frac{1}{8}$	4.14	1.88	2.26
18	1.00	$1\frac{1}{16}$	1059	25	$1\frac{9}{16}$	$3\frac{1}{2}$	$22\frac{3}{4}$	16	$1\frac{1}{8}$	0.694	2865	$1\frac{1}{4}$	4.44	2.09	2.35
20	1.07	$1\frac{1}{8}$	1111	$27\frac{1}{2}$	$1\frac{11}{16}$	$3\frac{3}{4}$	25	20	$1\frac{1}{8}$	0.694	2829	$1\frac{1}{4}$	3.91	2.09	1.82
22	1.13	$1\frac{3}{16}$	1158	$29\frac{1}{2}$	$1\frac{13}{16}$	$3\frac{3}{4}$	$27\frac{1}{4}$	20	$1\frac{1}{4}$	0.893	2660	$1\frac{3}{8}$	4.26	2.31	1.95
24	1.20	$1\frac{1}{4}$	1200	32	$1\frac{7}{8}$	4	$29\frac{1}{2}$	20	$1\frac{1}{4}$	0.893	3166	$1\frac{3}{8}$	4.62	2.31	2.31

Extra Heavy Flanges.

1	0.45	$\frac{1}{2}$	250	$4\frac{1}{2}$	$\frac{11}{16}$	$1\frac{3}{4}$	$3\frac{1}{4}$	4	$\frac{1}{2}$	0.126	389	$\frac{5}{8}$	2.29	1.00	1.29
$1\frac{1}{4}$	0.47	$\frac{1}{2}$	312	5	$\frac{3}{4}$	$1\frac{7}{8}$	$3\frac{3}{4}$	4	$\frac{1}{2}$	0.126	609	$\frac{5}{8}$	2.65	1.00	1.65
$1\frac{1}{2}$	0.49	$\frac{1}{2}$	375	6	$\frac{13}{16}$	$2\frac{1}{4}$	$4\frac{1}{2}$	4	$\frac{5}{8}$	0.202	547	$\frac{3}{4}$	3.17	1.21	1.96
2	0.51	$\frac{1}{2}$	500	$6\frac{1}{2}$	$\frac{7}{8}$	$2\frac{1}{4}$	5	4	$\frac{5}{8}$	0.202	972	$\frac{3}{4}$	3.53	1.21	2.32
$2\frac{1}{2}$	0.53	$\frac{9}{16}$	555	$7\frac{1}{2}$	1	$2\frac{1}{2}$	$5\frac{7}{8}$	4	$\frac{3}{4}$	0.302	1016	$\frac{7}{8}$	4.15	1.44	2.71
3	0.56	$\frac{9}{16}$	667	$8\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{5}{8}$	$6\frac{5}{8}$	8	$\frac{3}{4}$	0.302	731	$\frac{7}{8}$	2.53	1.44	1.09
$3\frac{1}{2}$	0.59	$\frac{9}{16}$	778	9	$1\frac{3}{16}$	$2\frac{3}{4}$	$7\frac{1}{4}$	8	$\frac{3}{4}$	0.302	995	$\frac{7}{8}$	2.77	1.44	1.33
4	0.61	$\frac{5}{8}$	800	10	$1\frac{1}{4}$	3	$7\frac{7}{8}$	8	$\frac{3}{4}$	0.302	1300	$\frac{7}{8}$	3.01	1.44	1.57
$4\frac{1}{2}$	0.64	$\frac{5}{8}$	900	$10\frac{1}{2}$	$1\frac{5}{16}$	3	$8\frac{1}{2}$	8	$\frac{3}{4}$	0.302	1646	$\frac{7}{8}$	3.25	1.44	1.81
5	0.67	$\frac{11}{16}$	909	11	$1\frac{3}{8}$	3	$9\frac{1}{4}$	8	$\frac{3}{4}$	0.302	2032	$\frac{7}{8}$	3.53	1.44	2.09
6	0.72	$\frac{3}{4}$	1000	$12\frac{1}{2}$	$1\frac{7}{16}$	$3\frac{1}{4}$	$10\frac{5}{8}$	12	$\frac{3}{4}$	0.302	1950	$\frac{7}{8}$	2.75	1.44	1.31
7	0.78	$\frac{13}{16}$	1077	14	$1\frac{1}{2}$	$3\frac{1}{2}$	$11\frac{7}{8}$	12	$\frac{7}{8}$	0.420	1909	1	3.07	1.66	1.41
8	0.83	$\frac{13}{16}$	1230	15	$1\frac{5}{8}$	$3\frac{1}{2}$	13	12	$\frac{7}{8}$	0.420	2493	1	3.36	1.66	1.70
9	0.89	$\frac{7}{8}$	1285	$16\frac{1}{4}$	$1\frac{3}{4}$	$3\frac{5}{8}$	14	12	1	0.550	2410	$1\frac{1}{8}$	3.62	1.88	1.74
10	0.94	$\frac{15}{16}$	1333	$17\frac{1}{2}$	$1\frac{7}{8}$	$3\frac{3}{4}$	$15\frac{1}{4}$	16	1	0.550	2231	$1\frac{1}{8}$	2.97	1.88	1.09
12	1.05	1	1500	$20\frac{1}{2}$	2	$4\frac{1}{4}$	$17\frac{3}{4}$	16	$1\frac{1}{8}$	0.694	2546	$1\frac{1}{4}$	3.46	2.09	1.37
14	1.16	$1\frac{1}{8}$	1555	23	$2\frac{1}{8}$	$4\frac{1}{2}$	$20\frac{1}{4}$	20	$1\frac{1}{8}$	0.694	2773	$1\frac{1}{4}$	3.17	2.09	1.08
15	1.21	$1\frac{3}{16}$	1579	$24\frac{1}{2}$	$2\frac{3}{16}$	$4\frac{3}{4}$	$21\frac{1}{2}$	20	$1\frac{1}{4}$	0.893	2473	$1\frac{3}{8}$	3.36	2.31	1.05
16	1.27	$1\frac{1}{4}$	1600	$25\frac{1}{2}$	$2\frac{1}{4}$	$4\frac{3}{4}$	$22\frac{1}{2}$	20	$1\frac{1}{4}$	0.893	2814	$1\frac{3}{8}$	3.52	2.31	1.21
18	1.37	$1\frac{3}{8}$	1636	28	$2\frac{3}{8}$	5	$24\frac{3}{4}$	24	$1\frac{1}{4}$	0.893	2968	$1\frac{3}{8}$	3.23	2.31	0.92
20	1.48	$1\frac{1}{2}$	1666	$30\frac{1}{2}$	$2\frac{1}{2}$	$5\frac{1}{4}$	27	24	$1\frac{3}{8}$	1.057	3096	$1\frac{1}{2}$	3.52	2.53	0.99
22	1.59	$1\frac{9}{16}$	1760	33	$2\frac{5}{8}$	$5\frac{1}{4}$	$29\frac{1}{4}$	24	$1\frac{1}{2}$	1.295	3058	$1\frac{5}{8}$	3.81	2.75	1.06
24	1.70	$1\frac{5}{8}$	1846	36	$2\frac{3}{4}$	$5\frac{3}{4}$	32	24	$1\frac{5}{8}$	1.515	3110	$1\frac{3}{4}$	4.18	2.96	1.22
26	1.81	$1\frac{13}{16}$	1793	$38\frac{1}{4}$	$2\frac{13}{16}$	$6\frac{1}{8}$	$34\frac{1}{2}$	28	$1\frac{5}{8}$	1.515	3126	$1\frac{3}{4}$	3.86	2.96	0.90
28	1.91	$1\frac{7}{8}$	1866	$40\frac{3}{4}$	$2\frac{15}{16}$	$6\frac{3}{8}$	37	28	$1\frac{5}{8}$	1.515	3629	$1\frac{3}{4}$	4.14	2.96	1.18

TABLE 122.

THE AMERICAN STANDARD OF 1914.



Standard Flanged Fittings. — Straight Sizes.

	Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6
A–A	Face to Face	7	7½	8	9	10	11	12	13	14	15	16
A	Center to Face	3½	3¾	4	4½	5	5½	6	6½	7	7½	8
B	Center to Face of Long Radius Ells	5	5½	6	6½	7	7¾	8½	9	9½	10¼	11½
C	Center to Face of 45-Deg. Ells	1¾	2	2¼	2½	3	3	3½	4	4	4½	5
D	Face to Face Laterals	7½	8	9	10½	12	13	14½	15	15½	17	18
E	Center to Face	5¾	6¼	7	8	9½	10	11½	12	12½	13½	14½
F	Center to Face	1¾	1¾	2	2½	2½	3	3	3	3	3½	3½
G	Face to Face Reducer	...	...	...	...	...	6	6½	7	7½	8	9
	Diameter of Flange	4	4½	5	6	7	7½	8½	9	9¼	10	11
	Thickness of Flange	7/16	½	9/16	⅝	11/16	¾	13/16	15/16	15/16	15/16	1
	Diameter of Bolt Circle	3	3⅜	3⅞	4¾	5½	6	7	7½	7¾	8½	9½
	No. of Bolts	4	4	4	4	4	4	4	8	8	8	8
	Diameter of Bolts	7/16	7/16	½	⅝	⅝	⅝	⅝	⅝	¾	¾	¾
	Minimum Metal Thickness of Body	7/16	7/16	7/16	7/16	7/16	7/16	7/16	½	½	½	9/16

	Size	7	8	9	10	12	14	15	16	18	20	22	24
A–A	Face to Face	17	18	20	22	24	28	29	30	33	36	40	44
A	Center to Face	8½	9	10	11	12	14	14½	15	16½	18	20	22
B	Center to Face of Long Radius Ells	12¾	14	15¼	16½	19	21½	22¾	24	26½	29	31½	34
C	Center to Face of 45-Deg. Ells	5½	5½	6	6½	7½	7½	8	8	8½	9½	10	11
D	Face to Face Laterals	20½	22	24	25½	30	33	34½	36½	39	43	46	49½
E	Center to Face	16½	17½	19½	20½	24½	27	28½	30	32	35	37½	40½
F	Center to Face	4	4½	4½	5	5½	6	6	6½	7	8	8½	9
G	Face to Face Reducer	10	11	11½	12	14	16	17	18	19	20	22	24
	Diameter of Flange	12½	13½	15	16	19	21	22¼	23½	25	27½	29½	32
	Thickness of Flange	1 1/16	1⅛	1⅛	1 3/16	1¼	1⅜	1⅜	1 7/16	1 9/16	1 11/16	1 13/16	1⅞
	Diameter of Bolt Circle	10¾	11¾	13¼	14¼	17	18¾	20	21¼	22¾	25	27¼	29½
	No. of Bolts	8	8	12	12	12	12	16	16	16	20	20	20
	Diameter of Bolts	¾	¾	¾	⅞	⅞	1	1	1	1¼	1⅛	1¼	1¼
	Minimum Metal Thickness of Body	⅝	⅝	11/16	¾	13/16	⅞	⅞	1	1 1/16	1⅛	1 3/16	1¼

Extra Heavy Flanged Fittings. — Straight Sizes.

	Size	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	6	7	8	9	10	12	14	15	16	18	20	22	24
A–A	Face to face	8	$8\frac{1}{2}$	9	10	11	12	13	14	15	16	17	18	20	21	23	26	30	31	33	36	39	41	45
A	Center to face	4	$4\frac{1}{4}$	$4\frac{1}{2}$	5	$5\frac{1}{2}$	6	$6\frac{1}{2}$	7	$7\frac{1}{2}$	8	$8\frac{1}{2}$	9	10	$10\frac{1}{2}$	$11\frac{1}{2}$	13	15	$15\frac{1}{2}$	$16\frac{1}{2}$	18	$19\frac{1}{2}$	$20\frac{1}{2}$	$22\frac{1}{2}$
B	Center to face of long radius ells	5	$5\frac{1}{2}$	6	$6\frac{1}{2}$	7	$7\frac{3}{4}$	$8\frac{1}{2}$	9	$9\frac{1}{2}$	$10\frac{1}{4}$	$11\frac{1}{2}$	$12\frac{3}{4}$	14	$15\frac{1}{4}$	$16\frac{1}{2}$	19	$21\frac{1}{2}$	$22\frac{3}{4}$	24	$26\frac{1}{2}$	29	$31\frac{1}{2}$	34
C	Center to face of 45-deg. ells	2	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{2}$	$3\frac{1}{2}$	4	$4\frac{1}{2}$	$4\frac{1}{2}$	5	$5\frac{1}{2}$	6	6	$6\frac{1}{2}$	7	8	$8\frac{1}{2}$	9	$9\frac{1}{2}$	10	$10\frac{1}{2}$	11	12
D	Face to face, laterals	$8\frac{1}{2}$	$9\frac{1}{2}$	11	$11\frac{1}{2}$	13	14	$15\frac{1}{2}$	$16\frac{1}{2}$	18	$18\frac{1}{2}$	$21\frac{1}{2}$	$23\frac{1}{2}$	$25\frac{1}{2}$	$27\frac{1}{2}$	$29\frac{1}{2}$	$33\frac{1}{2}$	$37\frac{1}{2}$	$39\frac{1}{2}$	42	$45\frac{1}{2}$	49	53	$57\frac{1}{2}$
E	Center to face, laterals	$6\frac{1}{2}$	$7\frac{1}{4}$	$8\frac{1}{2}$	9	$10\frac{1}{2}$	11	$12\frac{1}{2}$	$13\frac{1}{2}$	$14\frac{1}{2}$	15	$17\frac{1}{2}$	19	$20\frac{1}{2}$	$22\frac{1}{2}$	24	$27\frac{1}{2}$	31	33	$34\frac{1}{2}$	$37\frac{1}{2}$	$40\frac{1}{2}$	$43\frac{1}{2}$	$47\frac{1}{2}$
F	Center to face, laterals	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	3	3	3	$3\frac{1}{2}$	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	5	$5\frac{1}{2}$	6	$6\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{2}$	8	$8\frac{1}{2}$	$9\frac{1}{2}$	10
G	Face to face, reducer	...	...	...	...	...	6	$6\frac{1}{2}$	7	$7\frac{1}{2}$	8	9	10	11	$11\frac{1}{2}$	12	14	16	17	18	19	20	22	24
	Diameter of flange	$4\frac{1}{2}$	5	6	$6\frac{1}{2}$	$7\frac{1}{2}$	$8\frac{1}{4}$	9	10	$10\frac{1}{2}$	11	$12\frac{1}{2}$	14	15	$16\frac{1}{4}$	$17\frac{1}{2}$	$20\frac{1}{2}$	23	$24\frac{1}{2}$	$25\frac{1}{2}$	28	$30\frac{1}{2}$	33	36
	Thickness of flange	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{3}{16}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{3}{4}$
	Diameter of bolt circle	$3\frac{1}{4}$	$3\frac{3}{4}$	$4\frac{1}{2}$	5	$5\frac{7}{8}$	$6\frac{5}{8}$	$7\frac{1}{4}$	$7\frac{7}{8}$	$8\frac{1}{2}$	$9\frac{1}{4}$	$10\frac{5}{8}$	$11\frac{7}{8}$	13	14	$15\frac{1}{4}$	$17\frac{3}{4}$	$20\frac{1}{4}$	$21\frac{1}{2}$	$22\frac{1}{2}$	$24\frac{3}{4}$	27	$29\frac{1}{4}$	32
	No. of bolts	4	4	4	4	4	8	8	8	8	8	12	12	12	12	16	16	20	20	20	24	24	24	24
	Diameter of bolts	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$
	Minimum metal thickness of body	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{9}{16}$	$1\frac{5}{8}$

NOTES. — Figures given are for center to face and for face to face finished dimensions.
Where necessary manufacturers will make suitable allowances in patterns before casting.

Corrugated steel gaskets covering the entire annular area inside the bolt holes are highly satisfactory for high pressures and temperatures. In a number of recent plants the tips of the flanges are welded by an oxy-acetylene torch to insure tightness. See Fig. 466.

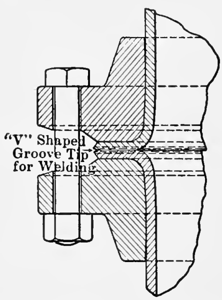


Fig. 466. Pipe Flange with Welded Tip.

The comparative costs of various flanges are given in Table 123.

Tables 121 and 122 give the dimensions of standard and extra heavy fittings as adopted by a joint committee of the manufacturers and of the American Society of Mechanical Engineers. This new schedule, "The American Standard of 1914," went into effect January 1, 1914.

The following explanatory notes refer to Tables 121 and 122:

(*a*) Standard and extra heavy reducing elbows carry same dimensions center to face as regular elbows of largest straight size.

(*b*) Standard and extra heavy tees, crosses and laterals, reducing on run only, carry same dimensions face to face as largest straight size.

(*c*) If flanged fittings for lower working pressure than 125 pounds are made, they shall conform in all dimensions, except thickness of shell, to this standard and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be standard dimensions.

(*d*) Where long radius fittings are specified, it has reference only to elbows which are made in two center-to-face dimensions and to be known as elbows and long radius elbows, the latter being used only when so specified.

(*e*) All standard weight fittings must be guaranteed for 125 pounds working pressure, and extra heavy fittings for 250-pound working pressure, and each fitting must have some mark cast on it indicating the maker and guaranteed working steam pressure.

(*f*) All extra heavy fittings and flanges to have a raised surface of $\frac{1}{16}$ inch high inside of bolt holes for gaskets.

Standard weight fittings and flanges to be plain faced.

Bolt holes to be $\frac{1}{8}$ inch larger in diameter than bolts.

Bolt holes to straddle center line.

(*g*) Size of all fittings scheduled indicates inside diameter of ports, except for heavy fittings 14 inches and larger when the port diameter is $\frac{3}{4}$ inch smaller than nominal size.

(*h*) The face-to-face dimension of reducers, either straight or eccentric, for all pressures, shall be the same face to face as given in table of dimensions.

(*i*) Square head bolts with hexagonal nuts are recommended.

For bolts $1\frac{5}{8}$ inch diameter and larger, studs with a nut on each end are satisfactory.

Hexagonal nuts for pipe sizes 1 inch to 46 inches on 125-pound standard and 1 inch to 16 inches on 250-pound standard can be conveniently pulled up with open wrenches of minimum design of heads. Hexagonal nuts for pipe sizes 48 inches to 100 inches on 125-pound and 18 inches to 48 inches on 250-pound standards can be conveniently pulled up with box or socket wrenches.

(*j*) Twin elbows, whether straight or reducing, carry same dimensions center to face and face to face as regular straight size ells and tees.

Side outlet elbows and side outlet tees, whether straight or reducing sizes, carry same dimensions center to face and face to face as regular tees having same reductions.

(*k*) Bull head tees or tees increasing on outlet will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet.

(*l*) Tees and crosses 9 inches and down, reducing on the outlet, use the same dimensions as straight sizes of the larger port.

Sizes 10 inches and up, reducing on the outlet, are made in two lengths depending on the size of the outlet as given in the table of dimensions.

Laterals $3\frac{1}{2}$ inches and down, reducing on the branch, use the same dimensions as straight sizes of the larger port.

(*m*) Sizes 4 inches and up, reducing on the branch, are made in two lengths depending on the size of the branch as given in the table of dimensions.

The dimensions of reducing flanged fittings are always regulated by the reductions of the outlet or branch. Fittings reducing on the run only, the long body pattern will always be used.

Y's are special and are made to suit conditions.

Double sweep tees are not made reducing on the run.

(*n*) *Steel flanges, fittings and valves are recommended for superheated steam.*

TABLE 123.

COMPARATIVE COST OF VARIOUS PIPE FLANGE FITTINGS, 12-INCH PIPE.

(Circular from the Crane Company.)

	Screwed.	Shrunk.	Lap Joint. Long Hub.	Lap Joint. Short Hub.	Lap Joint. No Hub.	Welded.	Rolled.	Single Riveted.
Cast iron	$ 7.40	$16.00	$18.00	......	......	......	$13.00	$21.00
Ferrosteel	8.70	18.40	20.00	......	......	......	16.00	23.40
Malleable iron	9.90	......	......	$22.00	......	......	18.00	......
Cast steel	22.40	28.40	34.00	......	$33.00	......	25.00	33.40
Weldless steel	26.40	32.40	38.00	......	37.00	$41.00	30.00	37.40

Any of the above screwed, shrunk, welded, rolled, or single-riveted flanges can be furnished with male or female face at $1.25 extra.

The screwed or welded flanges can be furnished with tongued or grooved face at $1.25 extra.

Any of the above screwed, shrunk, or single-riveted flanges can be furnished with calking recess at $1.25 extra.

In modern high-temperature, high-pressure practice all nozzles for connecting the leads are welded to the headers thereby insuring a minimum number of joints.

343. Loss of Heat from Bare and Covered Pipe. — Steam pipes, feed-water pipes, boiler steam drums, receivers, separators and the like should be covered with heat-insulating material to reduce heat losses to a minimum. By properly applying any good commercial covering

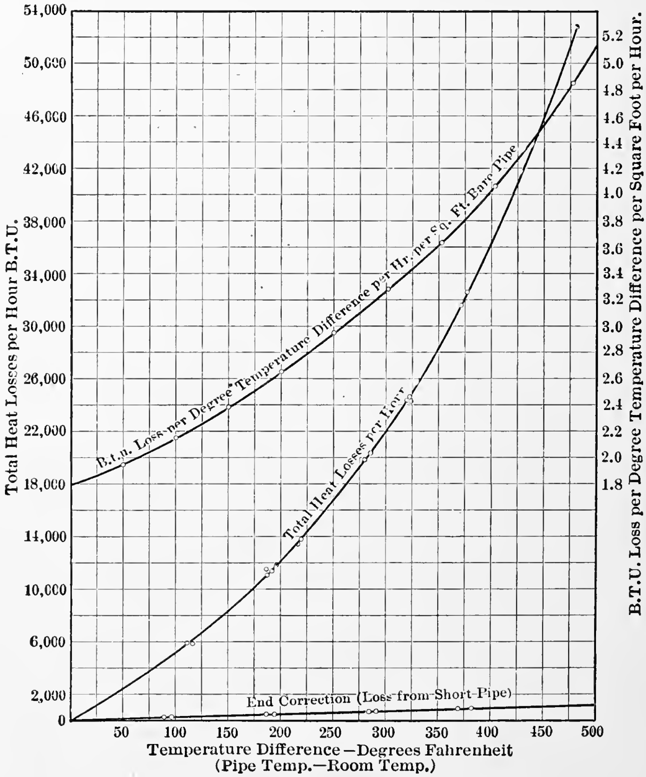


Fig. 467. Total Losses from Bare Pipe

from 75 per cent to 95 per cent of the heat loss may be prevented. Numerous investigations have been made relative to the heat losses from bare and covered pipes, but the results have been far from harmonious. The most trustworthy results appear to be those based upon the investigations of L. B. McMillan (Trans. A.S.M.E., Vol. 38, 1916). The loss of heat from bare pipes, as found by McMillan, is given in the curves of Fig. 467 and the insulating properties of a number of well-known pipe coverings are shown in Fig. 468. From the curves in Fig. 467 it will be seen that heat loss from bare pipes is so great that covering will pay for itself in a comparatively short time. The curves, Fig. 469, showing the relation of the rate of loss per sq. ft. of covering surface to the temperature difference between the covering

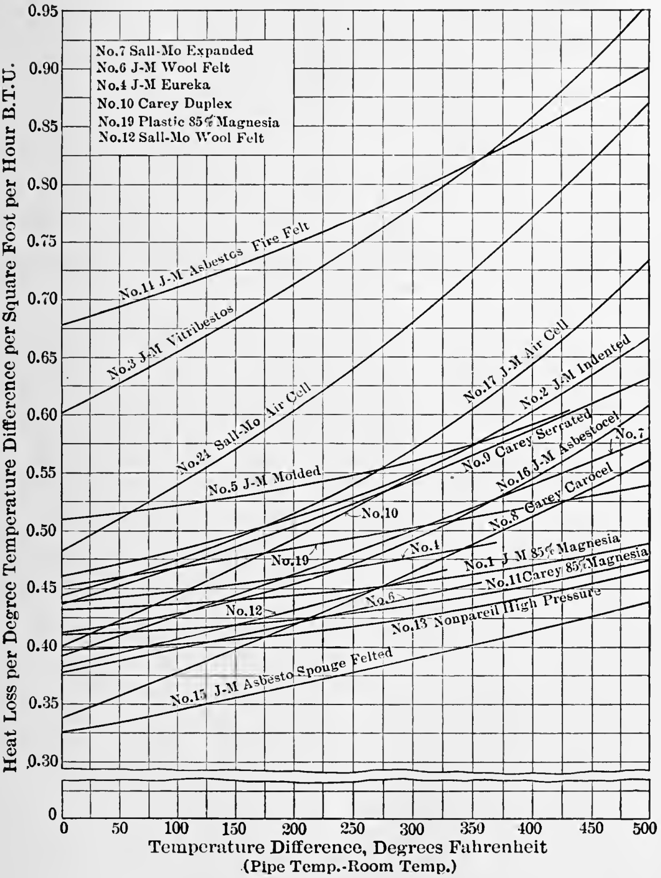


FIG. 468. Heat Loss Through Pipe Covering (Single Thickness).*

surface and the surrounding air is one of the most important results obtained by McMillan and furnishes the data required for calculating the heat loss from covered pipe having its surface finished in white canvas; thus, for finding the heat loss through any thickness of any

* The average thicknesses in inches of the coverings, Fig. 468, are as follows: 1 — 1.08; 2 — 1.12; 3 — 0.96; 4 — 1.04; 5 — 1.25; 6 — 1.10; 7 — 1.07; 8 — 0.99; 9 — 1.00; 10 — 0.96; 11 — 1.10; 12 — 1.16; 13 — 1.16; 14 — 0.99; 15 — 1.16; 16 — 1.10; 17 — 1.00; 19 — 1.05; 24 — 0.95.

material of which the conductivity is known, at any temperature difference between the pipe and room up to 500 deg. fahr.:

$$H_2 = \frac{k\,(t_1 - t - d)}{r_2\,(\log_e r_2 - \log_e r_1)}, \tag{268}$$

$$H_2 = \frac{r_1}{r_2} H_1, \tag{269}$$

in which

H_2 = heat loss per sq. ft. of outside covering surface, B.t.u. per hr.,
k = conductivity of the material, B.t.u. per hr. per sq. ft. per in. thickness per degree temperature difference,
t_1 and t = temperatures, respectively, of the pipe and of the air in the room, deg. fahr.,
r_2 and r_1 = radii, respectively, of the outer and inner surfaces of the covering, in.,
d = temperature difference between the covering and air corresponding to a rate of loss H_2,
H_1 = heat loss per sq. ft. of pipe surface, B.t.u. per hr.

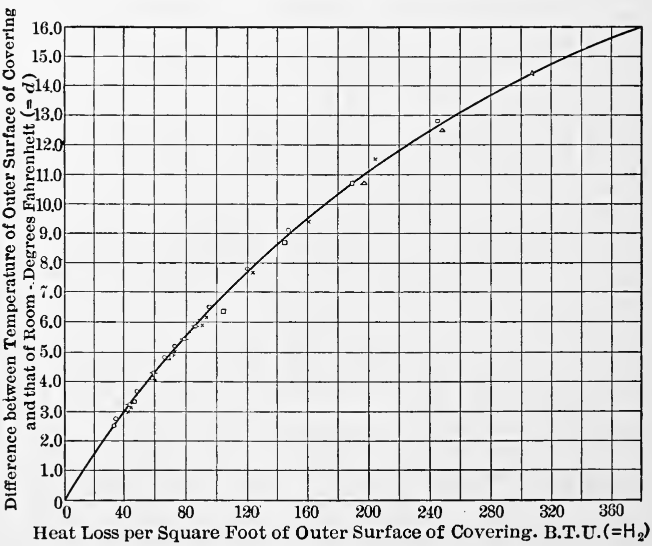


FIG. 469. Relation of Heat Losses to Temperature.

The conductivity may be calculated as follows:

$$k = \frac{H_1 r_1\,(\log_e r_2 - \log_e r_1)}{t_1 - t_2}, \tag{270}$$

in which

t_2 = temperature of the outer surface of the covering, deg. fahr.

Other notations as in equations (268) and (269).

These laws are best illustrated by examples 66 and 67.

Example 66. A steam pipe 5.6 in. outside diameter is covered with single-thickness J-M 85 per cent magnesia, 1.13 in. thick, temperature of the pipe 380 deg. fahr., room temperature 80 deg. fahr. Required the conductivity per inch thickness for the given conditions.

From Fig. 468 the rate of heat loss per hour per sq. ft. per deg. temperature difference is 0.455 B.t.u. Therefore, $H_1 = 300 \times 0.455 = 136.5$ and $H_2 = 136.5 \times \frac{5.6}{2} \div \left(\frac{5.6}{2} + 1.13\right) = 97.2$ B.t.u. From Fig. 469 the temperature difference between outer covering surface and air corresponding to a loss of 97.2 B.t.u. is 65 deg. fahr. Therefore, the temperature difference between inner and outer covering surfaces is $300 - 65 = 235$ deg. fahr. Substituting these values in equation (270) and solving for k,

$$k = \frac{136.5 \times 2.8\ (\log_e 3.93 - \log_e 2.8)}{235} = 0.551.$$

Example 67. If the pipe in Example 66 is covered with 3-inch thickness of material, other conditions remaining the same, calculate the heat loss per sq. ft. of pipe surface per hr. per degree temperature difference.

From equation (268)

$$H_2 = \frac{0.551\ (380 - 80 - d)}{(2.8 + 3)\ (\log_e 5.8 - \log_e 2.8)}$$
$$= 0.13\ (300 - d).$$

Now assume $d = 20$ deg. Then H_2 from Fig. 469 = 25.5 B.t.u. But H_2 from equation (268) = $0.13\ (300 - 20) = 36.4$. This shows that d must be greater than 20. Assume $d = 30$. Then H_2 from Fig. 469 = 39.5 B.t.u. and from equation (269) $H_2 = 0.13\ (300 - 30) = 35.1$. This shows that d must be less than 30. By cut and trial the correct value $d_2 = 27$ may be obtained. Then $H_2 = 0.13 \times (300 - 27) = 35.5$. Substitute this value of H_2 in equation (269) and solve for H_1.

$$35.5 = \frac{2.8}{5.8} \times H_1,$$

from which $H_1 = 73.5$ B.t.u. per hr. per sq. ft. Loss per sq. ft. per hr. per deg. temperature difference between the pipe surface and air in the room = $73.5 \div 300 = 0.245$ B.t.u.

Pipe covering is applied in sections molded to the required forr and held to the pipe by bands, or may be applied in a plastic form. The former is more readily applied and removed, and is usually adopted for pipes, while the valves and fittings are generally covered with plastic material. Piping should be tested under pressure before being covered, since leaks destroy the efficiency and life of the covering. If the surrounding atmosphere is moist the covering should be given two or three coats of good paint. Coverings are sometimes applied to cold water pipe to prevent sweating.

Identification of Power House Piping by Colors: Power and Engineer, April 26, 1910, p. 752.

TABLE 124.

COEFFICIENTS OF LINEAR EXPANSION PIPING MATERIALS.

Material.	Temperature Range.	Mean Coefficient per Degree F.
Wrought iron and mild steel	32–212	0.00000656
Wrought iron	32–572	0.00000895
Cast iron	32–212	0.00000618
Cast steel	32–212	0.00000600
Hardened steel	32–212	0.00000689
Nickel-steel, 36 per cent Nickel	32–572	0.00000030
Copper, cast	32–212	0.00000955
Copper, wrought	32–572	0.00001092
Lead	32–212	0.00001580
Cast brass	32–212	0.00001043
Brass wire and sheets	32–212	0.00001075
Tin cast	32–212	0.00001207
Tin hammered	32–212	0.00001500
Zinc cast	32–212	0.00001633
Zinc hammered	32–212	0.00001722

LINEAR EXPANSION OR CONTRACTION OF CAST IRON IN INCHES PER 100 FEET, — DEGREES F.

Temperature Difference.	Expansion.	Temperature Difference.	Expansion.
100	0.72	300	2.376
150	1.1016	400	3.360
200	1.5024	500	4.440
250	1.9260	600	5.616
....	....	800	7.872

Multiply by 1.1 for wrought mild steel.
Multiply by 1.5 for wrought copper.
Multiply by 1.6 for wrought brass.

344. Expansion. — One of the most difficult problems in the design of a piping system is the proper provision for expansion and contraction due to change in temperature. If a pipe is immovably fixed at both ends and under no strain when cold, and the temperature is increased, as by the admission of steam, it is subjected to a compression proportional to the rise in temperature (within the elastic limit). The axial force exerted due to expansion may be expressed

$P = EA\ (t_1 - t)\ \mu$ (Mechanics of Engng., Church, p. 218), (271)

P = force in pounds,

E = modulus of elasticity (average for steel pipe = 30,000,000),

t_1 = final temperature, deg. fahr. (the temperature of the pipe is practically that of the steam),

t = initial temperature,
μ = coefficient of expansion,
A = sectional area of the pipe material, sq. in.

Example 68. A 6-inch standard extra heavy steel iron pipe 200 feet long at 66 deg. fahr., heated to 366 deg. fahr. (the temperature corresponding to steam at 165 pounds per square inch absolute pressure), required the axial force exerted.

Here

$$E = 30{,}000{,}000; \quad t_1 = 366; \quad t = 66; \quad = 0.000007 \text{ (approx.)},$$
$$A = 8.5 \text{ sq. in.}$$

Substituting these values in equation (271),

$$P = 30{,}000{,}000 \times 8.5\,(366 - 66)\,0.000007$$
$$= 535{,}500 \text{ lb.}$$

Unless well braced throughout its entire length the pipe will buckle and become distorted. If free to expand its length would increase. The total increase in length is the sum of the elongation due to pressure and that due to increase in temperature. The increase in length due to pressure is negligible except for extremely high pressures and long lengths of thin pipe, but that due to temperature may be considerable.

TABLE 125.

SAFE EXPANSION VALUES OF 90-DEGREE WROUGHT STEEL BENDS IN INCHES.

(Full weight or extra heavy pipe.)

Sizes.	Mean Radius of Bend (in Inches).												
	12	15	20	30	40	50	60	70	80	90	100	110	120
1	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{3}{4}$	$3\frac{1}{8}$	...	...	...	...	...	...	...	...
2	$\frac{1}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	1	$1\frac{3}{4}$	$2\frac{3}{4}$	$3\frac{7}{8}$	$5\frac{3}{8}$	...	...	...	...	...
$2\frac{1}{2}$	...	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{7}{8}$	$1\frac{1}{2}$	$2\frac{1}{4}$	$3\frac{1}{4}$	$4\frac{1}{2}$	$5\frac{3}{4}$	...	...	...	...
3	...	$\frac{1}{8}$	$\frac{3}{8}$	$\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{7}{8}$	$3\frac{5}{8}$	$3\frac{5}{8}$	$4\frac{3}{4}$	6	...	...	...
3	...	...	$\frac{1}{4}$	$\frac{5}{8}$	1	$1\frac{5}{8}$	$2\frac{3}{8}$	$3\frac{1}{8}$	$4\frac{1}{8}$	$5\frac{1}{4}$	...	...	...
4	...	...	$\frac{1}{4}$	$\frac{1}{2}$	1	$1\frac{1}{2}$	2	$2\frac{7}{8}$	$3\frac{3}{4}$	$4\frac{3}{4}$	$5\frac{3}{4}$	...	...
$4\frac{1}{2}$	...	...	...	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{1}{2}$	$3\frac{3}{8}$	$4\frac{1}{4}$	$5\frac{1}{4}$	...	...
5	...	...	...	$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$2\frac{1}{4}$	3	$3\frac{3}{4}$	$4\frac{5}{8}$	$5\frac{5}{8}$	...
6	...	...	...	$\frac{3}{8}$	$\frac{5}{8}$	1	$1\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{1}{2}$	$3\frac{1}{8}$	$3\frac{7}{8}$	$4\frac{3}{4}$	$5\frac{5}{8}$
8	...	...	...	...	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{1}{2}$	3	$3\frac{5}{8}$	$4\frac{3}{8}$
10	...	...	...	...	...	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{1}{2}$	2	$2\frac{3}{8}$	$2\frac{7}{8}$	$3\frac{1}{2}$
12	...	...	...	...	...	...	$\frac{3}{4}$	1	$1\frac{3}{8}$	$1\frac{5}{8}$	2	$2\frac{1}{2}$	$2\frac{7}{8}$
14	...	...	...	...	...	...	...	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{1}{2}$
15	...	...	...	...	...	...	...	...	1	$1\frac{3}{8}$	$1\frac{5}{8}$	2	$2\frac{3}{8}$
16	...	...	...	...	...	...	...	...	$\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{1}{4}$
18	...	...	...	...	...	...	...	...	...	...	...	$1\frac{5}{8}$	$1\frac{7}{8}$
20	...	...	...	...	...	...	...	...	...	...	...	...	$1\frac{3}{4}$

For any compound expansion bend multiply the tabular value by the number of 90-degree bends, thus for a "U" bend multiply the tabular values by 2; for an "expansion U" bend multiply by 4.

The increase in length for both conditions may be expressed

$$l_p = \frac{paL}{EA}, \tag{272}$$

$$l_t = \mu (t_1 - t) L, \tag{273}$$

in which

l_p = increase in length due to the internal pressure, in.,
l_t = increase in length due to the temperature difference,
p = boiler pressure, lb. per sq. in. gauge,
a = inside area of the pipe, sq. in.,
L = length of the pipe, in.

Other notations as in equation (271).

Example 69. A 12-inch extra-heavy steel pipe is 100 feet long when cold (70 deg. fahr.): Required the increase in length when carrying superheated steam at 250-lb. gauge pressure, temperature 670 deg. fahr.

Here $p = 250$, $a = 108.4$, $L = 1200$, $E = 30{,}000{,}000$, $A = 19.25$, $t_1 = 670$, $t = 70$, $\mu = 0.0000075$ (the coefficient of linear expansion is known to increase with the temperature; the value assumed here is a purely arbitrary one).

Substituting these values in equations (272) and (273),

$$l_p = \frac{250 \times 108.4 \times 1200}{30{,}000{,}000 \times 19.25} = 0.056 \text{ in., which is negligible,}$$

$$l_t = 0.0000075\ (670\text{–}70)\ 1200 = 5.4.$$

Since the forces produced by expansion are practically irresistible the pipe is invariably allowed to expand and its movement is prevented from unduly stressing the fittings and connections by

1. Long radius bends.
2. Double-swing screwed fittings.
3. Expansion joints.

TABLE 126.

MINIMUM DIMENSIONS FOR PIPE BENDS.

Size of Pipe, In.	Radius of Bend, In.		Lengths of Straight Pipe on Each Bend, In.	Size of Pipe, In.	Radius of Bend, In.		Lengths of Straight Pipe on Each Bend, In.
	Full Weight Pipe.	Extra Heavy Pipe.			Full Weight Pipe.	Extra Heavy Pipe.	
2½	12.5	7	4	8	40	28	9
3	15.0	8	4	9	45	35	11
3½	17.5	10	5	10	50	40	12
4	20.0	12	5	12	60	50	14
4½	22.5	14	6	14	70	65	16
5	25.0	15	6	15	75	70	16
6	30.0	20	7	16	80	78	18
7	35.0	24	8	18	108	88	18

Where practical long radius bends will prove most satisfactory.* Fig. 470 shows a number of standard bends and Table 126 gives the minimum radii and lengths of straight pipe at the end of each bend as recommended by the Crane Company. The amount of expansion

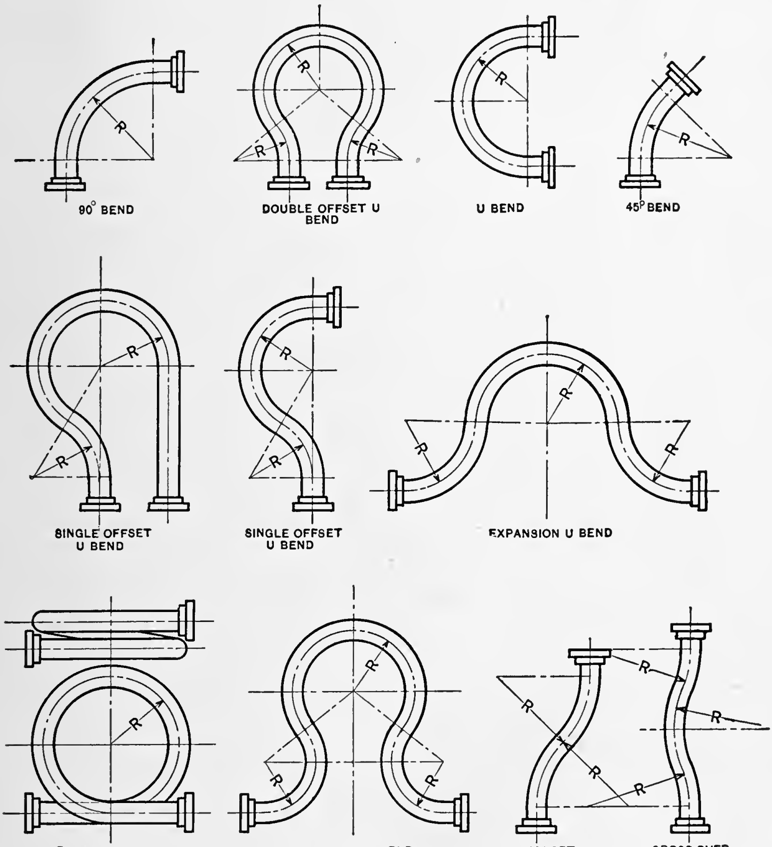


Fig. 470. Types of Expansion Bends.

absorbed by a standard 90-degree quarter bend and other shapes may be taken from Table 125.

Figs. 484 and 485 show applications of pipe bends to boiler and header connections.

* At the Essex Power Station of the Public Service Electric Co. of New Jersey there are no expansion joints in the headers. The headers are installed under a tension between anchorages, which causes an elongation equal to about one-half the expansion of the section from normal temperature to that of the steam. When the headers are at ordinary room temperature they are in tension, and when at the temperature of the steam they are in compression.

Fig. 471 shows a double-swing screwed joint in which expansion causes the fittings to turn slightly and thus relieve the strain. This method is usually adopted where long radius bends are not practicable on account of lack of space and where screwed fittings are used.

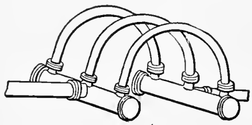


Fig. 470a. U Bends for Large Headers when Overhead Space is Limited.

Slip joints, Fig. 472, are now little used except with very large pipes and where space prohibits long radius bends. When slip joints are employed the pipe must be securely anchored to prevent the steam pressure from forcing the joint apart and at the same time permit the pipe in expanding to work freely in the stuffing box. Sagging of the pipe on either side, which might cause binding in the joint, is prevented by suitable supports.

Elasticity and Endurance of Steam Piping: Power, Feb. 23, 1915, p. 278.

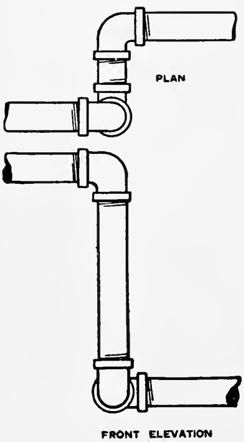


Fig. 471. "Double-swing" Expansion Joint.

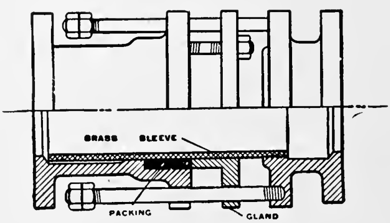


Fig. 472. Slip Expansion Joint.

345. Pipe Supports and Anchors. — Pipe lines must be supported to guard against excessive deflection and vibration. Supports are conveniently classified as (1) hangers, (2) wall brackets, and (3) floor stands.

Fig. 473 illustrates a type of hanger for suspending pipes from I beams. The supports being free to swing, no provision for expansion is necessary. A properly designed hanger may be readily removed without disturbing the pipe line, and should be adjustable to facilitate "lining up." If of rigid construction the lower end should be provided with a roller.

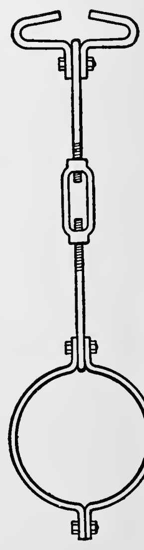


Fig. 473. A Typical Pipe Hanger.

Fig. 474 gives the details of a wall bracket with rolls and roll binder. Supports adjacent to long radius bends should be provided with roll binders as illustrated to prevent the pipe from springing laterally,

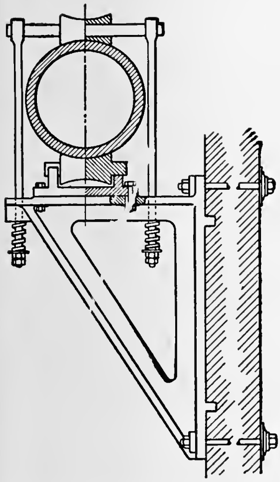


FIG. 474. A Typical Wall Bracket with Binding Roll.

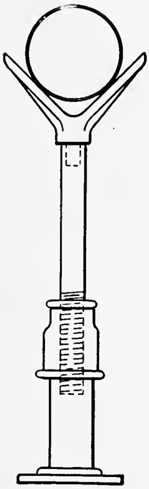


FIG. 475. A Typical Floor Stand.

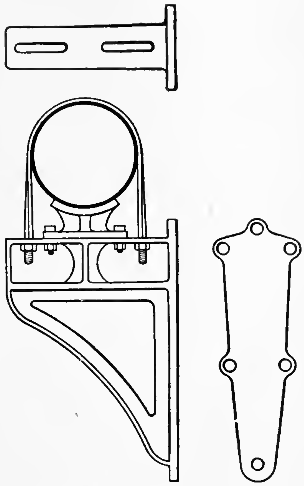


FIG. 476. A Typical Pipe Anchor.

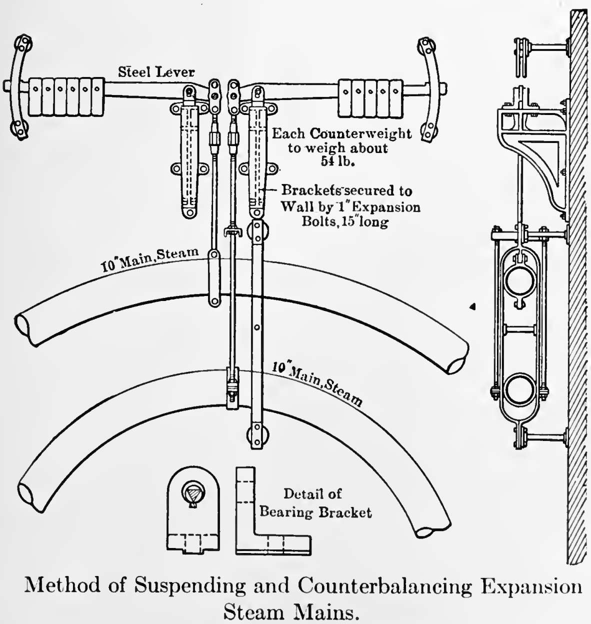


FIG. 477. Method of Suspending and Counterbalancing Expansion Loops in Steam Mains.

but they may otherwise be omitted. The rollers are often made adjustable to facilitate lining up.

Fig. 475 illustrates a typical floor stand. Pipe lines are usually securely anchored at suitable points in a manner similar to that illustrated in Fig. 476, the pipe resting on a saddle and being rigidly clamped to the bracket by a flat iron band with ends threaded and bolted. This limits expansion to one direction and prevents excessive strain on the fittings.

Fig. 477 illustrates a method of suspending and counterbalancing expansion loops in a main header and Fig. 478 a flexible support for a large vertical exhaust header.

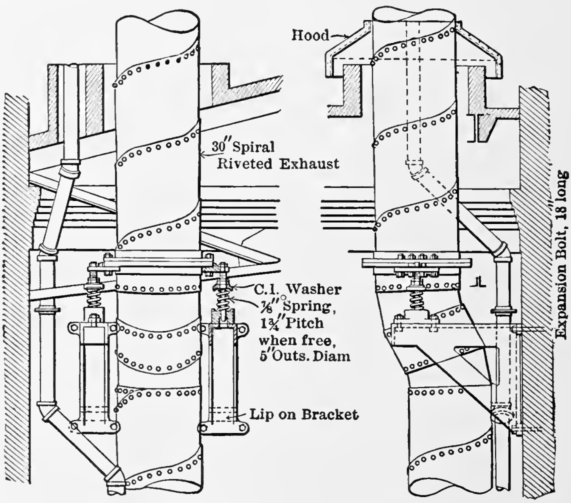


Fig. 478. Spring Support for 30-inch Exhaust Pipe.

346. General Arrangement of High-pressure Steam Piping. — The general arrangement of piping depends in a great measure upon the space available for engines and boilers.

The engine and boiler room may be placed

(1) Back to back, 480.
(2) End to end, 479.
(3) Double decked, 486.

The *back-to-back* arrangement is the most common and, other things permitting, is to be preferred on account of the short and direct connection between prime movers and boilers and the ease of enlargement. The engine and boiler rooms are separated by a wall, and as much of the piping as possible is located in the boiler room.

The *end-to-end* arrangement is ordinarily limited to situations where the distribution of space precludes the back-to-back system.

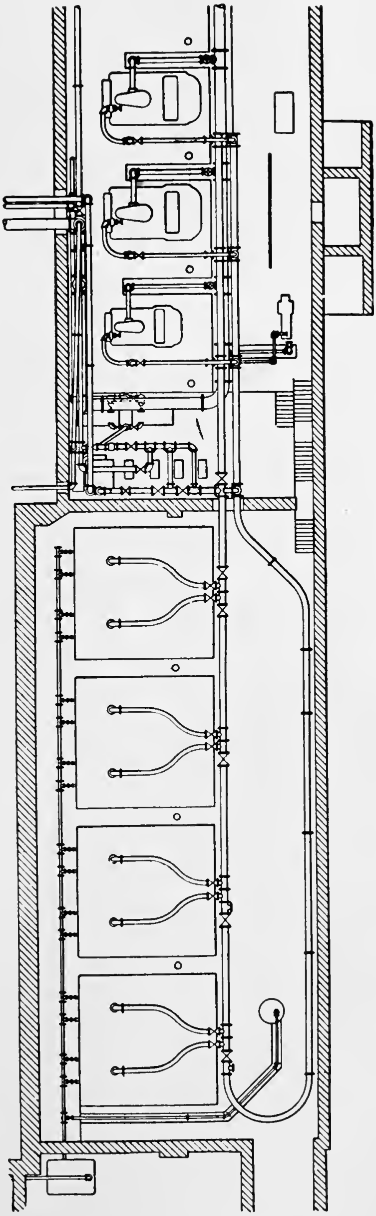


Fig. 479. Plan of High-pressure Piping, Princeton University Power Plant.

The *double-decked* arrangement is frequently used where ground space is limited or expensive.

Prime movers and boilers are connected in a variety of ways through steam headers as shown in Figs. 479 to 489:

1. Spider system, Fig. 480.
2. Single header, Fig. 481.
3. Duplicate system.
4. Loop or ring header, Fig. 483.
5. The "unit" system, Fig. 484.

The *spider* system is often used in small plants. In this arrangement all branch pipes are brought to one central header which is made as short as possible. The shortness of such a header minimizes danger from breakdowns, and brings all the principal valves close together.

The *single-header* system is perhaps the most common, since it embodies simplicity, low first cost, and provision for extension.

The *duplicate* system is losing favor, since experience shows that the extra cost of the duplicate mains will usually give better returns in continuity of operation and maintenance if invested in high-grade fittings on a single-pipe system. A small auxiliary header is occasionally used in plants where double mains are desired. In the new River Station of the Buffalo General Electric Company the steam main is in duplicate, see Figs. 487 and 488, but this arrangement is for the purpose of insuring flexibility and for keeping down the size of pipe and not as a protection against breakdown. Both headers are in use simultaneously.

The *loop header* is well adapted where a large number of steam engines, elevator pumps, air compressors, and miscellaneous steam-consuming appliances are crowded together in a comparatively small space.

Large modern power plants are, by the latest practice, divided into complete and independent units, as in Fig. 484, each prime mover

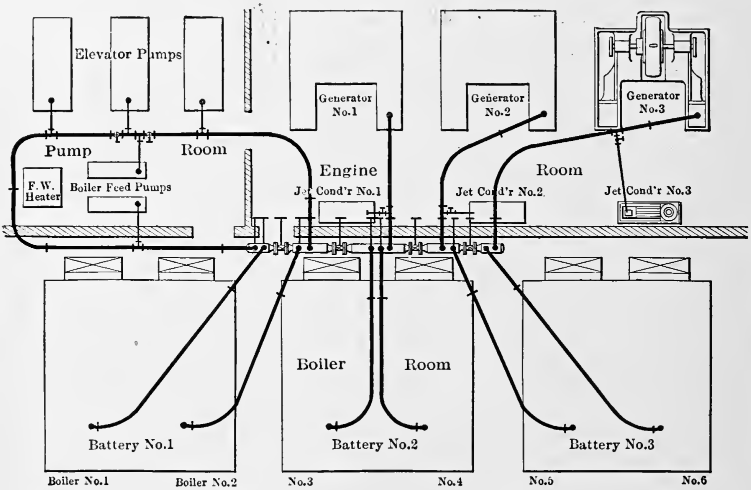


FIG. 480. "Spider" System.

having its own boiler equipment, coal and ash-handling machinery, feed pumps, and piping, operated independently of the rest of the plant.

The steam mains are usually cross connected so that steam from any boiler unit may be led to the adjacent prime mover.

Figs. 484 and 485 show the general arrangement of the steam piping at the Yonkers Power House of the New York Central, illustrating a typical "unit system." The turbines are connected in pairs by 14-inch loops, each turbine taking steam from either of two banks of four boilers. The high-pressure steam piping is of mild steel with modified reinforced "Van Stone" joints. The high-pressure valves are of the split-disk pattern with semi-steel bodies. Expansion is taken up by the long sweep bends.

Plants using superheated steam are sometimes piped to supply saturated steam to the auxiliaries as illustrated in Fig. 489. The boiler branch *E*, leading to the main header, normally supplies super-

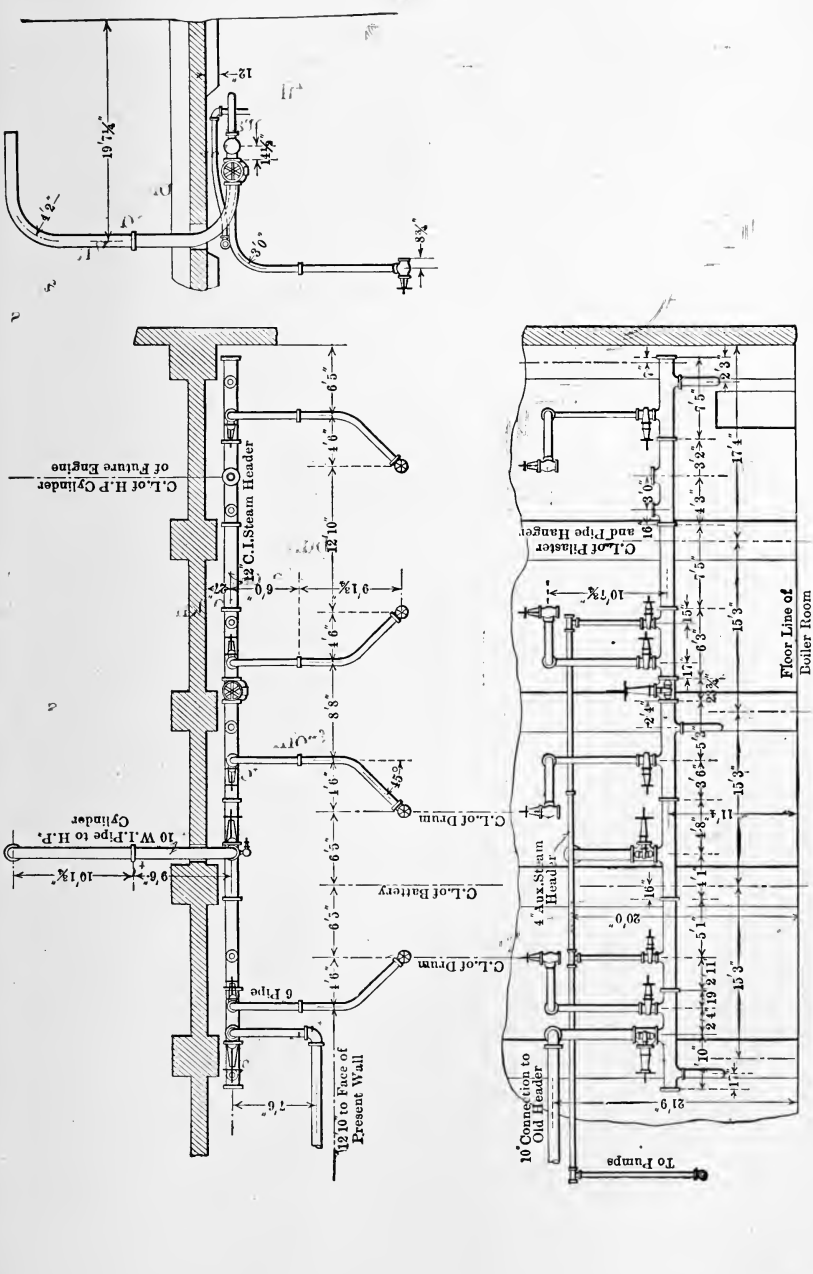


FIG. 481. Steam Header and Branches, Des Moines City Ry. Co.'s Power House (Typical Single Header).

heated steam to the engines. C is an auxiliary main supplying the air pumps, stoker engines, and other auxiliaries with saturated steam from branch pipe D.

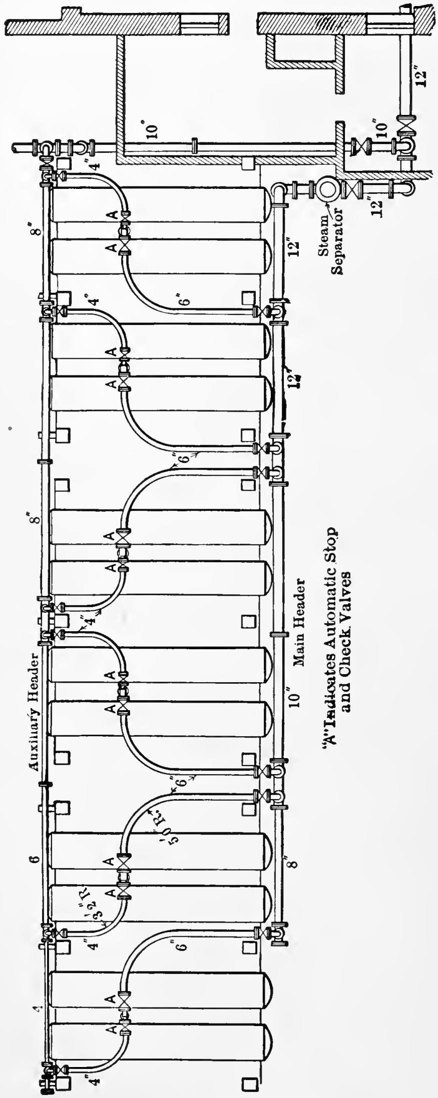


Fig. 482. Typical Auxiliary Header System.

347. Size of Steam Mains. — Until quite recently it was the usual practice to employ a common header running the entire length of the plant and to connect all boiler and engine leads with this header. With the low steam velocities used at that time headers as large as 24 inches in diameter were not uncommon. This type of station is rarely built at the present day except, perhaps, for very small plants. In the various large power houses recently built in this country with ultimate capacities of from 100,000 to 250,000 kilowatts, the largest steam headers are not over 18 inches in diameter. In some recent designs the pipes leading from the header to the engines are two sizes smaller than called for by the engine builders. In this case large receiver separators two to four times the volume of the high-pressure cylinder are provided near the throttle. The pipes between receiver and engine are full size. The object of the arrangement is to give (1) a constant flow of steam, (2) a full supply of steam close to the throttle, and (3) a cushion near the engine for absorbing the shock caused by cut-off.

With saturated steam and boiler pressures from 125 to 150 pounds a maximum velocity of 8000 feet per minute is allowed in the main and as high as 9000 feet per minute between header and receiver.

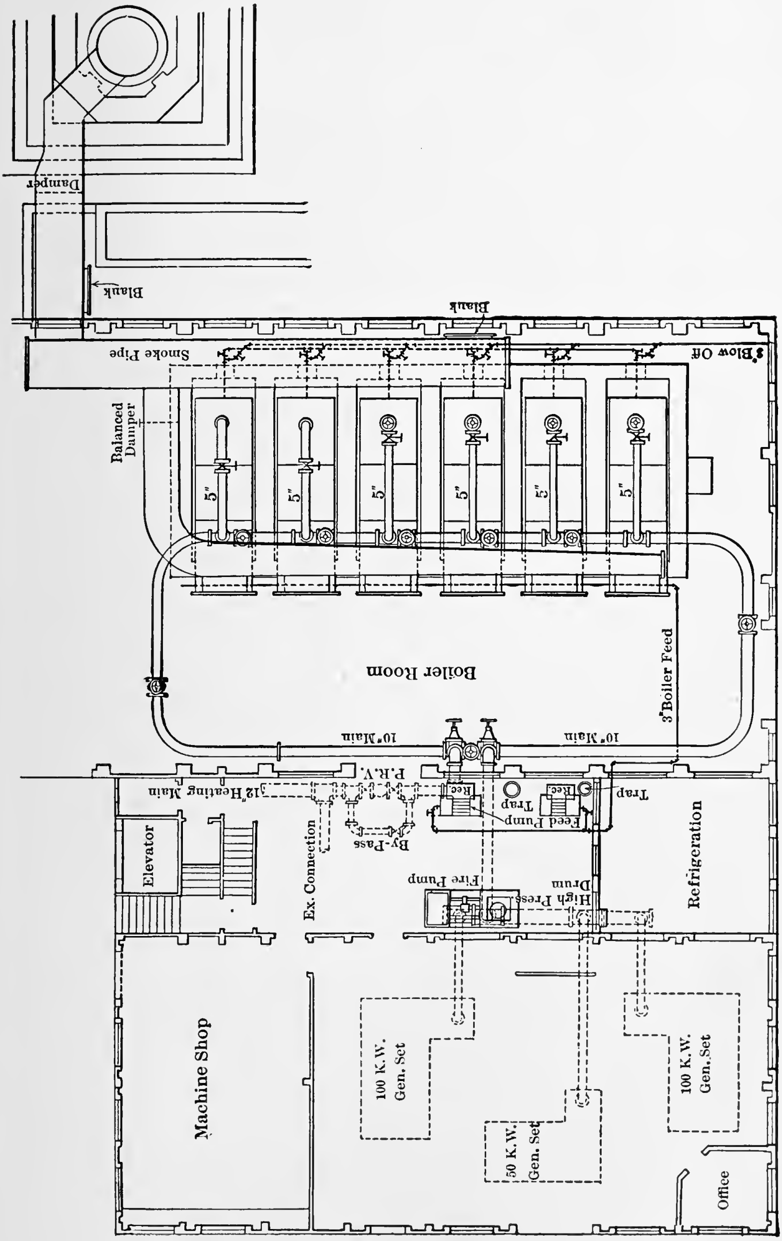


FIG. 483. Typical Loop Header.

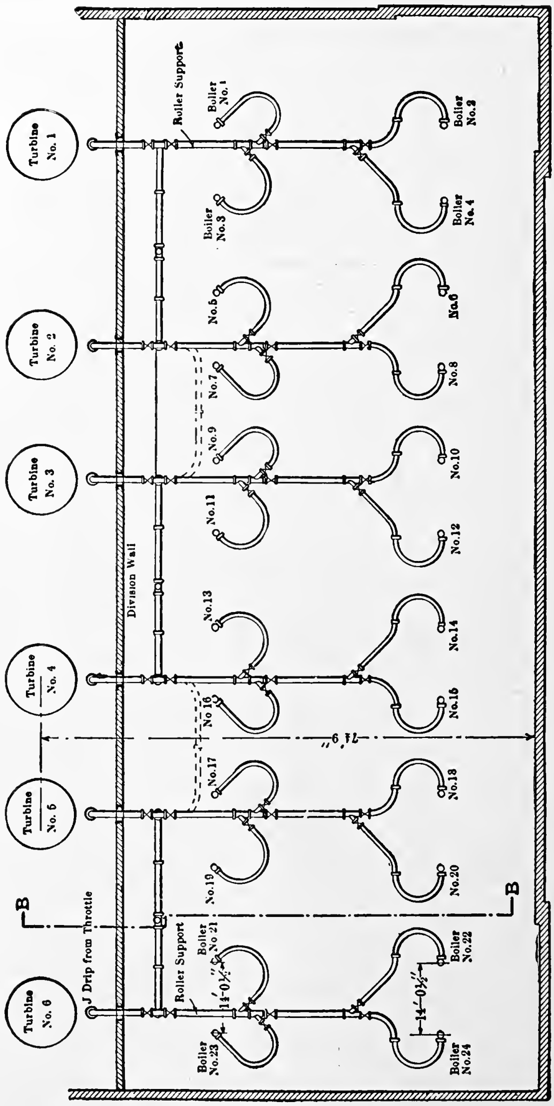


FIG. 484. Plan of High-pressure Piping, Yonkers Power House of the New York Central R.R. (Typical Unit System).

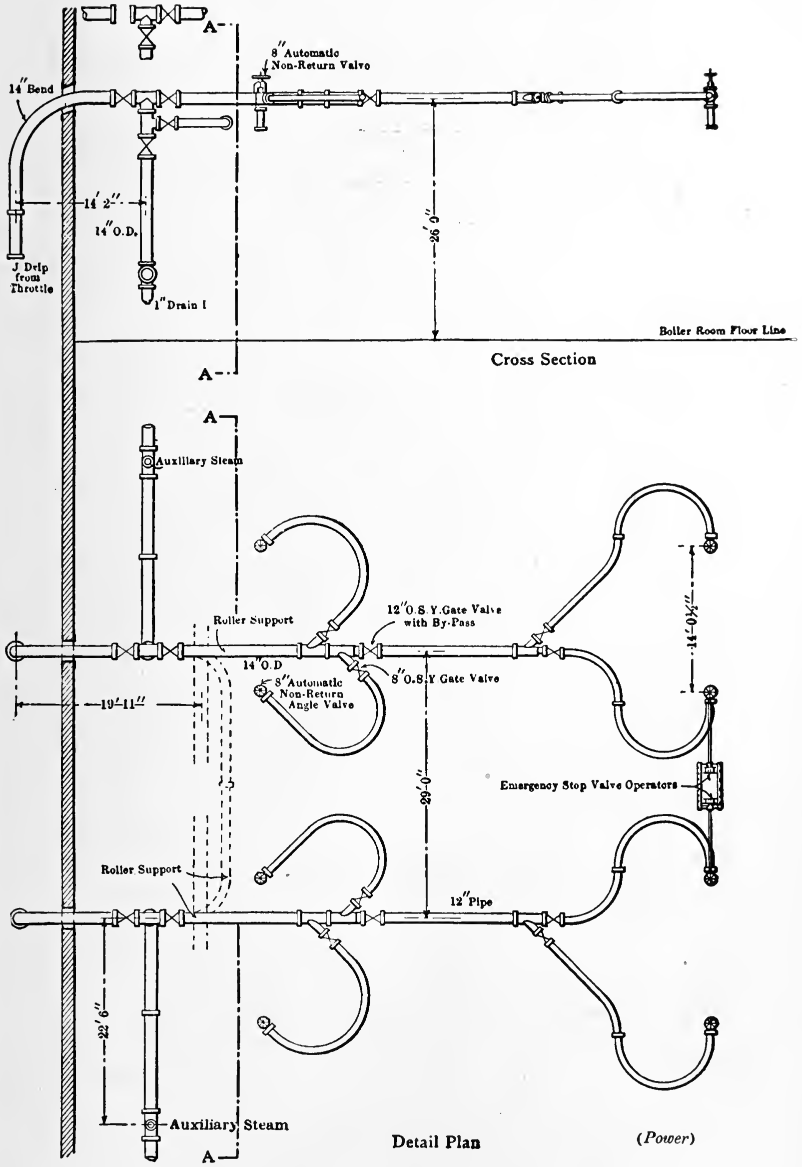


Fig. 485. Details of Boiler Steam Piping, Yonkers Power House of the New York Central R.R.

With steam turbines using highly superheated steam velocities as high as 16,000 feet per minute have been allowed during peak loads but the pressure drop between boiler and prime mover is apt to be ex-

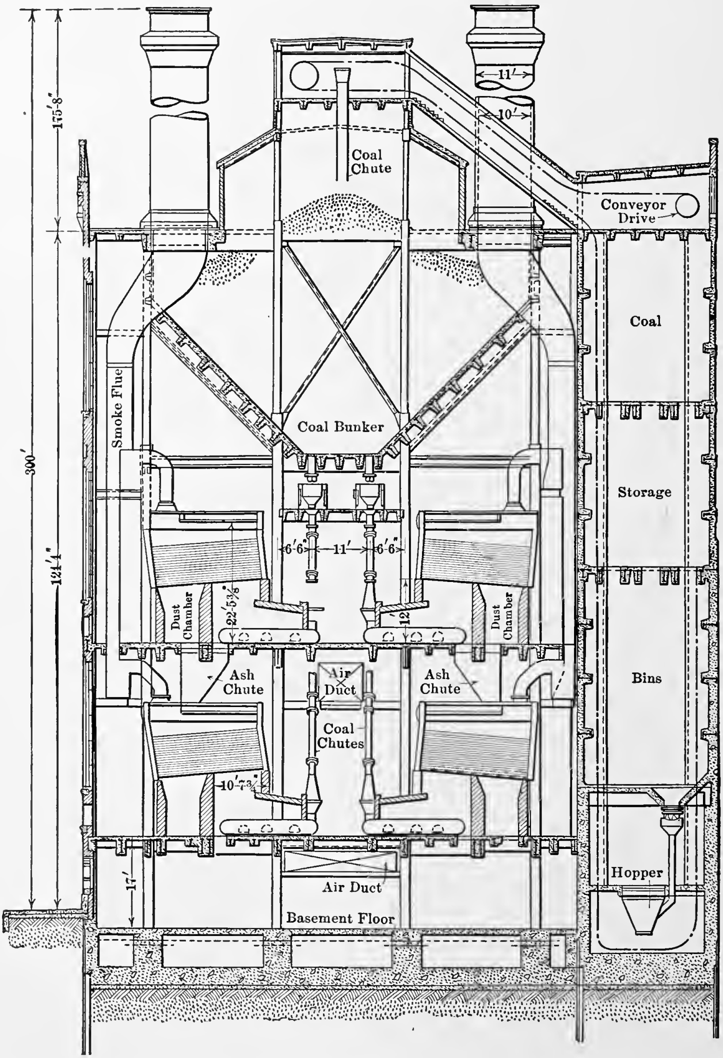


FIG. 486. Typical Double-deck Boiler Installation (New York Steam Co.).

cessive. Exhaust steam velocities range from 12,000 to 36,000 ft. per minute, depending upon the pressure of the steam and the length of the piping.

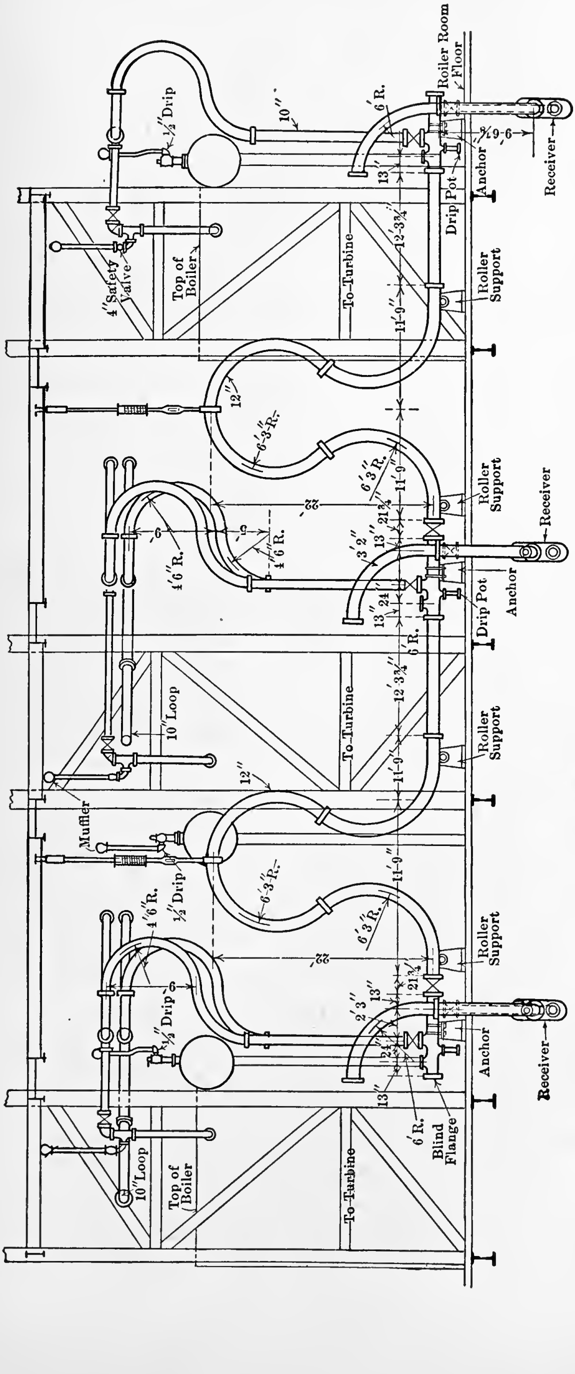


FIG. 487. Elevation of Steam Piping, River Station, Buffalo General Electric Company.

348. Flow of Steam in Pipes. — In designing a piping system the engineer is chiefly concerned with the size of pipe which will deliver a given weight of steam under given initial conditions to a distant point at a predetermined pressure drop. In small plants extreme accuracy in determining the size of pipe is not necessary; it is better to err in the installation of too large a pipe than one too small. In large stations where the pipes are large and the pressure is high the cost of piping increases rapidly with the size and greater accuracy is essential. Since the weight of steam discharged through any system

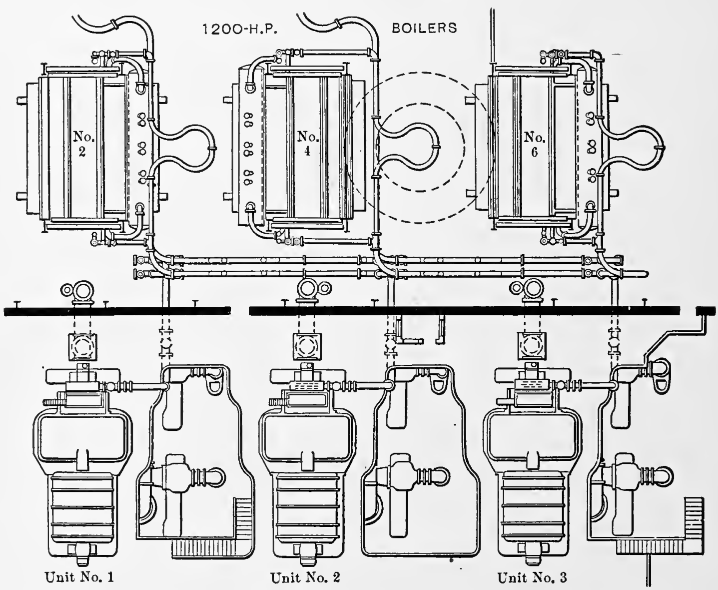


FIG. 488. Plan of Main Steam Piping, River Station, Buffalo General Electric Company.

of piping is a direct function of the drop in pressure it is evident that the greater the drop the larger will be the weight discharged. A large drop in pressure permits of a smaller pipe and lower radiation losses, but a point is soon reached where the economy in the size of pipe is more than offset by the loss in available energy due to the reduced pressure at the point of application. There seems to be no fixed rule for determining the drop most suitable for any given set of conditions. In reciprocating engine practice involving the use of saturated steam and in which the pipe leads directly to the inlet nozzle the maximum drop in pressure ordinarily varies from $\frac{1}{2}$ to $1\frac{1}{2}$ pounds per hundred feet of pipe, corresponding to a maximum velocity of approximately

6000 feet per minute. In a number of installations in which a large receiver is placed next to the inlet nozzle pressure drops of 1.5 to 2.5 pounds per 100 feet of pipe with corresponding maximum velocity of about 9000 feet per minute have given satisfactory results. For very long pipe lines the pressure drop per 100 feet must necessarily be small in order to avoid low pressures at the point of delivery. In steam turbine practice involving the use of high pressure and superheat pressure drops as high as 3.5 pounds per 100 feet of pipe have been allowed during periods of maximum discharge. Under the latter conditions pipe velocities as high as 16,000 feet per minute have been

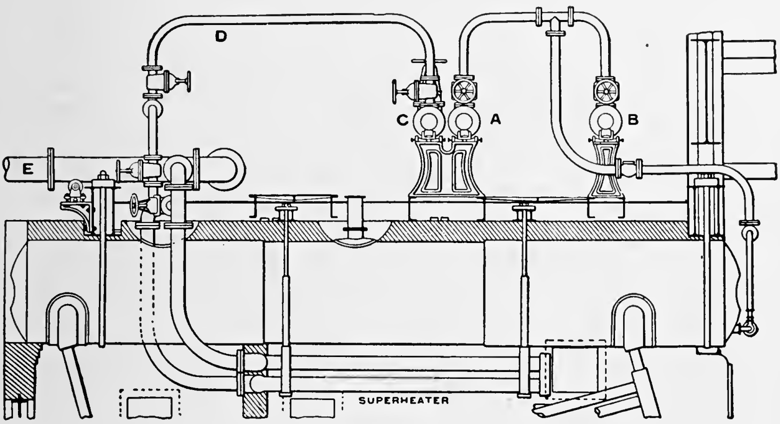


Fig. 489. Overhead Boiler Piping, Quincy Point Power Plant of the Old Colony St. Ry. Co., Quincy Point, Mass.

obtained. It must be remembered that the pressure drop through the piping is but a small portion of the total drop from boiler to prime mover because of the additional resistance of the dry pipe, superheater, valves, and fittings; consequently large pressure drops through the piping alone may cause excessive drops from boiler to prime mover. (See following paragraph for resistance of fittings, etc.) The average pressure drop in exhaust steam mains varies from 0.2 to 0.4 pound per 100 feet for non-condensing service and from 0.2 to 0.4 inch of mercury per 100 feet for a vacuum of 26 inches. In large steam turbine installations there is practically no exhaust piping and steam velocities of 300–400 feet per second are possible with a negligible pressure drop.

Notwithstanding the numerous investigations conducted on laboratory apparatus and on pipe lines under actual power plant conditions there is no trustworthy rule for accurately determining the behavior of the flow of steam in commercial piping.

Table 127 gives some of the rules commonly used in piping design and those classified under "Group I" have been given particular attention by various writers. For pressure drops under $\frac{1}{2}$ lb. per 100 feet

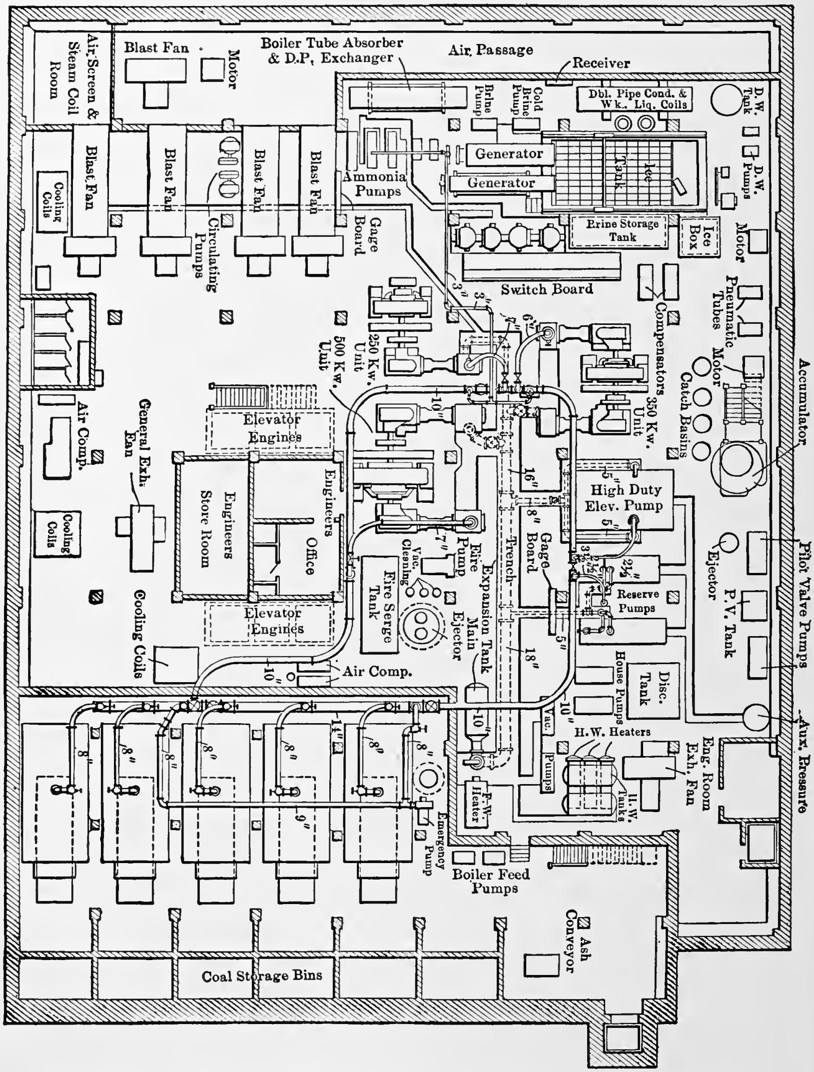


FIG. 490. General Arrangement of Steam and Exhaust Piping, La Salle Hotel, Chicago, Ill.

of pipe any of the equations in this group will give results which agree fairly well with practice but for greater pressure drops they may lead to serious error unless modified to suit the new conditions.

TABLE 127.

COMPARISON OF PIPE EQUATIONS FOR THE FLOW OF STEAM.

	Author.	V = Velocity, Feet per Minute.	W = Weight, Pounds per Minute.	P = Drop in Pressure, Pounds per Square Inch.	d = Diameter, Inches.
Group II.	Geipel and Kilgour	$V = 9240\sqrt{\frac{Pd}{yL}}$	$W = 50.2\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003960\frac{W^2L}{yd^5}$	$d = 0.2087\sqrt[5]{\frac{W^2L}{Py}}$
	Gutermuth . .	$V = 9722\sqrt{\frac{Pd}{yL}}$	$W = 53\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003557\frac{W^2L}{yd^5}$	$d = 0.2032\sqrt[5]{\frac{W^2L}{Py}}$
	Hawksley . . .	$V = 9976\sqrt{\frac{Pd}{yL}}$	$W = 54.4\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003370\frac{W^2L}{yd^5}$	$d = 0.2010\sqrt[5]{\frac{W^2L}{Py}}$
	Martin.	$V = 10{,}350\sqrt{\frac{Pd}{yL}}$	$W = 56.5\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003133\frac{W^2L}{yd^5}$	$d = 0.1990\sqrt[5]{\frac{W^2L}{Py}}$
	Hurst.	$V = 10{,}360\sqrt{\frac{Pd}{yL}}$	$W = 56.5\sqrt{\frac{Pyd^5}{L}}$	$P = 0.0003126\frac{W^2L}{yd^5}$	$d = 0.1990\sqrt[5]{\frac{W^2L}{Py}}$
Group I.	Babcock	$V = 15{,}950\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001321\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	. . . .
	Unwin.	$V = 16{,}050\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	. . . .
	Carpenter . . .	$V = 16{,}050\sqrt{\frac{Pd}{yL\left(1+\frac{3.6}{d}\right)}}$	$W = 87.5\sqrt{\frac{Pyd^5}{L\left(1+\frac{3.6}{d}\right)}}$	$P = 0.0001306\frac{W^2L\left(1+\frac{3.6}{d}\right)}{yd^5}$	. . . .
	Ledoux.	$V = 442\sqrt{\frac{(P_1^{1.94}-P_2^{1.94})\,d}{y^2L}}$	$W = 2.44\sqrt{\frac{(P_1^{1.94}-P_2^{1.94})d^5}{L}}$	$P_1^{1.94}-P_2^{1.94} = 0.1669\frac{W^2L}{d^5}$	$d = 0.699\sqrt[5]{\frac{W^2L}{P_1^{1.94}-P_2^{1.94}}}$

It has been shown* that all of the rules in Table 127 with the exception of "Ledoux" are based on the general equation

$$p = C \frac{v^2 \gamma L}{d} \tag{274}$$

and differ only with respect to the assumed value of the coefficient of friction.

In equation (274),

p = pressure drop, lb. per sq. in.,
C = a coefficient involving a number of reduction constants and the coefficient of frictional resistance,
v = velocity, ft. per second,
γ = mean density of the steam, lb. per cu. ft.,
L = length of straight pipe, ft., or its equivalent,
d = inside diameter of the pipe, in.

Equation (274) may be reduced to the form

$$w = k \sqrt{\frac{p \gamma d^5}{L}}, \tag{275}$$

in which

w = weight of steam flowing, lb. per sec.,
k = a coefficient involving the various reduction constants and the coefficients of frictional resistance.

Numerous experiments have been made with a view of determining the coefficient of frictional resistance but the results have been far from harmonious. The coefficients involved in the equation given in Table 127 are not applicable to the present practice of high velocities, pressures and temperatures.

Fritzsche's equation (Mitt. uber Forschungsarbeit, Vol. 60) has been mentioned as giving results more in accord with current practice but recent investigations made at the Berliner Elektrizitats Werke (Prac. Engr., March 15, 1916, p. 284) show that pressure drops calculated by means of this equation may be 50 per cent too low. In the light of the best evidence available at the present time preference should be given to the coefficient as determined by J. M. Spitzglass (Armour Engineer, May, 1917). Using the values of the coefficient of friction as determined by Spitzglass equation (275) may be reduced to the convenient form

$$w = K \sqrt{\frac{p \gamma}{L}}, \tag{276}$$

in which K is a coefficient with values as given in Table 128.

* See Author's paper, Power, June, 1907, p. 377.

The author has applied equation (276) to a number of cases in which pressure drops have been determined experimentally and the calculated values checked substantially with the test results. The values of K given in the table allow a sufficient factor of safety for all fittings which do not abruptly change the direction of flow or reduce the pressure by throttling. Attempts to include factors for condensation losses merely complicate the equation without adding to its accuracy. *All equations thus far established relative to the flow of steam in pipes are but approximations at the best and should be used accordingly.*

Example 70. Determine the diameter of pipe suitable for a 30,000-kw. steam turbine lead with operating conditions as follows: Initial absolute pressure 265 lb. per sq. in., superheat 200 deg. fahr., length of pipe 100 ft., maximum pressure drop in the pipe alone to approximate 3 lb. per sq. in. when delivering 330,000 lb. steam per hour.

$$w = \frac{330{,}000}{3600} = 91.66, \qquad L = 100, \qquad p = 3.$$

For small pressure drops the density may be assumed as that corresponding to initial pressure, thus $\gamma = 0.425$ for $p_1 = 265$ lb. abs. and 200 deg. fahr. superheat.

Substituting these values in equation (276)

$$91.66 = K\sqrt{\frac{3 \times 0.425}{100}},$$

from which $K = 811+$.

From Table 128 it will be seen that K for a 14-inch pipe is 800. A 14-inch pipe would therefore be the nearest commercial size which will fulfill the required conditions.

TABLE 128.

VALUES OF K FOR VARIOUS PIPE SIZES.

Nominal Pipe Diameter, In.	K.	Nominal Pipe Diameter, In.	K.
1.0	0.75	5.0	60.0
1.5	2.5	6.0	97.0
2.0	5.1	8.0	195.0
2.5	8.5	10.0	350.0
3.0	15.5	12.0	550.0
3.5	23.0	14.0	800.0
4.0	32.5	16.0	1100.0

349. Friction through Valves and Fittings. — Equations 275 to 276 and those outlined in Table 127 are strictly applicable only to well-lagged pipes, free from sharp bends or obstructions such as valves or fittings, which greatly increase the resistance of the flow of steam. If these obstructions must be considered, it is customary to allow for them by assuming an added length of straight pipe equivalent in resistance to the various fittings and bends. Unfortunately, the few tests which have been made for the purpose of determining the resistance of various pipe fittings give discordant results, and rules based on these investigations are limited to such a narrow range of operating conditions that their use for general design purposes is apt to lead to serious error.

It is definitely known that the value of the coefficient of resistance for smooth piping decreases with increasing diameter but with globe valves in short lengths of piping it appears to increase with increasing

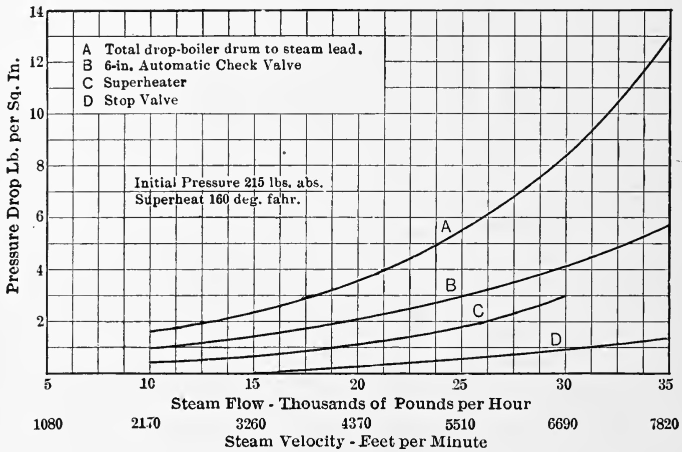


FIG. 491. Steam Pressure Drop, 500-hp. Babcock and Wilcox Boilers.

diameters. It seems probable that the placing of a globe valve in a limited length of piping produces an increasing diminution of the free steam passage for increasing valve diameters and thereby causes a whirling and friction which increases the resistance. The frequently observed fact that piping of large diameter always gives a higher pressure drop than is expected from calculations is probably due to allowing too small values for the resistance of valves, superheaters and fittings.

According to Briggs ("Warming Buildings by Steam") the length of straight pipe in inches equivalent to the resistance of one standard 90-degree elbow is

$$L = 75\,d \div \left(1 + \frac{3.6}{d}\right), \qquad (277)$$

and that of a globe valve

$$L = 114\,d \div \left(1 + \frac{3.6}{d}\right). \qquad (278)$$

These rules have been frequently quoted but results calculated from them are not in accord with the actual pressure drop in modern power plant practice. The curves in Figs.* 491 and 492 give some idea of the pressure drops in the piping system of a modern turbo-generator plant and serve to show that the actual drops are much higher than ordinarily supposed.

350. Equation of Pipes. — It is frequently desirable to know what number of one sized pipes will be equal in capacity to another pipe.

According to the equations in Group II, Table 127, the weights discharged for a given set of conditions vary with the square root of

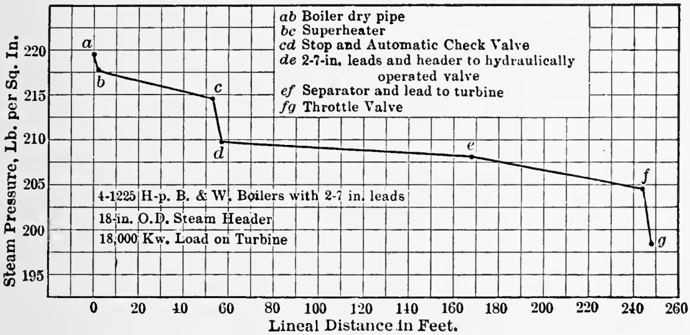


FIG. 492. Steam Pressure Drop from Boiler Drums to Turbine Throttle.

the fifth power of the diameter; that is, the number of pipes equal in capacity to any given pipe may be determined from the equation

$$N_1 = d^{\frac{5}{2}} \div d_1^{\frac{5}{2}}, \tag{279}$$

in which N_1 = number of pipes of diameter d_1 equal in capacity to a pipe of diameter d; d_1 and d in inches.

According to the equation (276)

$$N_1 = \left(\frac{d^5}{1 + \dfrac{3.6}{d}} \div \frac{d_1^5}{1 + \dfrac{3.6}{d_1}} \right)^{\frac{1}{2}} \tag{280}$$

Thus, one 8-inch pipe is equal in capacity to six 4-inch pipes, or (from Table 128) $N_1 = 195 \div 32.5 = 6$.

351. Exhaust Piping, Condensing Plants. — The exhaust piping in condensing plants is arranged either according to (1) the *independent* or (2) the *central condensing system.* In the former each engine is provided with an independent condenser and air pump. In case the

* Courtesy of A. D. Bailey, Engineer in Charge of Fisk Street and Quarry Street Stations, Commonwealth Edison Co., Chicago.

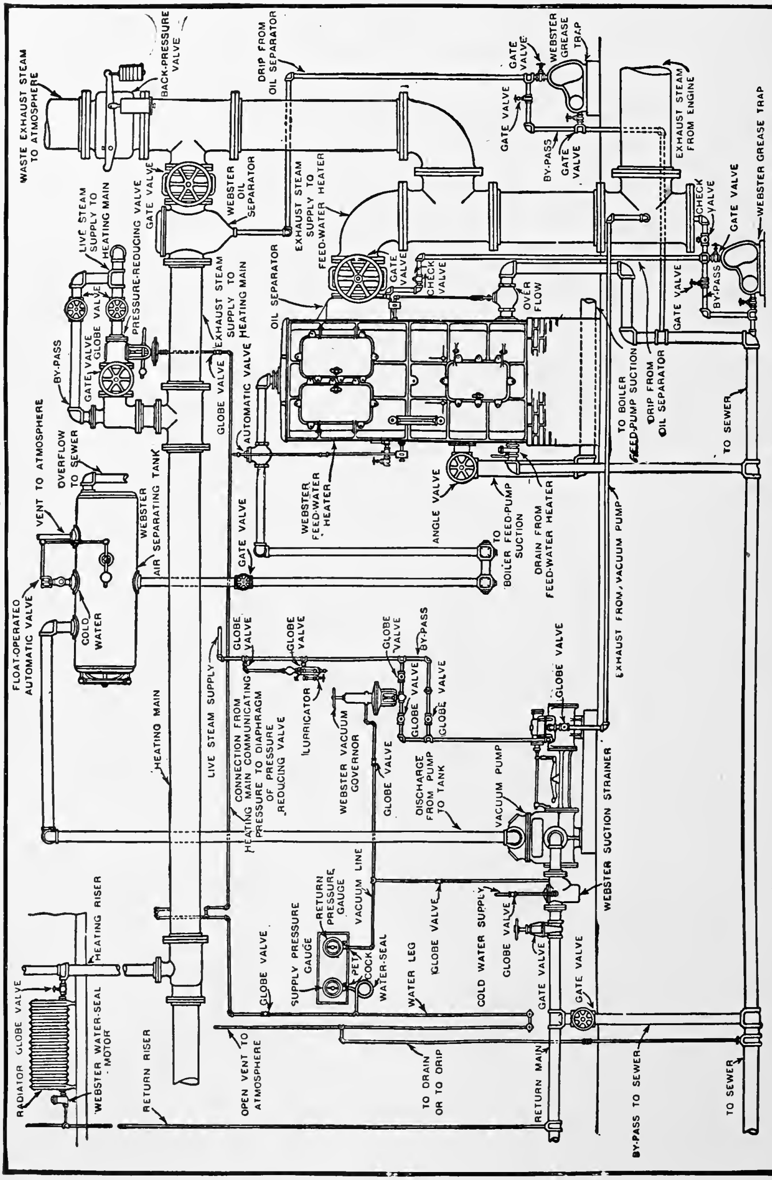


Fig. 493. Diagrammatic Arrangement of Webster Vacuum Heating System.

vacuum "drops" or it is desired to operate non-condensing, the steam is discharged through a branch pipe with relief valve to the atmosphere, Figs. 3 and 321. When there are a number of engines in one installation the atmospheric pipes lead to a common *free exhaust main*, which, on account of its great size, is ordinarily constructed of light-weight riveted steel pipe. The short connection between engine and condenser is usually made with lap-welded steel pipe, since riveted joints are apt to leak, due to the engine vibrations. In a central condensing plant, Fig. 330, the several engines exhaust through a common main into a single large condenser. An atmospheric relief valve is usually provided in connection with the condenser, and no free exhaust main is necessary. Several arrangements of condenser piping are illustrated in Figs. 321 to 330.

352. Exhaust Piping, Non-condensing Plant. Webster Vacuum System. — In the majority of non-condensing plants the exhaust steam is used for heating purposes. One of the best-known systems of exhaust steam heating, in which the back pressure on the engine is reduced by circulating below atmospheric pressure, is that known as the *Webster combination system*. The general arrangement is illustrated in Fig. 2 and the principles of operation are described in paragraph 3. It has the advantage of affording (1) minimum back pressure on the engine; (2) effective and continuous drainage of condensation from supply pipes and radiators; (3) continuous removal of air and entrained moisture from confined spaces; (4) independent regulation of temperature in each radiator; (5) continuous return of condensation to the boiler; (6) utilization of part of the exhaust steam for preheating the feed water; and (7) automatic regulation. Fig. 493 gives a diagrammatic arrangement of the piping and appurtenances in a typical installation. The characteristic feature of this system is the automatic outlet valve attached to each part requiring drainage, which permits both the water of condensation and the non-condensable gases to be removed continuously. The radiator temperature may be regulated by varying the quantity of steam supplied, either by hand or automatically by thermostatic control. The Webster valve, Fig. 493a, enables the vacuum to withdraw the water of condensation as fast as it is formed irrespective of the pressure in the radiator; hence the supply may be throttled to

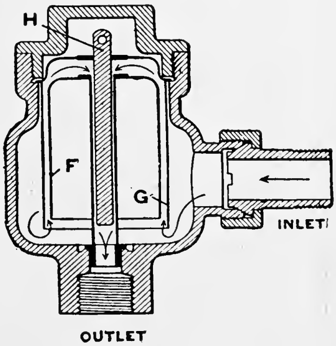


FIG. 493a. Webster Air Valve.

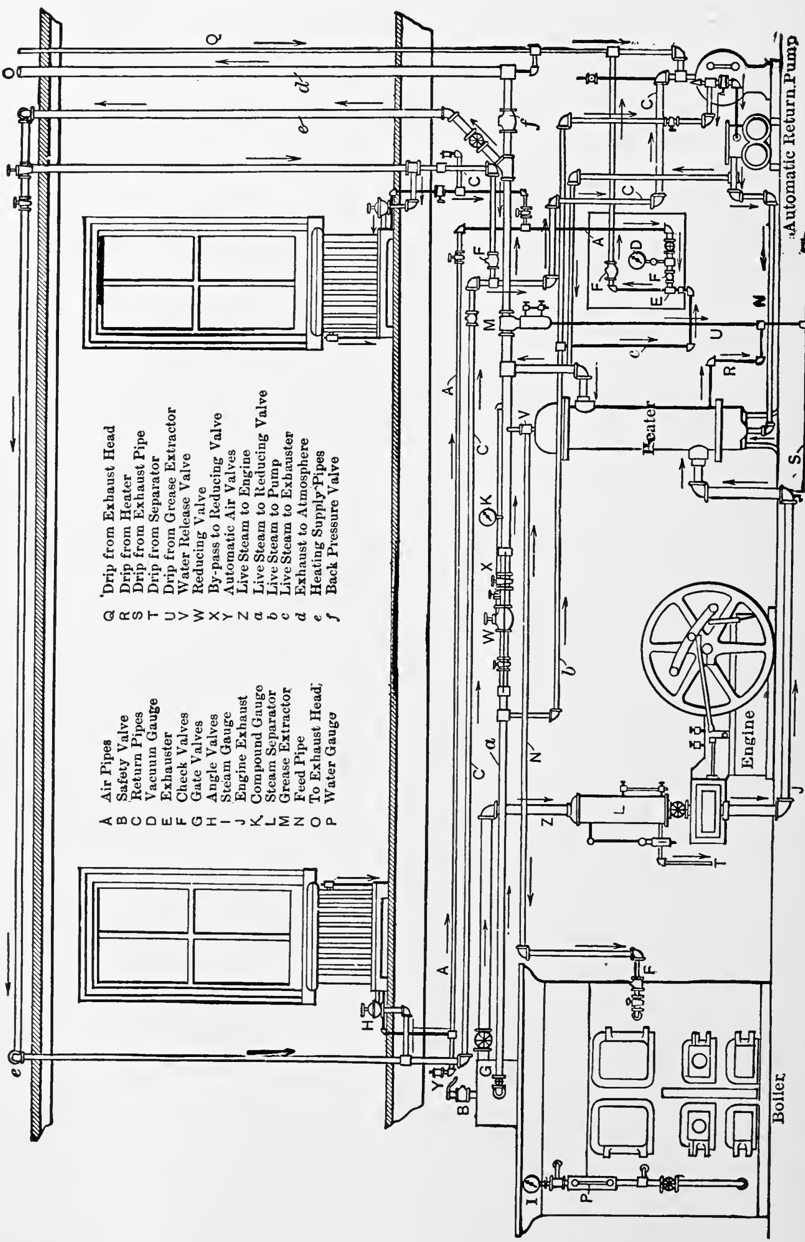


FIG. 494. Diagrammatic Arrangement of Piping in the Paul Vacuum Heating System.

such an extent that the temperature in the radiator is practically as low as that of steam corresponding to the pressure in the vacuum line. The small annular space between the inner tube of the float *F* and the guide *H* permits of a vacuum in the body of the valve. When the water from the radiator lifts the float the water is drawn into the returns pipe. The valve then returns to its seat and the escape of steam is prevented, except such as finds its way through the annular space around the guide stem *H*.

Automatic air valves are constructed in a variety of designs but space limitation prevents their description in this work. For a detailed description of a number of well-known devices consult "Mechanical Equipment of Buildings," by Harding and Willard, John Wiley & Sons, 1916.

353. Exhaust Piping, Non-condensing Plants. Paul Heating System. — The Paul vacuum system differs from the Webster in that the condensation, and the air and non-condensable gases are separately handled. Referring to Fig. 494, which gives a diagrammatic arrangement of the piping, the condensed steam gravitates to the *automatic returns tank and pump* and is pumped either directly to the boiler or through the heater to the boiler. Air and vapor are withdrawn from the upper part of the radiator by the *Paul exhauster* or ejector *E*, and discharged into the returns tank, which is vented to the atmosphere for the escape of the non-condensable gases. The exhauster receives its supply of steam through pipe *O*, Fig. 495, which shows the general arrangement of this apparatus. The piping is in duplicate to guard against failure to operate. The suction side of the exhauster is connected with the air pipes *A*, *A*, Fig. 494. Fig. 496 gives a section through the *Paul air* or *vacuum* valve which prevents steam from blowing into the air pipes and permits only air to pass. In Fig. 494 the heating system is

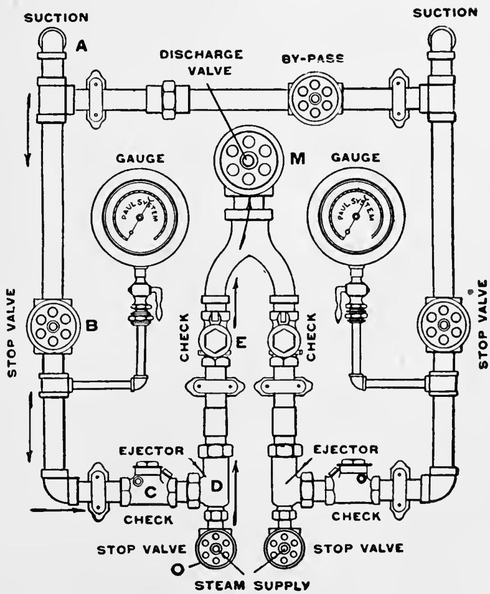


FIG. 495. Paul Exhauster.

piped on what is known as the "one-pipe down-feed" principle; i.e., the exhaust steam is first conducted to a distributing header in the attic, from which the various supply pipes are led to the radiators. The water of condensation returns through these same pipes and gravitates to the returns pump. Both the supply steam and the condensation flow in the same direction. This system is also piped on the "one-pipe up-feed," the "two-pipe up-feed," and the "two-pipe down-feed" principle. The "one-pipe up-feed" differs from the system just described in that the steam flows upward through the risers and does away with the attic piping. The returns, however, flow against the current of steam, and water hammer is more likely to occur than with the down-feed system. In the two-pipe systems the steam supply pipes or risers conduct steam only, and the returns carry the condensation. The one-pipe down-feed is cheaper and simpler and practically as efficient as the two-pipe system under normal conditions. It is objectionable, however, due to the difficulty of draining the radiator with closely throttled supply valve, since the velocity of the entering steam prevents the water from returning through the same orifice.

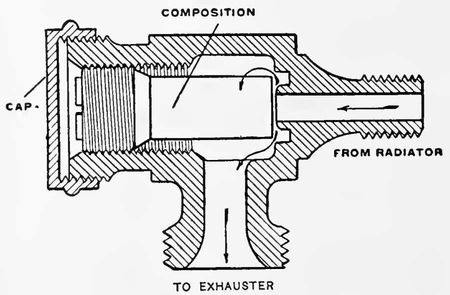


Fig. 496. Paul Vacuum Valve.

354. Automatic Temperature Control. — Experience shows that a considerable saving in fuel may be effected in the heating plants of tall office buildings and similar plants by automatically controlling the temperature. Hand-controlled valves are usually left wide open, and when the room becomes too hot the temperature is frequently lowered by opening the window, resulting in a waste of heat which may be considerable in modern buildings with hundreds of offices. Many successful methods of automatic temperature control are available, the usual system consisting of *thermostats* which control the supply of heat by means of *diaphragm valves,* the latter taking the place of the usual radiator supply valve.

Fig. 497 shows a Powers thermostat. The expansible disk U contains a volatile liquid having a boiling point of about 50 deg. fahr. The pressure of the vapor within the disk at a temperature of 70 degrees amounts to six pounds to the square inch, and varies with every change of temperature, causing a variation in the thickness of the disk. The disk is attached by a single screw O to the lever Q, which rests upon the screw F as a fulcrum. The flat spring R holds the lever and disk against the movable flange M. Connecting with the chamber N are two air passages H and I. The thermostat is attached by means of two screws at the upper end to a wall plate permanently secured to the wall. This wall plate has ports registering with H and I, one for supplying air under pressure and the other for conducting it to the diaphragm motor which operates the valve or damper. Air is admitted through H under a pressure of about fifteen pounds per square inch, and its passage into chamber N is regulated by the valve J, which is normally held to its seat by a coil spring under cap P. K is an elastic diaphragm carrying the flange M, with escape valve passage covered by the point of valve L. Valve L tends to remain open by reason of the spring. When the temperature rises sufficiently expansion of the disk U first causes the valve to seat, its spring being weaker than that above valve J. If the expansive motion is continued, valve J is lifted from its seat and compressed air flows into chamber N, exerting a pressure upon the elastic diaphragm K in opposition to the expansive force of the disk. If the temperature falls, the disk contracts and the overbalancing air pressure in N results in a reverse movement of the flange M, permitting the escape valve to open and discharge a portion of the air; thus the air pressure is maintained always in direct proportion to the expansive power (and temperature) of the disk U. The passage I communicates with a diaphragm valve, Fig. 498. The compressed air operates the diaphragm against a coiled spring resistance, so that the movement is proportional to the air pressure and the supply of steam controlled accordingly. The adjusting screw G, squared to receive a key, carries an indicator by means of which the thermostat can be set to carry any

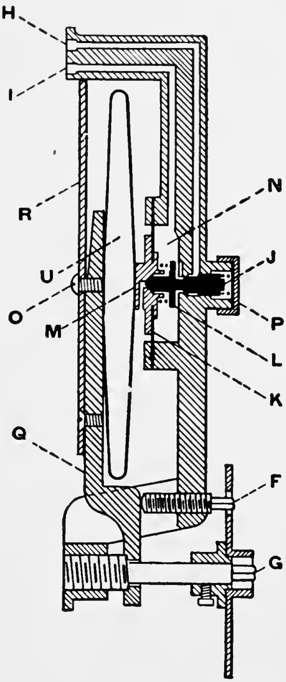


Fig. 497. Section through Powers Thermostat.

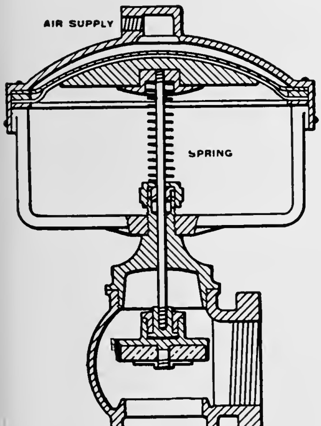


Fig. 498. A Typical Diaphragm Valve.

desired temperature within its range, usually from 60 to 80 degrees. In changing the temperature adjustment lever *Q* forces the disk *U* closer to or farther from the flange *M*.

In connecting up the system compressed air is carried to the thermostat and diaphragm valves, from a reservoir through small concealed pipes.

In the indirect system of heating the dampers are of the diaphragm type and the method of regulation is the same as with the direct system.

355. Feed-water Piping. — The simplest arrangement of feed-water piping may be found in non-condensing plants, in which the feed water is obtained under a slight head, such as is afforded by the average city supply, and is heated in an open heater by the exhaust steam from the engine to a temperature varying from 180 to 210 deg. fahr. The hot

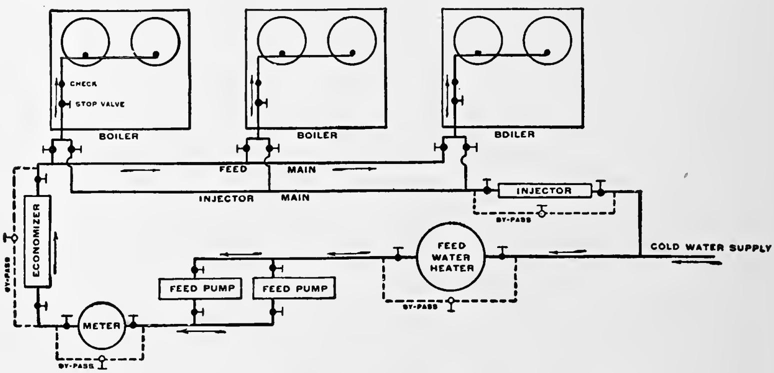


Fig. 499. Feed-water Piping; Non-condensing Plant.

feed water gravitates from the heater to the pump and then is forced to the boiler, or to the economizer if one is used. If a meter is used it is generally placed on the discharge side of the pump, and should be by-passed to permit it to be cut out for repairs (Fig. 499). Plants operating continuously should have feed pumps in duplicate. In some cases the returns from the heating system gravitate to the heater and only enough cold water is added to make up the loss from leakage, etc. In other cases the returns gravitate to a special "returns tank," from which they are pumped directly to the boiler without further heating. Occasionally a live-steam purifier is used, especially if the water contains a large percentage of calcium sulphate. The feed is then subjected to boiler pressure and temperature and the greater part of the impurity precipitated before it enters the boiler. Closed heaters are often used in place of open heaters. When the supply is not under head a closed heater is usually preferred and is placed between the pump discharge and the feed main.

In condensing plants the feed piping is similar to that in non-condensing plants, except that if exhaust steam is used for heating purposes it is supplied by the auxiliaries, such as feed pumps, stoker engines, condenser engines, and other steam-using appliances.

In plants having a number of boilers it is customary to run a feed main or header the full length of the boiler room and connect it to each boiler by a branch pipe. This main may be a simple header or in duplicate or of the "loop" or "ring" type. Horizontal tubular boilers are frequently arranged in one battery with the feed main run along the fronts of the boilers just above the fire doors. Water-tube boilers are generally set in a battery, and as the arrangement above would block the passageway between the batteries, the main is run either above or under the settings, the former being the more common. Where a

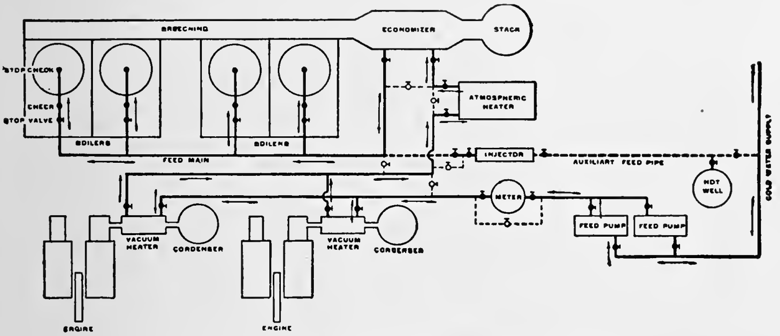


Fig. 500. Feed-water Piping; Condensing Plant.

single header is used, the feed pumps are sometimes placed so as to feed into opposite ends of the main, which is then cut into sections by valves. Another arrangement is to place the pumps so as to feed into the middle of the header. With the loop arrangement the main is ordinarily cut into sections by valves so that the water may be sent either way from the pumps and any defective section cut out. With duplicate mains a common arrangement is to place one main along the front of the boiler and the other at the rear or both overhead as in Fig. 489. Sometimes one main is placed in the passageway below the boiler setting and the other on top.

Standard wrought-iron pipe is usually used for pressures under 100 pounds and extra heavy pipe for greater pressures. The pipes and fittings from boiler to main are frequently of brass, and preferably so, since brass withstands corrosive action much better than iron or steel. Flanged joints should be used in all cases, since the pockets formed by the ordinary screwed joints hasten corrosion at those points. (Power, June, 1902, p. 4.)

Fig. 502, *A* to *E*, illustrates the various combinations of check valve, stop valves, and regulating valve in steam boiler practice. The simplest arrangement and one sometimes used in plants operating intermittently

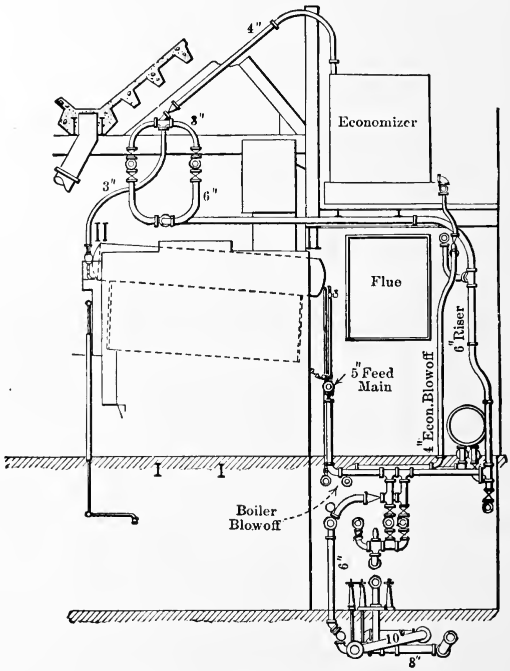


Fig. 501. Feed-water Piping.

is shown in *A*. Here there are but two valves between the boiler and the main, the check being nearest the boiler and the stop valve at the main. The stop valve performs both the function of cutting out the

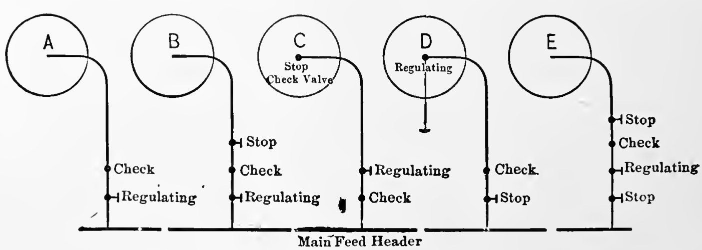


Fig. 502. Different Arrangements of Valves in Feed-water Branch Pipes.

boiler and of regulating the water supply. This arrangement is not recommended, as any sticking or excessive leaking of the check valve will necessitate shutting down the boiler. *B* shows the most common arrangement. Here the check valve is placed between the regulating valve and a stop valve as indicated. This permits a disabled check to be easily removed while pressure is on the boiler and the main. E shows an arrangement whereby both check and regulating valve may be removed, and is particularly adapted to boilers operating continuously where the regulating valve is subjected to severe usage. In this case the stop valves are run wide open and are subjected to no wear. The regulating valve most highly recommended is a *self-packing* brass globe valve with *regrinding disk*. The check valve is ordinarily of the *swing check* pattern with regrinding disk, Fig. 513 (C). Modern practice recommends an *automatic water relief valve* in the discharge pipe immediately adjacent to each pump (piston type only) to prevent excessive pressure in case a valve is accidentally closed in by-passing or in changing over.

356. Flow of Water through Orifices, Nozzles, and Pipes. — Bernoulli's theorem is the rational basis of most empirical equations for the steady flow of a fluid from an up-stream position n to a down-stream position m, thus ("Mechanics of Engineering," Church, p. 706):

$$\frac{P_m}{\gamma} + \frac{V_m^2}{2g} + Z_m = \frac{P_n}{\gamma} + \frac{V_n^2}{2g} + Z_n - \left\{ \begin{array}{l} \text{all losses of head} \\ \text{occurring between} \\ n \text{ and } m \end{array} \right\}, \tag{281}$$

in which

V = velocity in feet per second at the point considered,
P = pressure in pounds per square foot,
Z = potential head in feet of the fluid,
γ = density of the fluid, pounds per cubic foot,
g = acceleration of gravity.

Each loss of head will be of the form $K\,\frac{V^2}{2g}$ in which K is the *coefficient of resistance* to be determined experimentally. The loss of head due to *skin friction* is expressed:

$$H = 4f\,\frac{l}{d} \times \frac{v^2}{2g}, \tag{282}$$

in which

f = the coefficient of friction of the fluid in the pipe,
l = length of the pipe in feet,
d = diameter of the pipe in feet.

Other notations as in (281).

Discharge from a circular vertical orifice with sharp corners:

$$Q = CA\sqrt{2\,gh}, \tag{283}$$

in which

Q = cubic feet per second,
C = coefficient, varying from 0.59 to 0.65 (Merriman, "Treatise on Hydraulics," p. 118),
A = area of the orifice, square feet,
h = head of water in feet,
g = acceleration of gravity = 32.2.

Discharge from short cylindrical nozzles three diameters in length, with rounded entrance ("Mechanics of Engineering," Church, p. 690):

$$Q = 0.815\,A\,\sqrt{2\,gh}. \tag{284}$$

Discharge from short nozzles with well-rounded corners and conical convergent tubes, angle of convergence $13\frac{1}{2}$ *degrees* (Church, p. 693):

$$Q = 0.94\,A\,\sqrt{2\,gh}. \tag{285}$$

Discharge from cylindrical pipe under 500 *diameters in length* (Church, p. 712):

$$Q = 6.3\sqrt{\frac{d^5h}{(1 + 0.5)\,d + 4\,fl}}, \tag{286}$$

in which

f = coefficient of friction.

Other notations as above.

f varies with the nature of the inside surface, the diameter of the pipe, and the velocity of flow.

Discharge through very long cylindrical pipes ("Mechanics of Engineering," Church, p. 715):

$$Q = 3.15\sqrt{\frac{d^5h}{fl}}. \tag{287}$$

*Loss of head due to friction in water pipes.** Weisbach's equation is as follows:

$$H = \left(0.0144 + \frac{0.01716}{\sqrt{V}}\right)\frac{LV^2}{5.367\;d}, \tag{288}$$

in which

H = friction head in feet,
V = velocity in feet per second,
L = length of pipe in feet,
d = diameter of pipe in inches.

* See also, Friction Formulas for Commercial Pipe, by Ira N. Evans, Power, July 9, 1912, p. 54.

TABLE 129.

TABLE OF THE COEFFICIENT *f* FOR FRICTION OF WATER IN *CLEAN* IRON PIPES.

(Abridged from Fanning.)

Velocity in Ft. per Sec.	Diam. = ½ in. = .0417 ft.	Diam. = 1 in. = .0834 ft.	Diam. = 2 in. = .1667 ft.	Diam. = 3 in. = .25 ft.	Diam. = 4 in. = .333 ft.	Diam. = 6 in. = .50 ft.	Diam. = 8 in. = .667 ft.
0.1	.0150	.0119	.00870	.00800	.00763	.00730	.00704
0.3	.0137	.0113	850	784	750	720	693
0.6	.0124	.0104	822	767	732	702	677
1.0	.0110	.00950	790	743	712	684	659
1.5	.00959	.00868	.00757	.00720	.00693	.00662	.00640
2.0	.00862	810	731	700	678	648	624
2.5	795	768	710	683	662	634	611
3.0	.00753	.00734	.00692	.00670	.00650	.00623	.00600
4.0	722	702	671	651	631	607	586
6.0	689	670	640	622	605	582	562
8.0	663	646	618	600	587	562	544
12.0	630	614	590	582	560	540	522
16.0	.00618	.00600	.00581	.00570	.00552	.00530	.00513
20.0	615	598	579	566	549	525	508

Velocity in Ft. per Sec.	Diam. = 10 in. = .833 ft.	Diam. = 12 in. = 1.00 ft.	Diam. = 16 in. = 1.333 ft.	Diam. = 20 in. = 1.667 ft.	Diam. = 30 in. = 2.50 ft.	Diam. = 40 in. = 3.333 ft.	Diam. = 60 in. = 5. ft.
0.1	.00684	.00669	.00623	....	....	....	....
0.3	673	657	614	.00578	....	....	....
0.6	659	642	603	567	.00504	.00434	.00357
1.0	643	624	588	555	492	428	353
1.5	.00625	.00607	.00572	.00542	.00482	.00421	.00349
2.0	609	593	559	529	470	416	346
2.5	596	581	548	518	460	410	342
3.0	.00584	.00570	.00538	.00509	.00452	.00407	.00339
4.0	568	553	524	498	441	400	333
6.0	548	534	507	482	430	391	324
8.0	532	520	491	470	422	384	320
12.0	512	500	478	457	412	377	.00313
16.0	.00502	.00491	.00470	.00450	.00406	.00370	....
20.0	498	485	....	....	....	....	....

William Cox (American Machinist, Dec. 28, 1893) gives a simple rule which gives almost identical results:

$$H = \frac{(4\,V^2 + 5\,V - 2)\,L}{1200\,d}. \tag{289}$$

Notations as in (288).

Loss of head due to friction of fittings. Equations (286) to (289) are based on the flow of water through clean straight cylindrical pipes. Where there are bends, valves, or fittings in the line the flow is decreased on account of the additional resistance.

These frictional losses are conveniently expressed in feet of water, thus:

$$H = C \frac{V^2}{2g}, \tag{290}$$

C having the following values:

	Angles.		Class of Valve.		
	45 degrees.	90 degrees.	Gate.	Globe.	Angle.
C	0.182	0.98	0.182	1.91	2.94

Example 71. Determine the pressure necessary to deliver 200 gallons of water per minute through a 4-inch iron pipe line 400 feet long, fitted with four right-angle elbows and two globe valves. The water is to be discharged into an open tank.

A flow of 200 gallons per minute gives a velocity of

$\frac{200 \times 144}{7.48 \times 60 \times 12.72} = 5$ feet per second (7.48 = number of gallons per cubic foot, and 12.72 = internal area of the pipe, square inches).

From the preceding table, $f = 0.00618$ for $V = 5$.

From (290),

Resistance head of 4 elbows $= 0.98 \times \frac{25}{64.4} \times 4 = 1.52$ feet.

Resistance head of 2 globe valves:

$$1.91 \times \frac{25}{64.4} \times 2 = 1.48 \text{ feet.}$$

Resistance head of all fittings:

$$1.52 + 1.48 = 3 \text{ feet.}$$

Substitute $V = 5$, $L = 400$, and $d = 4$ in (289).

$$H = \left(\frac{4 \times 5^2 + 5 \times 5 - 2}{1200 \times 4}\right) 400$$

$$= 10.25 \text{ feet, resistance head of the pipe.}$$

Total resistance head $= 10.25 + 3 = 13.25$ feet of water, or 5.75 pounds per square inch.

Example 72. How many gallons of water will be discharged per minute through above line with initial pressure of 100 pounds per square inch, and what will be the pressure at the discharge end?

Since f depends upon the unknown V, we may put $f = 0.006$ for a first approximation and solve for V; then take a new value of f and substitute again, and so on.

Substitute $f = 0.006$, $d = \frac{4}{12}$, $h = 100 \times 2.3 = 230$, and $l = 400$ in (287):

$$Q = 3.15 \sqrt{\frac{0.33^5 \times 230}{0.006 \times 400}}$$

$= 1.95$ cubic feet per second, corresponding to a velocity of 22 feet per second.

From table,

$f = 0.00548$ (by interpolation) for $V = 22$ feet per second.

From (290) the friction of 4 elbows and 2 globe valves is found to be 58 feet for $V = 22$.

From (289) a resistance head of 58 feet of water for $V = 22$ is found to be equivalent to 136 feet of straight pipe, thus:

$$58 = \left(\frac{4 \times 22^2 \times 5 \times 22 - 2}{1200 \times 4}\right) L.$$

$$L = 136.$$

Substitute $f = 0.0548$, $l = 400 + 136 = 536$ in (287):

$$Q = 3.15 \sqrt{\frac{0.33^5 \times 230}{0.0058 \times 536}}$$

$= 1.74$ cubic feet per second, corresponding to a velocity of 19.3 feet per second.

$= 780$ gallons per minute.

If greater accuracy is necessary determine f and L for $V = 19.3$ and proceed as above.

The total friction head may be determined from (289), thus:

$$H = \left(\frac{4 \times 19.3^2 + 5 \times 19.3 - 2}{1200 \times 4}\right) 536$$

$= 177$ feet of water

$= 77$ pounds per square inch.

The pressure at the discharge end will be

$$100 - 77 = 23 \text{ pounds per square inch.}$$

Average power plant practice gives the following maximum velocities of flow in water pipes:

Size of Pipe in Inches.	Velocity, Feet per Minute.	Size of Pipe in Inches.	Velocity, Feet per Minute.
$\frac{1}{8}$ to $\frac{1}{2}$	50	3 to 6	250
$\frac{1}{2}$ to $1\frac{1}{2}$	100	Over 6	300–400
$1\frac{1}{2}$ to 3	200		

357. Stop Valves. — The valves used to control and regulate the flow of fluids are the most important element in any piping system. A good valve should have sufficient weight of metal to prevent distortion under varying temperature and pressure, or under strains due to connection with the piping; the seats should be easily repaired or renewed; there should be no pockets or projections for the accumulation of dirt and scale, and the valve stem should permit of easy and efficient packing. Stop valves are made in such a variety of designs that a brief description will be given of only a few fundamental types.

Fig. 503 shows a section of an ordinary *globe valve*, so called because of the globular form of the casing. This type of valve is the most common in use. Globe valves are designated as (1) *inside screw* and (2) *outside screw*, according as the screw portion of the stem is inside the casting, Fig. 503, or outside, Fig. 504. The top, or bonnet, may be screwed into the body of the valve, Fig. 503, or bolted, Fig. 504. The smaller sizes, three inches and under, are usually of the *screw-top* type and the larger of the *bolt-top* type. Valves with *outside yoke* and *screw*

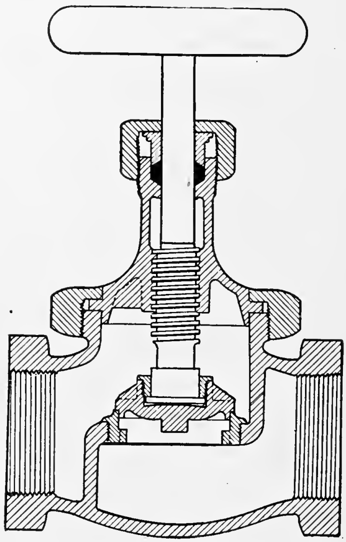


Fig. 503. A Typical Globe Valve, Screw-top, Inside Screw.

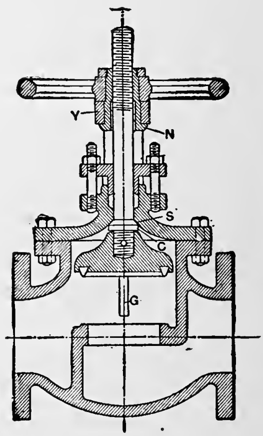


Fig. 504. A Typical Globe Valve, Bolt-top, Outside Screw.

are preferable to others in that they show at a glance whether the valve is open or closed, an advantage in changing from one section to another. The disks are made in a variety of forms, the material depending upon the nature of the fluid to be controlled. Thus, for cold water, hard rubber composition gives good results; for hot water and low-pressure steam, Babbitt metal; for high-pressure steam, copper or bronze; and for highly superheated steam, nickel. The valve bodies are of brass for sizes under three inches, cast iron for the larger sizes and ordinary pressures and temperatures, and cast steel or semi-steel for high temperatures and pressures. Globe valves should always be set to close against the pressure, otherwise they could not be opened if the valves should become detached from the stem. Globe valves should never be placed in a horizontal steam return pipe with the stem vertical, because the condensation will fill the pipe about half full before it can flow through the valve. Globe valves that are open all the time are preferably designed with a *self-packing spindle*, as in Fig. 504, in which the top of shoulder *C* can be drawn tightly against the under surface of bonnet *S*, thus preventing steam from leaking past the screw threads while the spindle is being packed.

Figs. 505 to 507 show different types of *gate* or *straightway* valves. These valves offer little resistance to the flow of steam or liquid passing

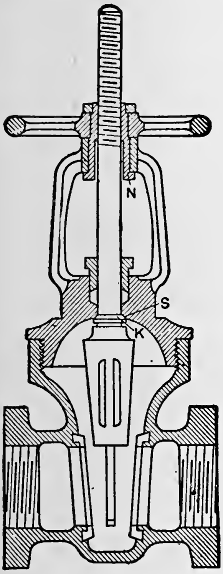


Fig. 505. A Typical Gate Valve, Solid-wedge, Screw-top, Outside Screw.

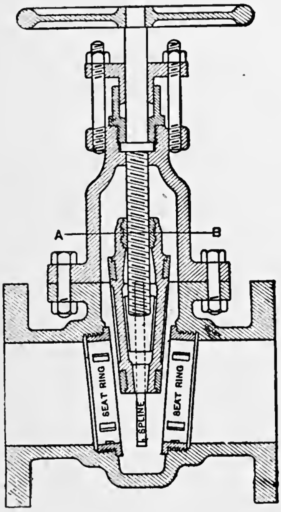


Fig. 506. A Typical Gate Valve, Solid-wedge, Bolt-top, Inside Screw.

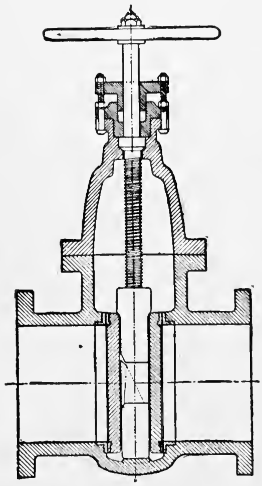


Fig. 507. A Typical Gate Valve, Split-wedge, Bolt-top, Inside Screw.

through them, and are generally used in the best class of work. Fig. 505 shows a section through a *solid-wedge* gate valve with outside screw and yoke. This form of outside screw and yoke with stem protruding beyond the hand wheel is a perfect indicator to show whether the valve is open or shut, as the hand wheel is stationary and the spindle rises in direct proportion to the amount the valve is opened. For these reasons outside screw valves are preferable for high-pressure work and especially for the larger sizes. The seats are made solid, or removable, and of various materials for different pressures and temperatures. Fig. 507 shows a section through a *split-wedge* gate valve with parallel faces and seats. For the sake of illustration this valve is fitted with inside screw.

In this design the spindle remains stationary so far as any vertical movement is concerned, and the gate or plug, being attached to it by means of a threaded nut, rises into the bonnet when the spindle is revolved. It is impossible to tell by its appearance whether this form of valve is opened or closed. Valves with inside screw are adapted to situations where there is considerable dirt and grit, since the screw is inclosed and protected, and excessive wear is thus avoided. Gate valves with split gates are more flexible than those with solid gates, and hence are less likely to leak. Fig. 508 shows the application of the

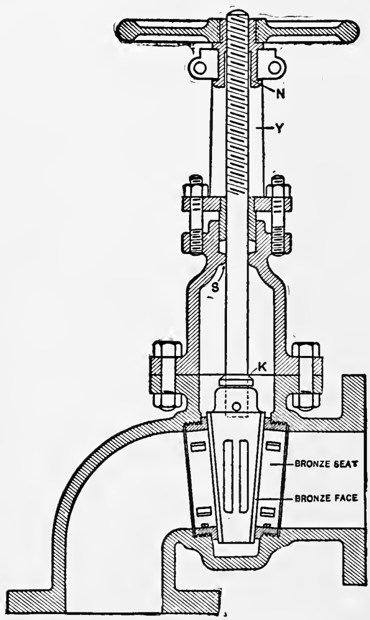


FIG. 508. Ludlow Angle Valve, Gate Pattern.

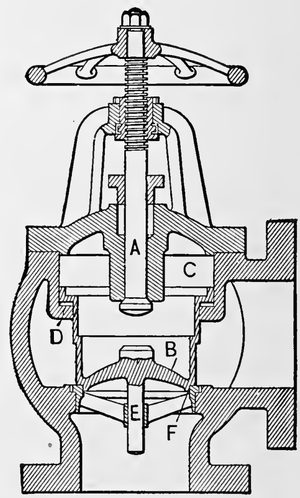


FIG. 509. Anderson Non-return Valve.

gate system to an angle valve. All high-pressure valves above 8 inches in diameter should be provided with a small by-pass valve, as the pressure exerted against the disk or gate is very great when the valve is closed and the force required to move it is considerable. The by-pass valve also facilitates "warming up" the section to be cut in and is more readily operated than the main valve.

358. Automatic Non-return Valves. — Fig. 509 shows a section through an *automatic non-return* valve as applied to the nozzle of a steam boiler. As will be seen from the illustration it practically amounts to a large check valve with cushioned disk. The object of this device is the equalization of pressure between the different units of the battery, the valve remaining closed as long as the individual boiler pressure is lower than that of the header. In case a tube blows out the valve closes automatically, owing to the reduction of pressure, and prevents the header steam from entering the boiler. It acts also as a safety stop to prevent steam being turned into a cold boiler while men are working inside, because it cannot be opened when there is pressure on the header side only. To be successful, such a valve should not open until the pressure in the boiler is equal to that in the header; it should not stick and become inoperative nor chatter and hammer while performing its work. Referring to Fig. 509, tail rod *E* insures alignment and hence prevents sticking; steam space *C* acts as a dashpot to prevent hammering of the valve as it rises, and steam space *D* acts as a cushion and prevents hammering at closing. Lip *F* is made to enter the opening in the seat and reduce wire drawing across the seat. Fig. 485 shows the installation of a number of non-return valves at the Yonkers power house of the New York Central Railway Company.

359. Emergency Valves and Automatic Stops. — In large power plants it is customary to protect the various divisions of the steam piping by *emergency valves* which may be closed by suitable means at any reasonable distance from the valve. The simplest form of emergency stop is a weighted "butterfly" valve, which is to all intents and purposes a weighted check, as illustrated in Fig. 513 (D). The weight when supported, say by a cord and pulley, holds the valve open; when the cord is cut or released the weight drops and forces the valve shut. The cord may lead to any convenient and safe distance from the valve. In applying this system of control to steam engines the valve is placed in the steam pipe just above the throttle and the weight held up by a lever controlled by the main governor or preferably by a separate governor. Should the engine exceed a certain speed, as in case of accident to the regular governor, the lever supporting the weight is tripped by the emergency governor and the valve is closed automatically. For high pressures a rotating plug valve or cock is preferred to the butterfly type, since it is balanced in all positions. Gate and globe valves may be converted into emergency valves by having the stems mechanically operated by electric motors, hydraulic pistons, and the like. Fig. 510 shows a section through a Crane hydraulically operated emergency gate valve.

Fig. 511 shows a partial section through an "Anderson triple-duty" emergency valve, and Fig. 512 a section through the pilot valve. A steam connection from the main line to the top of a copper diaphragm holds the pilot valve closed because of the large area above the diaphragm. A steam pipe connection from underneath the emergency piston of the triple-acting valve also leads to the pilot valve. In case a break occurs in the main steam line or branches, the pressure is removed from the top of the pilot valve, causing it to open, thus exhausting the pressure from beneath the emergency piston in the triple-acting valve. The boiler pressure on top of the emergency piston

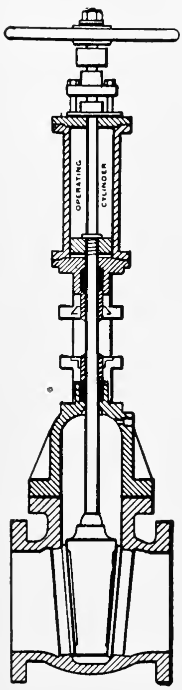


Fig. 510. Crane Emergency Valve, Hydraulic

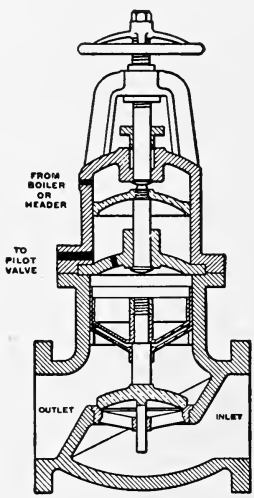


Fig. 511. Anderson Triple-duty Emergency Valve.

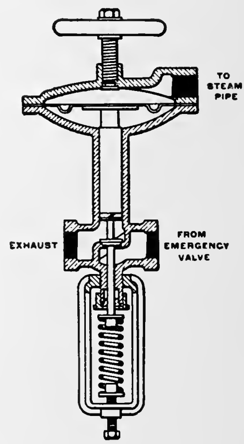
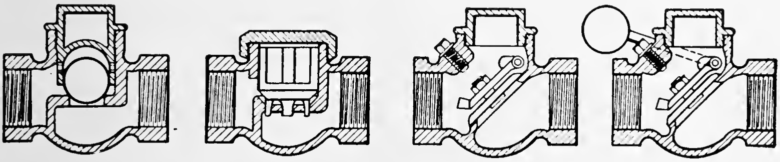


Fig. 512. Pilot Valve for Anderson Triple-duty Emergency Valve.

causes the valve to close. Pilot valves may be located at any desirable places, thus affording control from different points.

In the "Locke automatic engine stop system" the stop valve is operated by an electric motor which is controlled by contact points operated by a speed-limit device. (See Power, August, 1907, p. 471, for a detailed description.)

360. Check Valves. — Fig. 513, *A* to *D*, illustrates the different types of check valves in most common use. *A* is a *ball check*, *B* a *cup* or *disk check*, *C* a *swing check*, and *D* a *weighted check*. Occasionally the valve body is fitted with a valve stem and handle for holding the disk against its seat, in which it is designated as a *stop check*. In *A* and *B* the valve seat is parallel to the direction of flow and the valve is held in place by its own weight and by the pressure of the fluid in case of reverse flow. In the swing check the seat is at an angle of about 45 degrees to the direction of flow. The latter construction is preferred as it offers less resistance to flow and there is less tendency for impurities to lodge on the valve seat. By extending the hinge of the swing through the body of the valve, a lever and weight may be attached as in *D* and the check will not open except at a pressure corresponding to the resistance of the weight. It thus acts as a relief valve and at the same time prevents a reversal of flow. *Stop checks* are usually inserted in boiler feed lines close to the boiler, and, when locked, act as any ordinary stop



(A) (B) (C) (D)

Fig. 513. Types of Check Valves.

valve and permit the piping to be dismantled or the regulating valve to be reground without lowering the pressure on the boiler. Since the wear on check valves is excessive and necessitates frequent regrinding they are often mounted with *regrinding disks*, Fig. 513 (*C*), which may be "ground" against the seat without removing the valve from the line.

361. Blow-off Cocks and Valves. — The requirements of a good blow-off valve are that it shall furnish a free passage for scale and sediment, that it shall close tightly so as not to leak, and that it shall open easily without sticking or cutting. On account of the rather severe service to which such valves are subjected, they are made very heavy, with renewable wearing parts.

Fig. 514 gives a sectional view of a Crane ferrosteel valve. The bonnet is easily taken off and the disk removed to be refaced or replaced by a new one. The old disk is repaired by pouring in a hard Babbitt metal and facing it off flush. The seats are of brass and oval on top to prevent scale lodging between them and the disk, and are so made that they may be removed; but it has been found in practice that there is not much cutting of the seat, the damage usually being confined to the softer Babbitt metal which faces the disk.

Fig. 515 gives a sectional view of a Faber valve. When the disk, which makes a snug fit in the body of the valve, is in the position shown, the boiler discharge is practically shut off and any sediment lying on the seat is cleaned off by a jet of steam or water.

Fig. 516 shows a section through a typical *blow-off cock* of the straightway taper plug pattern with self-locking cam. Plug cocks are often used instead of valves on the blow-off piping.

Current practice recommends the use of two valves, or rather one valve and one cock, in the blow-off line of each boiler. In most of the

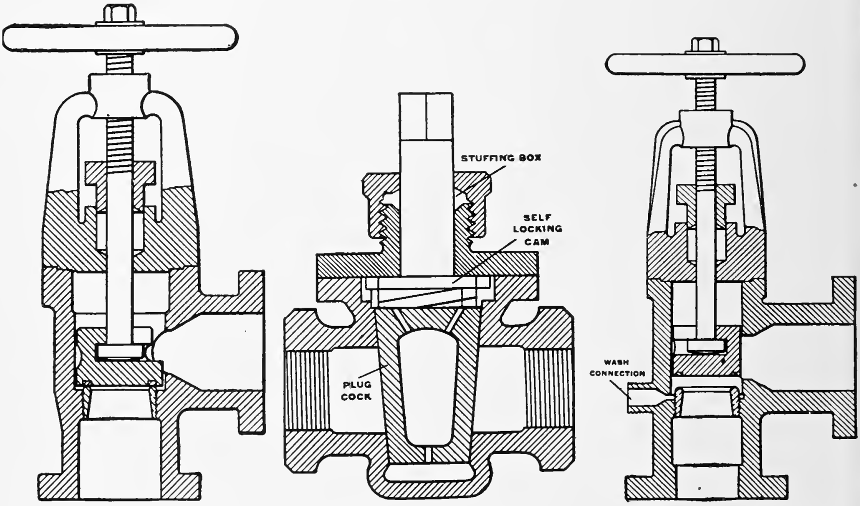


Fig. 514. Crane Ferro-steel Blow-off Valve.

Fig. 515. A Typical Blow-off Cock.

Fig. 516. Faber Blow-off Valve.

large stations a blow-off valve and a blow-off cock are installed as indicated in Fig. 517. The number and size of blow-off cocks are usually specified by city or state legislation. (For a description of various types of blow-off valves, see Power, Dec. 20, 1910, p. 2228.)

362. Safety Valves. — Fig. 518 shows a section through the simplest form of safety valve. The valve is held on its seat against the boiler pressure by a cast-iron weight as indicated. This type has the advantage of great simplicity, and can be least affected by tampering, since it requires so much weight that any additional amount which would seriously overload it can be quickly detected. For high pressure and large sizes of boiler this class of valve is entirely too cumbersome.

Fig. 519 shows the general details of the common *lever safety valve.* The valve is held against its seat by a loaded lever, thereby enabling the use of a much smaller weight than the "dead-weight" type, since the resistance is multiplied by the ratio of the long arm of the lever to the short one. The proper position of the weight is determined by simple proportion. Safety valves of the "dead-weight" or "lever" type are little used in modern practice, and their use is prohibited in U. S. marine service and in many states.

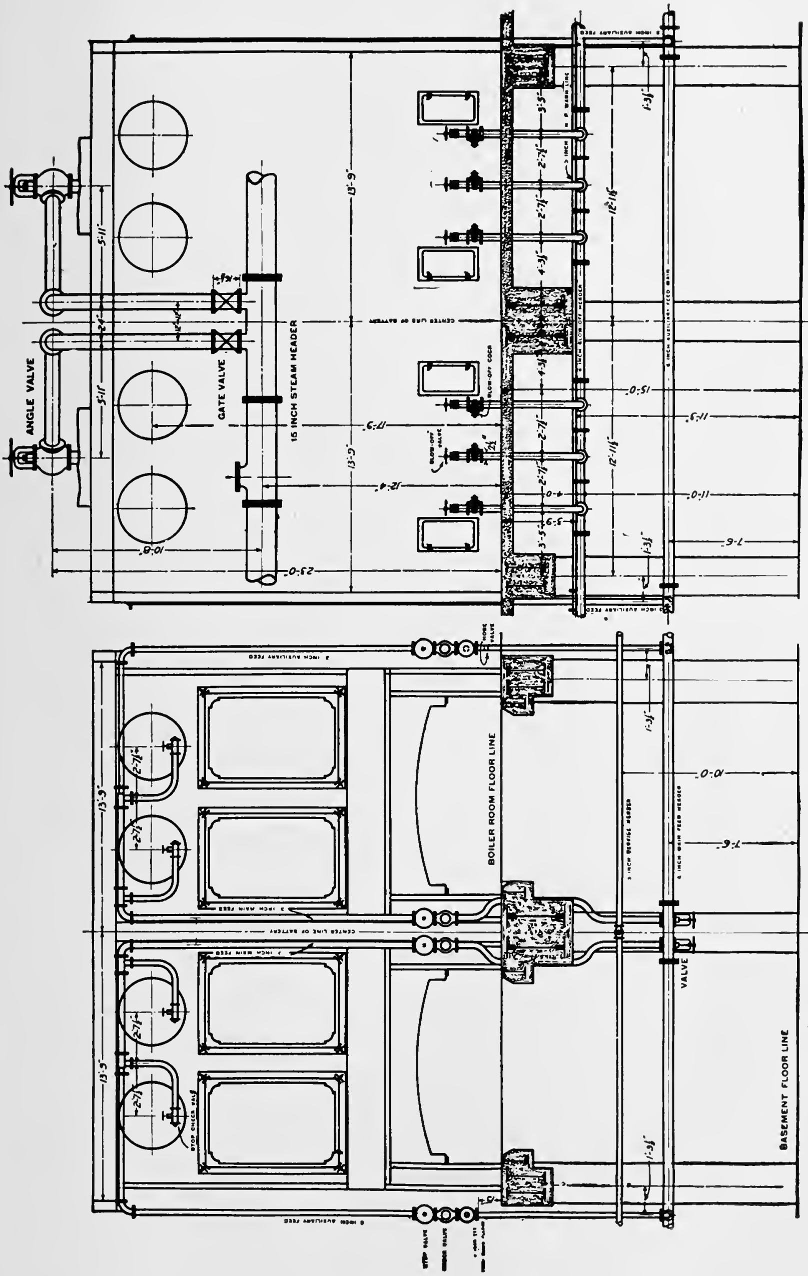


Fig. 517. Feed Water and Blow-off Piping, South Side Elevated Ry. Station, Chicago, Ill.

Fig. 520 shows a section through a typical *pop safety valve* in which the boiler pressure is resisted by a spring. This type of valve has practically supplanted all other forms. The boiler pressure acting upon the under side of valve V is resisted by the tension in spring S. As soon as the boiler pressure exceeds the resistance of the spring the valve lifts from its seat and the steam escapes through opening O. The static pressure of the steam plus the force of its reaction in being deflected from the surface A holds the valve open until the pressure in the boiler drops about 5 pounds below that at which the valve is lifted. The additional area of valve exposed to pressure when the valve lifts causes it to open with a sudden motion which has given it its name, and it also closes suddenly when the pressure has fallen. These valves are arranged so that the spring tension may be varied without taking them apart, and provision is made for lifting the seats by means of a lever. The seats are of solid nickel in the best designs, to minimize corrosion.

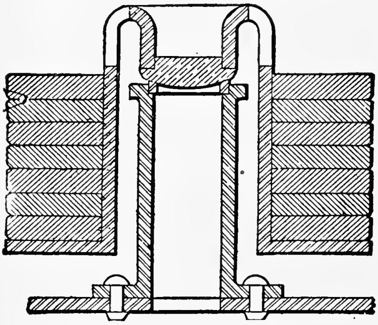


Fig. 518. "Dead-weight" Safety Valve.

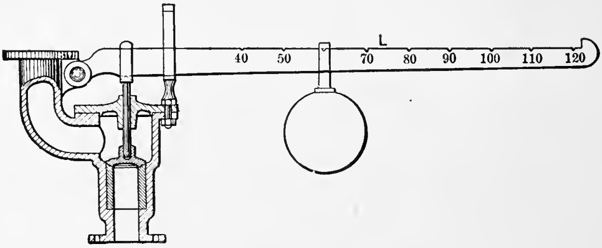


Fig. 519. Common Lever Safety Valve.

The commercial rating of a safety valve is based upon the area exposed to pressure when the valve is closed.

The number and size of safety valves for a given boiler are ordinarily specified by city or state legislation.

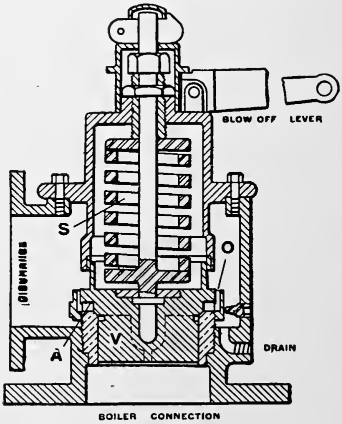


Fig. 520. Consolidated Pop Safety Valve.

The logical method for determining the size of safety valves is to make the actual opening at discharge sufficient to take care of all steam generated at maximum load without allowing the pressure to rise more than six per cent above the maximum allowable working pressure, thus:

Let W = maximum weight of steam discharged, pounds per hour,
A = effective discharge area, square inches,
P = boiler pressure, pounds per square inch absolute,
L = lift of valve, inches,
K = coefficient determined by experiment,
D = diameter of valve, inches.

According to Napier's rule for the discharge of steam through unrestricted orifices

$$W = \frac{3600}{70} PA = 51.4\,PA. \tag{291}$$

Allowing for restriction of orifice

$$W = 51.4\,KPA. \tag{292}$$

In the A.S.M.E. "boiler code" the value of K is taken as 0.96. Substituting this value K in equation (292),

$$W = 49.3\,PA. \tag{293}$$

For a flat-seated valve

$$A = \pi DL,$$

whence

$$W = 155\,PDL \tag{294}$$

and

$$D = 0.00645\,\frac{W}{PL}. \tag{295}$$

For the almost universal 45-degree seated valve

$$A = \pi DL \text{ sine 45 degrees}$$
$$= 0.707\,DL,$$

whence

$$W = 109.7\,PDL \tag{296}$$

and

$$D = 0.00911\,\frac{w}{PL}. \tag{297}$$

The present rule of the United States Board of Supervising Inspectors is

$$a = 0.2074\,\frac{w}{P}, \tag{298}$$

in which

a = area of the safety valve in square inches per square foot of grate surface per hour,
w = pounds of water evaporated per square foot of grate surface per hour.

Example 73. A boiler at the time of maximum forcing uses 2150 lb. of Illinois coal per hour; heat value 12,100 B.t.u. per lb.; boiler pressure 225 lb. per sq. in. gauge; feed water 200 deg. fahr. Required the size of safety valve.

Assuming a boiler efficiency of 75 per cent the total maximum evaporation is

$$W = \frac{2150 \times 12{,}100 \times 0.75}{1033} = 18{,}880 \text{ lb. per hour.}$$

(1033 = heat content of 1 lb. of steam at 225 lb. gauge above 200 deg. fahr.)

Assuming a lift of 0.1 in., we have, from equation (297),

$$D = 0.00911 \frac{18{,}880}{240 \times 0.1} = 7.17 \text{ in.}$$

According to the A.S.M.E. code two valves would be required. Considering two valves of the same size, the diameter of each for the given condition would be $\frac{7.17}{2} = 3.5$ (approx.).

The following rules pertaining to safety valves are taken from the A.S.M.E. Boiler Code:

Each boiler shall have two or more safety valves, except a boiler for which one safety valve 3-in. size or smaller is required.

One or more safety valves on every boiler shall be set at or below the maximum allowable working pressure. The remaining valves may be set within a range of three per cent above the maximum allowable working pressure, but the range of setting of all of the valves on a boiler shall not exceed ten per cent of the highest pressure to which any valve is set.

Each valve shall have full sized direct connection to the boiler. No valve of any description shall be placed between the safety valve and the boiler, nor on the discharge pipe between the safety valve and the atmosphere.

The complete A.S.M.E. Boiler Code may be purchased from the American Society of Mechanical Engineers, New York City.

363. Back-pressure and Atmospheric Relief Valves. — These valves are for the purpose of preventing excessive back pressure in exhaust pipes. In non-condensing plants such valves are designated as *back-pressure valves* and in condensing plants as *atmospheric relief valves.* In the former the valve is usually adjusted so that a pressure of one to five pounds above the atmosphere is necessary to lift it from its seat; in the latter the valve lifts at about atmospheric pressure. They are practically identical in construction, differing only in minor details. A slight leakage in the back-pressure valve is of small consequence, but in an atmospheric relief valve it may seriously affect the degree of vacuum and throw unnecessary work upon the air pump, hence it is customary to "water-seal" the latter. Fig. 521 shows a section through a typical back-pressure valve. The valve proper consists of a single disk moving vertically. The valve stem is in the form of a

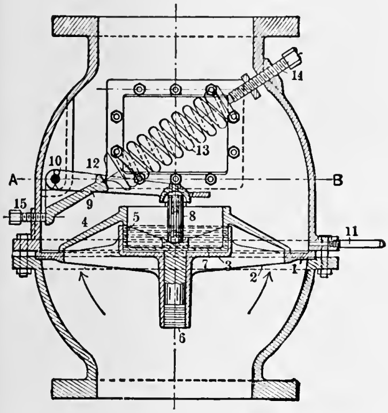


Fig. 521. Foster Back-pressure Valve.

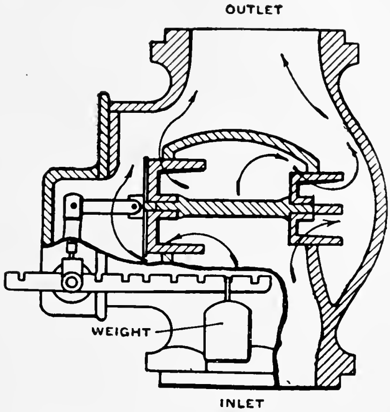


Fig. 522. Davis Back-pressure Valve.

piston or dashpot which prevents sudden closing or hammering. The pressure holding the valve against its seat is regulated by a spring. When the back pressure becomes greater than atmospheric plus that added by the spring, the valve raises from its seat and relieves it.

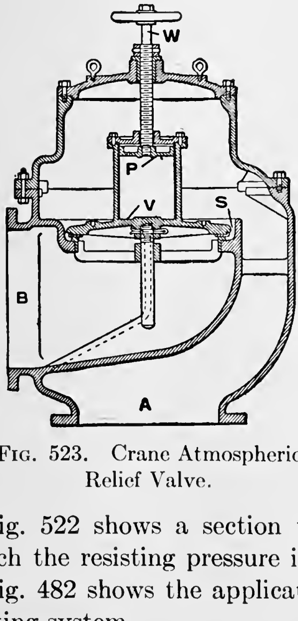


Fig. 523. Crane Atmospheric Relief Valve.

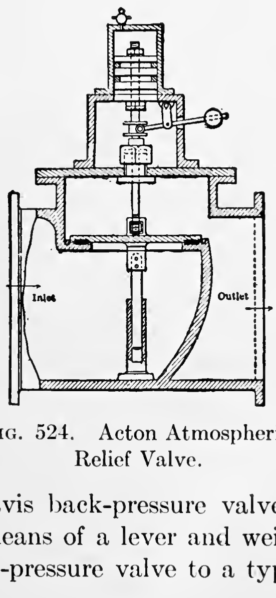


Fig. 524. Acton Atmospheric Relief Valve.

Fig. 522 shows a section through a Davis back-pressure valve, in which the resisting pressure is varied by means of a lever and weight.

Fig. 482 shows the application of a back-pressure valve to a typical heating system.

Fig. 523 shows a section through a typical *atmospheric relief valve.* Opening B is connected to the exhaust pipe and opening A leads to the atmosphere. Under normal conditions of operation atmospheric pressure holds valve V against its seat. Water in groove S "water-seals" the seat and prevents air from being drawn into the condenser. In case the pressure in pipe B becomes greater than atmospheric it lifts valve V from its seat and is relieved. Piston P acts as a dashpot and prevents the valve from slamming.

Fig. 524 shows a section through an atmospheric relief valve in which the weight of the valve is counterbalanced or even overbalanced by an adjustable weight and lever, thereby permitting the valve to open at or below atmospheric pressure, as may be desired.

364. Reducing Valves. — It is often necessary to provide steam at different pressures in the same plant, as in the case of a combined

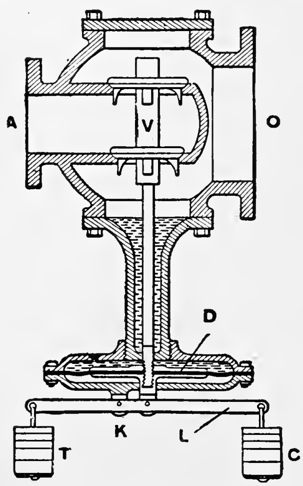


Fig. 525. Kieley Reducing Valve.

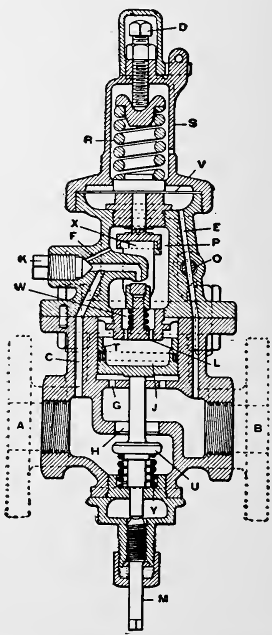


Fig. 526. Foster Pressure Regulator.

power and heating plant. To effect this result the reduction in pressure is accomplished by passing the steam through a *reducing valve,* which is but an automatically operated throttle valve. There are many different forms, the operation of all being based upon the same general principles.

In the Kieley valve, Fig. 525, the low-pressure steam acts upon the top of flexible diaphragm *D*, and the weighted lever *L* (which may be adjusted to give the desired reduction in pressure) acts upon the other side. The movement of the diaphragm causes the balanced valve *V* at the upper end of the spindle to open or close, as may be necessary to maintain the desired lower pressure. Inertia weights *T* and *C* prevent chattering.

Fig. 526 shows a section through a class G Foster *pressure regulator* or reducing valve. In operation, steam enters at *A* and passes through the main valve port *H* to the outlet *B*. Steam at initial pressure passes through port *C* to chamber *P* and thence to the top of piston *T* through port *L*, opening the main valve *U*. Steam at delivery pressure passes through *E* and raises the diaphragm *V* against the pressure of spring *R*, allowing spring *W* to close the auxiliary valve *X*. The pressure in chamber *J* is then equalized by the reduced pressure in ports *G* and the under side of piston *X*, and thus allows spring *Y* to close the main valve which is then held to its seat by the initial pressure. Any reduction in delivery pressure is transmitted to diaphragm *V*, and permits spring to open auxiliary valve *X*, thereby admitting steam to the top of piston *T*, as previously explained. The delivery pressure is adjusted by screw *D*; thus increasing the tension of spring *R* increases the discharge pressure; and *vice versa.* The adjustment once made, the delivery pressure will remain constant, regardless of any variable volume of discharge or of the initial pressure, so long as the latter is in excess of the delivery pressure. *W*, Fig. 494, shows the application of a reducing valve to an exhaust steam heating system. Live steam is led to the valve through pipe *A*. It will be noted that the pipe leading from the valve to the heating system is much larger than the high-pressure supply pipe on account of the increase in volume of the low-pressure steam. Reducing valves should always be by-passed to permit of repairs without shutting down the system. Care should be taken in not selecting too large a reducing valve, as the valve lift is very small and the larger the valve the less will be the lift for a given weight of flow and consequently the greater the wire drawing and erosion of the valve seat.

365. Foot Valves. — Whenever a long column of water is to be moved in either suction or delivery pipe it is customary to place a check valve near the lower end of the column to prevent the water from backing up when the pump reverses or shuts down. The check valve placed at the end of the suction pipe is called a *foot valve.* Any check valve may be used as a foot valve, though practice limits the choice to the disk or flap type as illustrated in Fig. 527. To prevent rubbish from destroying the action, a strainer or screen is generally incorporated with the body of the valve. *A*, Fig. 527, illustrates a *single-flap*, *B* a *multi-flap* and *C* a *disk* valve composed of a nest of small rubber valves. The single-flap are usually made in sizes $\frac{3}{4}$ to 6 inches, the multi-flap 7 to 16 inches, and the disk valve in all commercial sizes from $\frac{3}{4}$ to 36 inches.

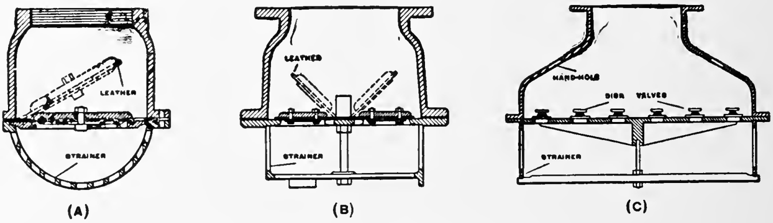


FIG. 527. Types of Foot Valves.

For large sizes, 16 to 36 inches, the multi-disk valve is given preference, since a number of the disks may be disabled without destroying its operation.

Blowoff Valves and Systems: Prac. Engr., July 1, 1916, p. 565.
Steam Stop Valves — A Survey of the Field of Design: Sibley Jour., Apr.–May, 1915.
Nonreturn Stop Valves: Power, Jan. 18–25, 1916, p. 72, 104.
The Use and Abuse of Globe Valves: Power and Engr., Jan., 1909, p. 10.
Gate Valves in Steam Pipe Lines: Power and Engr., Feb. 16, 1909, p. 320.
Types of Check Valves and Their Operation: Power and Engr., July 6, 1909, p. 11.

PROBLEMS.

1. Steam at 200 lb. abs. pressure is conducted through a bare pipe 3 in. nominal diameter, 500 ft. long. If the temperature of the room is 80 deg. fahr. calculate the total heat loss per hour.

2. If the pipe is covered with a single thickness of "Sall-Mo Air Cell" determine the saving in heat.

3. Determine the conductivity of the covering in Problem 2, per inch of thickness.

4. Determine the size of steam pipe suitable for a 10,000-kw. steam turbine using 14 lb. steam per kw-hr., initial pressure 215 lb. abs., back pressure 2 in. mercury, superheat 125 deg. fahr., if the pipe is 150 feet long and the pressure drop is not to exceed 2.0 lb. per sq. in. per 100 ft.

5. Saturated steam at 125 lb. abs. initial pressure is flowing at the rate of 20,000 lb. per hr. through a standard 6-in. pipe, 2000 ft. long. Calculate the probable pressure drop.

6. Determine the initial pressure necessary to deliver 400 gallons of water per minute through a 5-in. standard pipe 1500 ft. long, fitted with two right angle elbows and one globe valve. The water is to be discharged into an open tank.

7. How many gallons of water will be discharged through a straight length of 6-in. standard pipe 10,000 ft. long if the initial pressure is 100 lb. per sq. in., and what will be the pressure at the discharge end?

8. Determine the number and size of safety valves for a 500-hp. boiler designed to operate at a maximum load of 300 per cent above rating; boiler pressure 250 lb. abs.

CHAPTER XVI

LUBRICANTS AND LUBRICATION

366. General. — The losses due to the friction of the working part of machinery include considerably more than the mere loss of power, namely, the depreciation resulting from wear of bearings, guides, and other rubbing surfaces, and the expense arising from accidents traceable to excessive friction. The power absorbed in overcoming friction varies with the type of plant and the character of machinery and is seldom less than 5 per cent and often greater than 30 per cent of the total power developed. In large central stations these losses approximate 8 per cent and in weaving and spinning mills will average as high as 25 per cent. (Trans. A.S.M.E., 6–465.) These figures refer to properly lubricated plants operating under normal conditions. The proper selection of lubricant is therefore a very important problem, since, besides the cost of the lubricant itself, the loss in power and in wear and tear to machinery is no small item. A change of lubricant may frequently result in marked increase in economy of operation. Table 130 gives an idea of the saving effected in power by the proper selection of lubricants in a number of mills. (Power, May 12, 1908, p. 752.) The net financial gain depends, of course, upon the cost of the oil. As a general rule a 10 per cent reduction in friction horsepower will more than equal the cost of lubricants for one year. The lubricants most commonly met with in power plant practice are conveniently classified as oils, greases, and solids, and are of animal, mineral, or vegetable origin.

Reference books: Archbutt and Deeley, Lubrication and Lubricants; Redwood Lubricants; W. M. Davis, Friction and Lubrication; Gill, Oil Analysis; Robinson, Gas and Petroleum Engines; Thurston, Friction and Lost Work; Gill, Engine Room Chemistry.

367. Vegetable Oils. — Except for certain special purposes and for compounding with mineral oils these possess lubricating properties of little practical value, since they decompose at comparatively low temperatures and have a tendency to become thick and gummy. The vegetable oils sometimes employed are linseed, cottonseed, rape, and castor.

368. Animal Fats. — Many animal fats have greater lubricating power than pure mineral oils of corresponding viscosity but are objectionable on account of their unstable chemical composition. They decompose easily, especially in the presence of heat, and set free acids which attack metals. They are seldom used in the pure state and are usually compounded with mineral oils. The animal products used in this connection are tallow, neat's-foot oil, lard, sperm, wool grease, and fish oil, the first named being the most important. In cylinder lubrication, especially in the presence of moisture, the addition of 2 to 5 per cent of acidless tallow seems to make the oil adhere better to the metal surfaces and increases the lubricating effect, while the proportion is so small that ill effects from corrosion or gumming are scarcely perceptible. *Animal and Vegetable Oils*, Power, Nov. 3, 1914, p. 636.

TABLE 130.

EXAMPLES OF REDUCTION IN FRICTION DUE TO PROPER SELECTION OF LUBRICANTS.

No. of Test.	Country.	Plant.	Mill Oils. Test I.		New Oils. Test II.		Per Cent of Transmission to Full Load.		Power Reductions.	
			Full Load, I.H.P.	Transmission, I.H.P.	Full Load, I.H.P.	Transmission, I.H.P.	Test I.	Test II.	Full Load, Per Cent.	Transmission, Per Cent.
1	America......	Cotton........	543.21	192.70	481.75	168.90	35.4	35.0	11.31	12.35
2 A	America......	Worsted.......	611.60	....	596.30	....	......	......	2.50	....
B	America......	Worsted.......	702.90	....	648.70	....	......	......	7.80	....
3	America......	Cotton........	786.00	....	758.00	....	......	......	3.56	....
4 A	England......	Cotton........	1408.60	356.00	1301.80	319.30	25.3	24.5	7.60	10.30
B	England......	Cotton........	1428.40	357.90	1358.70	348.90	25.0	25.7	4.90	2.50*
5	England......	Worsted.......	348.10	111.10	327.50	99.50	31.9	30.4	5.90	10.40
6	England......	Weaving.......	495.00	146.60	453.60	127.50	29.6	28.1	8.40	13.00
7	Ireland.......	Linen..........	110.70	49.90	93.10	38.60	45.0	41.4	15.90	22.70
8 A	Scotland.....	Woolen........	177.70	61.80	164.60	56.10	34.7	34.0	7.40	9.20
B	Scotland.....	Woolen........	325.10	161.40	293.50	147.30	49.6	50.2	9.70	8.70
9	Germany.....	Cotton........	263.41	114.03	239.35	97.11	43.2	40.5	9.10	14.80
10 A	Germany.....	Worsted.......	341.36	118.24	290.53	95.67	31.7	32.9	14.90	19.10
B	Germany.....	Worsted.......	341.36	141.29	299.30	119.28	41.3	39.8	12.30	15.57‡
11	Germany.....	Jute...........	1135.20	362.60	1034.20	328.10	31.9	31.7	8.89	9.51
12	Russia.......	Cotton........	1238.80	....	1069.10	....	......	......	13.70	....
13	India.........	Cotton........	642.60	230.70	596.80	202.20	35.9	33.9	7.10	12.40
14	Japan.........	Cotton........	346.60	....	313.60	....	......	......	9.50	....
15	India.........	Flour..........	364.70	....	336.80	....	......	......	7.70	....
16	England......	Paper.........	465.40	....	390.40	....	......	......	16.20	....
17	Germany.....	Paper.........	511.37	....	482.43	....	......	......	5.60	....
18	England......	Brass shop.....	6.74x	1.77x	5.12x	1.53x	26.2	29.8	24.00	13.80
19	England......	Iron shop......	137.80	74.90	116.00	68.10	54.3	58.7	15.80	9.10
20	England......	Wood shop....	84.00	31.60	65.30	25.40	37.6	38.8	22.30	19.60

* Same oil after nine months' use. † Not full load of mill. ‡ Morning load.
x = Electrical units.

369. Mineral Oils. — These are all products of crude petroleum and form by far the greater part of all lubricants. They present a wider range of lubricating properties than those derived from animal or vegetable sources, the thinnest being more fluid than sperm and the thickest more viscous than fats and tallows. They are not easily oxidized, do not decompose, become rancid, or contain acids.

Mineral lubrication oils may be classified as

(1) *Distilled oils,* which are produced by distillation from crude petroleum and made pale, amber colored, and transparent by treatment with acid and alkali.

(2) *Natural oils,* which are prepared from crude petroleum, from which grit, suspended and tarry impurities have been removed. They are dark and opaque and are rich in lubricating properties.

(3) *Reduced oils,* or heavy natural oils, from which the lighter hydrocarbons have been evaporated and from which the tarry residue has been removed by filtration.

369a. Solid Lubricants. — Dry graphite, soapstone, and mica are sometimes used as lubricants, though they are usually mixed with grease or oils. They cannot easily be squeezed or scraped from between the surfaces, and are consequently suitable where very great weights have to be carried on small areas and when the speed of rubbing is not high. The coefficient of friction of such lubricants is high, and when economy of power is essential better results may be secured by the use of liberally proportioned rubbing surfaces and liquid lubricants. Under certain conditions of pressure and speed these lubricants will sustain, without injury to the surfaces, pressures under which no liquid would work.

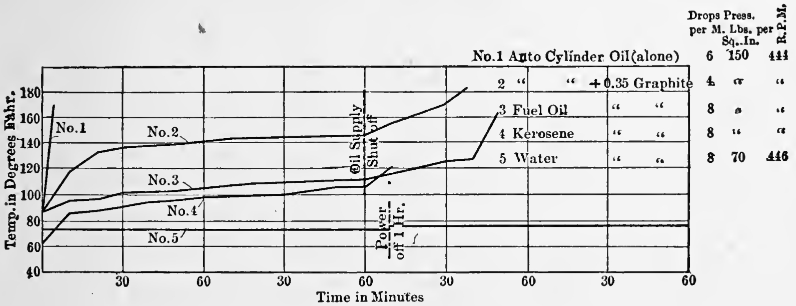


FIG. 528. Tests of Graphite Mixed with Various Lubricants.

Deflocculated graphite suspended in oil or water, and designated commercially as "oildag" and "aquadag" respectively, is finding favor with many engineers. Graphite in this deflocculated condition remains suspended indefinitely in water and oil, readily adheres to the journal, has great wearing properties, and is easily applied to the wearing surfaces. From numerous and long-continued trials it appears that 0.35 per cent serves adequately for all purposes. Temperature curves of deflocculated graphite in combination with various carrying fluids are given in Fig. 528. For further data pertaining to the curves in Fig. 528 and for an extensive discussion on the subject of lubrication consult *Lubrication and Lubricating,* by C. F. Maberg, Jour. A.S.M.E., Feb. and May, 1910.

370. Greases. — Under this name may be included the various compounds which consist of oils and fats thickened with sufficient soap to form, at ordinary temperatures, a more or less solid grease. Those usually employed are lime, soda, or lead soaps, made with various fats and oils. "Engine" greases are thickened with a soap made from tallow or lard oil and caustic soda, and often contain neat's-foot oil, beeswax, and the like. For exceptionally heavy pressures, graphite, soapstone, and mica are sometimes added to the grease. Table 131 gives an idea of the characteristics of a number of greases. (Prac. Engineer, U. S., Apr. 1911, p. 293.) The friction tests were made on a small Thurston oil testing machine, 320 r.p.m. and bearing pressure of 240 pounds per square inch of projected area. These results are purely comparative under the given conditions of rubbing surfaces, speed and pressure. For results of these greases tested on a large Olsen oil machine consult reference given above.

Commercial Lubricating Greases: Prac. Engineer, U. S., Apr., 1911, p. 293; *Tests of Grease Lubrication,* Ibid., p. 295; Am. Mach., Aug. 24, 1911, p. 356; Power, Nov. 8, 1910, p. 1998.

TABLE 131.

LUBRICATING CHARACTERISTICS OF A NUMBER OF GREASES.

Type.	Class.	Melting Point, Deg. F.	Per Cent Soap.	Kind of Soap.	Per Cent Free Acid as Oleic.	Average Coefficient Friction.
A Mineral	Summer	167	38	Lime	Trace	0.075
B Mineral	Summer	178	20	Lime	0.3	0.054
C Mineral	Winter	165	23	Lime	6.1	0.063
D Mineral	Winter	163	16	Lime	0	0.057
E Mineral	Winter	142	19	Lime	Trace	0.046
F Tallow No. 3	Winter	125	1.4	Potash	0	0.022
G Tallow No. XX	Summer	120	2.1	Potash	0	0.029
H Lard oil	....	41	0	....	....	0.011

Type.	Final Coefficient Friction After 3-Hr. Run.	Maximum Temperature of Bearing Above that of Room, Degs. F.	Final Temperature of Bearing Above that of Room at End of 3-Hr. Run, Degs. F.
A Mineral	0.075	70	68
B Mineral	0.050	70	58
C Mineral	0.063	76	65
D Mineral	0.054	69	58
E Mineral	0.046	58	50
F Tallow No. 3	0.012	38	18
G Tallow No. XX	0.018	45	32
H Lard oil	0.010	13	12

371. Qualifications of Good Lubricants. — A good lubricant should possess the following qualities:

(1) Sufficient "body" to prevent the surfaces from coming into contact under conditions of maximum pressure.
(2) Capacity for absorbing and carrying away heat.
(3) Low coefficient of friction.
(4) Maximum fluidity consistent with the "body" required.
(5) Freedom from any tendency to oxidize or gum.
(6) A high "flash point" or temperature of vaporization and a low congealing or "freezing point."
(7) Freedom from corrosive acids of either metallic or animal origin.

372. Testing Lubricating Oils. — There is no question but that the lubricant best suited for a given set of conditions can only be determined by an actual practical test under service conditions. Each plant is an individual problem since certain grades and qualities of oil which work perfectly in some cases have proved entirely unsatisfactory in others where the conditions appeared to be exactly the same. Nevertheless, in order to avoid needless experiment and to limit the number of acceptable lubricants to a minimum it is desirable to know certain characteristics which will indicate whether or not the particular lubricant under consideration is unfitted for the desired service. The small consumer must depend upon the reputation of the concern from which he is buying for reliable data pertaining to the qualifications of their products, since the cost of conducting a series of preliminary or identification tests is out of all proportion to the actual cost of the lubricant. The large consumer on the other hand may find it to be worth while to conduct an elaborate series of tests before drawing up contracts for the oil supply.

The complete test of an oil consists of three parts: Chemical, physical, and practical.

373. Chemical Tests of Lubricating Oils. — To pass the chemical tests of the Navy Department * "all oils should be neutral in reaction and should not show the presence of moisture, matter insoluble in petroleum ether (hard asphalt), matter insoluble in ether alcohol (soft asphalt), free sulphur, charring or wax-like constituents, naphthenic acids, sulphonated oils, soap, resin or tarry constituents, the presence of which indicates adulteration or lack of proper refining. Except in oil for engines without forced lubrication, no traces of fixed oils (animal or vegetable fats) should be found.

* *Lubricating Oils.* Lieut. J. L. Kauffman, U.S.N., Jour. Am. Soc. Naval Engrs., Aug., 1916, p. 692.

"In lubricating oil for main engines without forced lubrication, approved fixed oils, such as rapeseed, olive, tallow, lard and neat's-foot oil, may be used. When the foregoing fixed oils are used, they must be well refined with alkalies, unadulterated, containing a minimum of free fatty acids, with no moisture or gumming constituents. Olive oil should not have a high specific gravity. If satisfactory emulsifying results can be obtained with straight mineral oils on engines without forced lubrication, they may be submitted for service test."

The most satisfactory procedure is to have the various tests made by a competent chemist but since a number of plants are provided with the necessary equipment the tests stipulated by the Navy Department, and which are representative of current commercial practice, will be described in a general way.

Moisture. — Heat 3 to 4 cc. in a test tube (the walls of which have been thoroughly wet with oil) in a bath of liquid paraffin up to 300 deg. fahr. Oils containing water will form emulsions on the walls and cause foaming and spluttering. A test is also made with a mixture of oil and eosin to determine faint traces of moisture by changes of color. The presence of moisture is particularly undesirable in transformer oils, but there is danger of its forming objectionable emulsions in any straight mineral oil.

Sulphur. — Boil about 50 cc. of oil with a piece of bright metallic sodium for half an hour; add water, heat and stir until the sodium is dissolved; pour off the water and test the remainder with a fresh 1 per cent solution of sodium nitroprusside. If the mixture turns violet color, the oil contains sulphur. When sulphur is found, an additional test for sulphonated oils is made.

Acids or Alkalies. — Heat for one-half hour with frequent stirring 25 cc. of oil and 50 cc. of neutral distilled water. Test a few cubic centimeters of the mixture first with methyl-orange to determine the acids, and another portion with phenolphthalein for the determination of alkalies. Acids and alkalies cause emulsions. Acids also cause corrosion of journals and other metal parts.

Matter Insoluble in Ether Alcohol. — Shake 11 cc. of oil and 14 cc. of ether alcohol (8 parts ether and 6 parts alcohol). After standing 12 hours, note the precipitate, if any, at the bottom of cylinder. The precipitate will be asphalt, and even a trace would make the oil undesirable as a lubricant. Asphalt would cause scoring of journals and clogging of oil lines.

Matter Insoluble in High-grade Gasoline. — Shake 2 cc. of oil and about 300 cc. of high-grade gasoline (86–88 Baumé gravity). After standing 12 hours, note precipitate, if any, in the bottom of glass.

The precipitate will be soft asphalt or carbon particles, and a slight trace would make the oil undesirable.

Tarry or Suspended Matter. — Same as the foregoing, except using 5 cc. of oil and 95 cc. of gasoline and allowing it to stand for half an hour; then examine deposit, if any, for dirt or tarry matter.

To Detect Fixed Oils. — Heat 10 cc. of oil with a small piece of metallic sodium. If the mixture becomes gelatinized or a semisolid, it indicates the presence of fixed oils. If an equal volume of oil is heated alone to the same temperature, the viscosity of the two samples can be compared; if the oil contains fixed oils (animal or vegetable oils), the sample with sodium will be much heavier than the sample heated alone.

Effect of Heat. — Heat 5 cc. of oil in test tube over flame until vapors are evolved and compare the color of the heated oil with that of unheated oils. If the heated oil turns black, it shows the presence of undesirable carbon or hydrocarbons.

*Gumming Test.** — This is particularly applicable to petroleum oils and is used to indicate the extent to which the oil has been refined. It serves indirectly to indicate the extent to which the oil may be expected to change due to oxidation when in use. Numerous opportunities have been offered to check the results obtained with this test and results obtained in practice with the same oils, and all of this experience tends to show the great value of the gumming test.

This test is made by putting a small quantity of the oil to be tested in a small glass vessel, such as a cordial glass, and then mixing with it an equal quantity of nitrosulphuric acid. A properly refined oil will show little, if any, change, but a poorly refined oil will be indicated by the separation of large quantities of material of dark color. This color is due to the oxidation of the tarry matter contained in the lubricant. Experience has shown that oils containing large percentages of tar absorb the most oxygen, that is, they are mildly drying oils.

The results obtained by the gumming test agree well with carbon-residue tests made by distilling to dryness in a glass or a fused quartz flask. The carbon-residue test has been found of great assistance in choosing a satisfactory cylinder lubricant for gas engines, as a large amount of carbon means trouble in the engine cylinder. The lowest carbon content mentioned by the author was 0.11 per cent. The oil giving this test showed no tarry matter when tested with nitrosulphuric acid. In general, a gas-engine oil should not contain more than 0.5 per cent carbon as determined by the carbon-residue test.

374. Physical Tests of Lubricating Oils. — The physical characteristics usually involve (1) color; (2) odor; (3) specific gravity; (4) flash

* Prof. A. H. Gill.

point; (5) fire point; (6) cold point; (7) viscosity; (8) emulsion; (9) evaporation; and (10) friction. The following tests, unless otherwise indicated, refer specifically to the requirements of the Navy Department which, as previously stated, are representative of current commercial practice.

Color. — The color, although having no influence on the lubricating value, may be used to identify the sample. American oils fluoresce with a grass-green color. Russian oils have a blue sheen; oils containing distillation residues and unfiltered oils are brown to green-black in reflected light. Nearly all mineral machinery oils are distilled and filtered to some extent and are transparent in a test tube, the colors ranging from a yellowish white to a blood red. The color may be determined in a tinctometer by comparing with different-colored glasses or lenses. These glasses are numbered and for machinery oil extend from No. 1 (white) to No. 6 (red).

Odor. — The odor may be determined by heating in a test tube or by rubbing on the hand, by which means fatty oils, coal tar, rosin oils, etc., may be detected.

Specific Gravity. — The specific gravity is obtained by the use of the "pyknometer," this term signifying any vessel in which an accurately measured volume of liquid can be weighed. The bottle is first filled with distilled water at a temperature of 60 deg. fahr., and the weight of the water determined. The bottle is then filled with oil at a temperature of 60 deg. fahr. and the weight of the oil determined. The weight of the oil divided by the weight of the water gives the specific gravity at 60 deg. fahr. The Baumé gravity is obtained by using the Baumé hydrometer, which is simply an ordinary hydrometer with a certain arbitrary scale. Baumé gravity may be converted into specific gravity by the following formula:

$$\text{Sp. gr.} = \frac{140}{130 + \text{Baumé}}.$$

Baumé gravity is largely used in commercial practice.

The specific gravity does not affect the lubricating value of an oil, but indicates to the experienced oil man the locality from which the crude oil is obtained. For instance, the specific gravities of the lubricating oils tested at the Experiment Station vary from 0.864 to 0.945. A Baumé gravity of 32 corresponds to a specific gravity of 0.864, and a Baumé gravity of 18.1 to a specific gravity of 0.945, so that an increase in specific gravity is a decrease in Baumé gravity. The paraffin-base oils of Pennsylvania derivation have an average specific gravity of 0.875 with a corresponding Baumé gravity of 30. The asphaltic-base oils from Texas and California have an average specific gravity of 0.930 with a corresponding Baumé gravity of 20.

TABLE 132.

SPECIFIC GRAVITY AND GRAVITY BAUMÉ OF A NUMBER OF LUBRICANTS.

	Specific Gravity.	Gravity Baumé.	Flash Test, Degrees F.
Water	1.000	10	....
Cylinder oil	.9090	24.5	575
Cylinder oil	.8974	26	540
Heavy engine oil	.9032	25.5	411
Medium engine oil	.9090	24	382
Light engine oil	.8917	27	342
Castor machine oil	.8919	27	324
Lard oil	.9175	23	505
Sperm oil	.8815	29	478
Tallow oil	.9080	24.5	540
Cottonseed oil	.9210	22	518
Linseed oil	.9299	19	505
Castor oil (pure)	.9639	15	....
Palm oil	.9046	25	405
Rape-seed oil	.9155	23	....
Spindle oil	.8588	33	312

Flash Point. — The flash point is determined with both the Cleveland open cup and the Pensky-Martin closed cup. The flash point of all oils is determined as a measure of their volatility. The flash point of steam-cylinder oils is of primary importance, the required flash point depending on the temperature of the steam at the engine. With lubricating oils for bearings the flash point is important only in that it indicates the volatility of the oils and the presence of kerosene or naphtha fractions, with the accompanying fire risks. In the case of very low flash-point lubricating oils, it is desirable to run a special distillation or volatility test, mentioned under chemical tests. The flash point determined with the open cup is higher than with the closed cup, as the inflammable gases on the surface of the oil are disturbed by the air currents in the open cup. These differences range from 5 deg. to 40 deg. with the average at 20 deg. The presence of very light ends (kerosene, naphtha, etc.) may increase this difference to 100 deg.

Fire Point. — This is the temperature at which the oil burns and is determined by raising the temperature about 3 deg. a minute, applying the flame for about a second. The fire, or burning, point is from 30 deg. to 65 deg. higher than the flash point with all lubricating oils, the light oils having a difference of about 40 deg.

Cold Point. — Mineral oils become more viscous on cooling, and finally solidify. In lubricating oils refined from paraffin-base crudes, cooling first causes the paraffin particles to solidify, which gives the oil a cloudy appearance; with this class of oils this change is known as the cloud point.

The Committee on Lubricants of the American Society for Testing Materials uses the words "cold test" as a general term, with subheads of "cloud test" and "pour test." The method recommended by this committee is used at the Experiment Station, and in substance is as follows: Heat the oil to 150 deg. fahr. and cool by air to 75 deg. fahr. Take a bottle about $1\frac{1}{4}$ in. inside diameter and 4 to 5 in. high and pour in oil to a height of $1\frac{1}{4}$ in. from the bottom. Insert a cold-test thermometer (specially made, using colored alcohol, and with a long bulb) through a tight-fitting cork. A special jacket is used having an inside diameter about $\frac{1}{2}$ inch larger than the bottle. Ice or any other cooling medium is packed around this jacket. When the oil is near the expected cloud point, at every 2 deg. drop in temperature remove the bottle and inspect the oil, being careful not to disturb the oil. When the lower half becomes opaque, read the thermometer; this reading is taken as the cloud point. The cold, or pour, test is simply a continuation of the cloud test, except that the temperature is noted every 5 deg. and the bottle tilted till the oil flows. When the oil becomes solid and will not flow, the previous 5-deg. point is taken as the cold point of the oil.

Viscosity. — The viscosity of a lubricating oil is the most important factor to be determined. The viscosity of an oil is inversely proportional to its fluidity and is a measure of its internal friction or resistance to flow. Viscosity is sometimes called "body" and is determined by a viscosimeter. There are a number of different instruments for this purpose but no recognized standard instrument or method, so that "viscosity" conveys no meaning unless the name of the instrument, the temperature, and the amount of oil tested are given. Nearly all instruments are of the orifice type; that is, the viscosity of an oil is taken as the number of seconds required for a given amount to flow through an orifice at a given temperature. By "specific viscosity" is meant the ratio of the time required for the oil to run out to that of an equal quantity of water at 60 deg. fahr. The viscosity of engine oils is usually taken at 100 to 130 deg. fahr. and of cylinder oils at 210 deg. fahr. The absolute viscosity is determined from the amount flowing through capillary tubes, the results being given in C. G. S. units. The determination of the absolute viscosity is a very difficult operation requiring complex apparatus and a relatively long time. Several absolute viscosimeters have been invented; but to date none of them is considered practical enough for the routine testing of oil.

The accepted theory advanced by Ubhelohde * is that the absolute viscosity is directly proportional to the internal friction of the lubricant, and that the viscosity is a direct indication of the friction

* General Electric Review, November, 1915.

developed in a bearing. If Ubhelohde's conclusion is substantiated a very great advance will have been made and it will be possible to duplicate any friction results by duplicating the viscosity of the lubricant.

In general, the lower the viscosity the lower will be the friction, but since the rubbing surfaces should have as much lubricant between them as possible it is necessary to have sufficient viscosity to prevent them from "seizing." Under normal conditions of bearing lubrication the lightest oil that will prevent seizing should be used to obtain a minimum frictional loss.

Viscosity and its Relation to Lubricating Values: Power, Jan. 11, 1916, p. 37.

Emulsion Tests. — Emulsion tests are made on all straight mineral oils except cylinder oils. Four emulsion runs are made, using 40 cc. of oil in each case and (*a*) 40 cc. of distilled water; (*b*) 40 cc. of salt water; (*c*) 40 cc. of normal caustic-soda solution; (*d*) 40 cc. of boiling distilled water. The mixture is stirred with a paddle for five minutes at 1500 revolutions per minute and is kept at a temperature of 130 deg. fahr. during the stirring and while separating. On oils used with forced lubrication or on ice machines, the oil must completely separate from the mixture in less than 20 minutes. The emulsion is made with distilled and salt water, and a normal caustic-soda solution is also taken, as there is a possibility of water containing boiler compound getting into the system. Boiling distilled water is used in case gland steam or water runs into the oil system. These emulsion tests are considered of the greatest importance, as an oil on any type of forced lubrication system must not emulsify. If emulsions do occur, it will mean clogging of the oil lines, forming of residues in the base of the bearings, with a resultant loss of a large amount of oil.

Evaporation Tests. — It is advisable to include an evaporation test with the flash test of lubricants. The evaporation test is made by exposing about 0.2 gram of oil at a proper temperature and determining the loss by weight in a given time.

Friction Tests. — The coefficient of friction as determined from friction-testing machines is useful in obtaining a comparison of oils under the test conditions, but gives little information concerning the action of the oil under the widely different conditions found in actual practice.

Table 133 gives the physical properties of a number of lubricating oils, with their particular fields of application.

375. Service Tests. — These tests are the real proof of the commercial value of the lubricant for a given service. The lubricant is tested under actual operating conditions and that one selected which gives

TABLE 133.

PHYSICAL CHARACTERISTICS OF A NUMBER OF LUBRICANTS.

(*Power*, December, 1905, p. 750.)

Kind of Oil.	Use and Adaptation.	Gravity, Degrees.	Cold Test, Degrees.	Flash Test, Degrees.	Fire Test, Degrees.	Viscosity at 70 Degrees.
High-pressure cylinder oil.	For steam cylinders using dry steam at pressures from 110 to 210 pounds.	25 to 24.5	30	600 to 610	645 to 660	175 to 205
General cylinder oil . .	For steam cylinders using dry steam at 75 to 100 pounds. For air compressor cylinders when made from steam-refined mineral stock and when viscosity is 200.	26 to 25.5	30	550 to 585	600 to 630	180 to 190
Wet cylinder oil. (Remark 1.)	For use where the steam is moist, especially in compound and triple expansion engines.	25.8 to 25.3	30	560 to 585	600 to 630	150 to 185
Gas engine cylinder oil. (Remark 2.)	For gas engine cylinders. Neutral mineral oil compounded with an insoluble soap to give body.	26.5	30	320	350	300
Automobile gas engine oil. (Remark 3.)	For automobile gas engines and similar work.	29.5	30	430	485	195
Heavy engine and machinery oils.	For heavy slides and bearings, shafting, and horizontal surfaces.	30.5 to 29.5	30	400	440 to 450	170 to 195
General engine and machine oils.	For high-speed dynamos and machines.	30.8 to 30	30	400 to 420	450 to 470	175 to 190
Fine and light machine oils.	For fine work, from printing presses to sewing machines and typewriter oils. With a cold test of 25° to 28° and a viscosity of 140° this makes an excellent spindle oil.	32.5 to 30.2	30	400	440	110 to 160
Cutting and heat dissipating oils. (Remark 4.)	For cutting tools, screw cutting and similar work.	27 to 23	30	410 to 420	475 to 480	210 to 175
Refrigerating oils.	For ice machinery.	30.2	0	200	225	165
Wet service and marine oils. (Remark 4.)	For marine service, or where a great deal of moisture must be handled.	28	30	430	475	230
Greases.	They are used in special work and for heavy pressures moving at slow velocities.					

Remark 1. — May contain not over 2 to 6 per cent of refined acidless tallow oil in the high-pressure oils and not over 6 to 12 per cent in the low-pressure oils.

Remark 2. — The reason for using an insoluble soap such as oleate of aluminum is that it is impossible to decompose the soap with a high heat; the soap, although not a lubricant, is a vehicle for carrying some oil.

Remark 3. — Owing to a lack of body, this oil will not interfere with the sparking by depositing carbon on the platinum point.

Remark 4. — May contain 30 to 40 per cent of pure strained lard oil.

the best overall economy, such factors as first cost, quantity used, effect on the rubbing surfaces, maintenance and attendance being taken into consideration. Having determined the particular grade of lubricant which gives the best returns the tests previously mentioned are made and the results incorporated in the specifications so as to insure delivery of that particular grade of lubricant. Large consumers frequently employ the services of an experienced lubricating engineer under the supervision of the plant engineer or millwright for determining the lubricant best suited for the different classes of machinery.

Testing of Lubricating Oils: Power, Apr. 13, 1915, p. 522.

376. Atmospheric Surface Lubrication. — In a general sense all journals, slides, and "atmospheric" surfaces should be lubricated with straight mineral oils (as free from paraffin as possible), except when in contact with considerable water, in which case it is advisable to add 20 to 30 per cent of lard oil. Vegetable oils, paraffin oils, and animal oils (except lard oil as above stated) are not recommended for general engine and dynamo service. The test requirements of a number of classes of lubricants are outlined in Table 133 and represent current practice. Bearings, guides, and all external rubbing surfaces may be lubricated in a number of ways. (1) They may be given an *intermittent* application of oil, as, for example, with an oil can; (2) they may be equipped with oil cups with *restricted* rates of feed; and (3) they may be *flooded* with oil. The relative lubricating values of the systems have been estimated approximately as follows (Power, December, 1905, p. 750):

	Coefficient of Friction.	Comparative Value.
Intermittent	0.01 and greater	72 and less
Restricted feed	0.01 to 0.012	79 to 86
Flooded bearing	0.00109	100

377. Intermittent Feed. — Intermittent applications are ordinarily limited to small journals, pins, and guides which are subject to light pressures and which do not easily permit of oil or grease cups, as, for example, parts of the valve gear of a Corliss engine, governors, and link work. On account of the labor attached and the frequent doubt about the oil reaching the wearing surfaces this method of lubrication is limited as much as possible even in the smallest plants.

378. Restricted Feed. — In the average power plant the major part of the lubrication is effected by means of oil cups which are filled at intervals by hand or by mechanical means, the oil being fed from the cup by drops, according to the requirements.

379. Oil Bath. — In large power plants the principal journals and wearing parts are supplied with a continuous flow of oil which completely "floods" the rubbing surfaces. The oil is forced to the various parts either by gravity from an elevated tank or by pressure from a pump. After the oil leaves the bearings it flows into collecting pans, thence into a receiving and filtering tank, and finally is pumped back into an elevated reservoir and used over and over again. The little lost by leakage and depreciation is replenished by the addition of new oil to the system.

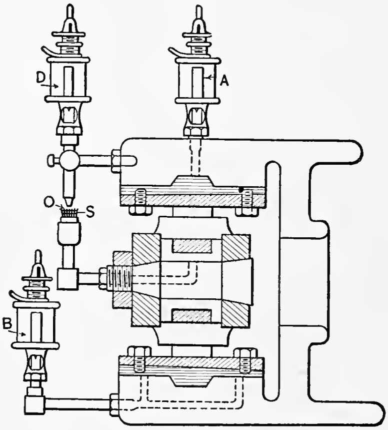


FIG. 529. Oil-cup Lubrication, Hand-filled.

380. Oil Cups. — Fig. 529 illustrates the application of *sight-feed* oil cups to the crosshead and slides of a reciprocating engine. The oil is fed into the cups by hand and gravitates to the rubbing surfaces, the rate of flow being regulated by

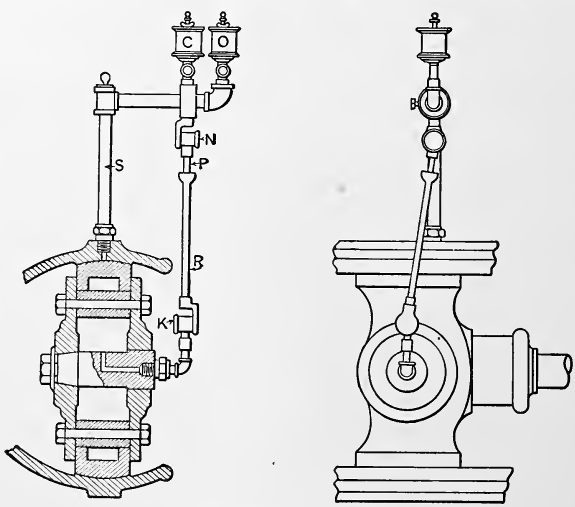


FIG. 530. Nugent's Telescopic Oiler.

a needle valve. Cups *A* and *B* feed directly to the crosshead guides, but the oil from cup *D* flows to the bottom orifice *O*, from which it is wiped by a metallic wick *S*, and carried by gravity to the wrist pin.

381. Telescope Oiler. — Fig. 530 shows the application of a *telescopic oiler* to a crosshead and guides. *O* and *C* are sight-feed oil cups, the former feeding directly to the top guide through the tube *S*. The oil from *C* flows by gravity through the swing joint into the telescopic tubes *P*, *R*, and thence to the pin through the lower swing joint as indicated. As the crosshead moves back and forth, the pipe *P* slides into and out of pipe *R*, the oil being thus conducted directly to the pin without wasting. A device of this type installed on a high-speed automatic engine at the Armour Institute of Technology has been in operation for five years without cost for repair or renewal.

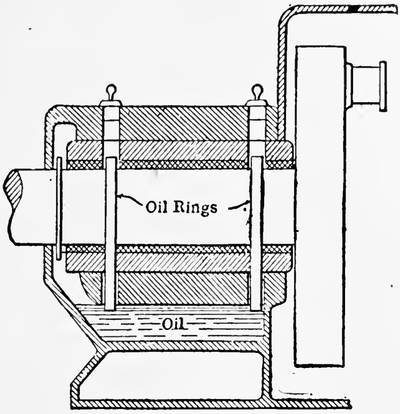


Fig. 531. Oil-ring Lubrication.

382. Ring Oiler. — Small high-speed engines are often oiled by the *oil-ring* system, as illustrated in Fig. 531. The shaft is encircled by several loose rings which dip into a bath of oil in the base of the pedestal or frame and, rolling on the shaft as it turns, carry oil to the top of the shaft where it spreads to the bearings. In some cases the rings are replaced by loops of chain.

Ring Lubrication: Power, Jan. 9, 1917, p. 42.

383. Centrifugal Oiler. — Fig. 532 illustrates the application of a *centrifugal oiler* to a side-crank engine. The oil supply is regulated by the sight-feed cup *C* and flows by gravity to the pipe *P* in line with the center of the crank shaft. Centrifugal force throws the oil outward through pipe *B* to the center of the pin *D*, which is drilled longitudinally and radially so as to distribute the oil upon the bearing surface.

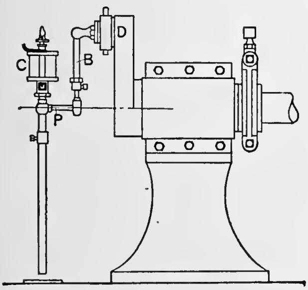


Fig. 532. Centrifugal Oiler.

384. Pendulum Oiler. — Fig. 533 illustrates the application of a *pendulum oiler* to the crank pin of a center-crank engine. Oil cups and pendulum *P* are fastened to the crank shaft *S* by trunnion *T*. The pendulum holds the cup vertical, since the friction of the trunnion is not sufficient to revolve it. Oil flows along the center of the crank shaft under the head of oil in cup *O* and is thrown outward to bearing *B* by centrifugal force.

385. Splash Oiling. — In some high-speed engines the crank, connecting rod, and crossheads are inclosed by a casing, the bottom of which is filled with oil to such a depth that at each revolution of the

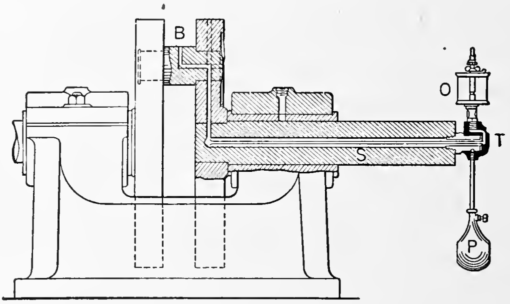


FIG. 533. Pendulum Oiler.

crank, the end of the connecting rod is partly submerged. The result is that the oil is splashed into every part of the chamber, and the crank pin, crosshead pin, and crosshead slides practically run in an oil bath.

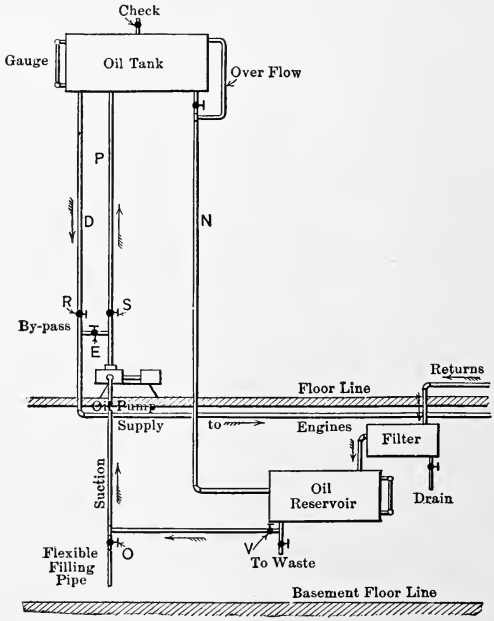


FIG. 534. Simple Gravity Feed System.

386. Gravity Oil Feed. — Fig. 534 illustrates a simple *gravity oil-feed* system. The oil to the engine is supplied from the oil tank by pipe D under pressure corresponding to the height of the tank above the oil cups. After performing its function the oil gravitates to the filter and from the latter to the oil reservoir, from which it is pumped back to the supply tank, the overflow being returned to the reservoir through pipe N. Operation is interrupted only when new oil is to be added to the system from the barrel through the flexible filling pipe. In case the oil tank is put out of commission, or the supply pipe becomes clogged, full pump pressure may be used by closing valves R and S and opening valve E. The make-up oil is small in amount compared to the quantity circulated. The reclaiming and purifying of the oil are essential if the bearings are to be flooded, otherwise the cost of oil would be prohibitive. At the power house of the South Side Elevated Railway the daily circulation (24 hours) of engine oil is approximately 1500 gallons. The make-up oil amounts to eight gallons.

An objection sometimes made to the above system is that the varying heights of oil in the supply tank may cause considerable variation in pressure at the oil cups, causing them to feed faster when the tank is full and slower when the tank is nearly empty. This applies only to installations where the supply tank is filled intermittently.

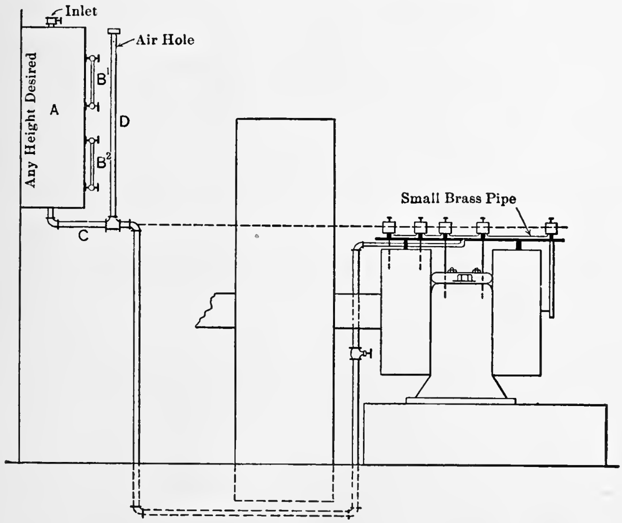


FIG. 535. Low-pressure Gravity Feed, Constant Head.

387. Low-pressure Gravity Feed. — Fig. 535 shows the application of a low-pressure oiling system in which the level in the sight feeds is kept constant. A is the main supply tank, B^1 and B^2 the upper and lower gauges indicating the oil level, C the supply pipe running to the engines, and D a small standpipe closed at one end and vented near the top. The reservoir is supplied with oil by the valve marked "inlet." When the tank is filled the oil rises in the standpipe D a corresponding height. The inlet valve is then closed and the oil in the standpipe feeds down to the level of the sight feeds or to a point where the air will enter the bottom of the tank. This will be the constant oil level, since oil flows from the tank only in proportion to the amount of air admitted. A head of 6 inches has been found to give the best results. (Engineer, U. S., March 16, 1903, p. 243.)

388. Compressed-air Feed. — Fig. 536 shows diagrammatically the arrangement of the oiling system at the First National Bank Building, Chicago. The storage tank containing the supply of engine oil is under air pressure at all times except during the short periods when it is being filled with oil from the filter. The air pressure on the surface

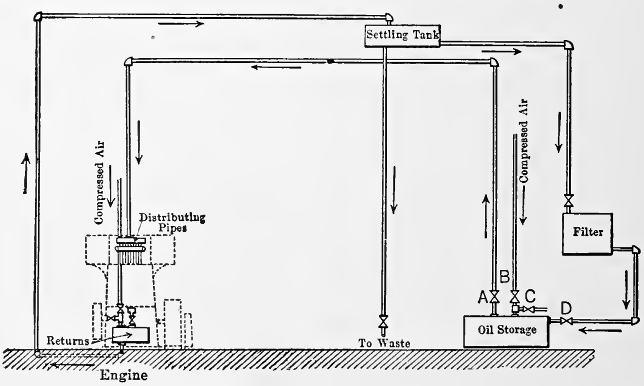


FIG. 536. Oiling System at the Power Plant of the First National Bank Building, Chicago.

of the oil forces it to a manifold on the engine from which it is distributed to the various oil cups. The oil flows from the different bearings to the returns tank located at the base of the engines. When the tank is filled air pressure is admitted and the oil forced to the settling tank, which has a capacity of about 400 gallons and is located near the ceiling. The oil is allowed to settle and the entrained water and foreign material are drained to waste. The oil gravitates from this tank to a series of Turner oil filters. When a new supply of oil is needed, valves *A* and *B* are closed and vent valve *C* opened, cutting off the supply of air and reducing the pressure to atmospheric. Valve *D* is then opened and oil flows from the filters to the storage tank.

389. Cylinder Lubrication. — The test requirements for cylinder oils are outlined in Table 133, from which it will be seen that pure mineral oil fulfils practically all requirements for dry steam. In connection with moist steam, as in the low-pressure cylinders of compound engines, an addition of from 2 to 5 per cent of acidless tallow oil is recommended.

Vegetable oils, beeswax, lard oil, degras (wool grease), and the like should never be used in compounding cylinder oils. The best cylinder oils are made from Pennsylvania stock. For data pertaining to the amount and grade of cylinder oil used in a large number of piston engine plants see Table I, p. 824, Jour. A.S.M.E., May, 1910. See also "Lubricants and Lubrication," by Dr. C. F. Mabery, Jour. A.S.M.E., Feb., 1910.

Cylinder oils must be forced to the parts requiring lubrication against the prevailing steam pressure, which is ordinarily accomplished by (1) *cylinder cups,* (2) *hydrostatic lubricators,* or (3) hand- or power-driven *force pumps.*

390. Cylinder Cups. — A cylinder oil cup consists essentially of a steam-tight brass vessel fitted at the bottom with a pipe connection and valve. A screwed cap offers a means of introducing the lubricant into the cup. After the cap is in place the valve is opened and the cup is subjected to full steam pressure. The pressure in the cup, being equal to that in the steam chest or cylinder, permits the lubricant to gravitate through the valve into the cylinder.

Fig. 537 shows a section through an improved form of oil cup in which the oil feeds from the top instead of the bottom as is the case with the common form of cylinder cup. The vessel is attached to the steam chest or to the supply pipe below the throttle valve. Steam is admitted through opening *B* and, condensing, settles through the oil to the bottom. This raises the level of the oil until it begins to overflow down the same passage by which the steam enters. This action is intensified by the fluctuation in steam pressure. The rate of feeding is regulated by valve *C* and tested by unscrewing plug *F*. If oil appears through opening *G*, the cup is feeding oil; if steam or water is emitted the cup is empty. The cup is filled by means of plug *E* and the water drained at *D*.

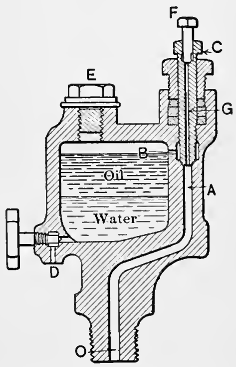


Fig. 537. Leyland Automatic Cylinder Cup.

391. Hydrostatic Lubricators. — The most common method of cylinder lubrication is by means of *hydrostatic* lubricators of the sight-feed class, Fig. 538. The principle of operation is as follows: The lubricator is filled with cylinder oil by removing cap *K*, the height of oil appearing in glass *L*. If water is present the oil floats on top as indicated. After the cap is screwed in place the valves in the condenser pipe are opened, subjecting the oil in the vessel to steam-pipe pressure.

Steam is condensed in pipe *C*, filling tube *B* and part of *C*, thus adding to the steam pressure the pressure due to the weight of the water column. Valve *F*, which communicates with the top of the vessel by means of tube *A*, is opened wide, as is also the regulating valve *I*. The pressure at *B* being greater than that at *A* by an amount equivalent to the height of the water column, forces the oil through *A* and the "sight feed" *S* to the steam pipe. The rate of flow is controlled by the

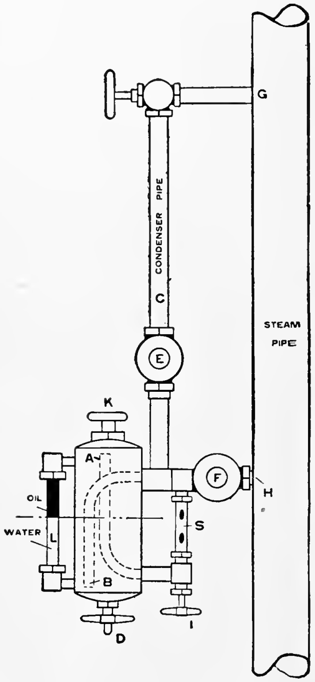


FIG. 538. Common Hydrostatic Lubricator.

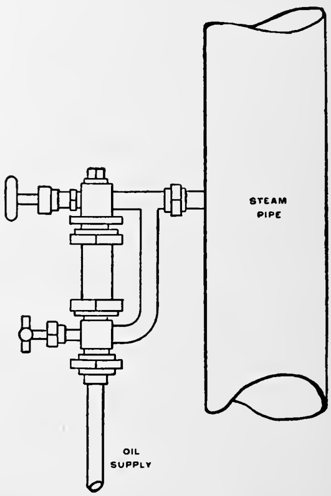


FIG. 539. Lunkenheimer Sight-feed Lubricator.

regulating valve *I*. As the oil flows from the vessel its space is occupied by condensed steam, the height of oil and water being visible in glass *L*. Owing to the small capacity of the lubricator it must be refilled frequently. To reduce the amount of labor required with the above apparatus, independent sight feeds, Fig. 539, are sometimes used in connection with a central reservoir. Such an installation is shown diagrammatically in Fig. 540. A condenser pipe leading from the steam main enters the bottom of the reservoir and the condensed steam fills up the reservoir as fast as the oil is fed out. The principle is the same as that of the simple hydrostatic lubricator. Oil is frequently injected by mechanical means under a steady pressure generated and governed independently of the steam. Two systems are in common use, direct mechanical pump pressure and air pressure.

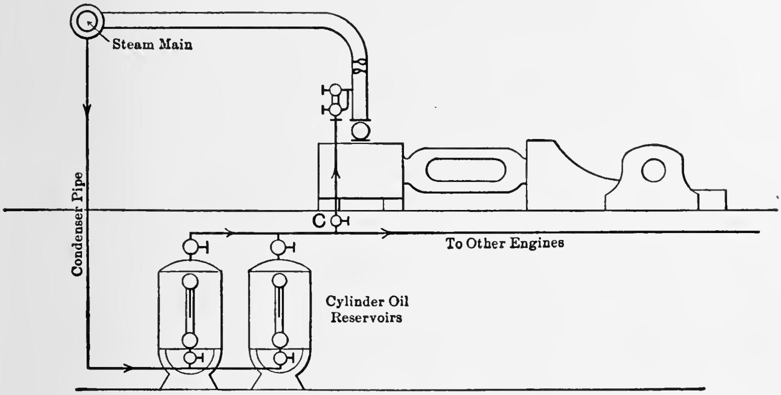


FIG. 540. Central Hydrostatic Lubricator.

392. Forced-feed Cylinder Lubrication. — Fig. 541 illustrates the "Rochester" simple feed automatic lubricating pump, which takes the oil by gravity from the reservoir through a sight-feed glass and forces it through a small pipe to the steam supply pipe. The pump entirely

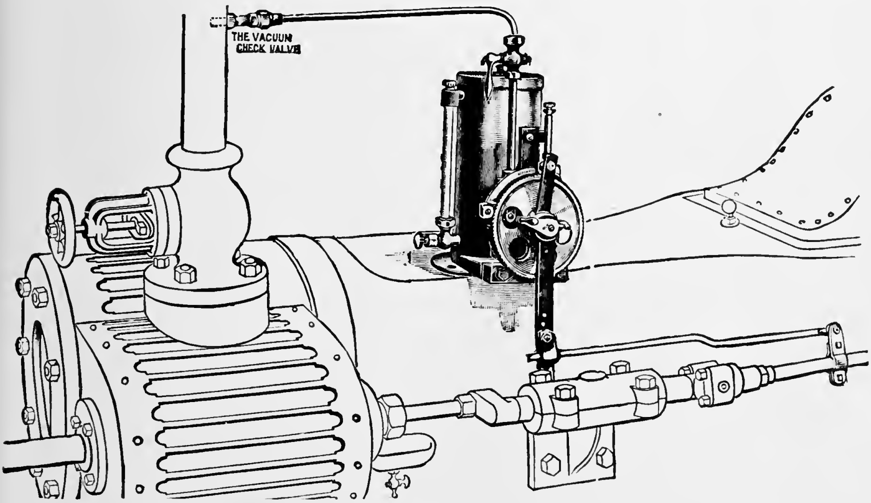


FIG. 541. Rochester Forced-feed Lubricator.

obviates the trouble due to intermittent feeding and, being directly driven from the engine, runs at constant speed. The feed is uniform and independent of the pressure pumped against. The rate is determined by the length of stroke of the pump piston, which is easily adjusted.

With large engines multi-feed pumps are sometimes used, which force oil to the various valves as well as to the steam pipe. Fig. 542 shows

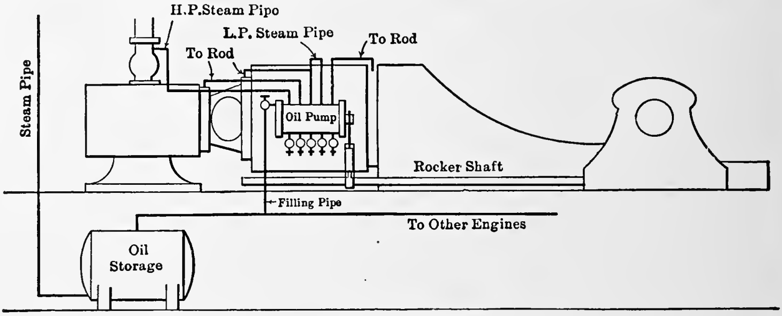


Fig. 542. Forced-feed Cylinder Lubrication.

an arrangement of storage tank in connection with pump reservoir to avoid the trouble of hand filling.

393. Central Systems. — Fig. 543 shows the piping for a large central system of cylinder and engine lubrication. There are two storage

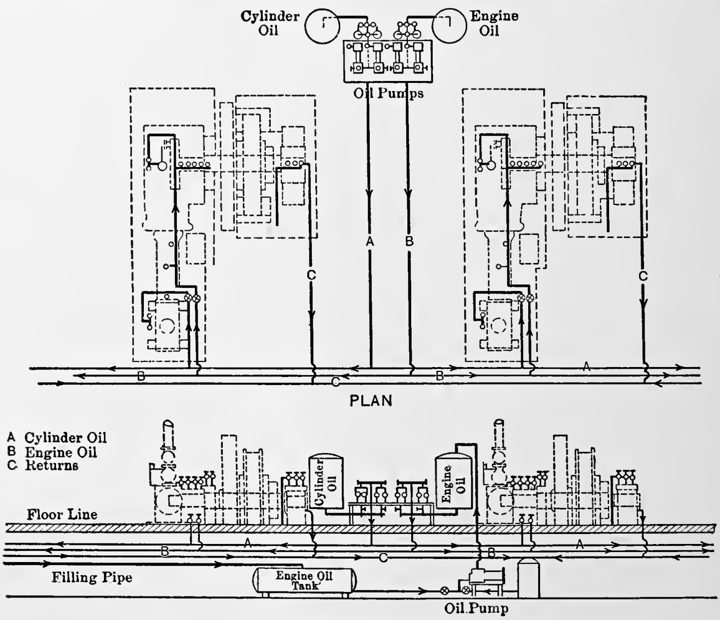


Fig. 543. Central System for Large Stations.

tanks on the engine-room floor, one for cylinder oil and the other for engine oil, the distributing arrangements being the same in each case. The oil is pumped from each tank into a main pipe extending the length of the engine room and provided with branches at each point requiring lubrication. The oil pumps are actuated by steam and are of the duplex direct-acting type, provided with automatic governors which regulate the speed to suit the demand for oil. The cylinder oil is forced through a special sight-feed lubricator, Fig. 544, under a pressure of about 25 pounds in excess of the steam pressure. Referring to Fig. 544, diaphragm valve D, in the bottom of the lubricator, is kept closed by the steam pressure admitted through pipes B. Thus the inlet pressure must be greater than that of the steam before the valve will open and admit oil to the engine. The oil, after entering, passes upward through the sight-feed glass and downward through the hollow arm A to the steam pipe. The engine oil is forced by the pump to the various points under a pressure of about 20 pounds. The waste oil is caught in suitable receptacles and, after being filtered, is returned to the storage tank by a steam pump. This pump is connected so that it can supply the storage tank either from the filter or with fresh oil from a large oil tank in the basement. By this arrangement all handling of oil in the engine room is done away with.

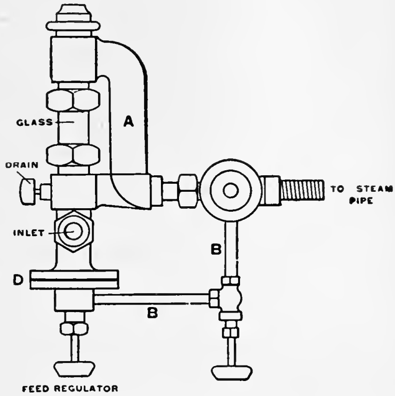


Fig. 544. Siegrist Sight-feed Lubricator.

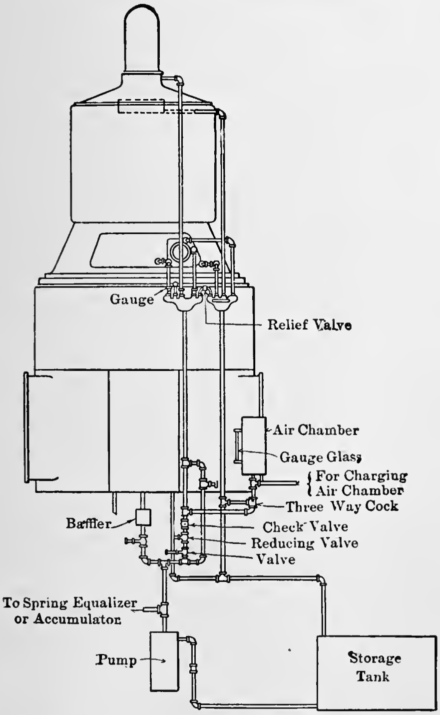


Fig. 545. Arrangement of Oiling System for Vertical Curtis Turbine.

Fig. 545 gives a diagrammatic outline of the oiling system for a vertical Curtis steam turbine. A tank, of sufficient capacity to contain all the oil and fitted with suitable straining devices and a cooling coil, is located at a level low enough to receive oil by gravity from all points lubricated. A pump draws oil from this tank and delivers it at a pressure about 25 per cent higher than that required to sustain the weight of the turbine in the step bearing. A spiral duct baffle connects the source of pressure to the step bearing and serves to regulate the oil supply to the lower end of the shaft. This source of pressure is also connected through a reducing valve to the upper oiling system of the machine, in which a pressure of about 60 pounds to the square inch is maintained. This system, which includes a storage tank partly filled with compressed air, operates the hydraulic governor mechanism and supplies oil to the upper bearings. Delivery of oil to these bearings is regulated by adjustable baffles designed to offer

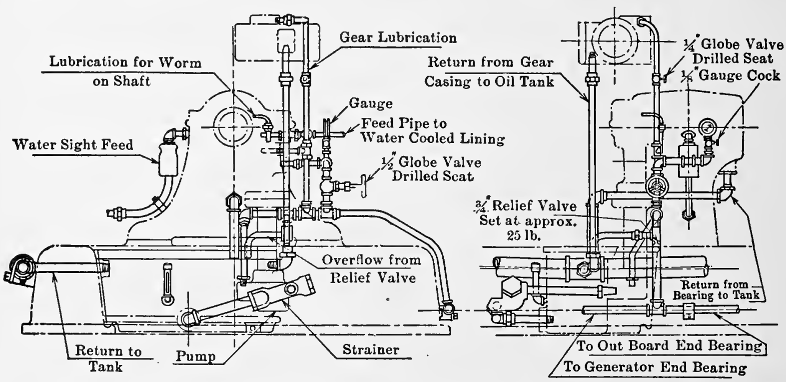


Fig. 546. Diagram of Oil Piping for Curtis Horizontal Turbine.

resistance to the oil flow without forcing the oil to pass through any very small opening which might easily become clogged. A relief valve is provided to prevent the pressure in the upper part of the oiling system from rising above a desirable limit. Drain pipes from the upper bearings and from the hydraulic cylinder and relief valve all discharge into a common chamber, in which the streams are visible, so that the oil distribution can always be easily observed. At some point in the high-pressure system adjacent to the pump it is desirable to install a device to equalize the delivery of oil from the pump, as is done by the air chamber commonly used with pumps designed for low pressure. A small spring accumulator is furnished for this purpose, except in cases where weighted storage accumulators are used. In large stations where several machines are installed, a storage accumulator is desirable and can be arranged advantageously so that it will normally remain full, but will discharge if pressure fails, and in doing so will start auxiliary pumping apparatus.

All modern steam turbines are equipped with forced feed lubricators. The oil pumps are either independently driven or geared to the turbine shaft. The different systems employed are described in paragraphs 207–213.

394. Oil Filters. — After oil has been applied to machinery its lubricating properties become impaired on account of (1) contamination with anti-lubricating material, such as dust, metallic particles from wear, gum, acid, and resin; and (2) exposure to heat and the atmosphere which drives off part of the more volatile constituents and decreases the fluidity of the oil.

In many small plants no attempt is made to reclaim oil that has once been used, since the quantity is so small that the cost and trouble involved would more than offset the gain. Where large quantities of oil are used, considerable saving may be effected by using it over and over again. To render the oil fit for reuse it must be thoroughly purified. The anti-lubricating matter is removed by precipitation and filtration.

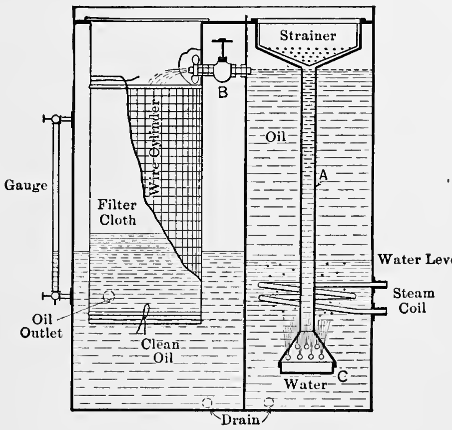


Fig. 547. White Star Oil Filter.

Fig. 547 shows a section through a "White Star" oil filter and purifier. The apparatus consists of a cylindrical sheet-iron vessel divided into two compartments by a vertical partition. These two compartments are connected near the top by valve *B*. The smaller chamber is provided with a funnel *A* and a steam coil for heating the contents. The large chamber contains a cylindrical wire screen covered with several folds of filtering cloth. Impure oil is poured into funnel *A*, the upper part of which is provided with a removable sieve or strainer, and is discharged below the surface of the water through holes in the foot of the tube. The thin streams of oil rise vertically to the surface of the water and the heavy particles of grit and dirt gravitate to the bottom. The steam coil heats the oil and water and facilitates precipitation of the solid matter by thinning out the streams of oil. When the oil in the smaller chamber reaches the level of valve *B* it flows into the filter bag, which removes the remaining impurities and permits the purified products to flow into the large compartments from which it may be drawn at will. All parts are accessible and readily removed for cleaning purposes. The accumulated sediment in the bottom of the small chamber is discharged to waste at intervals by means of a suitable drain. When the filter cloth is to be removed, valve *B* is closed and the wire cylinder is disconnected and lifted out. Any oil remaining in the filter is returned to funnel *A*. The filter cloth is held against the screen by cords and hence is readily removed.

Fig. 548 shows a section through a Turner oil filter, illustrating the type of filter usually installed in large stations where continuous filtration is desired. This apparatus consists of a rectangular tank divided into four compartments. The returns from the lubricating

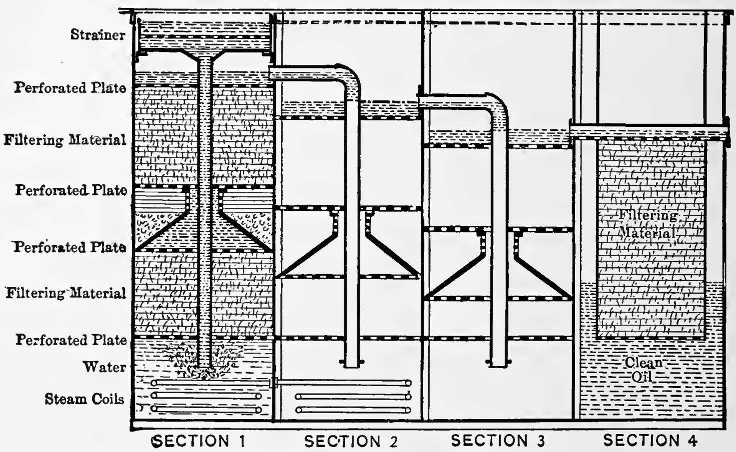


FIG. 548. Turner Oil Filter.

system flow into section 1 through a screened funnel and discharge into the water space at the bottom of the compartment. The oil rises through the water, passes, under pressure of the head in the funnel, through a layer of filtering material resting on a perforated plate, and collects in an inverted cone. Through perforations around the top of the cone it passes into a dirt chamber, where most of the heavy impurities are deposited, and then, still rising, passes through another perforated plate and more filtering material. The partially cleaned oil, which issues, overflows into the second compartment and thence into the third, the same cycle of operations being repeated in these two. The overflow from the third compartment descends through a final filter in the fourth compartment and collects at the bottom, from which it is withdrawn by the oil pump.

Cylinder Lubrication: Power, Apr. 11, 1916, p. 519, Feb. 15, 1910; Jour. A.S.M.E., Feb. and May, 1910.

Miscellaneous. — Measurement of Durability of Lubricants: Trans. A.S.M.E., 11–1013. *Valuation of Lubricant by Consumer:* Trans. A.S.M.E., 6–437. *Suitability of Lubricants:* Power, Nov., 1906, p. 673. *Oil Required for Lubricators:* Elec. World, May 5, 1906, p. 934. *Gumming Tests:* Jour. Am. Chem. Soc., April, 1902, p. 467. *Valuation of Lubricants:* Jour. Soc. Chem. Ind., April 15, 1905, p. 315.

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Oil Purification: Elec. World, Dec. 1, 1906, p. 1053.

Economy in Lubrication of Machinery: Trans. A.S.M.E., 4–315. *Theory of Finance of Lubrication:* Trans. A.S.M.E., 6–437.

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Lubricators and Lubricants: Power, Sept. 21, 1909, p. 486, Feb. 22, 1910, p. 347.

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Tests of Used Oil: Prac. Engr., Apr. 15, 1914, p. 469.

Laws of Lubrication of Journal Bearings: Trans. A.S.M.E., 37–1915, p. 534.

CHAPTER XVII

TESTING AND MEASURING APPARATUS

395. General. — The importance of maintaining a system of records is discussed in paragraph 419. The various items which may be recorded and the instruments and appliances used in this connection are outlined in the accompanying chart. In large stations a full complement of indicating, recording, and integrating instruments may prove to be a good investment if intelligently and closely studied by the operating engineer with a view to locating and eliminating unnecessary losses. The instruments should be inspected and calibrated at intervals, since many of them are delicately constructed and are apt to become inaccurate after a few months' service. Steam gauges, thermometers, and pyrometers, and particularly piston water meters are subject to appreciable error after considerable use. Voltmeters, ammeters, and other switchboard instruments are easily deranged, especially when subjected to continuous vibration or to high temperature.

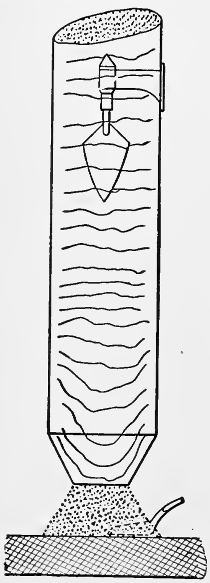


Fig. 549. Coal Meter.

396. Weighing the Fuel. — In most small plants the delivery tickets of the coal dealer are depended upon for the weight of coal used, no attempt being made to determine the evaporative value, and the economy of the plant is judged by the size of the coal bill. In such cases a considerable saving may be effected by keeping a daily record covering at least the coal and water consumption. The coal can be conveniently weighed on ordinary platform scales. In a number of large stations the weight of coal is determined by suspended weighing hoppers, which may be stationary, as in Fig. 141, or mounted on a traveling truck, as in Fig. 142. The scales of such devices are made indicating, autographic, integrating, or a combination of the three, the latter costing but little more than the simple indicating or recording devices.

A simple and inexpensive coal meter recently brought out is illustrated in Fig. 549. It consists essentially of a helical vane placed in a cylindrical conduit. The movement of the coal causes the vane to

TESTING AND MEASURING APPARATUS.

Steam Plant.

Weights	Fuel	Platform scales, indicating and autographic.		
		Suspension hoppers, indicating and autographic.		
		Coal meters, integrating.		
	Water	Platform scales and tanks.		
		Water meters	Piston	Integrating.
			Rotary	Integrating.
			Disk	Integrating.
			Venturi, indicating and autographic.	
		Weirs and volume displacement meters.		
	Steam	Weighing condensed steam.		
		Steam meters	Direct.	
			Indirect.	
Pressures	High	Bourdon gauge, indicating and autographic.		
		Manometers, mercurial, indicating.		
	Low	Manometers — mercurial, indicating, and autographic.		
		Manometers — water, indicating, and autographic.		
		Diaphragms, indicating and autographic.		
Temperatures	Up to 800 deg. fahr.	Mercurial thermometers, indicating.		
		Expansion thermometers, indicating and autographic.		
	800 to 2500 deg. fahr.	Expansion thermometers, indicating and autographic.		
		Resistance thermometers, indicating and autographic.		
		Thermo-electric thermometers, indicating and autographic.		
	Over 2500 deg. fahr.	Optical pyrometer, indicating and autographic.		
		Platinum or clay ball pyrometer.		
Power	Indicated	Indicators, hand manipulated.		
		Indicators, continuous autographic.		
	Developed	Rope brake.		
		Prony brake.		
		Absorption dynamometers.		
		Electric generator.		
Flue gas analysis	Orsat apparatus.			
	Hay's recorder.			
	Westover recorder, autographic.			
	Uehling gas composimeter, autographic.			
Moisture	In air	Hygrometer, indicating and autographic.		
	In steam	Calorimeters	Separating.	
			Throttling.	
Fuel analysis	Coal calorimeters	Mahler bomb.		
		Carpenter.		
		Thompson.		
		Parr.		
	Gas calorimeter	Junker.		

Electrical Plant.

Voltage	Voltmeters, A. C. and D. C., indicating and autographic.
Current	Ammeters, A. C. and D. C., indicating and autographic.
Output	Wattmeters, A. C. and D. C., integrating and autographic.
Power factor	Power factor meters, A. C. only, indicating and autographic.
Frequency	Frequency meter, A. C. only, indicating.
Synchronism	Synchronizers, A. C. only, indicating.

rotate and the number of revolutions is a measure of the weight of fuel passing. For hard coal of uniform size the meter gives consistent results agreeing within two per cent of scale weight, but with bituminous coal the results are somewhat erratic and particularly so with lumps of varying size. (For a detailed description of the device, see Prac. Engr., U. S., Apr. 15, 1912, p. 438.) With certain types of mechanical stokers it is possible to approximate the rate at which fuel is fed into the furnace by registering the speed of the stoker engine. In the new River Station of the Buffalo General Electric Co. "Electric stoker tachometers" are used for this purpose.

397. Measurement of Feed Water. — The quantity of water fed to the boiler may be determined by

1. Actual weighing.
2. Measurement of volume displacement.
3. Measurements by weirs and orifices.
4. Measurement by determining the velocity of flow in the feed pipe.

Some of these methods necessitate measurement on the suction side of the pump; others are applicable to either suction or pressure. The former, as a class, are the more accurate but involve bulky apparatus. The choice for any given case depends upon the quantity of liquid to be measured, the degree of accuracy required, space requirements, and first cost.

398. Actual Weighing of Feed Water. — The most accurate means of measurement is by the use of two or more tanks resting upon scales, arranged to be filled and emptied alternately. This method is limited to comparatively small quantities because of the great bulk of apparatus involved and is seldom used for continuous service. It is commonly employed in conducting special tests of short duration and for calibration purposes. For regular boiler service it involves considerably more time than is ordinarily at the disposal of the fireman and engineer. For temperatures above 150 deg. fahr., the weighing tanks should be covered, since evaporation may cause an appreciable error. See also "Rules for Conducting Boiler Trials," A.S.M.E., Code of 1915.

399. Worthington Weight Determinator. — Fig. 550 shows the general details of the Worthington weight determinator, illustrating a commercial means of continuously measuring and recording the *weight* of water fed to the boiler. The apparatus consists primarily of two tanks of equal size, A and B, each mounted on knife edges K and equipped at one end with a siphon S and at the other end with counterweight W. The liquid to be measured flows through inlet pipe P and along deflector D into either tank. Each tank remains in a horizontal position until the weight of liquid overcomes the counterweight when it tilts into the position shown by the dotted lines. Discharge now takes place through siphon S until the liquid reaches a certain level at which point the tank tilts back to its original position and the siphon

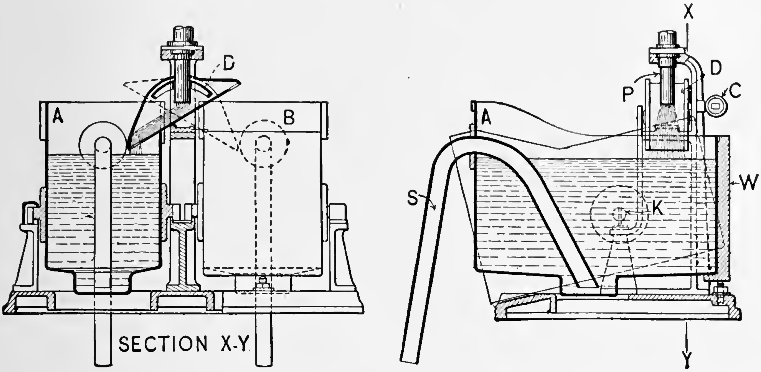


Fig. 550. Worthington Water Weigher.

continues its action until the vessel is emptied. The tanks operate alternately, one filling while the other is discharging. Since each tilt represents a definite *weight* of liquid irrespective of variations in volume due to specific gravity or changes in temperature, the number of tilts as recorded by counter C is a correct measure of the weight discharged. This apparatus operates at atmospheric pressure and is arranged to discharge into a storage tank from which the feed pump takes its supply.

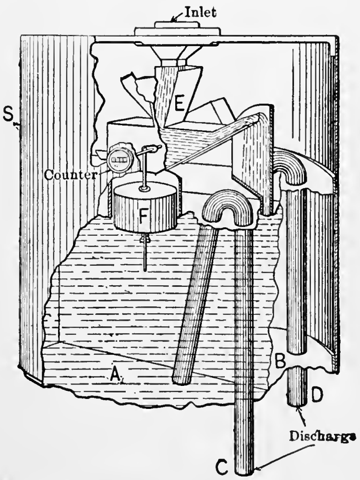


Fig. 551. Kennicott Water Weigher.

400. Kennicott Water Weigher. — This apparatus is used in many boiler houses and seems to give universal satisfaction. It consists of a cylindrical shell S, Fig. 551, the lower part of which is divided into two measuring compartments A and B, each fitted with a siphon for discharge and a float F for actuating the tripping mechanism. Tripping box E is divided into two sections which alternately fill with water and serves the double purpose of furnishing a sufficient quantity of water to start the siphons and to shift the supply from one compartment to the other. This tripping box is balanced on knife edges and is mounted directly above the measuring compartments. Water enters the inlet and passes to the tripping box where a small portion is intercepted, the remainder passing directly to the measuring compartment below. When this compartment is nearly filled the float tilts the tripping box, discharges its contents into the compartment, and starts the siphon. A counter registers each double charge. This apparatus discharges at atmospheric pressure, though with slight modification it may be installed on the pressure side of the pump. Kennicott water weighers are constructed in various sizes ranging from a capacity of 750 to one million pounds per hour and are guaranteed by the manufacturers to record the correct weight of water within one-half of one per cent of scale weight at any given temperature. Calibration for different temperatures is necessary since the apparatus is actuated by volume displacement. For example, the weight of one cubic foot of water at 60 deg. fahr. is 62.37 pounds and at 210 deg. fahr. it is 59.88, a difference of 2.49 pounds. Hence, if the device is calibrated to read correctly at 60 degrees it would be in error 4 per cent if used to measure water at 210 deg. fahr.

401. Willcox Water Weigher. — Another successful volume displacement meter is illustrated in Fig. 552. The device consists of a cylindrical tank divided into an upper and lower compartment by a horizontal partition. The water enters the upper compartment, passes to the lower, in which its volume is measured, and then out through the U-shaped discharge pipe. The operation, beginning with both compartments empty, is as follows: Water enters the upper compartment through the inlet pipe and rises to the top of the standpipe. (The latter is open at the top and bottom and is rigidly connected to the bell float, but when in its lowest position it is held against its seat by weight of the bell float.) Further admission of water causes it to overflow into and through the standpipe into the lower compartment. The water, rising in the lower compartment, seals the lower edge of the bell float and entraps a volume of air under the bell. Further rise compresses the air under the float,

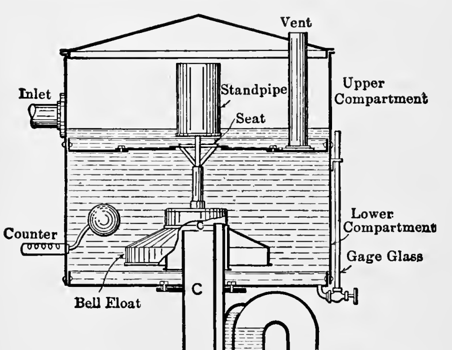


Fig. 552. Willcox Water Weigher.

in leg C of the discharge pipe and in leg A of the trip pipe AB. This compression causes the float to rise to its highest position and raises the standpipe from its seat, permitting the water in the upper chamber to pour into the lower vessel. Compression of air continues until the pressure becomes great enough to break the seal in the trip pipe. This action immediately reduces the pressure below the float, permits the latter to descend, sealing the upper chamber against further discharge, and allows the water in the lower compartment to siphon out through the discharge pipe. The number of discharges is recorded mechanically.

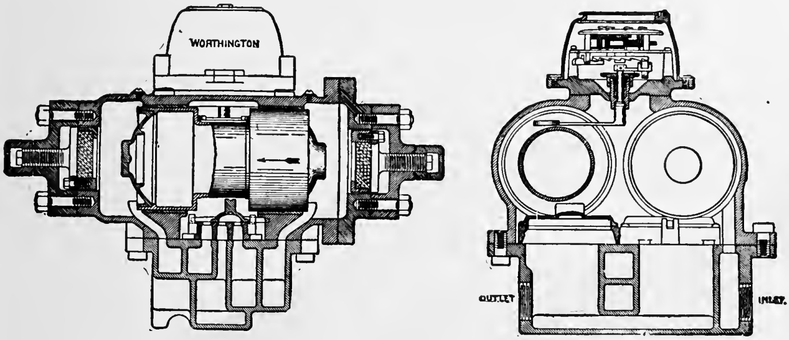


Fig. 553. A Typical Piston Water Meter. (Worthington.)

402. Weir Measuring Devices. — Feed water heaters or specially designed tanks fitted with V-shaped, cycloidal, or trapezoidal weir notches offer a simple means of measuring the rate of flow. The chamber is divided into vertical compartments arranged so that one may discharge through a calibrated weir notch into the other. The height of water above the bottom of the notch is a direct measure of the volume flowing. The height may be noted in an ordinary gauge glass or it may be transferred through a suitable float mechanism to an outside indicator. Commercial weir measuring devices are usually provided with autographic and integrating attachments for recording the rate of flow and for totaling the weight of water passing through the device. For the theory of weir notches, orifices, and nozzles consult "Experimental Engineering," Carpenter and Diederichs, 1911, Chapter XII. See also, Trans. A.S.M.E., 1915.

Weir Meters for the Power Plant: Power, May 1, 1917, p. 582.

403. Pressure Water Meters. — There are a number of reliable water meters on the market for hot or cold water which may be placed on the pressure side of the feed pump. Among them may be mentioned the Hersey, Crown, Nash, and Worthington. They are all based on volume displacement and consequently require correction for different temperatures if graduated to read in pounds. They are compact, comparatively inexpensive, and require considerably less space than the tank weighers of the Kennicott and Willcox type but are open to the objection that no particular provision is made against leakage and after considerable use they are subject to serious error. In many plants where meters of this type are installed the meter is by-passed and operated only for short periods. For continuous service meters of the tank-weighing or Venturi type are recommended. Fig. 554 illustrates

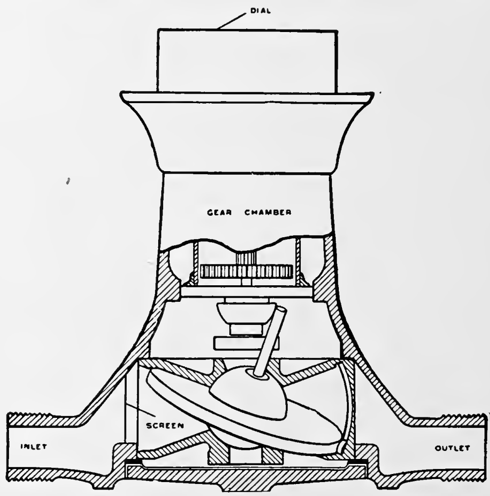


Fig. 554. A Typical Disk Water Meter. (Nash.)

the *piston* type of pressure meter, in which reciprocating pistons are displaced by a definite volume of water; Fig. 425, the *rotary* type, depending upon the displacement of rotary impellers; Fig. 555, the *disk* type, in which impellers are given a combined rotating and tilting motion. The capacities of pressure meters range approximately as follows:

Size of meter (pipe size)	$\frac{3}{8}$,	$\frac{1}{2}$,	$\frac{3}{4}$,	1,	$1\frac{1}{2}$,	2,	3,	4,	6
Maximum capacity, cubic feet per minute:									
Rotary or disk meters	1,	2,	4,	8,	12,	20,	36,	72,	120
Piston meters	$1\frac{1}{2}$,	3,	5,	6,	8,	23,	60,	120	

404. Venturi Meter. — The Venturi tube with indicating, autographic, and integrating mechanism, as constructed by the Builder's Iron Foundry of Providence, R. I., is one of the most satisfactory methods of measuring feed water under pressure. The total absence of working parts in the meter proper insures continuity of operation and freedom from wear, and the fact that the recording mechanism may be placed at a considerable distance from the meter is a great advantage. The Venturi tube, Fig. 555, is essentially the same in principle as an orifice placed in the pipe. The pressure difference H between A in the "upstream" portion of the tube and B at the "throat" is a measure of the velocity through the throat. The loss of head due

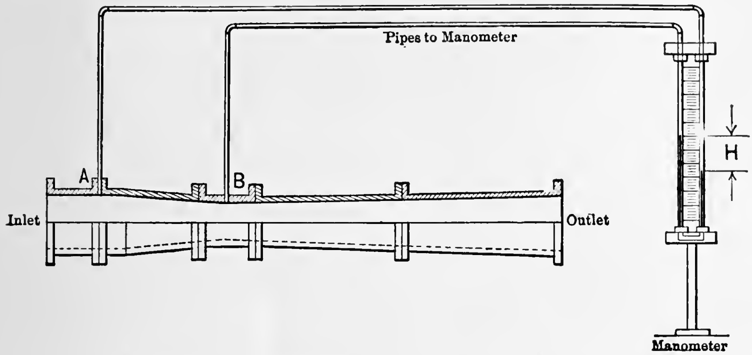


Fig. 555. Venturi Tube with Indicating Manometer.

to friction is negligible and the velocity may be calculated, within an error of 2 per cent, from the following modification of Bernouilli's theorem:

$$V_t = \frac{F_u}{\sqrt{F_u^2 - F_t^2}} \sqrt{2\,gH}, \tag{299}$$

in which

V_t = velocity at the throat, feet per second,
F_u = area of the upstream section, square feet,
F_t = area of the throat, square feet,
H = pressure difference, feet of water.

For accurate work the tube requires calibration. Once calibrated the error in weight readings for a given temperature should not exceed one per cent for capacities within the working range of the manometer. For very low throat velocities the error may be considerable because of the slight pressure difference between the points A and B. In situations where there are periods of very low and very high rates of flow, as in connection with combined heating and lighting plants, it is customary to install a small tube for the light loads and a large tube for the heavy loads, the same indicating mechanism being used in each case. The equipment illustrated in Fig. 555 is purely indicating and readings must be taken at frequent intervals in order to obtain the total flow for a given period. Where the size of the plant warrants the outlay the combined indicating, integrating, and recording instrument is often installed. With this device the instantaneous rate of flow is indicated by a pointer and dial, the variation in rate of flow for any given period is recorded on a clock-driven chart, and the total weight flowing is registered on a counter. (For a detailed description of this mechanism see Power, Jan. 23, 1912, p. 102.) Tests made at Armour Institute of Technology on a carefully calibrated tube and recorder with feed water at 210 deg. fahr. and constant rate of flow gave chart and counter readings agreeing substantially with scale weights; for irregular and fluctuating flow, as when feeding the boilers, the average error was about two per cent.

405. Orifice Measurements. — The appropriation of the great majority of small steam power plants does not permit of the installation of tank meters, Venturi meters, or other forms of reliable commercial appliances for measuring the weight of water fed to the boilers. For

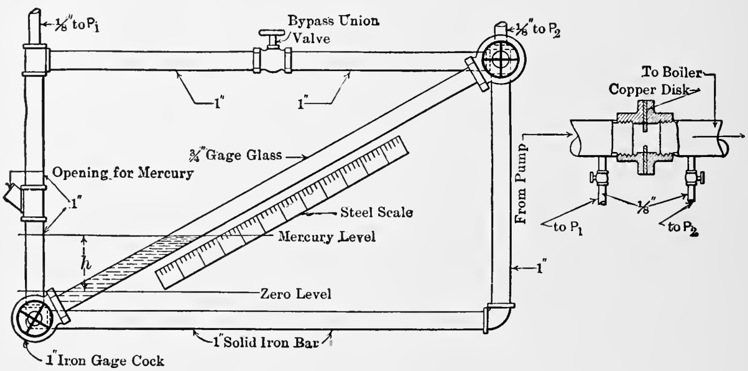


Fig. 556. Simple Indicating Water Meter, Orifice Type.

use in such cases an inexpensive and fairly accurate indicating meter may be constructed of ordinary pipe fittings, as illustrated in Fig. 556. A thin metal diaphragm with circular orifice is inserted on the pressure side of the feed pump and the pressure drop across the orifice is measured by inclined mercury manometer. The height of mercury h is an indication of the rate of flow. By calibrating the manometer against tank measurements the readings of the mercury column may be graduated to read directly in pounds per hour. If means are not available for calibration purposes the weight of discharge may be approximated from the formula

$$W = 1120\, a \sqrt{hd}, \qquad (300)$$

in which

W = weight flowing, pounds per hour,
a = area of the orifice, square inches,
h = vertical height of mercury column, inches,
d = density of the water, pounds per cubic foot.

For a fairly continuous flow and pressure drop corresponding to three inches of mercury or more this simple device gives results agreeing within four per cent of tank weights, but for widely fluctuating flow and small pressure drops the error may be considerably more.

For application of the Pitot tube for water measurements consult accompanying bibliography.

The Pitot Tube for Water Measurements: Trans. A.S.M.E., Vol. 30, 1908, p. 351, Vol. 25, 1904, p. 184, Vol. 22, 1901, p. 284.

The Pitometer: Proc. Am. Wks. Asso., 1907, p. 136; Jour. Frank. Inst., Dec., 1907, p. 425.

406. Measurement of Steam. — The quantity of steam passing through any device may be determined by (1) condensing and weighing the steam after it has passed through the apparatus and by (2) measuring the flow by means of steam meters before it enters. The first necessitates the use of surface condensers, and consequently has a limited field of application, whereas the latter may be used in both condensing and non-condensing service.

407. Weighing Condensed Steam. — The weight of condensed steam may be obtained by any of the devices used in connection with feed water measurements but such measurements are seldom made except for test purposes because of the expense or labor involved. The Wheeler Condenser and Engineering Company's "indicating hot well" offers a practical and simple solution of continuously measuring the condensed steam. The hot well is attached to the bottom of the condenser chamber in the usual way and differs from the ordinary hot well only in the addition of a vertical partition. This partition divides the hot well chamber into two compartments. Condensation from the condenser drains into one of these compartments and flows to the other through a calibrated orifice. The height of water above the orifice as shown in the gauge glass is an indication of the weight of condensation flowing. By means of suitable attachments the readings may be automatically recorded and totaled. The manufacturers guarantee an accuracy within 2 per cent of scale weight for readings over the whole range.

408. Steam Meters. — The weight of fluid flowing through an opening may be calculated by the equation

$$W = AyV, \tag{301}$$

in which

W = weight in pounds per second,
A = cross-sectional area in square feet,
y = density of the fluid, pounds per cubic foot,
V = velocity of flow, feet per second.

All steam meters for indicating or recording the weight of steam flowing through a pipe are based upon the law expressed in equation (301). Thus, for steam of constant density the opening through which it flows may be made constant and the variation in velocity will be an indication of the rate of discharge; or the velocity may be held constant and a variation in the amount of opening will be an indication of the weight discharged. Unfortunately, the density of steam is seldom constant under commercial conditions and herein lies the inherent defect of all steam meters which depend for their operation upon a variation in the area of efflux or a variation in velocity. The density of steam is a function of its pressure and quality and any variation in either will affect the weight of discharge as determined from equation (301). Pressure variations may be automatically compensated for, but corrections for quality must be made in each specific case.

CLASSIFICATION OF STEAM METERS.

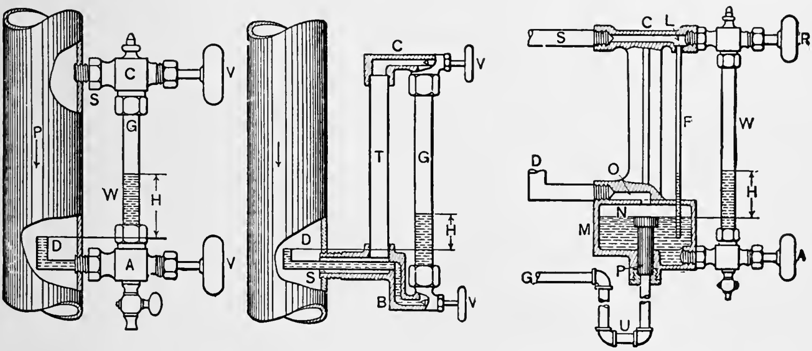
Indirect	Velocity	Pitot tube	Impeller	Lindenheim (1896)* Gebhardt (1908)†
			Water manometer	Burnham (1905)† Gebhardt (1910)†
			Mercury manometer	General Electric (1910)*†‡ Republic 1916*†‡
Direct		Current	Impeller	Holly (1877)*
	Throttling	Floating valve	Mechanical control	St. Johns (1893)†‡ Gehre (1896)†‡ Baeyer (1902)†‡ Bendemen (1902)†‡ Sargent (1908)† Lindmark†‡
		Stationary disk	Mercury manometer	Gehre-Hallwachs (1907–1910)*†‡ Sarco (1910)*†‡ Bailey (1910)*†‡
		Venturi tube	Bourdon manometer	Eckardts (1903)†‡
			Mercury manometer	Parenty (1886)†‡ Builders' Iron Foundry (1910)†‡

* Integrating. † Indicating. ‡ Autographic.

The different means adopted for transmitting this area and velocity variation to the indicating or recording devices overlap to such an extent as to render a classification of steam meters very unsatisfactory. The accompanying chart is offered as a guide in grouping the most commonly known devices. From this chart it will be seen that all meters may be grouped into general classes, *direct* and *indirect*. The direct meter is an integral part of the piping and the entire mass of fluid to be measured passes through the apparatus. It is not portable and cannot be readily applied to pipes of different sizes. In the indirect meter only a small part of the fluid to be measured is directed through the apparatus and the pipe line need not be disconnected for its installation. One instrument suitably calibrated may answer for any size of pipe.

The average high-grade steam meter is a reliable and accurate means of measuring the flow of steam in straight lengths of pipes, provided the flow is continuous or that the change in the rate of flow is gradual and the pressure and quality are practically constant. For interrupted or intermittent flow and for sudden variations in pressure or quality, the results are not reliable and may be considerably in error. The accuracy of all meters, provided they have been correctly calibrated and adjusted, depends largely upon the degree of refinement in reading the indicators and in integrating the charts. The commercial failure of many steam meters is due to the fact that they are not cared for or operated in strict accordance with the principles of design.

Only a few of the best-known meters will be described here. For a detailed discussion of the various types of steam meters see the author's paper "Various Types of Steam Meters," Power, Feb. 6 and 13, 1912.



FIGS. 557, 558, 559. Principles of the "Gebhardt" Indicating Steam Meters.

"Gebhardt" Steam Meters. — Figs. 557 to 560 illustrate various forms of indicating steam meters designed and tested at the Armour Institute of Technology, which are based on the principles of the Pitot tube. Referring to Fig. 557, A and C are two ordinary gauge cocks and G is a common gauge glass, C being connected with the static nozzle S and A with the dynamic tube D. The height of water H is proportional to the square of the velocity of steam flowing through pipe P and automatically adjusts itself to the variations in velocity; thus, for decreasing velocities, the water in glass G discharges through D until the water column H balances the velocity pressure in pipe P,

and for increasing velocities, condensation from the upper part of the instrument accumulates and the water column H rises until a balance is effected for the higher velocities.

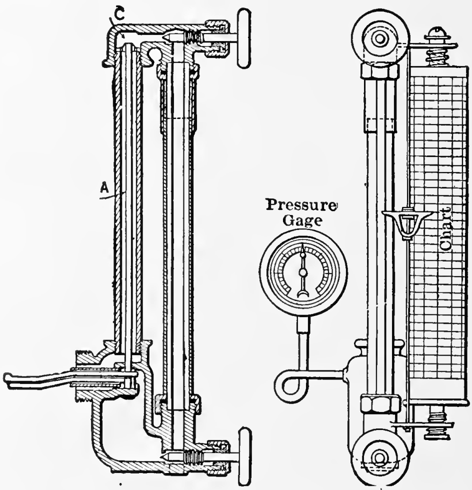


FIG. 560. Commercial Form of "Gebhardt" Steam Meter.

The relation between the height of the water column and the velocity of the steam in the main pipe at the entrance to the dynamic tube may be determined from the well-known equation

$$V = c\sqrt{2\,gh}, \tag{302}$$

in which

V = maximum velocity of flow, feet per second,
c = coefficient determined by experiment,
h = height of a column of steam equal in weight to the water column H.

The equation may be expressed

$$V = K\sqrt{H\frac{d_w}{d_s}}, \tag{302a}$$

in which

K = coefficient determined by experiment,
H = height of water column in inches,
d_w = density of water in gauge glass, pounds per cubic foot,
d_s = density of steam in the main pipe.

Because of the labor of determining the relationship between the mean and the maximum velocity for various conditions of flow and different pipe diameters it is more satisfactory to calibrate the gauge, by actual experiment, to read directly in pounds per hour.

This simple device in connection with a calibrated scale gives readings within 5 per cent of condenser measurements for continuous flow and constant pressure and quality of steam (for velocity pressures corresponding to $1\frac{1}{2}$ inch of water or more). For a considerable variation in pressure and quality or for marked changes in rate of flow the instrument is not reliable. Its sensitiveness is greater at high velocities, since the height of water column in the gauge glass increases with the square of the velocity of the steam in the main pipe. For interrupted flow, as when connected to a high-speed engine, the water column may be made to closely approximate the mean velocity of suitably throttling the gauge cocks.

Fig. 558 shows application of the same principle with only one connection to the main pipe. Under favorable conditions the commercial meter (Fig. 560) gives readings within 2 per cent of condenser weights for velocity pressures corresponding to 1 inch of water or more. Fig. 559 shows another form which may be placed below or above the point in the main pipe at which the Pitot tubes are placed. The operation is as follows: Velocity pressure is transmitted through tube D and opening O, into the body of the chamber M. This pressure, acting on the surface of the condensed steam in the chamber, forces the water into the glass W until a balance is effected. Condensation is discharged continuously through pipe P and the water seal U of the main pipe. Tests of this meter have given results agreeing within 2 per cent of condenser measurements for continuous flow for all velocities ranging from the equivalent of a 1-inch to a 10-inch water column. No provision is made for automatic correction of pressure and quality variation in any of these devices. (For the theory and results of tests of the Pitot type of steam meter see author's paper "The Pitot Tube as a Steam Meter," Trans. A.S.M.E., Vol. 31, p. 603.)

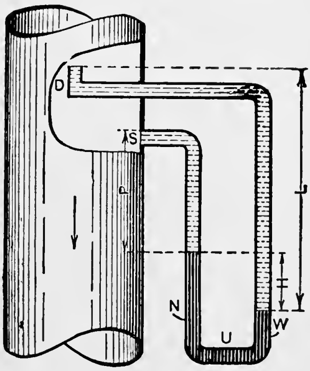


Fig. 561. Pitot Tube with Mercury Manometer.

G-E. Flow Meters. — All G-E. flow meters, with the exception of the "orifice tube" type for small pipe sizes, depend for their operation upon the displacement of a mercury column by the differential pressure action of a modified Pitot tube. The basic principle of operation is illustrated in Fig. 561: S is the static opening and D the dynamic opening; U is an ordinary U-tube manometer partially filled with mercury. When there is no flow the surface of the mercury in columns N and W

will be on the same level and the upper portion will be filled with condensed steam. When there is a flow, the mercury will be depressed as indicated and the difference H will be a measure of the velocity of flow at the point in the pipe where the dynamic tube is placed. This velocity may be expressed by the equation

$$V = K \sqrt{H \frac{d_m}{d_s}}, \tag{302b}$$

in which

d_m = density of mercury in lb. per cu. ft.

Other notations as in equation (302a).

A comparison of equations (302a) and (302b) will show that the mercury manometer is less sensitive than the water manometer by an amount equivalent to $d_m \div d_w$, or approximately 13.6. The variable heights of the water column above the mercury is usually included in the value of the coefficient K.

In all G-E. meters (the "orifice-tube" type excepted) the Pitot tube is given the form of a "nozzle plug" as shown in Fig. 562: TT are the

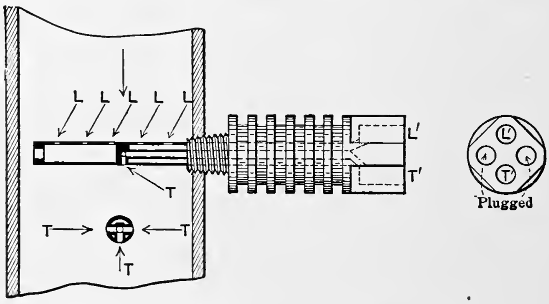


FIG. 562. Nozzle Plug; G-E. Steam Meter.

static openings or "trailing set" and LL the dynamic openings or "leading set." The plug is screwed into the pipe with the "leading set" directly facing the current and the connections to the manometer are made through the openings T and L. The manometer for the "portable indicating" or laboratory device is shown in Fig. 563. Adjustments for variations in pressure, quality, and pipe diameter are made by setting the chart cylinder C in accordance with the auxiliary scale attached to the instrument. The meter may be used to measure flow under normal conditions in any number of different pipe lines. It is only necessary to provide the pipes with the proper size and kind of nozzle plug or pipe reducer to which the meter can be connected.

Fig. 564 shows a section through the G-E. indicating flow meter which differs from the simple portable device in that the movement of the mercury column is magnified by suitable mechanism. A small float resting on the top of the mercury in one leg of the U-tube is attached to a silk cord passing over a pulley; this cord is kept taut by a counterbalance weight acting in the opposite direction. The shaft on which the pulley is mounted carries a small horseshoe magnet with its pole faces near and parallel to the inside surface of a copper plug fastened to the body of the meter. A small magnet is mounted on pivot bearings in such a manner that its poles are near and parallel to the outside surface of the copper plug, and its axis of rotation in line with the

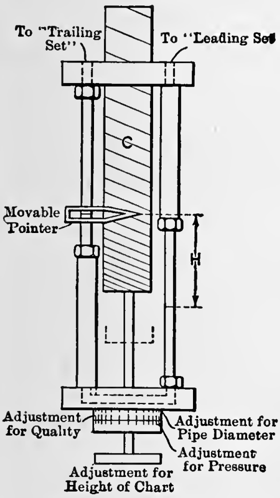


Fig. 563. General Principles of the G-E. Indicating-flow Meter.

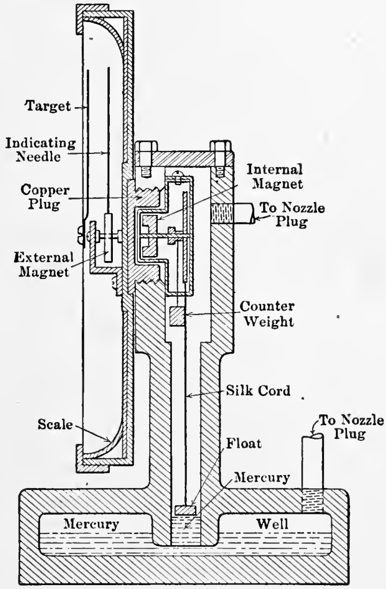


Fig. 564. Section through G-E. Steam-flow Meter.

shaft carrying the magnet inside the case. The indicating needle is attached directly to this magnet. By means of the float and cord, the pulley carrying the magnet inside the body is rotated in proportion to the change of level of the mercury. Any motion of this magnet is transmitted magnetically to the outside magnet carrying the indicating needle. In cases where the velocity is too low to be accurately measured with a normal velocity nozzle-plug, pipe reducers, as illustrated in Fig. 565, are employed.

The "G-E. Indicating Recording" meter differs from the simple indicating device just described only in minor detail. The movement of the float is transmitted to the indicating needle and recording pen through the agency of a rack and pinion in place of the cord and pulley.

The indicating needle is attached directly to the outside magnet but the recording pen is actuated by a sector which in turn is rotated by a small pinion on the shaft carrying the outside magnet.

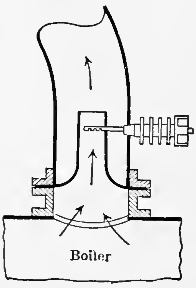


Fig. 565. Reducing Nozzle.

The "G-E. Indicating Recording, Integrating" meter is identical with the indicating-recording device with the exception that an integrating mechanism is attached to the sector actuating the recording pen.

For pipes 2 inches or less in diameter the nozzle plug is replaced by an "orifice tube," Fig. 566, which is to all intents and purposes a Venturi tube.

Fig. 567 gives a diagrammatic outline of the counting mechanism of a European steam meter which serves to illustrate the basic principle of the G-E. integrating attachment. R is a small friction wheel mounted on the pen arm a and connected to gears c and d by the small shaft m; P is a clock-driven disk in contact with the friction wheel R. As the pen arm moves the wheel R in and out from the center of disk P, the speed of the small friction wheel is decreased or increased accordingly. The revolutions of R are transmitted to the integrating mechanism e so that the total flow may be read directly from the dials.

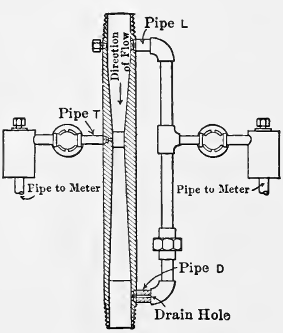


Fig. 566. G-E. Orifice-tube Steam Meter.

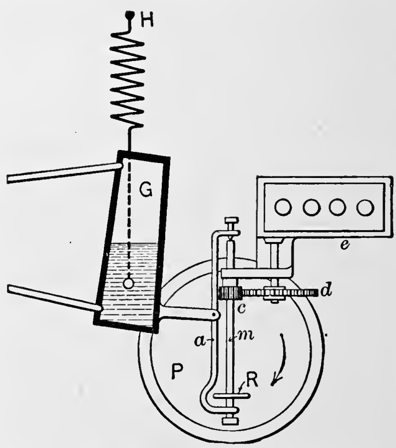


Fig. 567. Counting Mechanisms for Steam Meters.

Since the pen arm or its equivalent in the G-E. meter does not move directly proportional to the velocity of the steam it is necessary to correct its movement by means of a cam so that this result may be effected.

Republic Flow Meter. — This meter is of the Pitot tube and mercury manometer type but differs radically from the G-E. devices in the manner of utilizing the displacement of the mercury columns for indicating, recording, and integrating purposes.

The principles of operation are illustrated in Fig. 568; c_1, c_2, c_3 are electric conductors, of varying length. As the mercury in the static

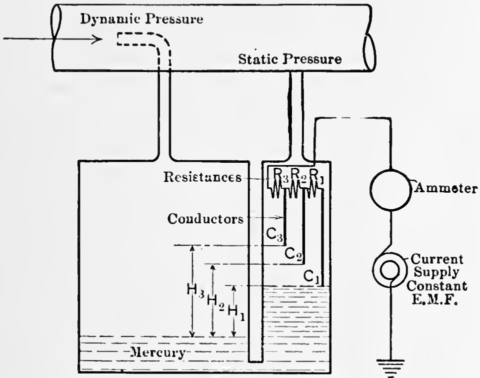


FIG. 568. Fundamental Principles of the Republic Flow Meter.

leg of the manometer rises it makes successive contact with these conductors. The resistance of each conductor is such that a constant electromotive force impressed upon the circuit will cause a current to flow through the conductor directly proportional to the flow of steam in the pipe. Any suitable ammeter and watthour meter may therefore be used for indicating, recording, and totaling the weight of steam flowing through the pipe.

Fig. 569 shows the general assembly of the Pitot tubes or "tube holder" as used in the commercial instrument, and Fig. 570 shows a

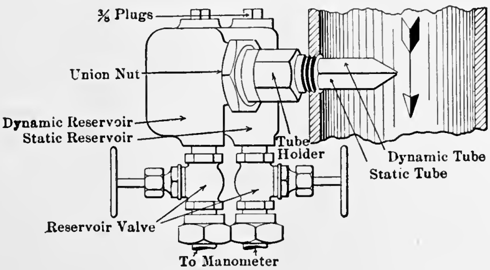


FIG. 569. "Tube Holder," Republic Flow Meter.

section through the meter body. Referring to Fig. 569 it will be seen that the dynamic and static elements are plain cylindrical tubes with beveled ends and placed side by side as indicated. This beveling of the ends insures the necessary pressure difference for actuating the manometer. Referring to Fig. 570, the conductors consist of a large number of small steel rods of varying length, the lower ends of which when in contact with the mercury form one end of the circuit; and the upper ends in series with individual resistance coils are connected to a common terminal post to form the other end of the circuit. The conductors and resistances are insulated by means of oil which entirely fills the "contact chamber" above the mercury and also the annular chamber between the meter body and contact chamber. This prevents water and foreign substances from reaching the contact rods. A small rotary converter (for direct-current supply) or a small transformer (for alternating-current supply) furnishes the necessary current under a pressure of 40 volts for actuating the various measuring instruments. The maximum current demand is approximately one ampere. The particular feature of this meter is

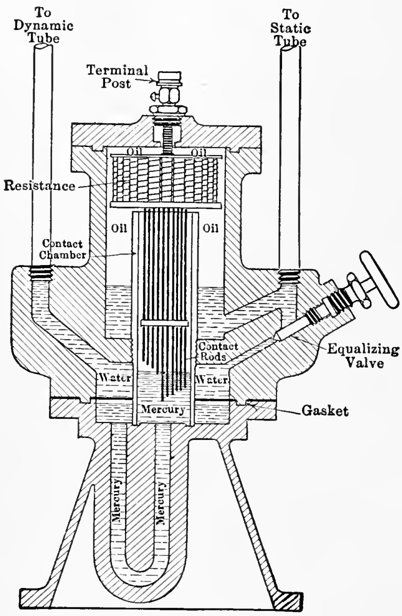


FIG. 570. Section through Body of the Republic Flow Meter.

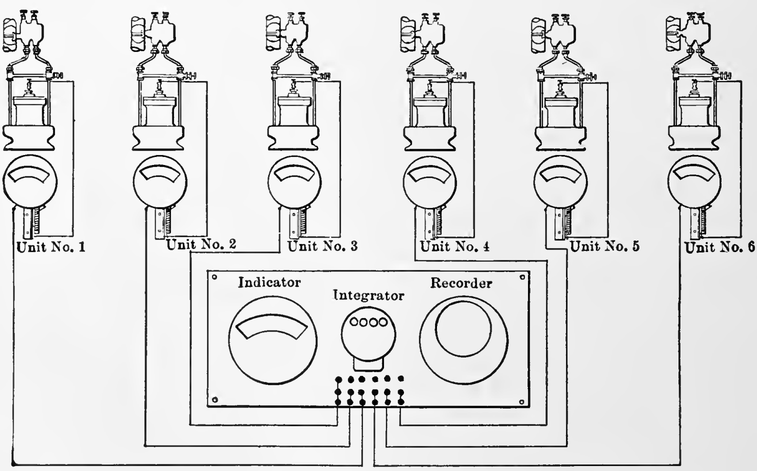


FIG. 571. Typical Arrangement of Republic Flow Meter in a Six-unit Boiler Plant.

that the reading dials can be located at any point with respect to the meter body and at any distance from the pipe. Fig. 571 shows a diagrammatic arrangement of a typical installation. The Pitot tubes and meter bodies are connected in the boiler outlet. The indicators are mounted on the boiler fronts and show the rate of evaporation. The board located in the office of the chief engineer includes one indicator, integrator, and recorder, and is equipped with suitable switches so that the performance of any boiler may be observed at any time.

St. Johns Steam Meter. — In the groups of meters described above the indicating and recording mechanism is actuated by the natural velocity of the steam. In the St. Johns, Bailey, Gehre-Hallwachs, Storrer, Eckardt, and Venturi steam meters the velocity is increased by throttling and the pressure drop is utilized in actuating the mechanism. The weight of steam flowing through the orifice may be calculated from the following modification of equations (301) and (302):

$$W = AK\sqrt{p_1 - p_2}, \tag{303}$$

in which

W = pounds discharged per second,
A = area of the orifice, square feet,
K = coefficient determined by experiment and includes the density of the steam,
p_1 and p_2 = pressure on the upper and lower side of the orifice, pounds per square inch.

In some of the meters the pressure drop $p_1 - p_2$ is maintained constant and the variation in the area A actuates the indicating mechanism, and in others the area is made constant and the variation in pressure drop operates the mechanism.

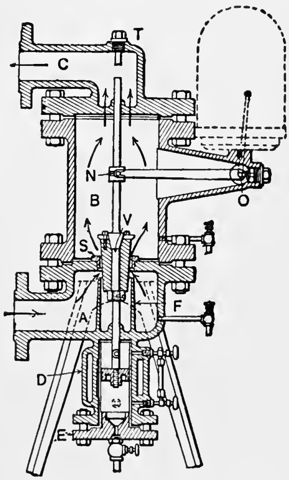


FIG. 572. St. Johns Steam Meter.

Fig. 572 represents a section through a St. Johns steam meter, illustrating the throttling type with a floating valve. This meter was placed on the market 20 years ago and still finds favor with many engineers. It records the weight of steam passing through the seat of an automatically lifting valve which rises and falls as the demand for steam increases or diminishes.

Referring to the illustration, valve V is weighted so that a pressure in space A of 2 pounds greater than in B is necessary to raise the valve off its seat. This pressure difference is constant for all positions of the valve. The plug is tapered so that the rise of the steam pressure is directly proportional to the volume of steam flowing through the seat. The movement of the valve is transmitted through suitable levers to an indicating dial and a recording pen so that the instantaneous and continuous rate of flow may be read at a glance. For a given pressure and quality of steam, the indicating dial and chart may be calibrated to read the weight of discharge directly, corrections being made for variations in pressure and quality. The manufacturers guarantee the readings of the chart to be within 2 per cent of condenser measurements for a total pressure range of 10 pounds from the mean pressure at which the chart is calibrated.

The chief drawback to this instrument is inherent to all meters of the direct type in that they are bulky and the steam line must be taken down for the installation. The total hourly flow may be obtained by integrating the curve. Tests of this meter made by the author were in accordance with the guarantee of the manufacturer for continuous flow and for moderate changes in the rate of flow. For rapid fluctuations in flow the results were not so satisfactory, the greater error lying in the difficulty of integrating the curve correctly.

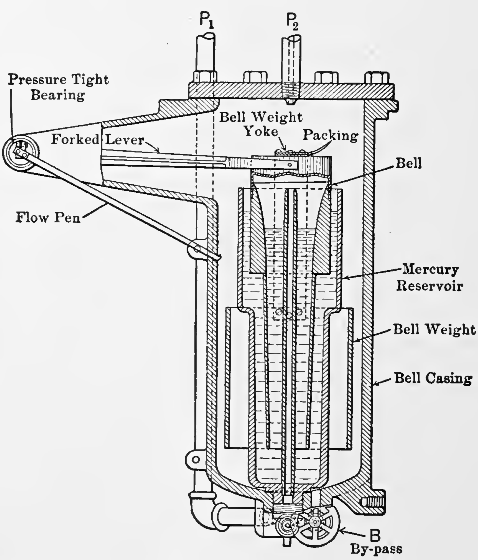


Fig. 573. Section through Meter Body of a Bailey Fluid Meter.

Bailey Fluid Meter. — Fig. 573 shows a section through the manometer body of a Bailey steam meter. An orifice placed in the steam line at a suitable point effects the necessary pressure drop for actuating the mechanism. The higher pressure is applied at P_1 and the lower pressure at P_2 through small tubes or pipes. The interior of the "bell casing" is subjected to pressure P_2 and the interior of the mercury sealed "bell" is subjected to the higher pressure P_1. This difference in pressure pushes the bell upward and as it rises from the mercury the change in the buoyant action of the mercury on the walls of the bell balances the force due to the pressure difference. By varying the area of the bell and the thickness of its walls any desired motion can be imparted to the bell. The displacement of the bell is a measure of the weight of steam flowing and its motion may be transmitted through suitable linkage to recording or integrating attachments.

The "Bailey Boiler Meter" is a combination of the Bailey draft gauge (Fig. 577), Bailey steam meter and a recording thermometer. By this combination the differences in draft between furnace and ash pit, furnace and uptake, temperature of the steam and rate of steam flow can be simultaneously recorded on a single clock-driven chart. This instrument is compact and easily applied. When correctly interpreted the records are of great assistance in regulating the rate of air supply to the furnace and in controlling the thickness of fire.

409. Pressure Gauges. — The Bourdon type of gauge, either autographic or indicating (Fig. 574), is the most familiar and satisfactory means of measuring pressures up to 1500 pounds per square inch or more, although diaphragm gauges are also used and both are employed as vacuum gauges. For the latter purpose, however, the mercurial vacuum gauge has the advantage of greater accuracy and is not subject to derangement. Bourdon gauges should be frequently standardized by comparison with a gauge of known accuracy, a mercury column, or a gauge tester.

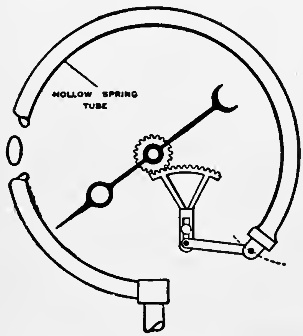


FIG. 574. Bourdon Pressure Gauge.

For measuring very low pressures, such as are found in boiler flues or gas mains, indicating or recording diaphragm gauges may be had, but some form of U-tube manometer is generally employed, the design best adapted to the purpose depending upon the accuracy required. The simple U-tube (Fig. 575), when filled with mercury, may be used for pressures limited only by the inconvenience due to length of tubes, or with water as the fluid, for pressures only a fraction of an ounce per square inch. Where greater accuracy is required than can be obtained with the simple U-tube, some modification may be employed, such as the Ellison draft gauge with one inclined leg which magnifies the reading several times. A form of sensitive gauge is sometimes used which depends upon the use of two fluids of different specific gravity, as oil and water.

The Blonck Boiler Efficiency Meter, Fig. 576, consists essentially of two differential draft gauges, one connected between the ash pit and

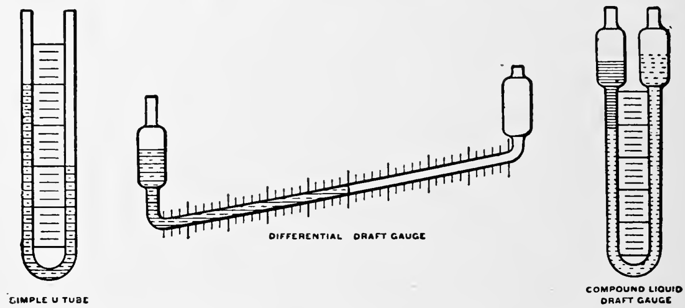


FIG. 575. Different Forms of Manometer Pressure Gauges.

furnace and the other between the furnace and the breeching on the boiler side of the damper. In the indicating device the lower gauge (showing the pressure drop through the fuel bed) is supplied with red colored oil, and the upper gauge (showing the pressure drop between furnace and damper) is supplied with blue colored oil. The readings of each gauge and the difference in readings between both gauges are indications of the furnace performance and offer a means of scientifically controlling the depth of fire, air supply, and rate of combustion.

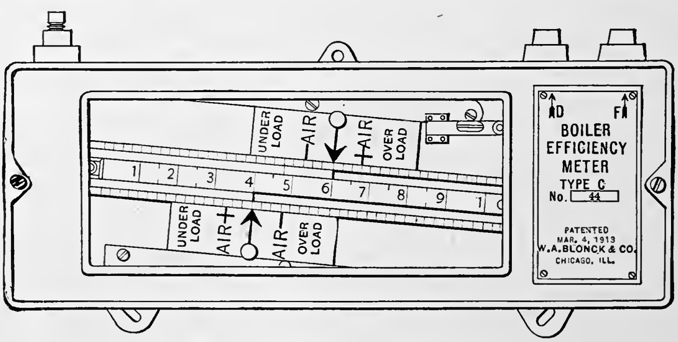


FIG. 576. Blonck Efficiency Meter.

Sliding pointers enable the fireman to fix the draft indications best suited for the particular equipment and conditions of operation. This device is also made recording.

A simple yet accurate instrument for measuring and recording very low pressures or a small pressure difference is shown diagrammatically in Fig. 577. Two bells A and B are suspended from opposite ends of a beam (which is pivoted on knife edge bearings) and are partly submerged in a light non-volatile oil as indicated. In measuring pressures less than atmospheric, connection is made at P_2 and P_1 is left open to the atmosphere. For pressures above atmospheric, connection is made at P_1 and P_2 is left open. For measuring the difference of two pressures the higher pressure is applied at P_1 and the lower at P_1. If a slight suction pressure is applied at P_2 it is effective over the inside area of bell A and pulls it down into the liquid. The relative motion between bells A and B is transmitted through levers L and pen arm P to the recorder pen. This instrument may be designed to record pressures or pressure difference as low as one one-thousandth of an inch of water.

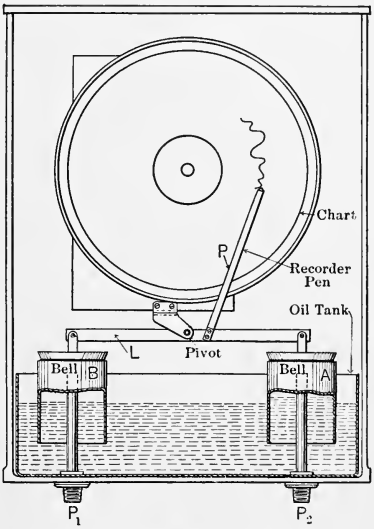


Fig. 577. Bailey Recording Draft Gauge.

410. Measurement of Temperature. — For power-plant purposes mercurial thermometers are most convenient for measuring temperatures up to 400 deg. fahr., and are inexpensive. For higher temperature, up to say 800 deg. fahr., they are also adapted, but must be made of special glass and the space above the mercury filled with nitrogen under pressure to prevent vaporization of the mercury. Such thermometers must be used intelligently and should be standardized from time to time, since they are subject to considerable change. The Bureau of Standards at Washington, D. C., is prepared to furnish certificates for which a nominal charge is made.

Fig. 578 shows a form of thermometer which is much used where a continuous autographic record is required. It depends for its operation upon the pressure produced by a fluid, liquid or gaseous, contained in a small bulb and exposed to the temperature to be measured. The pressure is transmitted to the recording mechanism through a flexible capillary tube which may be of considerable length. Such thermometers are suitable for feed water, flue gas, and temperatures not exceeding 1000 deg. fahr.

Fig. 579 illustrates a form of electrical pyrometer employing thermocouples which has come into wide use as a reliable means of measuring

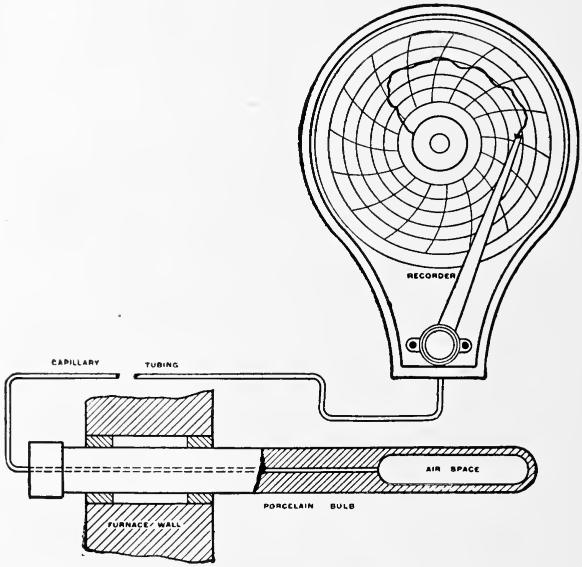


Fig. 578. Bristol Recording Pyrometer.

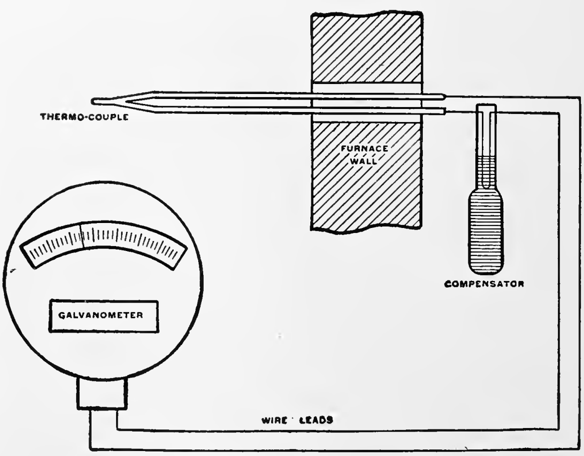


Fig. 579. Bristol Thermo-electric Pyrometer.

temperatures up to 2600 deg. fahr. The couples most frequently used are composed of platinum and platinum-rhodium, platinum and platinum-iridium, copper and copper-constantan, and copper and nickel,

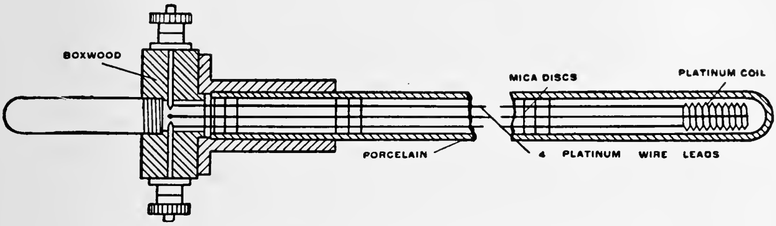


Fig. 580. Element for Callendar Resistance Pyrometer.

the first named being adapted to the higher ranges of temperature. The electromotive force set up, when the thermo-junction is heated, is proportional to the temperature and is measured by means of a sensitive millivoltmeter which is usually graduated to read temperature directly. Thermo-couples may be made to give an autographic record by means of a *thread recorder.*

Fig. 580 shows the element of an electrical thermometer based upon the change in resistance of a platinum wire when subjected to change

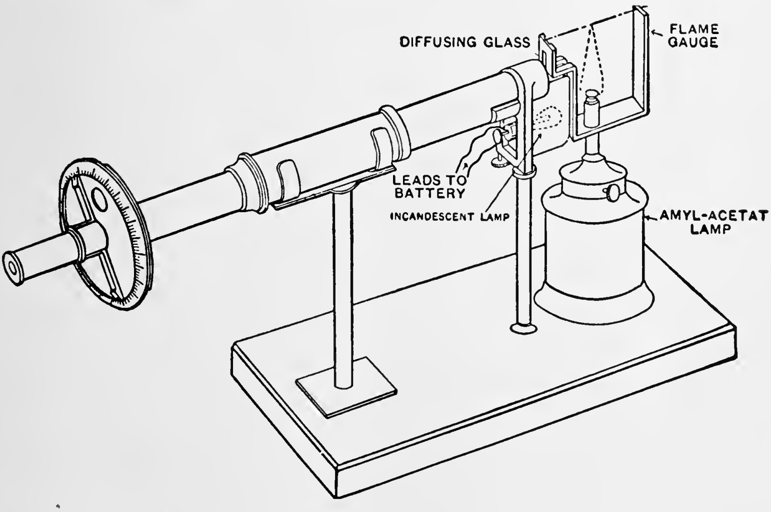


Fig. 581. Wanner Optical Pyrometer in Position for Standardizing.

in temperature. The resistance, in terms of temperature, is measured by a *Whipple indicator*, a convenient and portable form of Wheatstone bridge, or may be autographically recorded by means of a *Callendar recorder.* Resistance thermometers of this type are very sensitive and accurate, not easily deranged, and are limited in range only by the fusing points of the platinum and the porcelain protecting sheath.

For higher temperatures and for obtaining the temperatures of inclosed spaces above about 900 deg. fahr., such as boiler furnaces, annealing ovens, and kilns, various forms of *optical* and *radiation pyrometers* have been devised. In such devices no part of the instrument is exposed to the temperature to be measured and hence suffers no injury from this cause. Optical pyrometers are based upon the measurement of the brightness of the hot body by comparison with a standard. The Wanner optical pyrometer is shown in Fig. 581. After standardizing by comparison with an amyl-acetate lamp, it is only necessary to focus the instrument upon the source of heat to be measured and the temperature is read on the graduated scale.

Radiation pyrometers depend upon the measurement of the heat radiated from the hot body. The Féry radiation pyrometer, Fig. 582,

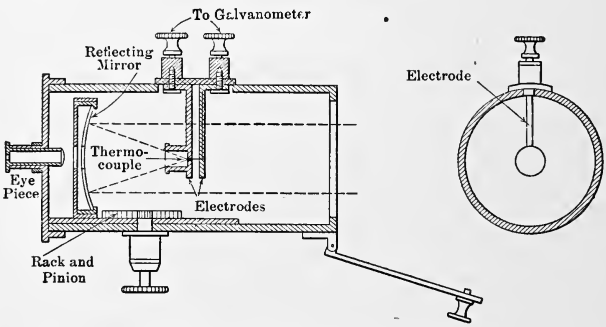


FIG. 582. Féry Radiation Pyrometer.

is the best-known instrument of this type. When focused upon the source of heat a cone of rays of definite angle is reflected by means of the mirror upon a thermo-couple located in its focus. The electromotive force set up is measured in terms of the temperature of the source of heat by a millivoltmeter. Neither the couple nor any part of the instrument is ever subjected to a temperature much above 150 deg. fahr. The indications are practically independent of the distance from the source of heat, and the range is without limit.

The *Uehling pyrometer* depends for its operation upon the flow of gas between two apertures, thus: Air is continuously drawn through two apertures by a constant suction produced by an aspirator. So long as the air has the same temperature in passing through these orifices there is no change in the partial vacuum in the chamber between them; if, however, the air passing through the first opening has a higher temperature than that passing through the second, the vacuum in the chamber will increase in proportion to the difference in temperature since the volume of air varies directly with the temperature. In the application of this principle, the first aperture is located in a nickel tube which is exposed to the heat to be measured, while the second aperture is kept at a uniform lower temperature. This style of pyrometer is made to indicate and record and the indicating and recording mechanism can be placed at a distance from the main instrument.

TABLE 134.

TYPES OF THERMOMETERS IN GENERAL USE.

Principle of Operation.		Type.	Range in Deg. Fahr. for which they can be used.
Expansion	Those depending on the change in volume or length of a body with temperature.	Gas	−400 to +2900
		Mercury, Jena glass, and nitrogen.	−35 to +950
		Glass and petrol ether.	−325 to +100
		Unequal expansion of metal rods.	0 to 950
Transpiration and viscosity.	Those depending on the flow of gases through capillary tubes or small apertures.	The Uehling	0 to 2900
Thermo-electric	Those depending on the electro-motive force developed by the difference in temperature of two similar thermo-electric junctions opposed to one another.	Galvanometric	−400 to +2900
Electric resistance	Those utilizing the increase in electric resistance of a wire with temperature.	Direct reading on indicator or bridge and galvanometer.	−400 to +2200
Radiation	Those depending on the heat radiated by hot bodies.	Thermo-couple in focus of mirror.	300 to 4000
		Bolometer	−400 to Sun
Optical	Those utilizing the change in the brightness or in the wave length of the light emitted by an incandescent body.	Photometric comparison.	1100 to Sun
		Incandescent filament in telescope.	
		Nicol with quartz plate and analyzer.	
Calorimetric	Those depending on the specific heat of a body raised to a high temperature.	Platinum ball with water vessel.	32 to 3000
Fusion	Those depending on the unequal fusibility of various metals or earthenware blocks of varied composition.	Alloys of various fusibilities. (Seger cones.)	32 to 3350

Table 134 embodies in outline the principles and temperature ranges of the various types of thermometers in use. Temperature ranges verified by U. S. Bureau of Standards.

Modern Methods of Temperature Measurements: Cassier's Mag., June, 1909, p. 99. *High Temperature Measurements:* Eng. and Min. Jour., Sept. 2, 1911, p. 447; Power, Aug. 2, 1910, p. 1376; Engineering, Feb. 9, 1912, Bul. No. 2, Bureau of Standards.

411. Power Measurements.—Instruments for the measurement of power may be divided into two general classes, *direct* and *indirect.* The former involve the direct measurement of force and linear velocity or torque and angular velocity and the latter give the equivalent in other forms of energy. Direct power measuring appliances include the various speed indicators, transmission and absorption dynamometers, and the indirect include ammeters, voltmeters, watt-hour meters, boiler flow meters, and the like. In all power measurements the time or speed factor is readily determined but the force or torque factor, or equivalent, often involves considerable labor and the use of costly and complicated apparatus. The various conversion factors for the measurement of work, power, and duty are given in Appendix F.

412. Measurement of Speed. — The following chart gives a classification of a number of well-known instruments for determining linear and angular velocities.

Counters	Hand	Worm and Wheel.
	Continuous	Gear Train. Electrical.
Tachometer or Speed Indicators	Centrifugal	Weights. Liquids.
	Electrical.	
	Resonance	Frahm's.
Chronograph	Electromagnetic Tuning Fork.	

The most commonly used device for speed determinations is the *hand speed counter,* consisting of a worm, worm wheel, and indicating dials. The errors to be corrected are principally those due to slipping of the point on the shaft, and to the slip of the gears in the counting device in putting in and out of operation. In some of the better grade of instruments the gears are engaged or disengaged with the point in contact with the shaft. In the latter design a stop watch, actuated by the disengagement gear, minimizes the error likely to occur in hand manipulation.

The *continuous counter* consists of a series of gears arranged to operate a set of indicating dials. It may be operated by either rotary or reciprocating motion. The rate of rotation is calculated from the readings of the counter.

All *tachometers* indicate directly the speed of the machine to which they are attached and are independent of time determination. The most commonly used devices depend upon the centrifugal force of revolving weights for their operation. The indicating needle is attached to the weights in such a manner that the number of revolutions per minute is read directly from the position of the needle on the dial. These instruments should be calibrated for accurate work because of the number of wearing parts.

Liquid tachometers consist essentially of small centrifugal pumps discharging into a vertical tube. The height of the indicating column is a function of the speed of rotation.

Electrical tachometers are miniature dynamos, the voltage being a measure of the speed of rotation. These instruments are accurate and readily attached but necessitate the use of a delicate and costly voltmeter. The indicating mechanism may be placed at any distance from the small dynamo and in this respect has a marked advantage over the other types of speed indicators.

The *resonance tachometer* affords a convenient method of measuring speeds over a wide range. It consists of a number of steel reeds of different periodicity mounted side by side on a suitable frame. When used to measure the speed of an engine or turbine the instrument is placed on or near the bed plates and the slight under or over balance causes the proper reed to vibrate in unison.

413. Steam-engine Indicators. — This subject has been extensively treated by various authorities and a general discussion would be without purpose. For *indicated horsepower, testing indicator springs,* and analysis or indicator diagrams see "Rules for Conducting Steam Engine Tests," A.S.M.E. Code of 1915.

414. Dynamometers. — Dynamometers for measuring power are of two distinct types, *absorption* and *transmission.* In the former the power is absorbed or converted into energy of another form while in the latter the power is transmitted through the apparatus without loss, except for minor friction losses in the mechanism itself.

The ordinary *Prony Brake* is the most common form of absorption dynamometer. In the various forms of Prony brakes the power is absorbed by a friction brake applied to the rim of a pulley. For low rubbing speeds and comparatively small powers it affords a simple and inexpensive means of measuring the actual output.

The Alden absorption dynamometer is a successful form of friction brake and has a wide field of application. It has been constructed in large sizes and is adapted to all practical ranges of speed. For a description of rope brakes and the Alden absorption dynamometers see Appendix No. 19, p. 179, A.S.M.E. Code of 1915.

Water brakes are finding much favor with engineers for high-speed service. There are two types, the Westinghouse and the Stumpf. In the former the rotor consists of a simple drum with serrated periphery revolving in a simple casing, the inner surface of which is serrated in a manner similar to the rotor. The resistance is produced by friction and impact, and the power is converted into heat which is carried away by the circulating water. The casing is free to turn about the shaft but is held against rotation by a lever arm. The torque of the lever arm is determined as in a Prony brake. A brake of this design, 2 feet in diameter and 10 inches wide, will absorb about 3000 horsepower at 3500 r.p.m. In the Stumpf type the rotor consists of a number of smooth disks mounted side by side on a common shaft. The casing is divided into a number of compartments corresponding to the division of the rotor. There is no contact between rotor and casing. The friction between the disks and water and the water and casing tends to rotate the latter and the torque is measured in the usual way. In either type the power output is readily controlled by the water supply.

Pump brakes and *fan brakes* are also used as absorption dynamometers. The latter are commonly used in connection with automobile engine testing.

Electromagnetic brakes are occasionally used for power measurements. They consist essentially of a metal disk or wheel revolving in a magnetic field. The resistance or drag tends to revolve the field casing and the torque is measured in the usual way.

An electric generator mounted on knife edges forms the basis of the Sprague electric dynamometer. The prime mover drives the armature of the generator and the reaction between armature and field is counterbalanced by suitable weights. The output is conveniently regulated by a water rheostat.

Transmission dynamometers are seldom used for testing prime movers and are ordinarily limited to small power measurements. In some instances, however, as in marine service, transmission dynamometers afford the only practical means of approximating the net power delivered to the propeller. For comparatively small power measurements may be mentioned the Morin, Kennerson, Durand, Lewis, Webber, and Emerson transmission dynamometers, and for large powers, the Denny and Johnson electrical torsion meter and the Hopkinson optical torsion meter. For detailed descriptions of these appliances consult "Experimental Engineering," Carpenter and Diederichs, Chap. X.

415. Flue Gas Analysis. — It has been shown (paragraph 22) that the products of combustion, commonly called flue gases, resulting from the complete oxidation of coal with theoretical air supply consist chiefly of nitrogen and carbon dioxide, with lesser amounts of water vapor and sulphur dioxide. It was also shown that with a deficient air supply the flue gases may contain carbon monoxide and varying amounts of hydrocarbon. If excess air was used in the combustion of the fuel free oxygen would be present in the gases. Evidently an analysis of the flue gases offers a basis for judging the efficiency of combustion. The first step in the analysis and the most important one is the obtaining of a representative sample. Since the gases in the breeching and flues may be far from homogeneous great care must be exercised in getting a true average sample. (See *Apparatus and Methods for Sampling and Analysis of Furnace Gases*, U. S. Bureau of Mines, Bul. No. 12, 1911.)

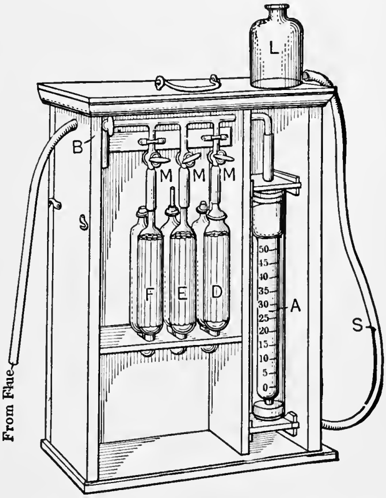


Fig. 583. Standard Orsat Apparatus for Flue Gas Analysis.

The analysis as ordinarily made in commercial practice is called volumetric, although in reality it is based upon the determination of partial pressures. According to Dalton's laws when a number of gases are confined in a given space each gas occupies the total volume at its own partial pressure, and the total pressure is the sum of all the partial pressures. When one of the gases is absorbed by a suitable medium and the remaining gases are compressed back to the original total pressure, a volume decrease is found, and if the temperature remains constant this decrease represents the volume absorbed.

The apparatus usually employed for volumetric analysis consists of a graduated measuring tube into which the gases are drawn and accurately measured under a given pressure, and a series of treating tubes, containing the necessary absorbing reagents, into which they are transferred until absorption is complete. The *Orsat apparatus*, Fig. 583, forms the basis of nearly all of the portable appliances on the market for analyzing flue gases and the ordinary products of combustion. In this apparatus a measured volume, representing an average sample of the gas, is forced successively through pipettes containing solutions of caustic potash, pyrogallic acid and cuprous chloride in hydrochloric acid, respectively, thus absorbing the carbon dioxide, the oxygen and the carbon monoxide, the contraction of volume being measured in each case. The apparatus as originally constructed is bulky and fragile and slow in its absorption of gas.

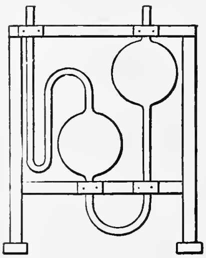


Fig. 584. Hempel Pipette.

The Hempel Apparatus works on the same principle as the simple form of Orsat apparatus described, so far as the latter is applicable, excepting that the absorption may be hastened by shaking the pipettes bodily, bringing the chemical into most intimate contact with the gas. It is less portable and in some particulars it requires more careful manipulation than the Orsat, while for general analysis it is not adapted unless used in a well-equipped chemical laboratory. The absorption pipettes are made in sets which are shaped in the form of globes, and a number of independent sets are required for the treatment of the different constituent gases. A simple pipette of the Hempel type is shown in Fig. 584.

The Williams Improved Gas Apparatus is a marked improvement over the standard Orsat in that the objections cited above are obviated. In addition to the elimination of these objectionable features

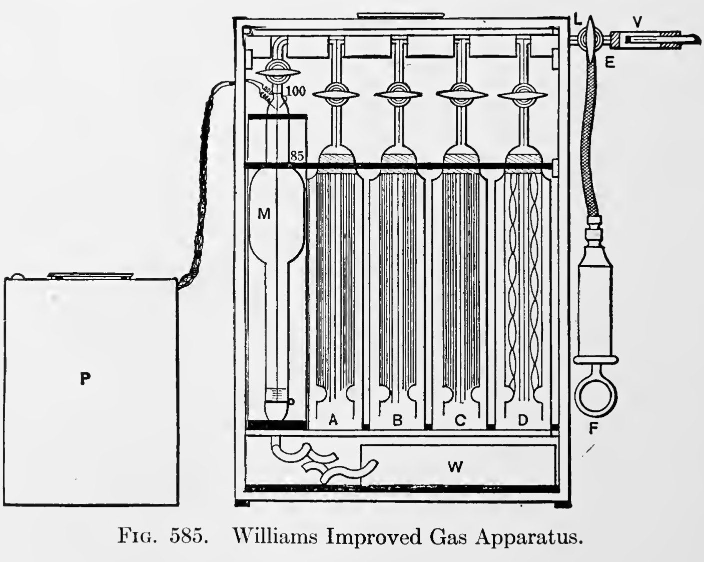


Fig. 585. Williams Improved Gas Apparatus.

provision is made in the "Model A" type for the determination of illuminants, hydrogen and methane along with the three gases mentioned above. Referring to Fig. 585, *A*, *B*, *C*, and *D* are pipettes containing the necessary reagents for absorbing, respectively, CO_2, illuminants, O_2, and CO. *M* is a graduated measuring flask connected at the bottom with water-level bottle *W* and at the top with the various pipettes. *F* is a hard rubber pump for taking gas sample directly from the source of supply, thereby eliminating the inaccuracy and annoyance of collection over water and transference. *P* is a portable case containing a spark coil and batteries for exploding the methane and hydrogen remaining in the burette after the other constituents have been removed. When extreme accuracy is desired mercury is used as the displacement medium in the leveling bottle since water absorbs CO_2 to a certain extent. For a complete description of this apparatus with sample calculations see paper read by F. M. Williams before Division of Industrial Chemists and Chemical Engineers, American Chemical Society, Washington, D. C., Dec. 28, 1911.

"Little" Modified Orsat Apparatus. — Fig. 586 illustrates a modified Orsat apparatus as used by the Arthur D. Little Company of Boston,

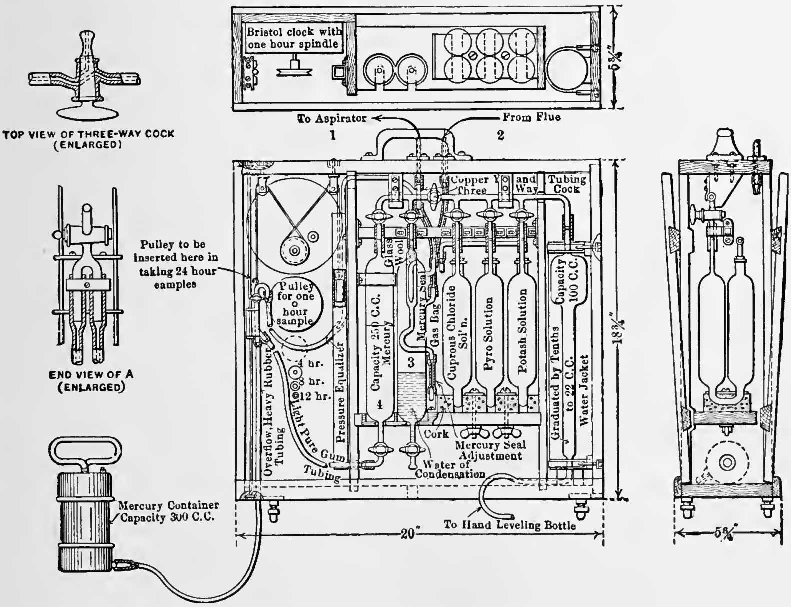


Fig. 586. Modified Orsat Apparatus. — Arthur D. Little Co.

Mass. The right half of the device is the ordinary Orsat apparatus and the left portion constitutes the sampling attachment. The gases are drawn from the source of supply through rubber tube (2) into the sampling pipette (3) and out through rubber tube (1) to the aspirator. The latter may be operated by steam or water. When a sample is being collected the three-way cock on the glass header is closed and the mercury in the sampling tube (4) is allowed to drain through the movable overflow into the mercury retainer. The overflow is lowered at a constant rate by clockwork. Two driving pulleys afford seven different rates of movement downward of the overflow, thereby enabling a continuous sample to be collected at constant rate over any period from ½ to 24 hours. Instantaneous samples may be drawn off and analyzed as often as desired and with practically no delay to the continuous sample. For further details see Power, July 16, 1912, p. 77.

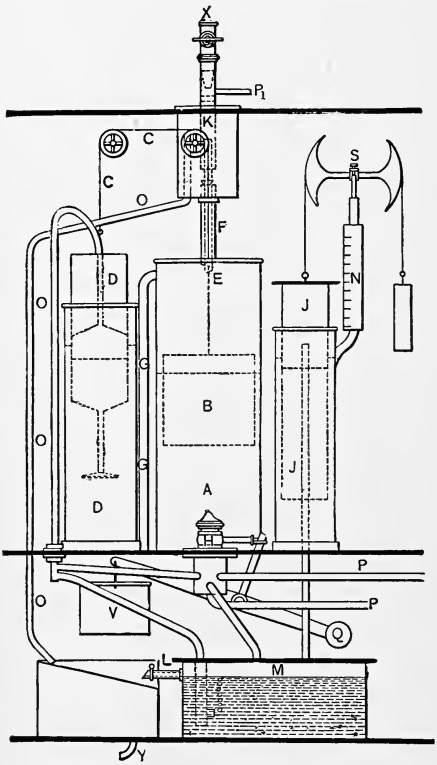


FIG. 587. Simmance-Abady CO_2 Recorder.

For many practical purposes it is sufficient to determine the carbon dioxide. A number of satisfactory appliances are on the market which give continuous autographic records of the percentage of CO_2 on clock-driven charts. These devices, however, are rather expensive and usually beyond the appropriation of small boiler plants.

Simmance-Abady CO_2 Recorder. — Fig. 587 illustrates the general principles of the *Simmance-Abady CO_2 Recorder.* The operation is as follows: A continuous stream of water enters reservoir *K* through inlet *X* and overflow at *O.* A portion of the stream flows into tank *A* through pipe *F* and causes bell float *B* to rise. As the float rises it permits bell *D* of the extractor to fall. When float *B* reaches the top of its stroke it raises valve stem *E,* trips the valve and causes the water to siphon out of tank *A* through siphon tube *G.* The lowering of the water level allows the bell to sink. As it falls it draws up the water-sealed extractor bell *D* and creates a partial vacuum under the latter. Flue gas then flows from the source of supply through *P* and *H* into the bell. The mass of water discharged from siphon tube *G* into the small vessel *V* beneath it overcomes the counterweight *Q* and closes the balance valve H, thereby entrapping a fixed volume of gas in the extractor bell. The stream of water which is continually flowing into tank A causes the float B to rise and the bell D to sink, as before. The lowering of bell D forces the entrapped flue gas through the caustic potash solution in vessel M into water-sealed recorder bell J. The displacement of bell J will be less than that of bell D by the volume of CO_2 absorbed in vessel M. The percentage of CO_2 in the flue gas is thus indicated by the position of the bell J with reference to the graduated scale N. The pen mechanism is attached to bell J and records the percentage of CO_2 by the length of lines on a clock-driven chart. These samples are analyzed and the lines are drawn at three-minute intervals. The small water aspirator at X is for the purpose of exhausting gas continuously from the pipes connecting the recorder to the boiler, thereby insuring true samples at the time of absorption. Auxiliary pipe P is connected to main gas lead P.

The Uehling Composimeter is another successful instrument for continuously recording the percentage of CO_2 in the flue gas. The principles of this apparatus are illustrated in Fig. 588. The device consists primarily of a filter, absorption chamber, two orifices, A and B, and a small steam aspirator. Gas is drawn from the usual source by means of the aspirator through a preliminary filter located at the boiler, and then through a second filter as illustrated in the diagram. From the latter the gas passes through orifice A, thence through the absorption chamber and orifice B to the aspirator where it is discharged. The CO_2 is absorbed by the caustic potash solution in the absorption chamber. This reduces the volume and causes a change in tension between the two orifices in proportion to the CO_2 content of the gas. This variation in tension is indicated by the water column, as shown, and is transmitted by suitable piping to the recording mechanism which may be placed at a considerable distance from the boiler room.

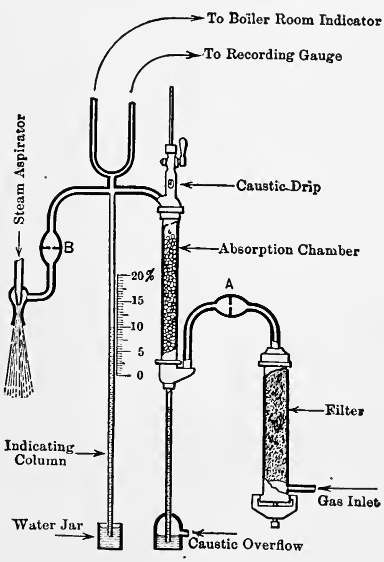


FIG. 588. Principles of the Uehling Gas Composimeter.

416. Moisture in Steam. — Several forms of calorimeters are available for determining the quality of steam. The simplest as well as the most satisfactory, if the percentage of entrained moisture is not beyond its range, is the *throttling* calorimeter, Fig. 589. In this device the sample of steam, which is taken from the steam pipe by means of the perforated nipple, is allowed to expand through a very small orifice into a chamber open to the atmosphere. The excess of heat liberated serves first to evaporate any moisture present and then to superheat the steam at the lower pressure. From the observed temperature and pressures it is easy to calculate, with the aid of steam tables, the percentage of moisture in the original sample.

The limit of the throttle calorimeter depends upon the steam pressure and is about 3 per cent of moisture at 80 pounds pressure and about

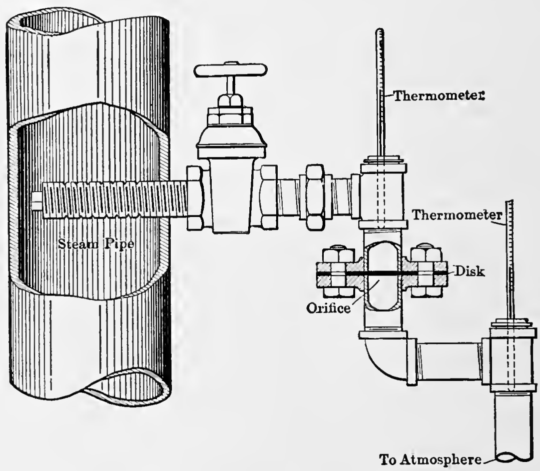


FIG. 589. A Typical Throttling Calorimeter.

5 per cent at 200 pounds. For steam containing greater percentages of moisture the *separating* calorimeter, Fig. 590, is sometimes used. This instrument is virtually a steam separator and mechanically separates the moisture from the sample of steam. The water thus separated collects in a reservoir provided with gauge glass and graduated scale, while the dry steam passes through an orifice to the atmosphere. The weight of dry steam per unit of time is indicated on the gauge, calculated according to Napier's rule, or may be determined by condensing and weighing. The accuracy of the moisture determination is greatly affected by the difficulty of obtaining true samples of steam containing large percentages of moisture.

Fig. 591 shows the Ellison *universal* steam calorimeter, which combines the separating and throttling principles and is adapted to steam of any degree of wetness. The separating chamber is provided with a gauge glass, not shown, for indicating the weight of water which accumulates only when the steam is too wet to be superheated.

Throttling Calorimeters: Power, Dec., 1907, p. 891; Trans. A.S.M.E., 17–151; 175, 16–448; Engr. U. S., Feb. 15, 1907, p. 219.

Separating Calorimeters: Trans. A.S.M.E., 17–608; Engr. U. S., Feb. 15, 1907, p. 219.

Universal Calorimeter: Trans. A.S.M.E., 11–790.

Thomas Electrical Calorimeter: Power, Nov., 1907, p. 791.

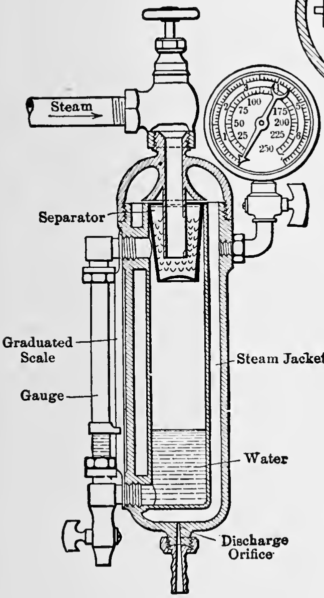


Fig. 590. Carpenter Separating Calorimeter.

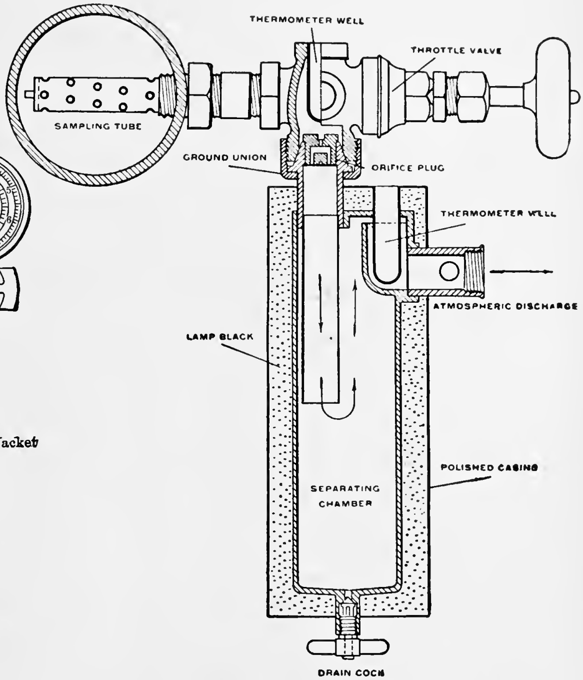


Fig. 591. Ellison Universal Steam Calorimeter.

417. Fuel Calorimeters. — The analysis and heat evaluation of fuel require considerable time and skill and much costly apparatus, hence in most power plants it is customary to depend upon a specialist to whom samples are submitted from time to time. In many large stations, however, the conditions often warrant the establishment of a testing laboratory equipped for the proximate analysis of coal and the determination of the calorific value of the solid, liquid or gaseous fuel used. The *Mahler bomb* calorimeter illustrated in Fig. 592 is the most accurate and satisfactory device for solid and liquid fuels but is comparatively expensive. The instrument consists of a steel shell or "bomb" of great strength, lined with porcelain or platinum, into which a weighed sample of the fuel is introduced and burned on a platinum pan in the presence of oxygen under a pressure of about 300

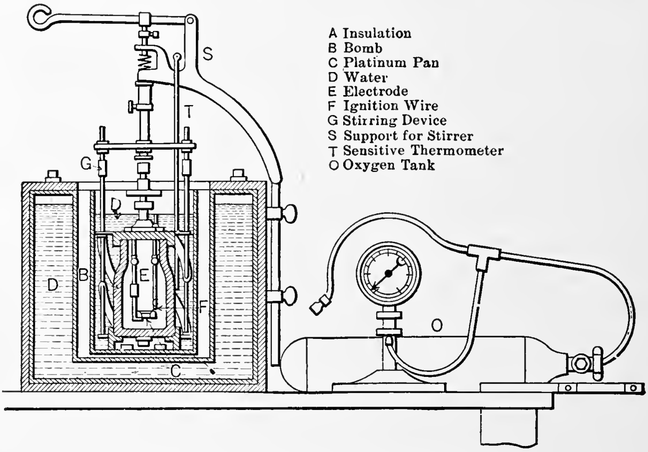


FIG. 592. Mahler Bomb Calorimeter.

pounds per square inch. The charge is ignited by an electric current. During combustion the bomb is submerged in a known weight of water which is kept constantly agitated. The calorific value is calculated from the observed rise in temperature due to the heat evolved, proper corrections being made for the water equivalent of bomb and appurtenances, heat given up by the igniting current, and for radiation or absorption of heat from the surrounding air.

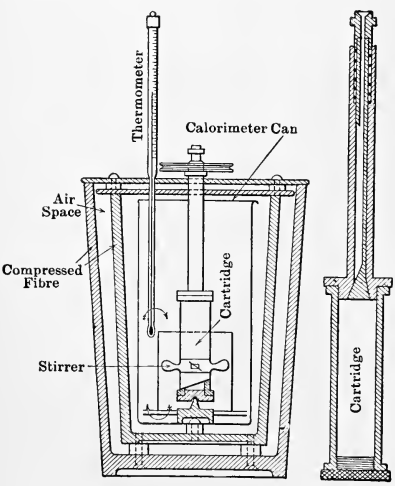


FIG. 593. Parr Fuel Calorimeters.

The *Parr* calorimeter, Fig. 593, is an inexpensive instrument, very simple in operation, and gives results which are sufficiently accurate for all practical purposes. The weighed sample of coal, together with a quantity of sodium peroxide which supplies the oxygen for combustion, is introduced into the cartridge. Means are provided for rotating the cartridge when submerged in the calorimeter, the attached vanes agitating the water to maintain uniform temperature. The charge is fired either electrically or by introducing a short piece of hot wire through the conical valve. The calorific value is calculated from the observed rise in temperature and the constants of the instrument. Among other forms of instruments, in more or less general use and which give very satisfactory results, may be mentioned the *Carpenter*, *Thompson*, *Atwater* and *Emerson* calorimeters.

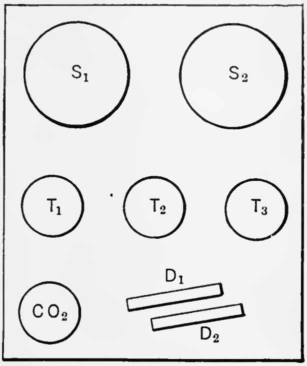


Fig. 594. Individual Boiler Control Board.

Comparison of Different Types of Calorimeters: Jour. Soc. Chem. Ind. (1903), 22–1230.

418. Boiler Control Boards. — In the modern large central station efficient operation of the various units composing the plant is greatly facilitated by grouping the testing instruments on a control board and by placing this board where it can be conveniently studied by the operating engineer. Fig. 594 shows the individual control board as installed before each boiler unit in the Northwest plant of the Commonwealth Edison Company of Chicago, and Fig. 595 shows the section control board for each turbine unit. The individual control board is mounted on the front of the boiler casing and the section board is placed at the end of the battery of boilers near the wall dividing the boiler from the turbine room. With reference to Fig. 594 the two instruments at the top are steam flow meters — one on each steam lead — with indicating, recording and integrating attachments. These meters show the amount of steam delivered at any time by the boiler and gives a complete record of its delivery. The three recording gauges below show the temperature in uptake from the boiler, the temperature of the feed water leaving the economizer and entering the boiler and the temperature of the flue gases leaving the economizers. Below and at the left is a CO_2 recorder, while at the right-hand corner are two indicating draft gauges, one connected to the furnace and the other to the uptake. With reference to the section control board, the two flow meters at the top measure the steam input to the turbine and the feed water input to the boilers, respectively. The recording thermometers immediately

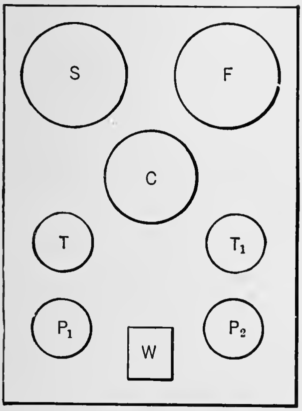


Fig. 595. Boiler Section Control Board.

below show the temperature of the steam entering the turbine and the temperature of the feed water entering the economizer, respectively. Below these are two recording pressure gauges showing the pressure on the steam header and on the boiler feed header, respectively, while in the center of the board is a clock and below that an indicating wattmeter showing the output of the turbo-generator unit which is direct connected to these boilers. Where automatic coal-weighing devices are in use the individual control board includes the fuel measuring dials. By the use of these instruments a very complete check is obtained of the performance of individual boilers and of the entire unit.

CHAPTER XVIII

FINANCE AND ECONOMICS.—COST OF POWER

419. General Records. — In all power plants, public or private, an itemized record of plant performance and cost of operation is of vital importance for the most economic results. In many states public utility corporations are required to submit an annual statement covering the various details of operation, and in order to insure uniformity ruled and printed forms are furnished by the state. The private plant owner, on the other hand, is free to use his own judgment and may adopt any system of cost accounting or dispense with them entirely.

The principal objects of keeping a system of records are (1) to enable the owner to compare the performance of his plant from time to time and to show him exactly what his plant is costing him, and (2) to enable the engineer to analyze the various records with a view of reducing all losses to a minimum. Power-plant records to be of value must be closely studied with a view to improvements. The mere accumulation of data to be filed away and never again referred to is a waste of time and money.

Records should cover not only the daily, monthly, and yearly operation of the plant but also, as permanent statistics, a complete analysis of each item of equipment. The value of such data cannot be overestimated. The engineer will frequently find it greatly to his interest to have available at a moment's notice the complete details of his engines, boilers, generators, and other machinery, especially when it is required to renew a broken or worn-out part in case of emergency.

The question of whether to purchase power or to generate it depends, chiefly, upon the relative cost of the two methods, although the absence of power-plant machinery and freedom from the coal- and ash-handling nuisance may be important factors. There is no doubt that the central station can generate power cheaper than the small isolated plant, but in most cases it is a question not only of power, but also of supplying steam for heating and other purposes, and a careful study of all of the items entering into the problem is necessary for an intelligent choice. The service department of the large central station with its carefully maintained system of records has a strong advantage in presenting its arguments over the average private plant with its ill-kept and faulty system of accounting, and in some instances central-station service has been adopted simply because the engineer in charge was not in a position to prove positively that his own plant was the better investment. R. J. S. Pigott, Jour. A.S.M.E., Dec. 1916, p. 947, shows the effects of modifying the operating conditions of power plants, and of changing the character of the auxiliary equipment by means of graphic analysis. From the study of such an analysis the cost of producing power for given conditions may be determined with little effort, and the effects of changes in the conditions or equipment may be predetermined with accuracy.

TABLE 135.

PERMANENT STATISTICS.

GENERAL INFORMATION.

Date of installation	....	Total cost of building	$5,000,000
Type of building	Office	Ground plan	191×231
Number of floors	18	Total office floor space, sq. ft.	400,000
Number of offices	900	Height of building	280
Volume of building, cu. ft.	10,860,000	No. of sides exposed	3
Type of heating system	Webster	Radiator surface, sq. ft.	100,000
Engine room, sq. ft.	6,840	Boiler room, sq. ft.	5,400
Height of chimney, ft.	318	Number of elevators	22
Draft, inches of water	3.5	Type of elevators	High pressure hydraulic
Kind of grate or stoker	Jones Underfeed	Capacity of elevators, lb., each	2,700
Kind of coal	Ill. screenings	Boiler pressure	150
Coal storage capacity, tons	450	Back pressure	Atmospheric
Capacity ice plant, tons in 24 hrs.	50	Part of bldg. lighted	All
Capacity storage battery, am. hrs.	None	Total cost of mechanical plant	$650,000

	Engines.	Generators.	Motors.	Boilers.
Type	Ball compd.	Crocker-Wheeler		
Number installed	5	5	25	5
Rated capacity	250 h.p.	150 kw.		375 h.p.

LIGHTS.

	Incandescent.		Arcs.
Type	Carbon	Tungsten	Inclosed
Number installed	150	30,000	15

A number of attempts have been made to standardize power-plant records but the results have been far from satisfactory because of the wide range in operating conditions. Each installation is a problem in itself and the items to be recorded must necessarily depend upon the size and character of the plant. A common mistake is to attempt too comprehensive a system with the result that after the novelty has ceased the labor of making the various entries becomes irksome, many of the items are omitted, guesses are substituted in place of actual observations, and the records are ultimately without value. A few properly selected items, accurately recorded, are of vastly more importance than an elaborate system of records indifferently maintained.

Walter N. Polakov, Jour. A.S.M.E., Dec., 1916, p. 966, has proposed a "standardization of power plant operating cost" by means of which the owners of power plants can judge, without the necessity of going into technical details themselves, how close the actual performance of the plant is to the possible minimum cost at any time or under any circumstances, all variable factors beyond operating control being automatically adjusted. Mr. Polakov shows the futility of attempting to judge any one plant by the performance of others having a different kind of equipment or of a different nature of service. Even where conditions appear identical such comparisons do not offer a true measure of excellence. It is not so important to know that one's plant is better than another as to know whether it is as good as it can be. Mr. Polakov shows how this can be determined by the use of curves of "standard costs" the plotting and application of which are explained in his paper before the American Society of Mechanical Engineers.

420. Permanent Statistics. — Tables 135 to 138 are taken from the records of a large isolated station in Chicago and serve to illustrate the make-up of the "permanent statistics." The complete file covers each item of equipment and includes the various drawings, specifications, and guarantees for the entire mechanical equipment. Since these records do not vary with the operation of the plant they require no further attention, once they are compiled, except of course for such changes as may be made from time to time in the plant itself.

420a. Operating Records. — The operating records of any plant bear the same relationship to the economical operation of that plant as the bookkeeping and cost accounting system bears to the manufacturing plant. The distribution of profit and loss in either case can only be obtained by itemizing the various factors involved and by grouping them in such a manner as to show at any time where improvement is possible. Commercial bookkeeping has been more or less standardized and entails very little need of originality on the part of the bookkeeper, but the selection and maintenance of a system of power-plant records may require considerable study and experimenting, since each installation is a problem in itself. The items included in the different forms depend upon the apparatus provided for weighing the coal and water,

TABLE 136.

PERMANENT STATISTICS.

BOILERS.

Make of boiler	Stirling
Total number in plant	5
Date of installation	....
Steam pressure, gauge	150
Safety-valve pressure	160
Type of safety valve	Pop
Area of grate, sq. ft	....
Heating surface, sq. ft	3,500
Superheating surface, sq. ft	None
Number of steam drums	3
Diameter of steam drums, in.	36
Distance between steam drums, ft	3
Thickness of shell, in.	$\frac{3}{4}$
Thickness of head, in.	$\frac{3}{4}$
Diameter of steam nozzle, in.	10
Diameter of safety valve	2–4 in.
Diameter of blow-off, in.	2.5
Diameter of feed pipe, in.	2
Temperature of flue, deg. Fah	450–490
Temperature of feed water, deg. Fah	210
Ratio of heating surface to grate area	41.6
Kind of fuel	Carterville, Ill., Screenings
Type of grate	Green chain grate
Rated horse power	375
Number in battery	1
Weight of boiler	62,186
Cost of boiler and fittings (each)	$5,400
Height of setting	17 ft. 9 in.
Length of setting	17 ft. 4 in.
Width of setting	15 ft. 3 in.
Weight of setting	272,000
Thickness of wall	Side 20 in.; back, 15 in.
No. of bricks, fire	6,590
No. of bricks, common	19,600
Dimensions of foundation	15 ft. 2 in. × 17 ft. 4 in.
Material of foundation	Stone and concrete
Cost of foundation and setting (each)	$1,500
Distance between batteries	4 ft. 6 in.
Distance back of boiler	17 ft. 6 in.
Distance in front of boiler	16 ft. 6 in.
Distance overhead	2 ft. 10 in.
Number of tubes	337
Diameter of tubes, in.	3.25
Length of tubes, ft.	12 to 14
Steam space, cu. ft.	96
Water space, cu. ft.	643
Kind of draft	Forced
Inches of draft (maximum)	3.5

TABLE 137.

PERMANENT STATISTICS.

FEED PUMPS.

Date of installation	
Make	Snow
Number in plant	2
Height, ft.	3
Length, ft.	12
Width, ft.	4
Weight of pump	5 tons
Cost, each	$965
Steam pressure	150
Back pressure	$\frac{1}{2}$
Number of valves	32
Character of valves	Rubber, brass lined
Area thro' valve seats, sq. in., per pump	12.13
Gallons of water per min., per pump	800
Pounds of water per 24 hrs., average, actual	479,400
Gallons of water per 24 hrs.	599.2
Volume of air chamber, cu. ft.	3
Shop number	24,572–3
Diameter of steam cylinder	16
Diameter of water cylinder	10
Stroke	12
Displacement per stroke, cu. ft.	0.5454
No. of strokes per min., average	12
Diameter of suction	8
Diameter of discharge	5
Diameter of steam pipe	2.5
Diameter of exhaust	4
Diameter of steam drips	$\frac{1}{2}$
Diameter of water drains	$\frac{1}{2}$
Suction head, lb. per sq. in.	$1\frac{1}{2}$
Discharge head, lb. per sq. in.	175
Kind of piston packing	Outside packed plunger
Size of piston packing	....
Kind of rod packing	Soft
Size of rod packing	$\frac{5}{8}$
Temperature of feed water	214

the type and number of instruments available for measuring temperature, pressure, and power, and the system adopted for keeping track of oil, waste, general supplies, and repairs. In large stations autographic recording and integrating appliances, which are to be found in nearly all strictly modern stations and represent but a small part of the first cost of the plant, greatly reduce the labor of keeping continuous records. In small plants the cost of autographic instruments may prove to be prohibitive and recourse must be had to the usual indicating devices. In the latter case, continuous records may be closely simulated by plotting the readings of the indicating appliances, say every 15 minutes, or even once every hour, and by connecting the points with a straight line. (See Figs. 601 to 606.) The oftener the readings are taken the smaller will be the error. Total quantities may be obtained by summing up the various items or by integrating the graphical chart by means of a planimeter. It is not sufficient to record monthly or yearly averages. Daily and even hourly records are absolutely essential for maximum economy. The various losses may be reduced to a minimum only by an intelligent analysis of daily records. A number of forms taken from the files of various power plants are reproduced in this chapter under the proper subheadings and serve to illustrate current practice.

Power Plant Records: Prac. Engr. U. S., Jan. 1, 1914, p. 80; March 1, 1912, p. 242; Jan. 1, 1912, p. 36; Power, May 28, 1912, p. 758; Nov. 11, 1913, p. 697.

Log Sheets at Delray Station: Power, Oct. 5, 1915, p. 182.

421. Output and Load Factor. — The output of a plant is usually stated in terms of the (1) average horsepower, or equivalent, for a given period of time. (2) Unit output — horsepower-hours, or equivalent.

When the plant is operating at practically constant load it is sufficiently accurate for most purposes to express the output in horsepower, or equivalent, per month or per year. When the output fluctuates as is the general case, it is best expressed in terms of unit output. For example, one horsepower per year, 24 hours per day, and 365 days per year is equivalent to $365 \times 24 = 8760$ horsepower-hours. If the full power is used throughout this time it matters little whether the charge is based on the *flat rate* (horsepower per year) or the *unit rate* (horsepower-hours); if, however, the power is used only half the time, the yearly cost per horsepower-hour will be just double.

The *yearly load factor* or simply *load factor* is the ratio of the *actual* yearly output to the *rated* yearly output measured on the twenty-four-hour basis. Thus:

$$\text{Load factor} = \frac{\text{Yearly output, horsepower-hours or equivalent}}{\text{Rated horsepower, or equivalent} \times 8760}.$$

The *curve load factor* or *station load factor* is the ratio of the yearly output to the rated output based upon the number of hours the plant is in actual operation. Thus, for an electric station:

$$\text{Curve load factor} = \frac{\text{Yearly output, kilowatt-hours}}{\text{Rated capacity} \times \text{hours plant is in operation}}.$$

Much confusion arises from the interpretation of the term "rated capacity." If rated below the maximum load it can sustain it is evident that a prime mover may operate with a load factor over 100 per cent, in which case the term is without purpose. The accepted definition of rated load in this connection is the maximum load which the prime mover can sustain continuously on a twenty-four-hour basis without overheating. Other definitions have been assigned to the term load factor and station factor, but the two stated above are more in accord with current practice.

In any plant the great desideratum is a high load factor with greatest return on the investment. All the factors of expense included in the cost of power are then operating at maximum economy. High peak loads and low average loads necessitate large machines which are but little used and greatly increase the fixed charges.

The *demand factor* is the ratio of the maximum demand to the connected load. There is a general tendency to overestimate the maximum electric demand, due, in a measure, to the possibilities of all the lights and motors being in use at one time. Practically speaking, such conditions are not likely to occur. Table 139 gives an idea of the value of the demand factor for various classes of service and may be used as a guide for determining the size of prime movers.

The *diversity factor* may be defined as the ratio of the sum of the individual maximum demands of a number of loads during a specified period to the simultaneous maximum demand of all these same loads during the same period. If all the loads in a group impose their maximum demands at the same time, then, the diversity factor of that group will be unity. See *Diversity and Diversity Factors,* Terrell Croft, Power, Feb. 6, 1917, p. 171.

TABLE 138.

LOAD FACTORS — LARGE STATIONS.

Plant.	Peak Load, Kilowatts (Thousands).	Yearly Output, Kw-hr. (Millions).	Yearly Load Factor, Per Cent.
Buffalo General Electric Company	65.5	299.3	57.0
Cleveland Electric Light Company	85.0	340.6	45.8
Duquesne Light Company	101.1	463.5	52.3
Edison Companies:			
Boston	80.5	238.5	33.7
Brooklyn	67.2	233.4	38.1
Commonwealth	369.7	1341.9	43.2
Detroit	130.2	546.9	47.8
New York	254.8	856.4	38.3
Southern California	60.9	300.0	56.0
Minneapolis General Electric Company	43.6	171.6	44.9
Philadelphia, Electric Company	142.3	444.8	35.6
Public Service, N. J.	174.0	608.0	39.8

TABLE 139.

CENTRAL STATIONS, DEMAND FACTORS.

Demand factors compiled by Commonwealth Edison Company of Chicago.

CLASS OF SERVICE.

	Demand Factor.
Lighting customers:	
Biliboards, monuments, and department stores	85.6
Offices	72.4
Residences and barns	60.0
Retail stores	66.3
Wholesale stores	70.1
Average	59.8
Motor customers:	
Offices	65.1
Public gathering places and hotels	28.7
Residences and barns	69.3
Retail stores	61.2
Wholesale stores and shops	58.2
Average	59.4

TABLE 140.

MAXIMUM DEMAND TABLE FOR INSTALLATIONS UNDER ONE KILOWATT CONNECTED LOAD.

Commonwealth Edison Co.

Connected Load.		Residence Lighting. (Monthly Basis.)		Commercial Lighting. (Monthly Basis.)	
Number of Sockets.	Wattage. Equivalent.	Kw-hr. at Full Rate.	Estimated Maximum Number of Sockets Used Simultaneously.	Kw-hr. at Full Rate.	Estimated Maximum Number of Sockets Used Simultaneously.
1	50	2	1	2	1
2	100	3	2	3	2
3	150	5	3	5	3
4	200	6	4	6	4
5	250	7	5	8	5
6	300	8	5	9	6
7	350	9	6	10	6.7
8	400	10	6.7	11	7.3
9	450	10	6.7	12	8
10	500	11	7.3	13	8.7
11	550	11	7.3	14	9.3
12	600	12	8	15	10
13	650	12	8	16	10.7
14	700	13	8.7	17	11.3
15	750	13	8.7	18	12
16	800	14	9.3	19	12.7
17	850	14	9.3	20	13.3
18	900	15	10	21	14
19	950	15	10	22	14.7

In alternating-current motor installations the Wright Maximum Demand Indicator is not applicable, so that the maximum demand is determined, except in special cases, by the following percentages of the rated capacity of the connected load:

	Per Cent.
Where installations are under 10 hp. and only one motor is used	85
Where installations are under 10 hp. and more than one motor is used	75
Where installations are from 10 hp. to 49 hp., both inclusive (irrespective of number of motors)	65
Where installations are 50 hp. or over (irrespective of number of motors)	55

TABLE 141.

TYPICAL OPERATING CHART.

DAILY POWER-HOUSE REPORT.

The United Light and Power Co.

..................Division

....19..

Weather — Noon

				Hr.	Min.
Engine No. 1	started...........M	stopped..........M	Total time run.........		......
Engine No. 2	started...........M	stopped..........M	Total time run.........		......
Inc. current on	..................M	off ....M	Total time on..........		......
Street arcs on	..................M	off..............M	Total time on..........		......

Noon AMPERE READINGS.

12 00	12 30	1 00	1 30	2 00	2 30	3 00	3 30	4 00	4 15	4 30	4 45	5 00	5 15	5 30	5 45	6 00	6 15	6 30
6 45	7 00	7 15	7 30	7 45	8 00	8 15	8 30	8 45	9 00	9 15	9 30	9 45	10 00	10 30	11 00	11 30	12 00	1 00
2 00	3 00	4 00	5 00	5 15	5 30	5 45	6 00	6 15	6 30	6 45	7 00	7 15	7 30	7 45	8 00	9 00	10 00	11 00

	Coal Received on Track.	Boilers in Service.
Coal used.....................lb.	Car No.........................	No. 1 from........m to.......m
Cylinder oil ....pt.	Initial.........................	No. 2 from........m to.......m
Engine oil ....pt.	Time placed ....m	No. 3 from........m to.......m
Waste.........................lb.	Time released................m	Washed No......................
Water......................cu. ft.	Weight......................lb.	Blew No........................
Carbons.........................	Ashes sold ....loads to	....
Globes..........outer....inner..		

Material Received for Power House Use.	Total Kilowatt Output. Read meter 12 o'clock noon
....	
....	Meter to-day............... Kw.
....	Meter yesterday............ Kw.
....	Diff..........................

Report here ANY interruption of service either arc or incandescent.

Time off..............Cause..

Arc lights out ....

Lights...

Location Reported by

TABLE 142.

TYPICAL OPERATING CHART.

(Large Chicago Department Store.)

..........................19..

Monthly Report.

Date.	Average Outside Temperature.	Fuel.					Supplies.			
		Coal.				Ash.	Oil Used, Gals.		Waste Pounds.	Total Water to Building, Cu. Ft.
		Kind.	Pounds Burned.	Cost Per Ton.	Cost Per Day.	Pounds Removed.	Engine.	Cylinder.		

Output.				Engine-Hours Run.					Boilers-Hours Run.						Breeching.	
Boilers.		Generators.														
Pounds of Water Evaporated.	Water Evaporated Per Lb. of Coal.	Ampere-Hours.	Kilowatt-Hours.	1	2	3	4	5	1	2	3	4	5	6	Draft.	Temperature.

Heating System.		Ventilating Plants, Hours Run.		Refrigerating Plant.			Repairs-Hours.		
Steam Pressure.	Live Steam-Hours.	Fan 1	Fan 2	Hours Run.	Gas Used, Pounds.	Ice Made, Pounds.	Engine Room.	Boiler Room.	Miscellaneous.

In the original copy all of these items are conveniently grouped on one large form ruled for 31-day entries with space at bottom for total quantities and costs. In the reproduction only the headings are included.

TABLE 143.

TYPICAL OPERATING CHART.

THE EDISON ILLUMINATING CO., *Detroit, Mich.*

TURBO-GENERATOR LOG

DELRAY POWER HOUSE NO. 2

For 24 hours ending midnight *Wednesday* *Dec. 9, 1914*

Day Date

Hour.	Barometer.	Turbine No. 6				Turbine No. 7.				Turbine No. 8.				Turbine No. 9.				Kilowatts Load.			Engineer on Watch.
		Started	*12-55*	*6-19*		Started	*6-12*			Started	*7-18*			Started	*5-14*						
		Stopped	*11-32*			Stopped				Stopped	*9-20*			Stopped	*12-15*						
		Running *17* hr. *46* min. Idle ... " *10* " Repair ... " ... "				Running *17* hr. *48* min. Idle ... " *22* " Repair ... " ... "				Running *14* hr. *2* min. Idle ... " *15* " Repair ... " ... "				Running *18* hr. *59* min. Idle ... " *22* " Repair ... " ... "				Power House No. 1.	Power House No. 2.	Total.	6:30 A.M. to 2:30 P.M. *Wm. Crawford* 2:30 P.M. to 10:30 P.M. *J. March* 10:30 P.M. to 6:30 A.M. *T. Callaghan*
		Pressures.				Pressures.				Pressures.				Pressures.							
		Throttle.	1st Stage.	Step.	Condenser in Abs.	Throttle.	1st Stage.	Step.	Condenser in Abs.	Throttle.	1st Stage.	Step.	Condenser in Abs.	Throttle.	1st Stage.	Step.	Condenser in Abs.				
A.M.																					Remarks.
1	*29 50*	*197*	*31*	*925*	*0 9*													*10 000*	*8 000*	*18 000*	*No. 9 turbine examined — there is no trace of oil working through anywhere.*
2		*200*	*42*	*950*	*0 9*													*7 000*	*8 900*	*15 900*	
3		*197*	*39*	*945*	*0.9*													*8 000*	*8 800*	*16 800*	
4		*196*	*40*	*948*	*0.9*													*6 800*	*8 800*	*15 600*	
5		*193*	*42*	*950*	*0 9*													*9 800*	*9 200*	*19 000*	
6		*200*	*42*	*950*	*1.0*									*194*	*40*	*975*	*0.9*	*12 800*	*19 400*	*32 200*	
7	*29.50*	*197*	*47*	*955*	*1.0*	*197*	*47*	*870*	*0 9*					*200*	*50*	*980*	*1.0*	*23 900*	*30 800*	*54 700*	*No. 9 Circulating engine and circulator. All bearings.*
8		*192*	*45*	*950*	*1 0*	*194*	*47*	*870*	*0.9*	*194*	*53*	*970*	*10*	*196*	*43*	*975*	*0.9*	*21 900*	*41 000*	*62 900*	

TABLE 144.

TYPICAL OPERATING CHART.

THE EDISON ILLUMINATING CO., *Detroit, Mich.* DELRAY POWER HOUSE NO. 2

CONDENSER ROOM LOG For 24 hours ending midnight *Wednesday Dec. 9,* 1914

Day Date

Hour.	Unit No. 6.				Unit No. 7				Unit No. 8.				Unit No. 9.				No. 8 Elect. B.F.P	Remarks.
	R.p.m. D. V. Pump.	*63*	R.p.m. Circul. Pump.	*160*	R.p.m. D. V. Pump.	*63*	R.p.m. Circul. Pump.	*162*	R.p.m. D. V. Pump.	*66*	R.p.m. Circul. Pump.	*151*	R.p.m. D. V. Pump.	*63*	R.p.m. Circul. Pump.	*162*		
	Water Temperatures.				Water Temperatures.				Water Temperatures.				Water Temperatures.					
	Entering Condenser.	Leaving Condenser.	Leaving Wet Pump.	Leaving B. F. Pump.	Entering Condenser.	Leaving Condenser.	Leaving Wet Pump.	Leaving B. F. Pump.	Entering Condenser.	Leaving Condenser.	Leaving Wet Pump.	Leaving B. F. Pump.	Entering Condenser.	Leaving Condenser.	Leaving Wet Pump.	Leaving B. F. Pump.		
A.M.																		
1	*39*	*49*															*156*	*No. 7 condenser*
2	*39*	*50*															*162*	*washed, tested.*
3	*39*	*50*	*63*														*166*	*Ferrel repacked.*
4	*39*	*50*	*62*														*149*	
5	*38*	*51*	*64*														*131*	
6	*38*	*51*											*38*	*50*			*170*	
7	*38*	*52*	*64*	*192*	*38*	*52*	*62*					*168*	*38*	*52*	*62*			
8	*40*	*53*	*64*	*190*	*40*	*54*	*64*		*40*	*53*	*60*	*172*	*40*	*53*	*62*			

TABLE 145.

TYPICAL WEEKLY OPERATING CHART.

THE EDISON ILLUMINATING COMPANY.

Delray Power Houses, Detroit, Michigan.

Date **Dec. 5-12 inc., 1914.**

Pounds of coal per kilowatt-hour delivered			**1.847**
B.t.u. per kilowatt-hour delivered		**25,102**	
Overall thermal efficiency, entire plant			**13.59**
Output:			
Generated	kw-hr.	**6,860,900**	
House Service,	kw-hr.	**170,500**	
Delivered	kw-hr.	**6,753,400**	
House Service to Total Generated	Per Cent		**1.574**
Average Output per Day	kw-hr.	**954,586**	
Coal:			
Coal in Bunkers, **Midnight, Dec. 5,**	lb.	**15,748,400**	
Coal to Bunkers, **Dec. 5–12 Inclusive**	lb.	**11,819,700**	
Coal Chargeable, **Dec. 5–12 Inclusive**	lb.	**27,568,100**	
Coal in Bunkers, **Midnight, Dec. 12th**	lb.	**15,094,000**	
Coal Consumed, **Dec. 5–12 Inclusive**	lb.	**12,474,100**	
Average Coal Consumed per Day,	tons	**891**	
Coal Analysis:			
Total Moisture	Per Cent		**2.953**
Ash	Per Cent		**8.190**
Heating Value, **As Fired (14004 dry)**	B.t.u.	**13,592**	
Ash Analysis:			
Carbon in Ash P. H. No. 1, **Stokers with periodic dumping**	Per Cent		**20.165**
Carbon in Ash P. H. No. 1, **Stokers with cinder grinders**	Per Cent		**10.770**
Carbon in Ash P. H. No. 2, **Stokers with periodic dumping**	Per Cent		**21.775**
Carbon in Ash P. H. No. 2, **Stokers with cinder grinders**	Per Cent		**9.775**

422. Cost of Power. General. — The actual cost of producing power depends upon the geographical location of the plant, cost of fuel and labor, the size of apparatus, the design, conditions of loading system, of distribution and the method of accounting. Comparisons based on the cost per hp-hr. or per hp-yr., or the equivalent are without purpose because of the many variables entering into the problem. It is impossible to intelligently compare costs or to obtain a true understanding of what costs for power really mean without a thorough knowledge of the various items entering into the unit cost such as costs of fuel, oil, waste, repairs, labor, insurance, taxes, management, distribution, maintenance and allowance for depreciation. In addition to these an understanding must be had of the operating conditions, such as size of plant, load factor, variation in load, ratio of the maximum load to the economic full load, number of hours a day the plant is operated and the like. With each plant having an individuality distinctly its own, in so far as the charges which go to make up the ultimate cost is concerned, it is practically impossible to arrive at any definite conclusion as to the manner in which the real cost of power may be correctly determined for purposes of comparison. Perhaps the best method of stating station economy is to give the average yearly heat units supplied by the fuel per kw-hr. delivered to the switchboard, and the load factor. This eliminates price and quality of fuel and offers a fairly satisfactory criterion of the efficiency of operation.

In any case the cost of power is based upon the expense which is independent of the output of *fixed charges* and that which is a function of the output or *operating costs.* In the small plant the items included in the fixed and operating costs are comparatively few in number and require but an elementary knowledge of bookkeeping, but in large industrial organizations or central stations the number of separate items to be considered may run into the hundreds and necessitate a complex system of accounting. Some idea of the different systems employed with examples of cost of power in specific cases may be gained from an inspection of Tables 151 to 160.

423. Fixed Charges. — These cover all expenses which do not expand and contract with the output. In the privately owned plant the fixed charges are usually limited to interest on the investment, rental, depreciation, taxes, insurance and sometimes maintenance, though the latter is ordinarily included in the operating costs. The accounting systems for public electric light and power companies are usually prescribed by the Public Utility Commission of the state in which the plant is located and the various charges must necessarily conform with the rules formulated by this Commission.

In any system the total fixed charges per year are constant irrespective of the load factor, since interest, taxes, depreciation, insurance, and maintenance go on whether the plant is in operation or not. The total fixed charges for a specific case are illustrated in Fig. 596 by a straight line. The cost per kilowatt-hour, however, decreases as the load factor increases. For example, with the plant operating continuously at rated load (100 per cent load factor) the fixed charges per kilowatt-hour are

$$\frac{65{,}000}{5000 \times 8760} = \$0.00148.$$

With 30 per cent load factor these charges are

$$\frac{65{,}000}{0.3\ (5000 \times 8760)} = \$0.00445 \text{ kilowatt-hour.}$$

The higher the load factor the greater is the amount of power produced and the longer does the apparatus work at best efficiency. But the greater the power produced the larger will be the fuel consumption and the oil and supply requirements. The labor charges will be practically constant. The total operating cost per year increases as the load factor increases, but not directly. (See Fig. 596.) The cost per

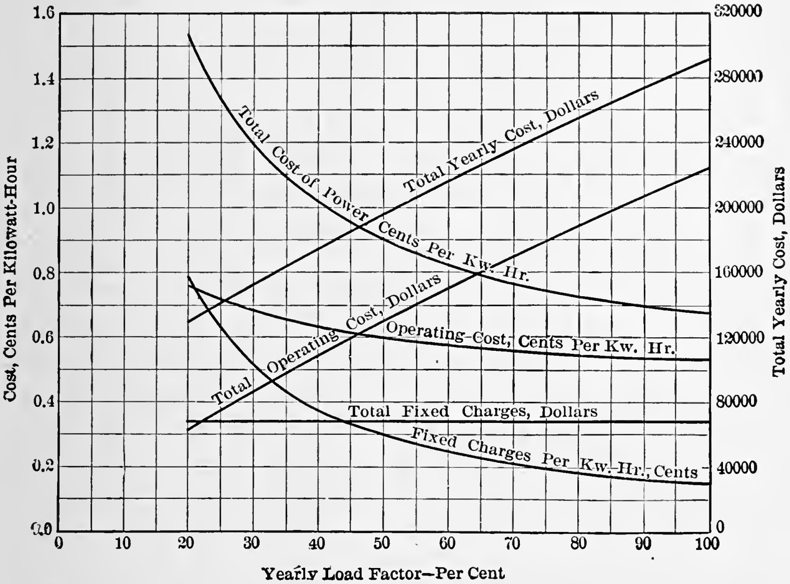


Fig. 596. Influence of Load Factor on the Cost of Power at the Switchboard. (5000-kilowatt Electric Light and Power Station.)

kilowatt-hour, however, decreases as the load factor increases. For example, the operating costs per year with plant operating continuously at full load are \$230,200. This gives

$$\frac{230{,}200}{5000 \times 8760} = \$0.00525 \text{ per kilowatt-hour.}$$

With 30 per cent load factor the yearly operating charges are \$87,890, which gives

$$\frac{87{,}980}{0.3\ (5000 \times 8760)} = \$0.0067 \text{ per kilowatt-hour.}$$

In general, the higher the load factor the greater becomes the ratio of the operating to the fixed charges, and extra investment may become advisable to secure the greatest economy possible.

On the other hand, when the load factor is low the fixed charges are the governing factor in the cost of power, and extra expenditures must be carefully considered, particularly if fuel is cheap.

Fixed Costs in Industrial Power Plants: Engineering Digest, Apr., 1911, p. 293.

TABLE 146.

AVERAGE INITIAL COST.*

Steam Engine Power Plants.

Horse Power.	Dollars per Horse Power.	Horse Power.	Dollars per Horse Power.
Simple Non-Condensing.			
10	225.00	60	180.00
20	200.00	70	177.00
30	195.00	80	175.00
40	190.00	90	170.00
50	185.00	100	165.00
Compound Condensing.			
100	170.00	700	76.00
200	146.00	800	69.00
300	126.00	900	64.00
400	110.00	1000	60.00
500	96.00	1500	58.00
600	85.00	2000	55.00
Triple Condensing.			
1000	62.00	4000	52.00
2000	58.00	5000	50.00
3000	54.00	6000	48.00

* Includes cost of buildings and entire equipment erected.

424. Interest. — The rates of interest on borrowed money vary with the nature of the security. In the case of power plants the form of security is usually a mortgage on the plant and equipment. If a builder has sufficient funds to construct the plant without borrowing, he should charge against the item "interest" the income which the sum involved would bring if placed out at interest or if invested in his own business. In estimating the interest charges 6 per cent of the capital invested is ordinarily assumed unless specific figures are available. Initial costs for various types of plants are to be found in the accompanying tables.

TABLE 147.

COST OF MECHANICAL EQUIPMENT — ISOLATED STATIONS.*

	Per Kilowatt of Plant Capacity.
Boilers (erected and set in masonry):	
Horizontal-tubular	$14–$18
Water-tube	16– 20
Steam engines:	
High-speed, simple direct-connected	20– 25
Medium-speed, compound non-condensing direct-connected	28– 35
Low-speed, compound condensing, belted	20– 25
Low-speed, simple, belted	25– 30
Gas engines	50– 60
Oil engines	75– 85
Gas producers	15– 20
Dynamos:	
Direct-connected to high-speed engine	13– 16
Belt-connected to engine	12– 15
Direct-connected to Corliss engine	16– 20
Switchboard	5– 10
Foundations	5– 10
Steamfitting—including auxiliary apparatus—such as feed heater, grease separator, exhaust head, tanks, covering, etc	20– 30

* By P. R. Moses before the A.I.E.E., Jan. 12, 1912.

425. Depreciation. — Depreciation may be defined as a decrease in value occasioned by wear or age, change of conditions rendering the plant inadequate for its particular functions, or changes in the art which renders it obsolete as compared with recent installations. Depreciation may be conveniently classified as:

Complete depreciation, or the gradual decrease in value occasioned by wear and age. This may be largely offset by maintenance.

Obsolescence, *inadequacy* or *destruction* by any cause. A thing is obsolete when it has been rendered valueless as the result of change in the art and this may occur where no physical deterioration has taken place. Inadequacy indicates that a thing is incapable of fully performing the function for which it is intended. It indicates neither physical depreciation nor obsolescence. Inadequacy may result from expansion of markets, community growth and the like. Obsolescence, inadequacy, and destruction cannot be predicted and charges against this class of depreciation are naturally conjectural.

Incomplete depreciation due to wear and tear likely to fall in large amounts and at irregular intervals.

There are several methods of dealing with depreciation; among the more common may be mentioned: *

* Report of the Committee on Gas, Oil, and Electric Light, City of Chicago, May, 1913.

(1) To charge to earnings in good years and credit to depreciation reserve such amounts as the profits from operation permit.

(2) To charge to earnings the depreciation as it matures and necessitates renewals.

(3) To charge to earnings and credit to depreciation reserve annually a certain percentage of the cost determined by the average weighted life of the property.

In general power plant practice it is customary to make an average annual depreciation allowance, based on the original cost of the property less salvage or junk value, spread over a period of years approximating the weighted life of the plant. If depreciation is considered to include

TABLE 148.

COST OF MECHANICAL EQUIPMENT — STEAM TURBO-ELECTRIC GENERATING STATIONS.*

2,000 to 20,000-kilowatt Capacity, Based on Maximum Continuous Capacity of Generators at 50° Rise.

	Dollars per Kilowatt.	
	High.	Low.
Preparing site — Dismantling and removing structures from site, making construction roads, tracks, etc.	$0.25	$....
Yard Work — Intake and discharge flumes for condensing water, railway siding, grading, fencing sidewalks	2.50	1.00
Foundations — Including foundations for building, stacks, and machinery, together with excavation, piling, waterproofing, etc.	6.00	1.00
Building — Including frame, walls, floors, roofs, windows and doors, coal bunker, etc., but exclusive of foundations, heating, plumbing, and lighting	12.00	4.00
Boiler-room Equipment — Including boilers, stokers, flues, stacks, feed pumps, feed-water heater, economizers, mechanical draft, and all piping and pipe covering for entire station except condenser water piping	24.00	12.00
Turbine-room Equipment — Including steam turbines and generators, condensers with condenser auxiliaries and water piping, oiling system, etc	22.00	12.00
Electrical Switching Equipment — Including exciters of all kinds, masonry switch structure with all switchboards, switches, instruments, etc., and all wiring except for building lighting	5.00	2.00
Service Equipment — Such as cranes, lighting, heating, plumbing, fire protection, compressed air, furniture, permanent tools, coal- and ash-handling machinery, etc	5.00	2.50
Starting Up — Labor, fuel, and supplies for getting plant ready to carry useful load	1.00	.50
General Charges — Such as engineering, purchasing, supervision, clerical work, construction, plant and supplies, watchmen, cleaning up	6.00	3.00
Total cost of plant to owner, except land and interest during construction	$83.75	$38.00

* By O. S. Lyford, Jr., and R. W. Stoval, of Westinghouse, Church, Kerr & Company, before the Engineer's Society of Western Pennsylvania.

the maintenance which is charged to expense directly, it would be proper to set aside as a reserve a fixed percentage of the decreasing value of the plant to represent the unmatured decadence. This ideal situation would equalize the total burden over the life by making the depreciation allowance largest when the repairs are smallest, and conversely the depreciation allowance smallest when the repairs are largest at the end of the useful life of the plant. If the system were composed of many small units not requiring renewal at or near the same time no special reserve would be necessary, as all replacements could be charged directly to operating expenses because of these inconsiderable amounts in any one year. In the large central station, however, a considerable portion of the plant is composed of large units which the rapid development of the art and growth of business may render inadequate long before their natural life has expired. As a result of and to provide for this condition, depreciation reserves are accumulated either on the "straight line" or "sinking fund" method.

Straight-line Method. — This method is based on the assumption that if the total investment, less salvage, is divided by the weighted life of the plant the resulting quotient expresses the amount which should be allowed each year to cover the accrued depreciation. This is the simplest of the several methods that have been suggested to determine the probable depreciation and make proper allowance for it in the records. No interest computations of any kind are involved, thus:

$$D = \frac{C - S}{n}, \tag{304}$$

$$V = (C - S)\left(1 - \frac{m}{n}\right), \tag{304a}$$

$$d = 100\frac{D}{C}, \tag{304b}$$

$$v = 100\frac{V}{C}, \tag{304c}$$

$$A = C - V, \tag{304d}$$

$$a = 100\frac{A}{C}, \tag{304e}$$

in which

D = accrued depreciation,
C = original cost,
S = scrap value of salvage,
n = assumed life, years,
V = present value,

m = age of the property, years,
d = rate of depreciation, per cent of original cost,
b = ratio of scrap value to original cost,
v = present value, per cent of original cost,
A = accrued depreciation,
a = accrued depreciation, per cent of original value.

The straight-line law is shown graphically in Fig. 597. The original cost is composed of the net cost (labor and material) plus the overhead (the extra charge intended to cover engineering and architects' fees; fire and liability insurance, and interest on the investment during

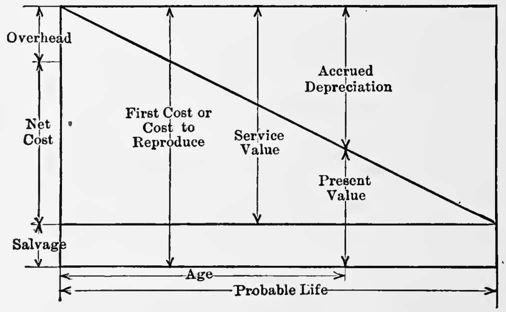


Fig. 597. Straight-line Method of Depreciation.

construction; contractors' profits on the portion of the work not done by the company itself; legal organization and incidental expense. The "probable life" of the plant is a purely theoretical quantity and is supposed to represent the weighted average period of usefulness of the various units composing the plant). It is determined by dividing the sum of the original costs of the various units composing the plant, less salvage, by the aggregate annual depreciation charge of these items. The actual life of the various units composing the plant can only be approximated since everything depends on the grade of material, workmanship and upkeep. Table 149 gives the average useful life of various portions of a steam power plant equipment, but so much depends upon the design and the conditions of operation that no fixed values can be definitely assigned and the values given should be used with caution. Most power plant appliances become obsolete long before the limit of their useful life is reached.

In the Report (May, 1913) to the Committee on Gas, Oil, and Electric Light on the investigation of the Commonwealth Edison Company, Chicago, depreciation is calculated on a 3 per cent sinking fund basis, giving an average weighted life of 18.5 years to the company's depreciable property.

TABLE 149.

APPROXIMATE USEFUL LIFE OF VARIOUS PORTIONS OF STEAM POWER PLANT EQUIPMENTS.

	Years.
Buildings, brick or concrete	40
Buildings, wooden or sheet iron	15
Chimneys, brick	40
Chimneys, self-sustaining steel	30
Chimneys, guyed sheet-iron	10
Boilers, water-tube	25
Boilers, fire-tube	20
Engines, slow-speed	25
Engines, high-speed	20
Turbines	25
Generators, direct-current	20
Generators, alternating-current	20
Motors	20
Pumps	20
Condensers, jet	25
Condensers, surface	20
Heaters, open	25
Heaters, closed	20
Economizers	20
Wiring	15
Belts	8
Coal conveyor, bucket	15
Coal conveyor, belt	10
Transformers	20
Rotary converters	20
Storage batteries	15
Piping, ordinary	12
Piping, first class	20

NOTE. — So much depends upon the design and the conditions of operation that no fixed values can be definitely assigned and the above figures should be used with caution. Practice shows that most power-plant appliances become obsolete long before the limit of their useful life is reached.

Example 75. A condenser equipment is 10 years old and cost originally \$3500.00. Assuming that its useful life is 25 years and that its junk value is \$350.00, determine the annual depreciation, the present value and the accrued depreciation on the straight line basis.

$$D = \frac{C - S}{n} = \frac{3500 - 350}{25} = \$126, \text{ annual depreciation charge.}$$

$$d = 100 \frac{D}{C} = 100 \frac{126}{3500} = 3.5 \text{ per cent.}$$

$$V = (C - S)\left(1 - \frac{m}{n}\right) = (3500 - 350)\left(1 - \frac{10}{25}\right) = \$1890, \text{ present value.}$$

$$v = 100 \frac{V}{C} = 100 \frac{1890}{3500} = 54 \text{ per cent.}$$

$$A = C - V = \$3500 - \$1890 = \$1610, \text{ accrued depreciation.}$$

Sinking Fund Method. — By the sinking fund method a fixed sum is placed aside each period and allowed to accumulate at compound interest. The amounts thus set aside plus the interest accumulations must be equal to the original cost less salvage at the end of the assumed period. The rate of depreciation in terms of interest and useful life is a simple problem in compound interest and may be expressed

$$d = 100 \frac{r\,(1 - b)}{(1 + r)^n - 1}, \tag{305}$$

$$D = \frac{dC}{100}, \tag{305a}$$

$$a = 100 \frac{(1 + r)^m - 1}{(1 + r)^n - 1}, \tag{305b}$$

$$A = \frac{a}{100}(C - S), \tag{305c}$$

$$V = C - A, \tag{305d}$$

$$v = 100 \frac{V}{C}, \tag{305e}$$

in which

r = rate of interest,

a = accrued depreciation, per cent of original cost less salvage.

Other notations as previously designated.

Example 76. Taking the data in Example 75 determine the annual depreciation charge, accrued depreciation, and present value on the sinking fund basis, assuming an annual interest rate of 5 per cent.

$$d = 100 \frac{r\,(1 - b)}{(1 + r)^n - 1} = 100 \frac{0.05\,(1 - 0.1)}{(1 + 0.05)^{25} - 1} = 1.97 \text{ per cent.}$$

$D = 0.0197 \times \$3500 = \$68.95$, annual payment to the sinking fund which at the end of 25 years will equal $\$3500 - \$350 = \$3150$.

$$a = 100 \frac{(1 + r)^m - 1}{(1 + r)^n - 1} = 100 \frac{(1 + 0.05)^{10} - 1}{(1 + 0.05)^{25} - 1} = 26.35 \text{ per cent.}$$

$A = \$3150 \times 0.2635 = 830.15$

$V = \$3500.00 - \$830.15 = \$2669.15$, present value.

$$v = 100 \frac{2669.15}{3500} = 76.4 \text{ present value, per cent of original cost.}$$

Table 150 may be conveniently used in this connection. At the intersection of vertical column 5 and horizontal columns 10 and 25 we find 7.95 and 2.09 respectively. Dividing 2.09 by 7.95 gives 0.2635 or 26.35 per cent, the accrued depreciation.

TABLE 150.

RATE OF DEPRECIATION.

(Per Cent of First Cost.)

Assumed Useful Life of Apparatus.	Rate of Interest, per Cent.												
	2	2.5	3	3.5	4	4.5	5	5.5	6	7	8	9	10
2	49.50	49.37	49.27	49.14	49.02	48.90	48.78	48.66	48.54	48.31	48.07	47.84	47.62
3	32.67	32.51	32.35	32.19	32.03	31.87	31.72	31.56	31.41	31.10	30.80	30.51	30.21
4	24.26	24.08	23.90	23.72	23.55	23.39	23.20	23.03	22.86	22.52	22.19	21.84	21.55
5	19.21	19.02	18.83	18.65	18.46	18.28	18.10	17.91	17.73	17.40	17.04	16.73	16.37
6	15.85	15.65	15.46	15.26	15.08	14.89	14.70	14.52	14.33	13.97	13.63	13.29	12.96
7	13.45	13.25	13.05	12.85	12.66	12.46	12.28	12.09	11.91	11.15	11.20	10.87	10.55
8	11.65	11.44	11.24	11.05	10.85	10.66	10.47	10.28	10.10	9.74	9.40	9.06	8.74
9	10.25	10.04	9.84	9.64	9.45	9.26	9.07	8.88	8.70	8.34	8.00	7.68	7.36
10	9.13	8.92	8.72	8.52	8.33	8.14	7.95	7.76	7.58	7.23	6.90	6.58	6.27
11	8.21	8.01	7.80	7.61	7.41	7.22	7.04	6.85	6.68	6.33	6.00	5.69	5.40
12	7.45	7.25	7.04	6.85	6.65	6.46	6.28	6.10	5.92	5.60	5.27	4.97	4.69
13	6.81	6.60	6.40	6.20	6.01	5.83	5.64	5.47	5.29	4.96	4.65	4.36	4.08
14	6.26	6.05	5.85	5.65	5.46	5.28	5.10	4.93	4.75	4.49	4.13	3.84	3.58
15	5.78	5.57	5.37	5.18	4.99	4.81	4.63	4.46	4.29	3.97	3.68	3.40	3.15
16	5.36	5.16	4.96	4.77	4.58	4.40	4.22	4.06	3.89	3.58	3.30	3.03	2.78
17	4.99	4.79	4.59	4.40	4.22	4.04	3.87	3.70	3.54	3.24	2.96	2.71	2.47
18	4.67	4.46	4.27	4.08	3.90	3.72	3.55	3.39	3.23	2.94	2.66	2.42	2.19
19	4.37	4.17	3.98	3.79	3.61	3.44	3.27	3.11	2.96	2.67	2.47	2.17	1.95
20	4.11	3.91	3.72	3.53	3.36	3.19	3.02	2.87	2.71	2.44	2.18	1.95	1.95
25	3.12	2.92	2.74	2.56	2.40	2.24	2.09	1.95	1.82	1.58	1.36	1.18	1.75
30	2.46	2.27	2.10	1.93	1.78	1.64	1.50	1.38	1.26	1.06	0.88	0.73	0.61
35	2.00	1.82	1.65	1.50	1.36	1.23	1.10	0.99	0.89	0.72	0.58	0.46	0.37
40	1.65	1.48	1.32	1.18	1.05	0.93	0.83	0.73	0.64	0.50	0.38	0.29	0.22
45	1.39	1.22	1.07	0.94	0.82	0.72	0.62	0.54	0.47	0.35	0.26	0.19	0.14
50	1.18	1.02	0.88	0.76	0.65	0.56	0.42	0.40	0.34	0.25	0.17	0.12	0.09

It is not supposed that an owner will regularly lay aside an annual amount, or take the trouble to arrange for its investment at current rates in the market or savings bank, since the money is probably worth more to him in his business. In practice it is retained in his business or investments and is earning the rate of interest obtainable therein, but in determining the net profit or loss this depreciation item is nevertheless accounted for just as if it were actually placed in outside investments.

The *expectancy* or remaining life of any article is the probable time during which it may reasonably be expected to render efficient service. It is determined from the actual condition of the article and all local circumstances which may affect its continued use and not by subtracting age from probable life. Thus an article may have a probable life of 25 years and yet be in first-class condition and as good as new when it reaches the end of this term. The value of this article is not written off the books nor should it be regarded as good as new. Its value is ascertained by determining its probable additional years of usefulness and the probable cost of replacing it at the end of this term.

The term "depreciation" is frequently used when the term "amortization" would be more appropriate. *Amortization* deals with the retirement of the invested capital. This may be in instalments in uniform or in unequal annual amounts, or in a lump sum at the end of useful life. The replacement may mean the substitution of a new identical plant, but at a cost dependent on new conditions, new prices of labor

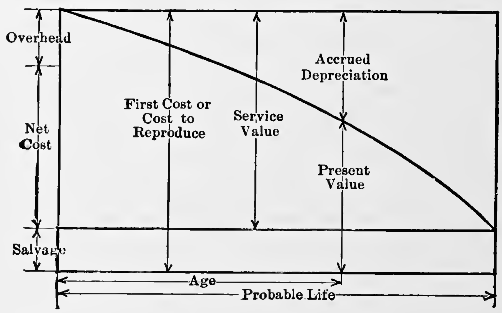


FIG. 598. Sinking Fund Method of Depreciation.

and material, or it may mean the substitution of new devices rendering equivalent service. In either event the replacement may be at a greater or less cost than the original cost, with, therefore, a corresponding increase or decrease of capital invested. Expenditures for new parts of a plant, which take the place of old parts which are retired for any cause, should be charged to replacement only to the extent of capital represented by the part of the plant thus retired. Any excess of the expenditure for replacement over the cost of the discarded part of a plant should be treated as an addition to, and any less cost as a deduction from, the invested capital. The term "replacement" should not be used in the sense of retirement of invested capital, which deals with the cost of the replaced part and not with the cost of the new equivalent installation. (Valuation Depreciation and the Rate-Base, Grunsky, 1917.)

The term *going value* may be properly taken to mean a value attaching to a public utility property as the result of its having an established revenue-producing business. Going value may be determined from a consideration of the amounts of money actually expended in the cost of producing the business or it may be determined from consideration of the present cost of reproducing the present revenue. (Value for Rate-Making, Floy, 1917.)

For purpose of design and comparison it is customary to assume a single fixed percentage for depreciation, obsolescence, inadequacy, etc. An average figure is 5 per cent.

426. Maintenance. — Maintenance usually refers to the expense of keeping the plant in running order over and above the cost of attendance, although the term is frequently used in place of "repairs." It includes cost of upkeep, replacement, and precautionary measures. This latter item includes the renewal of working parts, painting of perishable or exposed material, and replacing worn-out and defective material. Many engineers make no allowance for maintenance in the fixed charges and include these costs under supplies, attendance, or repairs. In a general way, when maintenance is included under the fixed charges, an annual charge of 2 per cent is considered a liberal allowance, since most of the repair work comes under attendance. In street-railway practice maintenance is divided among the several parts of the system as follows: Buildings, steam appliances, electrical equipment, and miscellaneous. In this connection the maintenance becomes a part of the operating charges, since the various items vary widely from month to month.

427. Taxes and Insurance. — Taxes vary from a fraction of one per cent to 2 per cent, depending upon the location of the plant. An average figure is $1\frac{1}{2}$ per cent of the actual value of the investment. Buildings and machinery are ordinarily insured against fire loss and boilers against accidental explosions, and accident policies are sometimes carried on all operating machinery. A fair charge for this item is one-half per cent.

428. Operating Costs. — General Division. — The distribution of the operating costs depends largely upon the size and nature of the plant. In the small isolated station the term "operating costs" without qualification refers to the generating or station operating costs, exclusive of fixed charges. These costs are commonly divided as follows:

1. Labor and attendance.
2. Fuel and water.
3. Oil, waste, and supplies.
4. Repairs and maintenance.

In some of the larger isolated stations a more extensive division is often made but there appears to be no accepted standard.

In large central stations the operating costs are divided under the major headings of

1. Production expenses.
2. Transmission expenses.
3. Electric storage expenses.
4. Utilization expenses.
5. Commercial expenses.
6. New business expenses.
7. General and miscellaneous expenses.

The extent of the subdivisions under each subheading depending upon the size and nature of the plant. See Table 151.

TABLE 151.

TOTAL EXPENSE (EXCLUSIVE OF DEPRECIATION) FOR THE CALENDAR YEAR 1912.

Commonwealth-Edison Co., Chicago.

	Total Expense.	Per Cent of Total.	798,677,000 Kw-hr. Purchased and Generated. Cost in Cents Per Kw-hr.
Production:			
Station wages	$352,053	4.34	0.044080
Fuel expense, including storage and shrinkage	2,137,076	26.38	0.267577
Station supplies and expense	54,155	0.67	0.006781
Building and property maintenance	37,887	0.47	0.004744
Maintenance of equipment	256,552	3.17	0.032122
Purchased power	359,311	4.44	0.044988
Total production	$3,197,034	39.47	0.400291
Transmission and distribution:			
Meter department expense	$237,090	2.92	0.029685
Substation operating and repairs	210,107	2.59	0.026307
Storage battery operating and repairs	110,088	1.36	0.013784
Maintenance of overhead and underground lines	429,871	5.31	0.053823
Install, remove, exchange meters	60,503	0.75	0.007575
Total transmission and distribution	$1,047,659	12.93	0.131174
Utilization:			
Maintenance tungsten fixtures and posts; Maintenance signs; Maintenance arc lights	$224,538	2.77	0.028114
Repairs to customers' installations	77,618	0.96	0.009718
Renewal of incandescent lamps	345,465	4.26	0.043255
Inspection of customers' premises	43,364	0.54	0.005429
Total utilization	$690,985	8.53	0.086516
New business:			
Contract department expense	$209,669	2.59	0.026252
Advertising	211,635	2.61	0.026498
Wiring and appliances	61,042	0.75	0.007643
Total new business	$482,346	5.95	0.060393
Commercial expense:			
Collecting and bookkeeping	$152,007	1.88	0.019032
Claim department expense	11,025	0.14	0.001380
Information bureau	9,072	0.11	0.001136
Billing department expense	68,581	0.85	0.008587
Customers' statistics	22,206	0.27	0.002780
Total commercial expense	$262,891	3.25	0.032916
General expense:			
Executive and legal and loss and damage account	$483,717	5.97	0.060565
Maintenance and rental of offices and miscellaneous buildings	152,633	1.88	0.019111
Telephone and telegraph and general office sundries	51,069	0.63	0.006394
Purchasing and stores department expense	89,954	1.11	0.011263
Engineering and operating supervision	179,618	2.22	0.022489
General office departments, accounting and statistics	103,055	1.27	0.012903
Net profit on mercantile sales	69,850	0.86	0.008746
Total general expense	$990,196	12.22	0.123980
Miscellaneous:			
Transportation department undistributed and miscellaneous	$22,627	0.28	0.002833
Miscellaneous operating — steam	90,509	1.12	0.011332
Conduit rental	5,436	0.07	0.000681
Municipal compensation	460,195	5.68	0.057620
Taxes	714,000	8.81	0.089398
Insurance	277,017	3.42	0.034684
Interest, discount, and exchange	117,225	1.45	0.014677
Profit on stores	193,316	2.38	0.024205
Pension fund	72,000	0.89	0.009015
Discount on bonds	39,114	0.48	0.004897
Bad debts	70,077	0.87	0.008774
Total miscellaneous expense	$1,429,562	17.65	0.178991
Grand total	$8,100,673	100.00	1.014261

A number of large central stations limit the major headings to

1. Generation.
2. Administration.
3. Distribution.

Some companies include all or part of the fixed charges under the major heading, others limit the operating costs to expense which is dependent only on the output. Because of this diversity in bookkeeping comparisons of the cost of power based on the annual report are without purpose. An excellent system is that prescribed by the State Board of Public Utility Commissioners of New Jersey, a discussion of which is to be found in Power, Nov. 11, 1913, p. 697. A few annual reports illustrating the different systems of accounting are reproduced in the accompanying tables.

429. Labor, Attendance, Wages. — The minimum number of men required to handle a given plant is approximately a fixed quantity and it is seldom possible to so arrange the work that any material reduction can be effected. Until very recently it has been the universal custom to pay wages on a "flat rate" basis, that is, the attendant is given a fixed sum per day or month irrespective of the amount of work required or the economy of operation. In many cases, however, the bonus system has been successfully adopted. For example, in the boiler room the coal consumption is determined for a given period of time with ordinary careful firing, and the fireman is offered a reasonable percentage on the saving of coal which he is able to effect over this record by special care and attention to the keeping of fires always in the best condition, avoiding the blowing off of steam, using as little coal as needed for banking fires, and in other ways. Where careful records are kept of supplies, repairs, and renewals, the bonus is also applicable to electricians, oilers, and other employees.

Labor should always be estimated or recorded as so many dollars per month or per year and not merely in terms of the output unless the load factor is definitely known, otherwise comparisons are misleading. For example, consider two plants of 500 kilowatts capacity, each with labor charges, say, of $400 per month. Suppose the output of one is 100,000 kilowatt-hours per month and that of the other 40,000 kilowatt-hours per month. The monthly charges are evidently the same, viz., $400, but the cost per kilowatt-hour differs widely, being 0.4 cent in the first case and 1 cent in the latter.

The cost of labor varies so much with the location of the plant and the conditions of operation that general figures are of little value except as a rough guide. Specific figures will be found in the accompanying tables.

For a summary of labor costs in large central stations see "Central-Station Labor Costs," Electrical World, Nov. 16, 1912, p. 1031.

430. Cost of Fuel. Tables 151 to 160 give specific examples of the cost of fuel in different sizes and types of steam power plants. It will be noted that this item varies considerably even with plants of the same general class. So much depends upon the grade and market price of the fuel, type, and size of plant and conditions of operation that no single item can afford a means of comparing fuel costs in different plants. Such items as "lb. coal per kw-hr.," "cost of fuel per kw-hr.," or the equivalent have their value in any accounting system, but fail utterly as a measure of the economy of operation unless accompanied by a statement of the qualifying conditions. For example, an inefficiently operated plant using a high-grade fuel may show a lower fuel consumption, lb. per kw-hr., than an economical plant using a low-grade fuel, and an uneconomical plant using a very cheap fuel may show a lower "cost of fuel per kw-hr." than an efficiently operated plant using costly fuel. Similarly, two plants of the same size and type, and

TABLE 152.

FUEL CONSUMPTION IN MASSACHUSETTS CENTRAL STATIONS.

(Year ending 1915.)

Company.	Long Tons Used.	Cost per Ton.	Total Coal Cost.	Cents Per Kw-hr. Generated.	Lb. Coal Per Kw-hr.
Cambridge El. Lt. Co.	15.251	$4.022	$61,339	0.403	2.246
Easthampton Gas Co.	4,170 2.35 coke	4.351 3.50	18,150	0.641	3.301
Edison Elec., Ill., Boston	182,679	3.902	712,734	0.359	2.063
Edison Co., Brockton	15,625	4.778	74,660	0.458	2.149
Fall River El. Lt. Co.	14,871	3.673	54,621	0.390	2.378
Fitchburg Gas & El. Co.	8,436 4,328 coke 348 gas coal	4.167 4.48 4.55	56,127	0.723	3.785
Greenfield El. Lt. Co.	4,494	4.623	20,778	0.693	3.359
Haverhill Electric Co.	9,600	4.664	44,777	0.64	3.075
Lawrence Gas Co.	4,574 39 coke 1,561 dust	4.667 4.000 2.000	24,627	0.995	5.588
Lowell El. Lt. Co.	18,584	4.698	87,316	0.667	3.178
Lynn Gas & Electric Co.	16,589	3.545	58,799	0.461	2.913
Malden Electric Co.	14,436 1,884 coke	4.6 4.0	73,951	0.693	3.424
New Bedford Gas & El. Lt.	10,935	3.608	39,457	0.472	2.931
No. Adams Gas Lt. Co.	7,532 96 gas coal	4.115 5.6	31,531	0.618	3.351
Salem El. Lt. Co.	8,973	4.074	36,557	0.55	3.023
Springfield United El. Lt.	31,954	4.277	136,686	0.567	2.971
Webster & S. Gas & El. Lt.	7,300	4.534	33,096	0.579	2.86
Worcester El. Lt. Co.	37,462	4.142	155,180	0.48	2.593

The Cambridge, Boston, Fall River, Lynn, New Bedford, and Salem companies are located on tidewater and enjoy the advantage of cheaper fuel transportation than those located inland.

TABLE 153.

POWER COSTS IN CENTRAL STATIONS.

Station A. 10–500 hp. boilers; 5000 hp. piston engines; Ill. screenings; no coal-handling apparatus; hand-fired furnaces.

Station B. Modern steam turbine plant; stoker equipment; coal- and ash-handling system; economizers; superheaters; Ill. screenings.

Station C. 5400 hp. boilers; 14,000-kw. turbines and engines; coal- and ash-handling system; stoker equipment; Ill. screenings.

June, 1913.

	A.	B.	C.
Kw-hr. generated	1,061,000	1,210,750	1,404,605
Tons of coal	2,775	2,437.37	3,981.40
Tons of ash	555	322.10	603
Lb. water evaporated	40,600,000	35,359,500	58,100,000
Lb. water evaporated per lb. coal	7.32	7.25	7.5
Lb. coal per kw-hr	5.23	4.03	5.55
Lb. water per kw-hr	....	29.20	41.6
Gal. engine oil per 1000 kw-hr	3.62	0.59	1.94
Gal. cylinder oil per 10,000 kw-hr	1.74	0.39	1.22

Total Cost, and Cost Per Kw-hr. in Cents.

	A.	Kw-hr.	B.	Kw-hr.	C.	Kw hr.
Superintendence	122.42	0.014	250.10	0.020	246.47	0.018
Repairs:						
Dynamos and appliances	171.33	0.019	10.84	0.001	12.94	0.001
Engines	......	......	......	......	......	......
Boilers	1017.48	0.115	299.81	0.024	......	......
Pumps, pipes, fittings and miscellaneous	8.80	0.001	22.15	0.002	1332.34	0.094
Operating boilers	880.92	0.100	392.13	0.033	794.96	0.057
Operating engines and dynamos	693.66	0.079	390.00	0.032	689.79	0.049
Supplies	5.47	......	44.80	0.004	101.10	0.007
Water	482.21	0.055	99.75	0.008	......	......
Lubricants and waste	220.12	0.025	42.50	0.004	95.74	0.007
Miscellaneous expense	291.24	0.033	60.08	0.005	177.68	0.013
Total, except fuel	3893.65	0.441	1612.16	0.133	3451.02	0.246
Coal	2635.75	0.298	2177.44	0.180	4906.44	0.349
Coal, labor, car to boiler room	198.62	0.022	114.62	0.009	105.48	0.008
Total cost	6728.02	0.761	3904.22	0.322	8462.94	0.603
Average cost of coal per ton on floor of boiler room	1.0214	......	0.94	......	1.344	......

October, 1913.

	A.	B.	C.
Kw-hr. generated	1,356,610	1,215,360	1,704,596
Tons of coal	3,052.4	2,838.5	4,900.72
Tons of ash	610.5	456.53	1,080
Lb. water evaporated	39,681,000	35,625,500	64,484,866
Lb. water evaporated per lb. coal	6.5	6.28	6.58
Lb. coal per kw-hr	4.5	4.67	5.75
Lb. water per kw-hr	....	29.31	37.83
Gal. engine oil per 10,000 kw-hr	1.24	0.41	0.95
Gal. cylinder oil per 10,000 kw-hr	7.07	0.41	0.40

Total Cost, and Cost per Kw-hr. in Cents.

	A.	Kw-hr.	B.	Kw-hr.	C.	Kw-hr.
Superintendence	121.67	0.010	243.74	0.020	201.14	0.012
Repairs:						
Dynamos and appliances	245.18	0.020	21.93	0.002	469.43	0.028
Engines	......	......	......	......	66.78	0.004
Boilers	559.11	0.046	484.51	0.040	833.61	0.049
Pumps, pipes, fittings and miscellaneous	16.32	0.001	9.00	0.001	595.97	0.025
Operating boilers	608.64	0.050	396.15	0.033	843.68	0.049
Operating engines and dynamos	718.88	0.059	390.00	0.032	673.29	0.039
Supplies	41.65	0.004	16.50	0.001	116.25	0.007
Water	354.97	0.029	98.16	0.008	......	......
Lubricants and waste	228.82	0.019	37.50	0.003	150.66	0.009
Miscellaneous expense	78.39	0.007	41.89	0.003	246.18	0.014
Total, except fuel	2973.63	0.245	1739.38	0.143	4,196.99	0.246
Coal	2899.91	0.239	2469.50	0.203	6,150.62	0.361
Coal labor, car to boiler room	187.60	0.015	135.60	0.011	183.20	0.011
Total cost	6061.14	0.499	4344.48	0.357	10,530.81	0.618
Average cost of coal per ton on floor of boiler room	$1.0113	......	$0.9178	......	$1.255	......

using the same fuel may show considerable difference in both "lb. of fuel per kw-hr." and "cost of fuel per kw-hr." because of difference in load factor even though both plants are efficiently operated for the given conditions. In a number of recent installations the station operating records include the heat supplied by the fuel per kw-hr. generated ("B.t.u. per kw-hr.") and the cost of the fuel on a heat basis (cents per 10,000 B.t.u.). These two items in connection with the load factor offer a satisfactory criterion of the fuel economy for plants of the same general design. Large central stations with individual units of 20,000 to 35,000 kw. rated capacity and yearly load factor of 50 per cent or more, have been credited with a yearly performance of 20,000 B.t.u. per kw-hr. generated, corresponding to an overall thermal efficiency of 17 per cent. With Illinois screenings this is equivalent to approximately 2 lb. coal per kw-hr. and with the better grades of bituminous coal, about 1.5 lb. coal per kw-hr. Much better results than this have been obtained for brief periods of operation but when averaged over a considerable period of time the standby losses, such as coal burned in banking fires, heat lost in blowing down boilers, lower efficiency in operating at underloads and overloads and the like, reduce the overall efficiency to substantially that given above. The coal consumption per kw-hr. for a number of medium size central stations in Massachusetts is given in Table 152. This table does not offer a fair basis of comparison since the calorific value of the fuel and the yearly load factor are not given.

In estimating the cost of fuel for a proposed installation the logical procedure is as follows:

1. Construct load curves for the probable power requirements.
2. Calculate the total weight of steam supplied from the load curve.
3. Transfer the total steam requirements to the unit water rate basis.
4. Reduce the average unit water rate to "B.t.u. supplied by the steam per unit output."
5. Divide the average B.t.u. supplied by the steam per unit output by the estimated overall boiler efficiency, considering all standby loss. This gives the B.t.u. supplied by the fuel per unit output.
6. Reduce the cost of fuel to "cost per 10,000 B.t.u."
7. Multiply item 5 by item 6 and divide by 10,000. This gives the average cost of fuel per unit output for the required period.

The construction of the load curves is the most important item since the cost of the fuel per unit putput is primarily a function of the load factor. See paragraph 434.

The total weight of steam is calculated from the load curve by considering the unit water rate of the prime mover and steam-driven auxiliaries at the variable loads, and the time element.

TABLE 154.

DISTRIBUTION OF STATION OPERATING COSTS.

Steam Turbine Plants. (Medium Size.)
(Year Ending 1915.)

	Brockton.	Fall River.	New Bedford.	United Elec. Light.
Rated boiler capacity, hp..	3300	2800	3416	9300
Rated turbine capacity, kw.	9000	10,000	9400	13,600
Output, kw-hr. (million)..	16.28	14.00	8.35	24.09
Load factor, per cent......	....	32.4	....	....
Tons of coal (thousands)..	15.62	14.87	10.96	32.0
Coal per kw-hr., lb........	2.12	2.38	2.93	2.82
Cost of coal per ton.......	$4.78	$3.67	$3.60	$4.27
Men employed............	26	20	....	....

Operating Costs, Cents per Kw-hr.

	Actual.	Per Cent Total.	Actual.	Per Cent Total.	Actual.	Per Cent Total.	Actual.	Per Cent Total.
Fuel.....................	0.458	53.9	0.390	65.4	0.472	52.7	0.570	62.5
Oil, waste, and packing...	0.005	0.6	0.005	0.8	0.003	0.3	0.005	0.6
Water....................	0.019	2.2	0.016	2.7	0.038	4.2	0.004	0.5
Wages....................	0.179	21.0	0.124	20.8	0.309	34.5	0.160	17.6
Station tools and appliances..................	0.023	2.7	0.011	1.9	0.025	2.8	0.008	0.8
Station structure repairs..	0.079	9.2	0.011	1.9	0.017	1.9	0.050	5.5
Steam plant repairs.......	0.069	8.1	0.024	4.0	0.027	3.0	0.081	8.9
Electric plant repairs......	0.020	2.3	0.015	2.5	0.005	0.6	0.033	3.6
Total................	0.852	100.0	0.596	100.0	0.896	100.0	0.911	100.0

TABLE 155.

STATION OPERATING COSTS (1915).

Massachusetts Steam Power Plants.

Plant.	Fuel.	Oil, Waste and Packing.	Water.	Wages.	Station Tools and Appliances.	Station Structure Repairs.	Steam Plant Repairs.	Electrical Station Repairs.	Total.
Cambridge........	0.403	0.012	0.026	0.274	0.003	0.017	0.040	0.046	0.821
Easthampton......	0.641	0.004	0.003	0.307	0.005	0.001	0.049	0.010	1.020
Edison, Boston....	0.359	0.002	0.010	0.161	0.017	0.009	0.051	0.060	0.687
Edison, Brockton..	0.458	0.005	0.019	0.179	0.023	0.079	0.069	0.020	0.852
Fall River.........	0.390	0.005	0.016	0.124	0.011	0.011	0.024	0.015	0.596
Haverhill..........	0.640	0.016	....	0.209	0.011	0.055	0.071	0.004	1.006
Lowell.............	0.667	0.006	0.007	0.193	0.013	0.032	0.062	0.005	0.986
Lynn...............	0.461	0.015	0.032	0.194	0.003	0.051	0.152	0.014	0.922
Malden............	0.693	0.011	0.057	0.177	0.015	0.002	0.058	0.006	1.019
New Bedford......	0.472	0.003	0.038	0.309	0.025	0.017	0.027	0.005	0.896
Salem.............	0.550	0.013	0.020	0.233	0.010	0.002	0.059	0.006	0.903
Worcester.........	0.480	0.004	0.004	0.108	0.003	0.018	0.046	0.014	0.680

The heat supplied by the steam is measured above the temperature of the feed water. In plants where exhaust is used for heating or manufacturing purposes only the difference between the heat supplied to the prime movers and steam-driven auxiliaries and that of the exhaust utilized for heating is charged to power. See paragraph 177.

Current practice gives an average efficiency (based on yearly operation) of boiler and furnace of 70 per cent for pumping stations running at practically full load, 68 per cent for large lighting and power stations

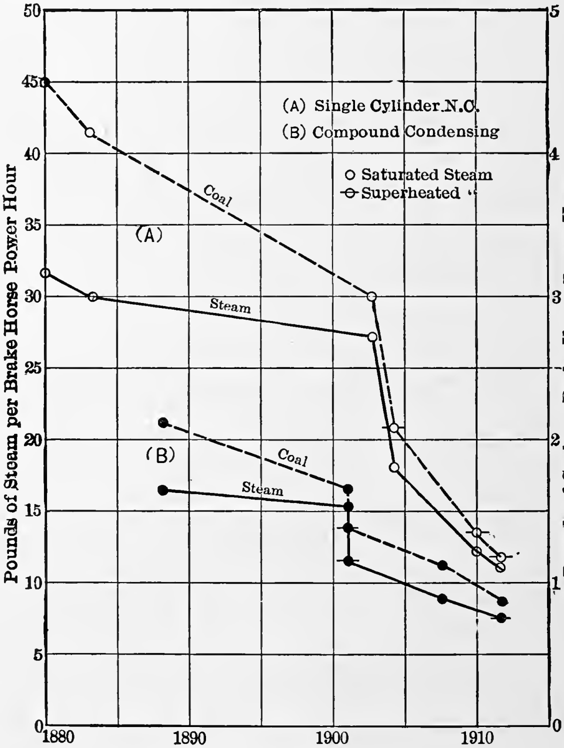


FIG. 599. Development of the Steam Power Plant. (Locomobile Type.)

with yearly load factor of 0.45 or more, and 65 per cent for similar stations with load factor between 0.35 and 0.40. For very low load factors, 0.25 and under (as in connection with large manufacturing plants, tall office buildings, and other plants operating on a 12-hour basis), this efficiency seldom exceeds 60 per cent. With these figures as a guide the cost of fuel per unit output may be roughly approximated.

In Europe the "locomobile" type of steam power plant has attained an extremely high degree of heat efficiency as will be seen from the curves in Fig. 599. The most economical result shown, namely 0.87

pound of coal per developed horsepower-hour, is equaled only by our best gas-producer plants.

431. Oil, Waste, and Supplies. — These items approximate from a fraction to 5 per cent of the total operating expenses. Tables 153 to

TABLE 156.

YEARLY COST OF OPERATION.

Fort Wayne Municipal Plant.

(1915–1916.)

Equipment: 2–500, 1–1500, 1–300 = 5500 kw. turbo-generators.
1–725, 2–500, 1–400, 3–300 = 3025 hp. boilers.

	Total.	Unit.
Investment cost:		
Boiler-plant equipment	$79,363.75	$26.00 per b. hp.
Boiler-plant buildings, fixtures and grounds	26,302.34	8.70 " "
Steam power plant equipment	182,773.75	33.00 " kw.
Steam power plant building, fixtures and grounds	39,469.54	7.20 " "
Total power plant	327,909.38	59.50 " "
Distribution system and other expenses	407,138.19	74.00 " "
Grand total	735,047.57	134.00 " "
Total output 6,520,670 kw-hr.		Lb. coal per kw-hr., 5.55
Total coal burned 18,100 tons		
Yearly load factor 24.7 per cent		

	Cost per Year.	Per Cent of Total.	Cents per Kw-hr.
Station operating costs:			
17 men, 3–8 hr. shifts, labor	$17,296.11	30.2	0.265
Coal, $1.80 per ton delivered	32,578.48	56.8	0.499
Supplies and sundries	514.71	0.8	0.008
Maintenance	6,980.40	12.2	0.107
Total	$57,369.70	100.00	0.879
Total expense:			
Steam power generation	$57,369.70	37.6	0.879
Distribution	10,269.01	6.7	0.158
Consumption	17,730.74	11.6	0.272
Commercial	11,472.04	7.5	0.175
General	12,144.63	7.9	0.185
Depreciation	29,658.99	19.5	0.455
Undistributed	9,850.30	6.4	0.149
Contingencies	4,391.98	2.8	0.067
Grand total	$152,887.39	100.0	2.340

160 give some idea of current practice in different classes of power plants.

432. Repairs and Maintenance. — This item ordinarily refers to the cost of keeping the plant in running order over and above the cost of labor or attendance, and depends upon the age and condition of the plant and the efficiency of the employees. Tables 153 to 160 give the cost of repairs and maintenance for a wide range in power-plant practice.

433. Cost of Power. — The actual cost of producing power depends upon the geographical location of the plant, the size of apparatus, the design, conditions of loading, system of distribution, and the method of accounting. Tables 151 to 160 compiled from various sources give the detailed costs of a large number of central and isolated stations.

TABLE 157.

COST OF GENERATING 1000 LB. STEAM.*

N. Y. Buildings — Steam Heating Only.

(1915.)

No. of Building.	1	2	3	4	5	6
Type of building	O	L	O	L	D	O
No. of floors	25	12	25	....	....	....
Building vol. cu. ft. (million)	....	4	6.5	....	15	....
Duration of test, days	15	4	5	4	46	151
Steam generated, 1000 lb.	1611	362	485	124	10,310	36,890
Tons of coal, gross	117	27.6	30.3	11.3	783	2540
Rate of evaporation	6.15	5.84	7.13	4.94	5.89	6.31
Average outside temperature	30.7	34.2	39.6	37.0	34.8	40.9
Boiler capacity, hp.	384	600	600	800	1200	900
Maximum boiler, hp.	280	300	330	150	600	850
Average boiler, hp.	100	80	.150	50	235	350
Cost per 1000 Lb. Steam.						
Coal	$0.191	$0.201	$0.165	$0.238	$0.203	$0.187
Labor	0.049	0.085	0.079	0.251	0.052	0.056
Ash removal	0.010	0.011	0.009	0.021	0.008	0.007
Water (makeup)	....	....	....	....	0.007	0.001
Electric current (forced draft)	0.014	0.005	0.007	0.021	0.008	0.006
Electric current (boiler feed pump)	....	....	0.007	....	....	....
Supplies	0.004	0.011	0.006	0.002	0.004	0.006
Repairs and miscellaneous	0.004	0.004	0.002	0.001	0.003	0.002
Total	$0.272	$0.317	$0.275	$0.535	$0.285	$0.265
Fixed charge on investment	0.029	0.051	0.054	0.084	0.044	0.033
Total cost per 1000 lb.	$0.301	$0.368	$0.329	$0.619	$0.329	$0.298

" O," Office building; " L," Loft building; " D," Department store. Coal, $2.50 per ton in all buildings.

* From report of the Station Operating Committee, National District Heating Association, read June 3, 1915, at Chicago.

TABLE 158.

SOME POWER COSTS FROM A MODERN APARTMENT HOUSE.

(New York.)

Original cost of plant on the foundation, 1909		$113,424
Present value at 10 per cent charged off each year		60,279
Average Cost per 24 hr. for 1916:		
Labor		$39.59
Coal		54.13
Ashes		1.66
Oil		1.27
Supplies		7.01
Repairs		4.61
Improvements		.65
Depreciation (10 per cent on $60,279)		16.51
Total cost per 24 hr.		$125.43
Average cost per hr.		5.22+
Quantities and Costs for Year Ended Dec. 31, 1916:		
Water consumed in boilers per 24 hr. (venturi-meter measured), lb.	376,911	
Coal consumed per 24 hr., lb	38,828	
Ashes put out per 24 hr., lb.	6,538	
Average horsepower-hr. developed per 24 hr	10,925.04	
Average horsepower-hr. developed per hr.	455.21	
Water evaporated per pound of coal (actual conditions), lb	9.271	
Water evaporated per pound of coal (from and at 212), lb.	9.707	
Coal (No. 3 Buck.) consumed per hp.-hr., lb.	3.55	
Ash, per cent	16.8	
Ash per analysis (commercial)	12.48+	
Cost per hp-hr., dollars	0.0114	
Kw-hr. delivered to board for 1916	788,129	
Average kw-hr. per 24 hr	2,159	
Average kw-hr. per hr.	90	
Electric load was 21 per cent of total load and		
Cost per 24 hr., dollars	26.34	
Cost per hr., dollars	1.10 --	
Cost per kilowatt-hours, dollars	0.012+	
Income from store lighting per 24 hr., dollars	9.06	
Net operating cost per 24 hr., dollars	116.37	
Net operating cost per hr., dollars	4.85	
Net hp-hr. cost, dollars	0.0107	
Net kw-hr. cost, dollars	0.0114	

Year.	Average B.t.u.	Average Ash, Per Cent.	Average Moisture, Per Cent.	Average Coal per Hp-hr., Lb.	Average Coal Cost per Hp-hr., Dollars.
1912	12,672.22	14.70	6.81	3.891 No. 1	0.0059
1913	12,538.46	14.687	7.05	4.427 Nos. 1, 2 & 3	0.0060
1914	12,826.43	14.425	6.386	3.487 No. 3	0.0051 −
1915	12,825.01	13.57	6.75	3.329 No. 3	0.0043
1916	12,796.40	12.48	7.05	3.554 No. 3	0.0049+

TABLE 159.

COST OF ONE HORSE POWER PER YEAR, SIMPLE ENGINES, NON-CONDENSING, 10-HOUR BASIS, 308 DAYS PER YEAR.

(Wm. O. Webber, Engineering Magazine, July, 1908, p. 563.)

Size of plant................horse power	20	40	60	80
Cost of plant per horse power...........	$200.00	$190.00	$180.00	$175.00
Fixed charges at 14 per cent	28.00	26.60	25.20	24.50
Coal per horse-power hour, in pounds....	12.00	10.00	9.00	8.00
Cost at $4.00 per ton	66.00	55.00	49.50	44.00
Attendance, 10-hour basis...............	30.00	20.00	15.00	13.00
Oil, waste, and supplies................	6.00	4.00	3.00	2.60
With coal at $5.00 per ton..............	146.50	119.35	105.07	95.10
With coal at $4.50 per ton..............	138.25	112.47	98.80	89.60
With coal at $4.00 per ton..............	130.00	105.60	92.70	84.10
With coal at $3.50 per ton..............	121.75	98.72	86.51	78.60
With coal at $3.00 per ton..............	113.50	91.85	80.32	73.10
With coal at $2.50 per ton..............	105.25	84.97	74.13	67.60
With coal at $2.00 per ton..............	97.00	78.10	67.95	62.10

TABLE 160.

COST OF ONE HORSE POWER PER YEAR, COMPOUND CONDENSING ENGINES, 10-HOUR BASIS, 308 DAYS PER YEAR.

(Wm. O. Webber, Engineering Magazine, July, 1908, p. 564.)

Size of plant.........horse power	100	200	300	400	500	600
Cost of plant per horse power....	$170.00	$146.00	$126.00	$110.00	$96.00	$85.00
Fixed charges at 14 per cent.....	23.80	24.40	17.65	15.40	13.45	11.90
Coal per horse-power hour, pounds	7.0	6.5	6.0	5.5	5.0	4.5
Cost of fuel at $4.00 per ton......	38.50	35.70	33.00	32.00	27.50	24.70
Attendance, 10-hour basis........	12.00	10.00	8.60	7.25	6.20	5.40
Oil, waste, supplies.............	2.40	2.00	1.72	1.45	1.24	1.08
Total	76.70	68.10	60.97	56.10	48.39	43.08
With coal at $5.00 per ton........	86.40	77.10	69.22	61.90	55.29	49.28
With coal at $4.50 per ton........	81.50	72.60	65.07	58.10	51.79	46.18
With coal at $4.00 per ton........	76.70	68.10	60.97	56.10	48.39	43.08
With coal at $3.50 per ton........	71.90	63.70	56.82	50.50	45.04	39.98
With coal at $3.00 per ton........	67.00	59.20	51.67	46.70	41.49	36.88
With coal at $2.50 per ton........	62.30	54.75	48.59	43.00	38.83	33.83
With coal at $2.00 per ton........	57.45	50.25	44.47	40.10	34.64	30.73
Size of plant.........horse power	700	800	900	1000	1500	2000
Cost of plant per horse power....	$76.00	$69.00	$64.00	$60.00	$58.00	$56.00
Fixed charges at 14 per cent.....	10.65	9.65	8.95	8.40	8.12	7.85
Coal per horse-power hour, pounds	4.0	3.5	3.0	2.5	2.0	1.5
Cost of fuel at $4.00 per ton......	22.00	19.20	16.50	13.75	11.00	8.25
Attendance, 10-hour basis........	4.70	4.15	3.75	3.50	3.25	3.00
Oil, waste, supplies.............	0.94	0.83	0.75	0.70	0.65	0.60
Total	38.29	33.83	29.95	26.35	23.02	19.70
With coal at $5.00 per ton........	43.79	39.73	34.05	29.80	25.77	21.75
With coal at $4.50 per ton........	41.04	36.28	32.00	28.05	24.39	20.72
With coal at $4.00 per ton........	38.29	33.83	29.95	26.35	23.02	19.70
With coal at $3.50 per ton........	35.54	31.48	27.87	24.60	21.64	18.67
With coal at $3.00 per ton........	32.79	29.03	25.80	22.90	20.27	17.65
With coal at $2.50 per ton........	30.04	27.18	23.75	21.20	18.89	16.60
With coal at $2.00 per ton........	27.29	24.23	21.70	19.47	17.52	15.57

TABLE 161.

COST OF POWER.

PACIFIC GAS AND ELECTRIC COMPANY.

Kilowatt-hours generated by steam		85,707,854
Kilowatt-hours generated by transmission		7,787,959
		93,495,813
Kilowatt-hours sold		68,797,090
Kilowatt-hours lost in distribution		24,698,723
Per cent loss, 26.5.		
TOTAL COSTS.		
Revenue from sales		$2,730,248.00
Cost of generation	$729,315.00	
Cost of distribution	347,182.00	
Cost of administration	943,363.00	2,019,860.00
Net earnings		$710,388.00

UNIT COSTS, CENTS PER KILOWATT-HOUR.

Generation:	
Labor	0.225
Materials	0.731
Repairs	0.104
	1.060
Administration:	
Labor	0.271
Materials	0.082
Legal Expenses	0.021
Fire Insurance	0.005
Bad Debts	0.026
Advertising	0.008
Damages to persons	0.005
Rental	0.005
Taxes	0.153
	0.576

Distribution:	
Labor	0.216
Materials	0.098
Repairs	0.191
	0.505
Summary of Unit Costs:	
Generation	1.060
Distribution	0.505
Administration	0.576
Interest	0.006
Depreciation	0.789
	2.936

434. Elements of Power-plant Design. — The real problem which confronts the designing engineer is not so much the selection and arrangement of apparatus for a given set of conditions as it is to foresee the conditions under which the plant is likely to operate. For this reason the plans for the station should be examined and approved by an experienced designing engineer, in case expert service is not employed at the outset. It is not sufficient to have a mechanically perfect plant, though of course proper installation is of prime importance. The choice of fuel, selection of type of prime mover, size of units, provision for future expansion, and similar factors bear considerable weight upon the economy of operation. Each proposed installation is likely to be a problem in itself, and though similar plants may be used as patterns, each case should be worked out on its own merits.

The most important factor in the design of a power station is the determination of the probable load curve. This refers not only to the average yearly load but also to the maximum daily load which is likely to occur, the minimum daily load, temporary peak loads, and probable

TABLE 162.

CENTRAL STATION STATISTICS FOR THE STATE OF IOWA.

Commercial Stations.

Station Number.	Population of District.	Number of Consumers per 100 Population.	Station Rating, Kw.	Station Rating per Capita, Watts.	Station Rating per Employee, Kw.	Station Rating to Connected Load, Per Cent.	Average Load During Operation, Kw.	Ratio, Station Rating to Average Load, Per Cent.	Annual Load Factor, Per Cent.	Operating Load Factor, per Cent.	Investment per Kilowatt of Rating.	Investment per Capita.	Gross Income per Kilowatt of Rating.	Gross Income per Consumer.	Gross Income per Capita.	Gross Income per $100 Invested.	Gross Income per Kw-hr. Made, Cents.	Ratio of Expense to Gross Income. Per Cent.	Earnings from Operation per $100 Invested, Interest and Depreciation not Deducted.	Total Hours of Operation per Year.
Factor No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Group I:																				
1	340	13.2	20	59	10	111	1.7	8.2	12.8	18.0	$250	14.71	$78.00	$34.70	$4.58	$31.20	15.1	92.9	$2.20	6250
2	530	23.6	25	47	12	25	4.0	16.0	20.0	20.0	240	11.32	120.00	24.00	5.66	50.00	8.6	80.0	10.00	8760
3	600	25.0	35	58	17	....	....	....	....	....	343	20.00	77.10	18.00	4.50	22.50	....	77.8	5.00	560
4	600	26.7	75	125	37	94	5.9	7.8	13.3	13.3	93	11.67	60.00	28.10	7.50	64.30	8.8	55.6	28.57	8760
5	714	16.8	75	105	25	250	....	....	....	....	200	21.01	42.70	26.70	4.48	21.30	....	69.2	6.57	3000
6	725	20.3	40	55	13	62	....	....	....	....	324	17.88	172.50	46.90	9.52	53.30	....	55.2	23.82	8240
7	750	25.3	75	100	30	48	14.1	18.8	....	....	309	30.87	96.10	37.90	9.61	31.10	7.4	57.0	13.39	6900
8	1100	11.2	82	75	33	24	4.0	4.8	....	....	212	15.82	49.00	32.70	3.65	23.10	11.6	68.4	7.30	8758
9	1300	21.6	75	58	25	75	....	....	....	....	227	13.10	120.00	32.00	6.92	52.90	....	66.7	17.64	8300
10	1300	15.6	145	111	49	82	14.7	10.1	17.0	17.0	161	17.91	61.60	44.00	6.87	38.30	6.9	60.8	15.02	8760
11	1450	10.3	145	100	45	135	6.0	4.1	....	....	74	7.40	37.10	35.80	3.71	50.00	10.3	90.1	4.98	8760
12	1500	21.0	50	33	25	62	11.4	22.8	25.3	25.3	500	16.70	204.30	32.40	6.81	40.90	10.2	68.1	13.00	8760
13	1500	22.9	90	60	15	34	14.3	15.9	23.7	25.0	167	10.00	98.70	25.80	5.92	59.20	7.5	80.6	11.53	8300
14	1500	16.5	110	73	37	47	15.0	13.6	31.2	38.0	209	15.33	81.10	36.10	5.95	38.80	8.3	56.3	16.96	7200
15	1534	15.9	195	127	65	62	19.9	10.2	15.9	15.9	308	39.10	60.80	48.60	7.73	19.80	6.8	61.3	7.63	8760
16	1550	23.2	70	45	18	....	....	....	....	....	314	14.18	200.00	38.90	9.03	63.70	....	83.6	10.46	8200
17	1700	15.5	175	103	43	67	24.0	13.7	21.5	27.0	131	13.50	73.00	48.60	7.52	55.60	7.6	81.8	10.13	7000
18	1775	22.5	125	70	18	....	....	....	....	....	240	16.90	151.00	47.20	10.64	62.90	....	90.6	5.97	8760
Average: 1915	1137	19.3	89	78	29	79	11.3	12.1	20.1	22.2	239	17.08	99.06	35.47	6.70	43.27	9.1	72.0	11.68	7502
1914	1040	20.1	82	76	34	82	10.0	15.0	16.3	24.3	189	13.00	85.40	30.40	6.03	48.74	9.4	78.4	10.46	6943
1913	1290	15.9	86	67	34	70	17.0	20.3	19.8	32.6	219	13.15	93.70	34.70	5.35	42.80	10.3	75.8	11.06	5960
Group II:																				.....
19	2500	19.1	140	68	35	31	27.0	19.3	25.9	27.0	185	10.40	124.60	36.60	6.98	67.10	7.7	57.4	28.61	8400
20	3300	23.0	170	52	34	....	45.2	26.6	25.4	25.4	300	15.45	130.80	29.30	6.74	43.60	5.6	60.7	17.15	8760
21	3500	17.1	200	57	40	....	25.1	12.5	....	....	500	28.60	100.00	33.30	5.71	20.00	9.1	....	....	8760

Average: 1915	3,100	19.7	170	59	36	31	32.4	19.5	25.7	26.2	295	18.15	118.47	33.07	6.48	43.57	7.5	59.0	22.88	8640
1914	2,720	17.1	208	75	58	58	29.0	16.0	23.8	24.2	208	14.15	90.64	37.43	6.18	47.96	8.6	66.2	15.73	8629
1913	2,640	16.9	154	58	32	34	28.2	19.6	26.7	27.0	238	13.77	101.90	33.80	5.67	44.00	6.8	72.6	12.40	8620
Group III:																				
22	4,500	19.0	225	50	45	45	77.4	34.4	35.5	37.0	215	10.75	85.30	22.40	4.27	39.60	3.0	71.5	11.30	8400
23	4,500	13.2	400	89	67	53	70.0	17.5	36.0	36.0	355	31.60	66.20	44.50	5.89	18.70	4.3	62.3	7.04	8760
24	4,700	17.1	275	59	14	....	58.1	21.1	33.0	33.0	284	16.60	106.60	36.40	6.24	37.50	5.7	70.4	11.15	8760
25	5,100	15.5	460	90	58	91	40.0	8.7	28.0	28.0	130	11.80	53.20	31.10	4.80	40.80	7.0	65.3	14.17	8760
26	5,600	12.7	375	67	21	65	118.0	31.5	33.0	33.0	402	27.00	87.20	46.10	5.84	21.60	3.2	74.3	5.56	8760
27	7,000	22.1	700	100	58	58	109.3	15.6	36.6	36.6	107	10.70	80.00	36.10	8.00	74.60	5.8	57.3	31.89	8760
Average: 1915	5,233	16.6	406	76	44	62	78.8	21.5	33.7	33.9	249	18.07	79.75	36.10	5.84	38.80	4.8	66.8	13.52	8700
1914	5,494	17.4	367	67	33	65	74.5	21.8	26.4	26.6	302	19.17	106.09	38.74	6.59	39.77	6.1	66.5	11.38	8701
1913	5,680	14.5	489	87	47	45	67.4	14.5	23.2	23.5	261	21.30	80.10	41.50	5.94	34.10	7.2	66.1	11.59	8660
Group IV:																				
28	10,000	9.1	900	90	100	58	126.9	14.1	35.2	35.2	182	16.45	45.10	44.60	4.06	24.70	3.7	65.9	8.42	8755
29	11,700	14.2	750	64	24	44	133.0	17.7	31.2	31.2	468	29.90	97.40	44.00	6.24	20.90	6.3	68.2	6.63	8760
30	12,560	18.1	810	64	14	....	227.0	28.0	37.0	37.0	267	17.20	115.10	41.10	7.42	43.20	4.7	49.0	22.00	8760
31	12,850	12.1	1,300	101	118	43	302.9	23.3	30.3	30.3	395	39.90	57.40	48.10	5.81	14.60	2.8	56.7	6.30	8760
32	13,000	15.4	1,950	150	72	118	171.3	8.3	29.3	29.3	377	56.50	47.30	46.10	7.09	12.54	6.1	47.5	6.59	8760
33	15,785	6.3	1,850	117	28	112	182.1	9.8	41.1	41.1	430	50.70	53.30	98.70	6.26	12.30	6.2	81.8	2.25	8760
Average: 1915	12,649	12.5	1,260	98	59	75	190.5	17.0	34.0	34.0	353	35.11	69.27	53.77	6.15	21.37	5.0	61.5	8.70	8759
1914	11,850	12.5	1,217	103	68	93	187.9	14.3	31.1	31.1	362	30.89	64.42	51.20	6.19	19.47	4.7	59.4	7.86	8760
1913	12,690	12.5	1,225	103	53	86	171.6	15.0	32.9	32.9	423	44.10	68.20	45.18	5.96	15.62	4.6	54.6	7.32	8760
Group V:																				
34	25,000	11.1	1,600	64	64	31	553.3	34.7	35.8	35.8	204	13.04	90.10	51.60	5.76	44.20	3.0	39.7	26.65	8760
35	26,000	10.4	3,500	135	167	95	500.0	14.3	21.7	21.7	137	18.46	45.70	59.30	6.16	33.30	3.7	55.9	14.69	8760
36	30,000	16.8	2,420	81	50	32	450.0	18.6	26.5	26.5	194	15.67	76.60	36.80	6.18	39.50	4.7	61.3	15.25	8760
37	35,000	14.8	4,200	120	79	44	748.0	17.8	35.7	35.7	298	35.70	57.90	47.00	6.94	19.40	3.7	45.2	10.64	8760
38	42,000	11.5	9,500	226	111	90	1230.0	13.0	30.0	30.0	157	35.50	27.30	53.50	6.17	17.40	2.4	59.9	6.97	8760
39	105,538	12.1	10,500	100	51	42	2332.0	22.2	32.8	32.8	285	28.50	76.00	62.40	7.57	26.50	3.9	43.0	15.12	8760
Average: 1915	43,923	12.8	5,287	121	87	56	969.0	20.1	30.4	30.4	213	24.48	62.27	51.77	6.46	30.05	3.6	50.8	14.89	8760
1914	48,500	10.8	4,942	110	96	74	922.5	23.8	32.7	32.7	298	27.02	75.73	57.13	6.23	25.25	3.7	45.0	13.88	8760
1913	46,830	9.5	4,130	91	87	47	748.3	19.6	32.3	32.3	261	22.40	72.50	69.20	5.80	28.20	3.9	47.1	14.78	8760
Grand Ave.: 1915	10,272	16.9	1,124	86	45	70	232.2	16.8	28.3	29.0	258	21.23	87.34	40.70	6.43	37.20	6.4	65.5	12.59	8135
1914	8,920	17.1	856	82	46	74	177.9	17.7	24.5	26.9	242	18.52	86.10	38.70	6.19	40.99	7.2	66.7	11.73	7998
1913	9,470	14.6	819	77	45	56	152.0	18.3	24.9	29.5	251	18.45	87.30	41.00	5.78	37.25	7.6	67.8	11.60	7610

future increase. The station load factor and the yearly load factor which have such a marked bearing on the cost of operation may be closely approximated from the daily load curves. Steam requirements for heating and industrial purposes, water supply, and other forms of energy requirements should be considered simultaneously with the

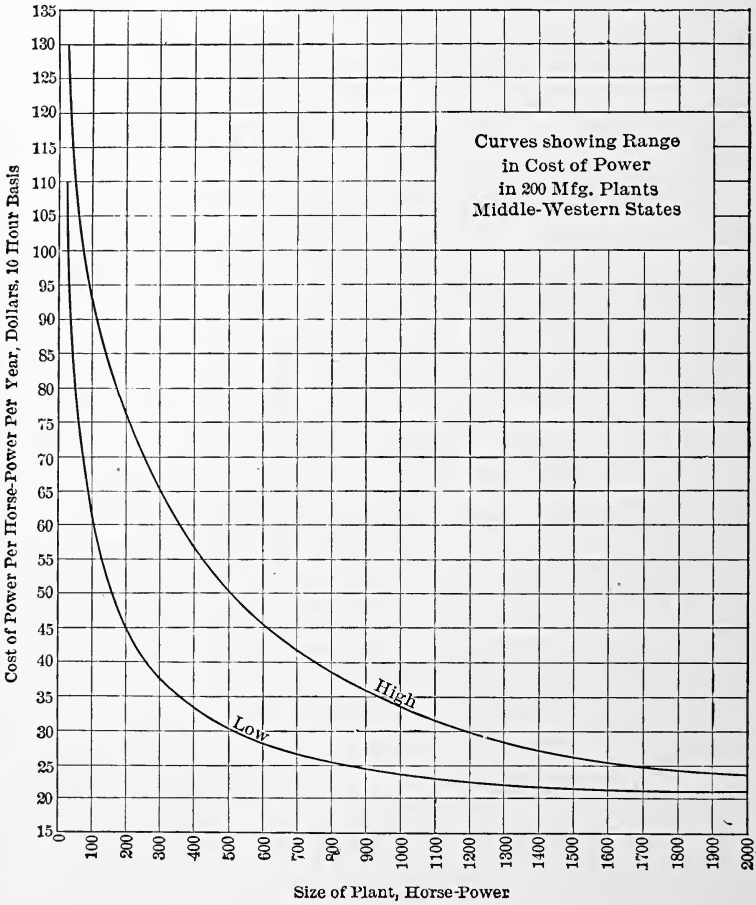


Fig. 600.

electrical demands since these factors largely influence the choice of prime mover. The curves in Figs. 601 to 603 are taken from the daily records of large power stations in Chicago and serve to illustrate the great variation in the electrical power demands for different days in the year. It is quite evident that at equipment based solely upon the average yearly requirements may not be adapted to the best economical operation.

The load curves for manufacturing plants may be predetermined with a fair degree of accuracy since the power demands for various purposes may be readily segregated and analyzed, but with public

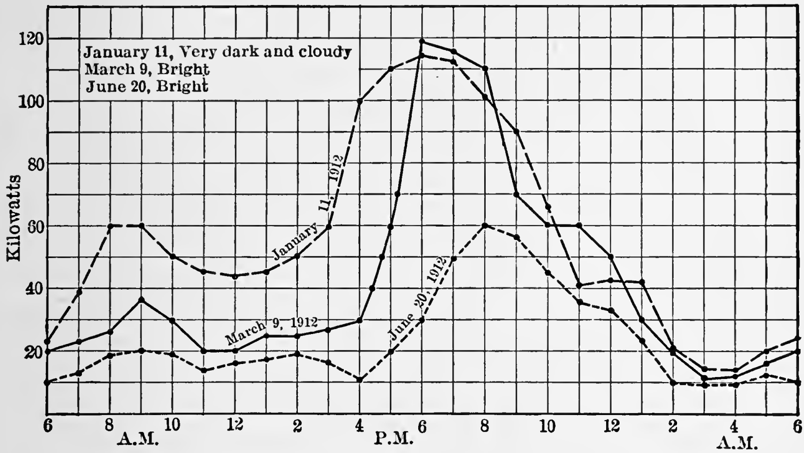


Fig. 601. Typical Daily Load Curves, Large Apartment Building.

utility concerns and certain classes of isolated stations the problem is largely a matter of judgment. Thus, in the case of an industrial plant, the power requirements for lighting, manufacturing purposes, heating, ventilation, and sanitation may be closely approximated since

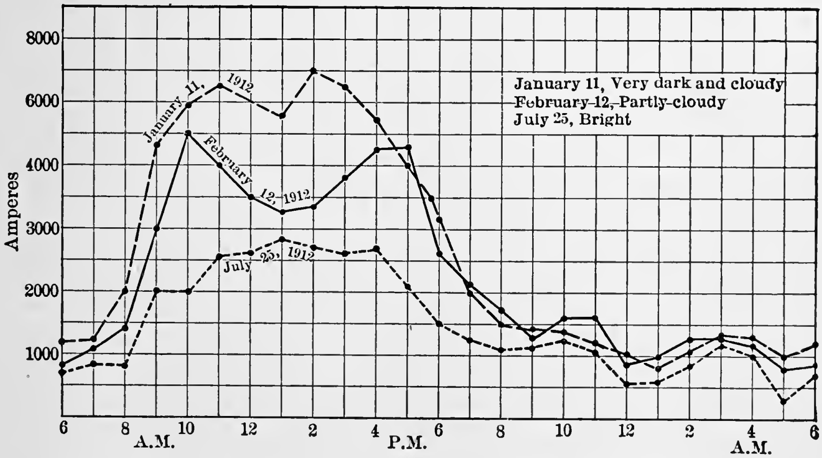


Fig. 602. Typical Daily Load Curves, Tall Office Building, Chicago.

the size of building, exposure, number of floors, and the number of elevators afford a definite basis for analysis; but with public utility concerns the probable load depends largely upon the business acumen of the management in securing customers, the location of the plant and future demands. In the latter case the load curve is based chiefly upon the experience of similar plants under comparable conditions of operation.

In any case the greatest care should be exercised in estimating the maximum peak load which is likely to occur. High peak loads with low daily average necessitate the installation of large machines which are idle or operate uneconomically the greater part of the time and

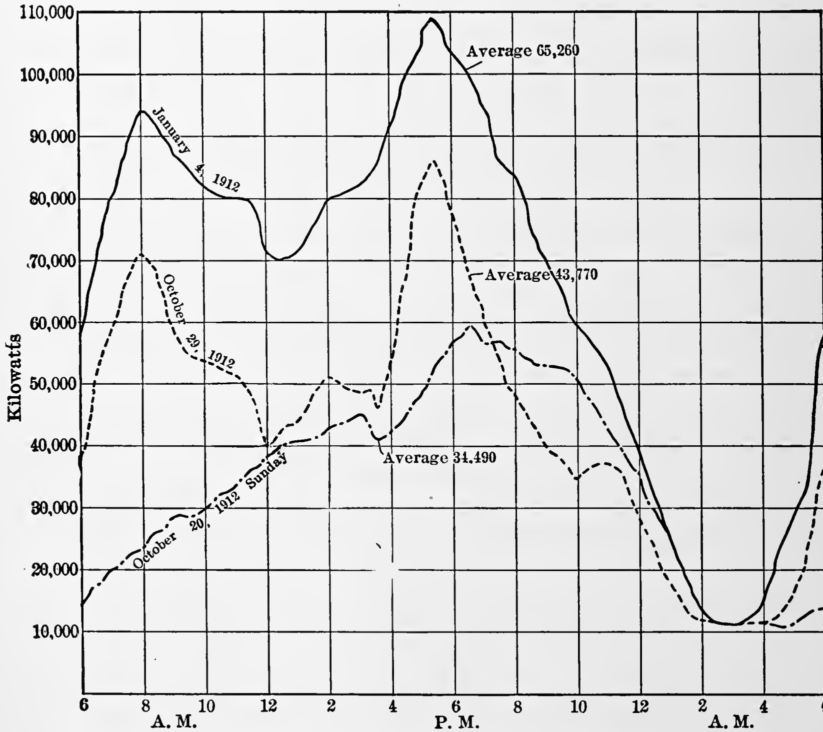


FIG. 603. Typical Daily Load Curves, Large Central Station, Chicago.

result in heavy fixed charges. The financial failure of many electric light and power plants is directly traceable to the failure to consider the influence of maximum peak loads on the ultimate cost of operation. In connection with central-station service every customer represents a certain investment, regardless of the amount of power used. Even should he consume no power, his account would have to be carried on the books and a certain amount of equipment would have to be held in readiness to serve him. In order that every customer shall incur his share of the expense, the expense of the plant must be apportioned between the capacity and output costs. The heavier the peak loads the greater will be this charge, and, as is the case with many small lighting plants where current is used but three or four hours a day, the

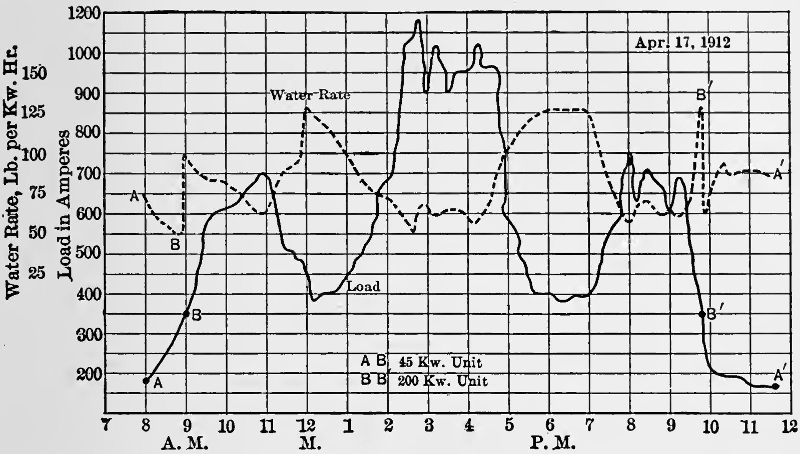


FIG. 604. Daily Load Curve showing Influence of Variable Generator Load on Steam Economy.

readiness to serve charge becomes excessive and either the station must operate at a loss or the unit cost will appear to be prohibitive.

The curves in Fig. 604 are taken from recording ammeter and recording steam meter readings of a 200-kilowatt direct-connected and a 45-kilowatt belted generator set installed at the power plant of the Armour Institute of Technology and serve to illustrate the influence of

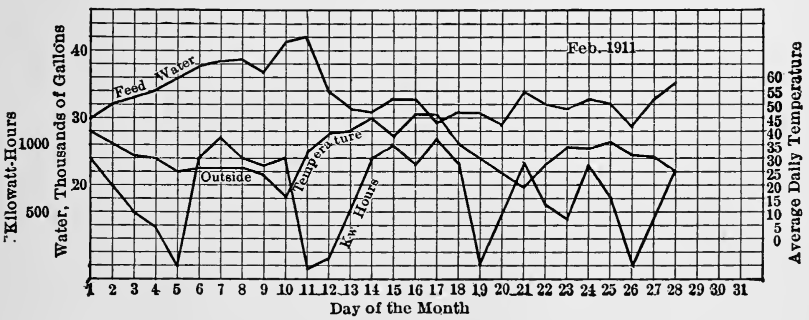


FIG. 605. Monthly Load Curves, Combined Heat and Power Plant, Armour Institute of Technology.

load on economy for very unfavorable conditions. At 8:00 A.M. the small unit is started up with initial load of about 150 amperes. As the load increases the water rate decreases, as is shown by the curve AB. At 9:00 A.M. the load is beyond the capacity of the small machine and the large unit is put into service. The increased water rate of the large unit over the requirements of the smaller is apparent by the sudden rise in the water-rate curve. This is due to the fact that the large unit is operating at only 20 per cent of its rating, against full load for the small one. The fluctuation of the water rate with the load variation is clearly shown. Evidently the two units are not of the proper size for the particular load conditions illustrated in Fig. 590. During the heating months when live steam is necessary for "make-up" purposes

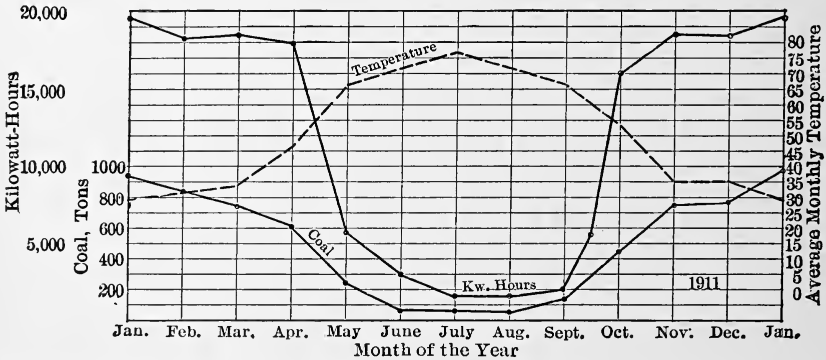


FIG. 606. Yearly Load Curve showing Influence of Temperature on Coal Consumption, Combined Heat and Power Plant, Armour Institute of Technology.

the unfavorable engine load has little effect on the ultimate economy, but during the summer months the loss from this cause may be a serious one.

The curves in Figs. 604 to 606 show that during the winter months in a combined heat and power plant the fuel requirements may be practically uninfluenced by the electrical demands and increase in electrical output does not effect an appreciable increase in fuel consumption, but the influence of the outside temperature is clearly indicated.

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PROBLEMS.

1. The rated capacity of a steam turbine station is 2000 kw. If the annual output is 6,380,000 kw-hr., required the yearly load factor.

2. If the plant in Problem 1 operates 18 hr. per day for 300 days in the year, required the station or curve load factor.

3. If the plant in Problem 1 cost $65.00 per kw. of rated capacity and the annual fixed charges amount to 14 per cent, required the fixed charges per kw-hr.

4. A plant cost originally $100,000.00. It is proposed to establish a sinking fund on a 3 per cent basis. If the weighted life of the plant is assumed to be 20 years and the junk value of the apparatus at the expiration of this period is estimated at 15 per cent of the original cost, how much money must be placed in the reserve fund each year.

5. What will be the accumulated fund in Problem 4 at the end of 15 years?

6. A steam plant erected 10 years ago at a cost of $250,000.00 is to be appraised for rate making. The average weighted life of the equipment is estimated as 25 years. What is the accrued depreciation and the present value of the plant on the "straight-line" basis. Salvage assumed to be 10 per cent of the original cost.

7. The average fuel consumption of a 30,000-kw. turbo-generator plant is 2.2 lb. coal (11,000 B.t.u. per lb.) per kw-hr. for a yearly load factor of 0.42. The cost of coal is 2.00 per ton of 2000 lb. and the fuel cost is 45 per cent of the total station operating costs. What is the total cost of operation, dollars per year?

8. A 20,000-kw. turbo-generator uses 14 lb. steam per kw-hr., initial pressure 215 lb. absolute, superheat 150 deg. fahr., vacuum 27.5 in. referred to a 30-in. barometer, feed water 180 deg. fahr. If the average overall boiler and furnace efficiency is 70 per cent and the calorific value of the coal is 12,500 B.t.u. per lb., required the average B.t.u. supplied by the fuel per kw-hr. generated. Determine also the average weight of coal used per kw-hr.

9. During the winter months all of the exhaust steam from a 500-hp. non-condensing engine is used for heating purposes. Engine uses an average of 60 lb. steam per kw-hr., initial pressure 125 lb. abs., back pressure 17 lb abs., initial quality 98 per cent, feed water 210 deg. fahr. If the average overall boiler and furnace efficiency is 65 per cent and the coal costs $3.00 per ton of 2000 lb. (calorific value 12,000 B.t.u. per lb.), what is the actual cost of fuel for power only, cents per kw-hr?

CHAPTER XIX

TYPICAL SPECIFICATIONS

435. Specifications for a Horizontal Tubular Steam Boiler. * — The following specifications for one 72-inch horizontal return tubular steam boiler, pressure 150 pounds, were prepared by the Hartford Steam Boiler Inspection and Insurance Company for the Armour Institute of Technology, Chicago:

This specification is intended to cover the construction of one horizontal tubular boiler designed to operate at a maximum pressure of 150 pounds per square inch. Each bidder must submit a proposal for doing the work exactly as specified but alternate proposals involving slight modifications will also receive consideration provided such modifications are fully described.

The Boiler Contractor shall furnish the various accessories mentioned herein and he shall also provide all the necessary miscellaneous iron or steel work as hereinafter enumerated. The Contractor under this specification will not be required to construct foundations, brickwork or other masonry.

Drawings. — Drawings prepared by The Hartford Steam Boiler Inspection and Insurance Company accompany this specification and are made a part hereof; the drawings and specification are intended to supplement each other and to be mutually co-operative, and, unless otherwise noted, the Boiler Contractor shall follow all details and shall furnish all parts and fittings which may be required by the drawings and omitted by the specification, or vice versa, just as though required by both. The said drawings are identified respectively by Nos. 6260 and 4890.

General Data. — The boiler with its fittings shall be constructed and furnished in accordance with the following general data and dimensions: —

Diameter measured on inside of largest course............72 inches.
Number of courses..Three.
Thickness of material: Heads, $\frac{9}{16}$ inch. Butt-straps, $\frac{7}{16}$ inch. Shell-plates, $\frac{17}{32}$ inch.
Girth seams: Single-riveted lap-joints with rivets spaced $2\frac{3}{8}$ inches on centers.
Longitudinal seams..................Quadruple-riveted butt-joints.
Diameter of rivets for all seams...............$\frac{7}{8}$ inch ($\frac{15}{16}$-inch holes).
Tubes: Number, 70. Diameter, four inches. Length, 18 feet. Thickness, 0.134 inch.

* Paragraphs pertaining to properties of steel plates, rivets, and tubes have been greatly abridged because of space limitation.

Braces above tubes: Number on each head, 20. Least diameter, $1\frac{1}{8}$ inches. Diameter of rivet holes for attaching, $\frac{7}{8}$ inch. Least cross-sectional area through sides at each rivet hole on head end, 0.55 square inch; ditto on shell end 1.10 square inches.

Through-braces below tubes: Number, 2. Least diameter, two inches. Least diameter of upset on front end, $2\frac{1}{2}$ inches. Diameter of pin, $1\frac{3}{4}$ inches. Least cross-sectional area through center of eye, 3.83 square inches.

Size of blow-off pipe . . . . $2\frac{1}{2}$ inches.

Diameter of nozzles: Steam opening . . . . 6 inches.
Safety valve connection . . . . 6 inches.

Size of feed-pipe . . . . $1\frac{1}{2}$ inches.

Manholes: One in front head below tubes and one in top of shell.

Size of grates . . . . 72 inches long by 66 inches wide.

Height from grates to bottom of shell, at front end . . . . 40 inches.

Smoke-Box: Bolted to front head by clip angles. Smoke opening 60 inches by 14 inches.

Style of Front . . . . Flush.

Fittings to be furnished with the boiler as follows: — One ten-inch steam gauge graduated from 0 to 225 pounds, brass siphon and union-cock for gauge, two $2\frac{1}{2}$-inch safety valves with minimum lift of 0.08 inch, flanged Y-base for safety valves, three $\frac{3}{4}$-inch gauge cocks, one combination water-column, one $\frac{3}{4}$-inch gauge glass 14 inches long.

Method of Support. — The boiler shall be suspended by means of U-bolts and steel hangers, from a framework made up of four I-beams and four columns. I-beams shall be eight inches deep and shall weigh 18 pounds per foot; they shall be assembled in pairs by means of tie-bolts and separators, spaced near each end and at intervals of not more than four feet, in such manner that the adjacent edges will be three inches apart. If cast-iron columns are used they shall be round with an outside diameter of eight inches and a thickness of $\frac{7}{8}$ inch, or square with a width of eight inches and a thickness of $\frac{3}{4}$ inch. Six-inch rolled steel H-beams, weighing 23.8 pounds per foot, may be used for columns but no other form of structural steel column will be approved unless it can be shown that the safe load (figured in the usual manner with regard to length and radius of gyration) will be equal to that which can be allowed on the H-beams specified above. Steel columns shall have suitable base-plates and cap-plates riveted on and cast-iron columns shall be made with top and bottom flanges of proper design. Details of hangers, U-bolts, etc., are shown on the accompanying drawing.

Properties of Steel Plates. — (Chemical requirements have been omitted.) Complete tests must be made to show that each plate will fulfill the above requirements in regard to tensile strength, elastic limit, chemical composition, elongation, bending, and homogeneity; and any plates failing to meet the said requirements shall be rejected. One tension, one cold-bend, and one quench-bend test shall be made from each plate as rolled. All details in regard to size and shape of specimens, method of making tests, etc., shall be in strict accordance with the "Requirements for Testing Steel," as adopted by The Hartford Steam Boiler Inspection and Insurance Company.

All tests and inspections of material may be made at the place of manufacture prior to shipment. Certified copies of reports of all tests must be approved by a representative of The Hartford Steam Boiler Inspection and Insurance Company before any of the material covered thereby is used for any portion of the work contemplated by this specification.

Stamping. — (Omitted.)

Rivets. — (Omitted.)

Details of Riveting. — Longitudinal seams shall be of the butt-joint type with double covering straps and the details shall be as specified herein and as shown on the accompanying drawing, except that the pitch of rivets in the outer row may be increased or decreased (with corresponding changes in the pitch of rivets in the other rows) in cases where such changes are desirable in order to secure a proper spacing of rivets between girth seams. It must be understood, however, that no such change can be made without the consent and approval of the inspector having jurisdiction and no such change shall be allowed if it will result in a factor of safety lower than 5.00 or if it will produce a pitch too great for proper calking. Except for rivet holes in the ends of butt-straps, the distance from the center of the rivet to the edge of the plate must never be less than one and one-half ($1\frac{1}{2}$) times the diameter of the rivet hole. The seams must be arranged to come well above the fire-line and to break joints in the separate courses.

Rivet holes shall either be drilled full size with plates, butt-straps and heads bolted up in position or else they shall be punched at least one-quarter inch ($\frac{1}{4}''$) less than full size. If the latter method is used, plates, straps, and heads shall be assembled and bolted together after punching and the rivet holes shall be drilled or reamed in place one-sixteenth inch ($\frac{1}{16}''$) larger than the diameter of the rivets. After reaming or drilling, plates and butt-straps shall be disconnected and the burrs removed from the edges of all rivet holes. If any holes are out of true more than one-sixty-fourth inch ($\frac{1}{64}''$), they must be brought into line with a reamer or drill; evidence that a drift-pin has been used for this purpose will be sufficient cause for the rejection of the entire work. The plates must be rolled to a true circle before drilling and the butt-straps and ends of plates forming the longitudinal joints must be formed to the proper curvature by pressure, — not by blows. Particular care must be used to secure proper fitting where the courses telescope together at girth seams. This is a matter of the utmost importance and the results obtained will be considered as a criterion of the general character of the workmanship throughout.

Rivets must be of sufficient length to completely fill the rivet holes and form heads equal in strength to the bodies of the rivets. Rivets shall be machine driven wherever possible, and always with sufficient pressure to entirely fill the rivet holes; the authorized inspector of The Hartford Steam Boiler Inspection and Insurance Company shall have the privilege of cutting out rivets to see if satisfactory results have been obtained and all such work of cutting rivets and replacing them shall be done at the expense of the Contractor. Rivets shall be allowed to cool and shrink under pressure.

Calking and Flanging. — All lacking edges shall be beveled to an angle of about fifteen degrees (15°) and every portion of such edges shall be planed or milled to a depth of not less than one-eighth inch ($\frac{1}{8}''$). Bevel-shearing will not be acceptable in place of planing or milling but chipping will be allowed in special cases provided the workmanship will meet with the inspector's approval. All seams must be carefully calked with a round-nosed tool.

Flanging must be performed in such manner that the flange will stand accurately at right angles to the face of the sheet and the straight portion of the flange must be long enough to allow for making a perfect joint with the shell plate. The radius of the bend, on the outside, shall be at least equal to four times the thickness of the head.

Tubes. — (Chemical requirements and method of testing have been omitted.) Each tube must be legibly stenciled with the name or brand of the manufacturer, the material from which it is made (steel or charcoal iron), and the words "Tested at 1,000 lbs."

All tests and inspections shall be made at the place of manufacture and the Boiler Contractor shall require the tube manufacturer to certify that the tubes have been tested and have met the requirements stated above. Tubes shall be rejected when inserted in the boiler if they fail to stand expanding and beading without showing cracks or flaws, or opening at the weld.

Tube holes may either be drilled full size or punched so as to have a diameter at least one-half inch ($\frac{1}{2}''$) less than full size and then drilled, reamed, or finished full size with a rotating cutter. The full size diameter of the hole shall be $\frac{1}{32}$ inch greater than the outside tube diameter. Edges of tube holes shall be properly chamfered.

Tubes shall be set with a Dudgeon expander and all ends shall be substantially beaded.

Staying. — The number, size, arrangement, and general details of stays or braces are specified on page — and shown on the drawing. No changes shall be made in the number and location of braces without the approval of The Hartford Steam Boiler Inspection and Insurance Company. All braces shall be made of solid, weldless mild steel.

Braces above the tubes shall be of the diagonal crowfoot form and none of them shall be less than three feet, six inches (3′ 6″) long. Each brace shall be attached by means of four rivets, two at each end; rivets of a larger diameter than specified on page — may be used if preferred, but the cross-sectional area through the brace at the sides of the rivet holes must be maintained as called for. Braces having a rectangular cross-section may be used provided the cross-sectional area of each brace is equal to that of each of the round braces specified, and provided also that the requirements regarding size of rivets and net area through rivet holes are fulfilled. Braces must be carefully set to bear uniform tension.

Through braces shall be used below the tubes, extending from head to head. Each brace shall be upset on the rear end to form an eye and the eye shall be inserted between the outstanding legs of a pair of angle-irons and held in place by a turned bolt passing through holes drilled in both angles and in the eye. The angles shall be securely riveted to the rear head in the manner shown on the drawing, being held at a distance of three inches from the head by means of spacers made of extra heavy pipe. Spacers must be accurately squared on both ends so that they will all be of the same length and will furnish a rigid and uniform bearing for the angles. Through braces shall be upset and threaded on the front ends and shall pass through the front head, being secured with nuts and washers both inside and outside. The center line of the braces at the front head must not be lower than the center line of the manhole.

Manholes. — Manholes shall be oval or elliptical in shape, not smaller than fifteen inches long by eleven inches wide, and shall conform to the following requirements: —

The manhole in the top of the shell shall be placed with its long dimension crossways of the boiler. The frame shall be made of pressed steel formed to the proper curvature, and it shall be riveted to the inside of the shell with two rows of rivets symmetrically spaced. Based on the allowance of 44,000 pounds per square inch the size and number of the rivets must be such that their total shearing strength will not be less than twice the tensile strength of the plate removed, as figured from the cross-sectional area in a plane passing through the center of the manhole and the axis of the shell; the net cross-sectional area of the manhole frame, as cut by such a plane, must not be less than the cross-sectional area of the plate removed in the same plane.

The manhole in the front head shall be formed by flanging the head inwardly to a depth of not less than three times the thickness of the head all around the opening and a steel band shall be shrunk on, pinned in position, and properly machined for the gasket bearing; the band will not be required if a recessed manhole plate is used.

All necessary manhole plates, yokes, bolts, and gaskets shall be furnished to make the installation complete, the various parts being proportioned so as the have a strength equal to that of manhole frames. Manhole plates and yokes shall be made of pressed steel. Gasket bearings shall be at least one-half inch ($\frac{1}{2}''$) wide and the thickness of gaskets shall not exceed one-quarter inch ($\frac{1}{4}''$).

Nozzles. — Nozzles shall be made of pressed or cast steel and shall be of heavy and substantial design properly adapted to the pressure to be carried. They must be accurately shaped to fit the curvature of the shell and must be carefully and securely riveted in place in such manner that the face of each flange after erection will lie in a horizontal plane parallel with the upper surface of the tubes. The flange of each nozzle must be properly faced.

Feed Piping. — Feed piping must be firmly supported in the boiler in such manner that no portion of the piping can be in contact with any of the tubes or other parts of the boiler. The feed-pipe shall enter the boiler through the front head by means of a brass or steel bushing placed on the left-hand side of the boiler, three inches (3″) above the top of the upper row of tubes as shown on the drawing. The feed-pipe shall extend back from the bushing to approximately three-fifths the length of the boiler, crossing over to the center and discharging above the tubes. The pipe must not discharge in proximity to any riveted joint.

All external feed-piping will be furnished under separate contract but the Boiler Contractor must leave the threads in proper condition so that the piping can be readily connected.

Blow-off Pipe Connection. — A connection for blow-off pipe shall be provided on the bottom of the shell near the rear end, as shown on the drawing. It shall consist of an extra-heavy pressed steel flange, properly tapped for the blow-off pipe and securely riveted to the boiler shell.

Fusible Plug. — A fusible plug shall be placed in the rear head, on the vertical diameter, and the center of the plug must not be less than two inches (2″) above the upper surface of the tubes. The plug must project through the sheet not less than one inch (1″).

Fusible plugs shall be filled with pure tin the least diameter of which shall be one-half inch ($\frac{1}{2}$″).

Safety Valves. — Safety valves shall be of the direct spring-loaded pop type with seats and discs of nickel or other non-ferrous material. Valves must operate without chattering and must be set and adjusted to close after blowing down not more than six pounds (6 lb.). Springs must not show a permanent set exceeding $\frac{1}{32}$ inch ten minutes after being released from a cold compression test closing the spring solid; no spring shall be used for a pressure in excess of ten per cent (10%) above or below that for which it was designed.

Each safety valve shall have a substantial lifting device with the spindle so attached that the valve disc can be lifted from its seat through a distance not less than one-tenth of the nominal diameter of the valve, when there is no pressure on the boiler.

The following items shall be plainly stamped or cast upon the body:

(*a*) The name or identifying trade-mark of the manufacturer.
(*b*) The nominal diameter with the words "Bevel Seat" or "Flat Seat."
(*c*) The steam pressure at which the valve is set to blow.
(*d*) The lift of the valve disc from its seat, measured immediately after the sudden lift due to the pop.
(*e*) The weight of steam discharged in pounds per hour at the pressure for which it is set to blow.
(*f*) The letters A.S. M.E. Std.

Safety valves having a lower lift than that specified on page 0 may be used but the diameter must be increased proportionately as directed by The Hartford Steam Boiler Inspection and Insurance Company.

In the absence of any specific directions from the Purchaser, the Boiler Contractor shall state in his proposal the make and style of valve which he intends to furnish. It is understood that failure to do this will give the Purchaser the right to specify the make of valve after the contract is awarded and, in such event, the Contractor agrees to furnish any make the Purchaser may select.

Fittings. — The foregoing in regard to choosing the make and style of safety-valves shall apply in the same manner and with equal force to the make of gauge-cocks, water-column, steam-gauge, etc.

The combination type of water-column shall be used and openings for water and steam connections must be tapped for one-and-one-quarter-inch ($1\frac{1}{4}''$) pipes. Brass pipe shall be provided for the water connection and the piping shall be made up with plugged fittings to facilitate cleaning.

The Boiler Contractor shall properly drill and tap all holes required for the installation of the various fittings, including also a one-quarter-inch ($\frac{1}{4}''$) pipe with valve for the connection of test gage. The sizes of steam-gauge, gauge-cocks, and gauge-glass are specified on page 0.

All nozzles, flanges, fittings, etc., furnished under this specification must correspond in diameter, drilling, and other details with the "American Standard" for the stipulated pressure.

Front. — The front shall be constructed of sectional plate steel or of cast iron and the Contractor must state in his proposal which form he intends to furnish. If made of steel, the plates must not be less than three-eighths inch ($\frac{3}{8}''$) thick (except for moldings, etc.) and they must be straight and smooth with all edges machined and properly fitted to make good joints. Heavy cast-iron door-frames with planed surfaces shall be securely bolted to the plates and the front shall be further reinforced against warping by means of channel irons or other suitable braces placed on the back.

If made of cast-iron, the front must be of heavy and substantial design and all castings must be smooth, true, and free from cracks, blow-holes, or other defects.

The usual fire-doors, ash-pit doors, and doors for giving access to the tubes shall be provided as shown on the accompanying drawings. All doors must be of heavy design and all contact surfaces must be carefully machined so that the doors will fit closely. Each flue door must be provided with a suitable fastening at top and bottom, designed to clamp the door tightly in the closed position and prevent warping. All doors shall be furnished complete with handles, catches, hinge-bolts, etc., and fire-doors shall have liner plates.

The Boiler Contractor shall furnish all necessary anchor bolts for holding the front in position and shall see that the holes for the same are properly located in the steel plates or castings. Anchor bolts shall have a diameter of at least seven-eighths inch ($\frac{7}{8}''$) and shall be threaded and provided with nuts.

All parts must be carefully made so that the front will present a neat appearance after erection. Open joints, loosely-fitting hinges or other indications of careless workmanship will be sufficient cause for rejection and the Purchaser shall have the option of making any necessary modifications and deducting the cost thereof from the contract price or of requiring the Contractor to furnish new parts which will be satisfactory.

Grates. — The Boiler Contractor shall figure on furnishing stationary grates of suitable design and shall base his proposal thereon. If requested by the Purchaser, he shall submit an alternate proposal for furnishing, shaking, rocking, or dumping grates of a type which the Purchaser will specify.

Miscellaneous Iron Work. — Arch-bars for rear connection shall be made as shown on the accompanying drawings or in accordance with some detail which will meet with the approval of The Hartford Steam Boiler Inspection and Insurance Company. The Company will not approve any arch-bar the metal of which is exposed to the action of the flames and hot gases.

The rear connection door must fit closely and the frame must be provided with means for anchoring into the brickwork. The door must not be smaller than sixteen inches by twenty-four inches (16″ × 24″).

The Boiler Contractor shall furnish all necessary bearer-bars for grates, all buckstays, tie-rods, lintels for clean-out doors, bolts, etc., and any other iron-work, not specifically mentioned herein, which may be needed to complete the installation in the brick setting. Buckstays must be made of pressed steel or its equivalent; cast-iron will not be accepted.

Tests. — The Boiler Contractor shall at all times afford all facilities to The Hartford Steam Boiler Inspection and Insurance Company, and its authorized representatives, for the test and inspection of all materials and workmanship entering into the work covered by this specification.

Hydrostatic tests shall be made in the presence of the authorized inspector of The Hartford Steam Boiler Inspection and Insurance Company and in a manner which will meet with the approval of the said inspector. The pressure for such tests shall not exceed one and one-half ($1\frac{1}{2}$) times the maximum working pressure as hereinbefore stated.

Local or State Laws. — All details of construction and installation shall be made in strict accordance with any local or State ordinances which may apply and nothing in this specification shall be interpreted as an infringement of such rules or ordinances. If any discrepancy should arise, the Contractor shall immediately report it to The Hartford Steam Boiler Inspection and Insurance Company for settlement.

436. Specifications for Steam, Exhaust, Water, and Condenser Piping for an Electric Power Station.* — The work referred to in this contract shall be conducted under the general supervision of ——————— (referred to as the Engineers), who shall interpret the Specifications and the Drawings that may accompany the Specifications, and shall arbitrate any controversies between the parties hereto, that may arise under this contract, their decision to be final and binding upon both of the contracting parties.

The Contractor shall comply with all laws, statutes, ordinances, acts, and regulations of the town or city, the state and the government in which the work is to be performed, and shall pay all fees for permits and inspections required thereby.

The Contractor shall, at an early date, communicate with other contractors employed by the Purchaser, and shall work in harmony with them, any differences of opinion between contractors being arbitrated by the Engineers or their representative.

The Contractor shall begin work as soon as possible, and complete same, free of all liens and charges, on or before the time mentioned herein. If, in the opinion of the Engineers, the Contractor fails to prosecute the work with the necessary means and diligence to insure

* From the files of a prominent Chicago engineering firm.

its completion within the time limit, then the Engineers shall notify the Contractor by written notice to that effect, and the Purchaser may order the Contractor to employ more men, machinery, and tools to be put upon the work, specifying the additional force required, and if the Contractor fails to comply with such written demand within six (6) days from the date thereof, or within such time as the Engineers in writing prescribe, then the Purchaser may employ necessary means to complete the work within the time required, and such additional cost caused by either the employment of additional men, machinery, or otherwise, shall be deducted from any funds due, or that may become due the Contractor on account of this contract. The Contractor shall remove any particular workman or workmen from the work, if in the judgment of the Engineers it will be for the best interest of the work.

The Engineers shall have the right to make any changes in the Drawings or Specifications that they deem desirable. Should any additional labor or material be involved in such changes, the Contractor shall be paid for supplying same; on the other hand, should such changes reduce the amount of labor or material from that originally specified, the Contractor shall sustain an equivalent reduction in the contract amount and the Engineers shall be the arbiters in determining rates of increase or reduction. No claim shall be allowed for extra labor or material above the contract amount, unless same shall have been ordered in writing, with remuneration stipulated, by the Engineers. Acceptance by the Contractor of final payment on the contract price shall constitute a waiver of all claims against the Purchaser.

All material and workmanship furnished under this contract must be of the best quality in every particular and the Contractor must remedy any defects which develop during the first year of actual service, due to faulty material or workmanship, free of expense to the Purchaser. The Purchaser, the Engineers, or their representative may inspect any machinery, material or work to be furnished under this contract and may reject any which is defective or unsuitable for the uses and purposes intended, or not in accordance with the intent of this contract, and may order the Contractor to remedy or replace same; or the Purchaser may, if necessary, remedy or replace same at the expense of the Contractor.

Until accepted in its entirety by the Purchaser, all work shall be done at the Contractor's risk, and if any loss or damage should occur to the work from fire or any other cause, the Contractor shall promptly repair or replace such loss or damage free of all expense to the Purchaser. The Contractor shall be responsible for any loss or damage to material, tools or other articles used or held for use in or about the work.

The work shall be carried on to completion without damage to any work or property of the Purchaser or of others, and without interfering with the operation of their machinery or apparatus.

The Contractor shall furnish all false work, tools and appliances that may be required to accomplish the work and shall remove all débris after erection.

The Contractor must be responsible for the safety of the work until finished and accepted by the Purchaser and must maintain all lights, guards, and temporary passages necessary for that purpose. In case of any accident causing injury to person or property, the Contractor shall obtain acquittance from or pay the injured person (whether such person be an employee, a fellow-contractor, an employee of a fellow-contractor, or otherwise) the amount of damages to which he or she may be legally entitled on account of any act or omission of the Contractor or of any agent or employee of the Contractor, during the performance of the work referred to herein, and shall provide adequate insurance to protect the Purchaser from all claims arising therefrom. The Contractor shall, further, insure the compensation provided for in any workman's compensation act which may affect the work, to all its employees or their beneficiaries, and the Contractor shall carry insurance in a company satisfactory to the Purchaser, insuring said compensation to its employees or their beneficiaries. The Contractor shall notify his insurance company and cause the name of the Purchaser to be incorporated in the compensation policy, the policy or a copy thereof to be deposited with the Purchaser upon request. The Contractor must save the Purchaser harmless from all claims for damages set up by reason of any such injury and from all expenses resulting therefrom.

No certificates given or payments made shall be considered as conclusive evidence of the performance of this contract, either wholly or in part, nor shall any certificate of payment be construed as acceptance of defective work or improper materials. The Contractor agrees to furnish the Purchaser or the Engineers, if requested, at any time during the progress of the work, a statement showing the Contractor's total outstanding indebtedness for material and labor in connection with the work covered by this contract, such statement to be certified to by a notary public. Before final payment is made the Contractor shall satisfy the Purchaser by affidavits or otherwise, that there are no outstanding liens for labor or materials against the Purchaser's premises by reason of any work done or materials furnished under this contract.

If, during the progress of the work, the Contractor should allow any indebtedness to accrue for labor or material to sub-contractors or others, and should fail to pay and discharge same within five (5) days after demand made by any person furnishing such labor or material, then the Purchaser may withhold any money due the Contractor until such indebtedness is paid, or apply same toward the discharge thereof.

All royalties for patents, or charges for the use or infringement thereof, that may be involved in the construction or use of any machinery or appliance referred to herein, shall be included in the contract price, and the Contractor must satisfy all demands of this nature that may be made against the Purchaser at any time.

This contract shall not be assigned nor shall any part of the work be sub-let by the Contractor without the written consent of the Engineers being first obtained, but such approval shall not relieve the Contractor from full responsibility for the work included in this contract and for the due performance of all the terms and conditions of this contract; and in no case shall such approval be granted until such Contractor has furnished the Purchaser with satisfactory evidence that the Sub-contractor is carrying ample workmen's compensation insurance to the same extent and in the same manner as is herein provided to be furnished by the Contractor.

General Data.

The work herein referred to comprises the furnishing of all material and labor for the complete installation of Piping Systems for two (2) ——— kw. units to be installed in the Power Station being erected by

Each of the two (2) units is comprised of the following machinery:

(List of machinery omitted.)

All of the above machinery will be installed on the foundations by their respective contractors, and this Contractor shall make all piping connection to same unless otherwise mentioned.

Drawings. (These have been omitted.)

This contractor shall take such measurement at the building and allow for such make-up pieces as shall be necessary to make his work come true, as the Purchasee and its Engineers cannot be responsible for the exact accuracy of the dimensions given on Drawings.

The Drawings and Specifications must be taken together and any work called for in the one or indicated in the other, or such work as can be reasonably taken as belonging to the Piping Connections and necessary to complete the system, is to be included.

Live Steam Piping.

Connections from Boilers. — Each of the eight (8) boilers will be provided with two (2) 8-inch steam outlets to which this Contractor shall connect an 8-inch angle automatic stop and check valve with 7-inch outlet. From these valves Contractor shall provide 7-inch boiler leads connecting to the steam mains with gate valve at the mains, all arranged as indicated on Drawings, Nos. — and —.

Connections to Turbines. — Contractor shall provide a cast-steel manifold at rear of each of the two boilers on each unit on both sides of boiler room and connect to these manifolds the two 7-inch leads from the four boilers on each unit. From manifold at rear of boilers on north side of boiler room on each unit a 14-inch connection shall be run across the basement of firing room and connected together with 14-inch lead from manifold at rear of boilers on south side of boiler room of each unit into an 17-inch pipe, which shall be connected to the turbines. A 14-inch hydraulically operated valve shall be provided on each 14-inch line where they connect together into the 17-inch turbine lead; a gate valve shall be provided on turbine lead.

Connections shall be provided complete with cast-steel manifolds, valves, drip pockets, pipe lengths and bends, all of sizes and arranged as indicated on Drawings, Nos. — and —.

Steam Loops. — Contractor shall provide the 12-inch steam loops between the steam leads to turbines complete with pipe bends and a hydraulically operated gate valve on each end of loop. Hydraulically operated gate valves shall also be provided for connecting the future loop, all as indicated on Drawings, Nos. — and —.

Steam to Auxiliaries. — This Contractor shall install a 4-inch auxiliary steam header along division wall between turbine and boiler rooms, with connections to manifolds at rear of boilers on south side of boiler room with gate valve at each manifold, all arranged as indicated on Drawings, Nos. — and —. From the auxiliary header connections shall be made to one service pump in condenser well, three feed pumps in boiler room, exciter in turbine room, two auxiliary oil pumps on turbines and to tempering coils on air washers, as shown on Drawings. The steam connection to each of the pumps must be provided with angle or globe throttle valve at pump. A gate valve must be provided on each connection near header, as indicated on Drawings. Each of the three (3) turbine-driven feed pumps will be provided with a 3-inch pressure governor by Pump Contractor, which this Contractor shall install in the steam line. The steam-driven service pump will be provided with a 2-inch pressure governor by Pump Contractor, which this Contractor shall install, providing a by-pass with three valves around same, one of which is to be the throttle valve, the other two gate valves. This Contractor shall also provide a 3-inch steam connection to the exciter, providing a globe valve at turbine and gate valve at header.

On the steam connections to the oil pumps and air washers this Contractor must provide a 1-inch extra heavy pressure-reducing valve with by-pass around same for each unit. These shall reduce from 250 pounds to 100 pounds, and a second reducing valve shall be provided on connections to air washers reducing from 100 pounds to 10 pounds.

Steam from Turbines to Heaters. — The Contractor shall furnish and install the 5-inch steam connections from outlet on intermediate stage of each turbine to the auxiliary exhaust line connecting to feed-water heaters with automatic stop and check valve, regulating valve operated by thermostat in feed-water heater, set so as to heat water to about 120 deg. fahr., pressure-reducing valve and gate valve at header, as shown on Drawings. The exhaust from steam-driven auxiliaries will go to the heaters, and it is the intention to take necessary additional steam from second stage of turbine to heat the feed water to required temperature.

Steam Connections to Soot Ejectors. — Contractor shall provide a $1\frac{1}{2}$-inch steam header lengthwise on each side of boiler room, with connections to cast-steel manifolds in main steam connections with gate valve at north side of boiler room and to auxiliary steam header with valve on south side of boiler room. From these $1\frac{1}{2}$-inch headers a 1-inch connection with globe valve having extended stem shall be run to the ejectors in basement, for each of the two divisions of each of the eight economizers, all arranged as indicated on Drawings, Nos. —, — and —.

Steam Ejectors on Condenser Discharge Pipes. — Contractor shall provide a 4-inch ejector on top of each of the two (2) 54-inch condenser discharge pipes. These shall be of Schutte & Koerting or other make that Engineers may approve. To each of these ejectors Contractor shall provide a 1-inch steam connection with valve on both ends of line; also run a 4-inch discharge connection to 6-inch bilge pump discharge line with gate and check valve on each line.

Supports for Live Steam Piping. — The main supporting beams upon which the manifolds and fittings are supported will be provided by contractor for building steel, but this Contractor shall furnish the steel brackets framing to the main members above mentioned; also all roller and anchor bearings, complete with base castings, rollers, straps, spring, etc., all as indicated and detailed on Drawings. He shall provide the steel frames for supporting the 14-inch steam load across the boiler room basement. He shall also provide the bearings for supporting the pipes on those supports. This Contractor shall also provide the main anchor bearings for the 17-inch steam loads to turbines; also the roller bearings and brackets for the 17-inch steam load to Unit No. 2.

The steel brackets for supporting the auxiliary steam header will be provided by Contractor for Building Steel, but this Contractor shall provide the roller and anchor bearings on these brackets, all as indicated on the Drawings.

Contractor shall also provide such additional hangers, braces and supports for the steam piping as may be necessary to properly support the steam piping, and keep same free from vibration. These must in all cases be of steel or iron, and made subject to the approval of the Engineers.

Steam Drips and Drains. — The main steam headers shall be drained to the 10-inch drip pockets in boiler room basement. This Contractor shall provide and install a $1\frac{1}{2}$-inch steam trap for each unit for draining the drip pocket and must connect up same with a $1\frac{1}{2}$-inch pipe. The discharge from the trap shall be connected to the feed-water heater. Connections at trap shall be arranged with by-pass with three valves, so trap can be cut out of service.

Each of the 7-inch gate valves on steam leads from boilers shall have a boss tapped for $\frac{1}{2}$-inch drain above seat, which this Contractor shall connect into a $1\frac{1}{4}$-inch line for each unit and connect same with stop and check valve to the feed-water heater, also to the clear water reservoir; $1\frac{1}{4}$-inch lines to be cross connected with valves. Contractor shall provide a boss tapped for $\frac{3}{4}$-inch drain on the 12-inch hydraulically operated gate valves on steam loop, also on the two 14-inch valves on lead from manifolds at rear of boilers for each unit, and connect same with a $1\frac{1}{4}$-inch pipe to their respective steam traps, providing by-pass with valves as indicated diagrammatically on drawings. The 12-inch gate valve for future steam loop shall also have boss tapped for $\frac{3}{4}$-inch drain and connected to the $1\frac{1}{4}$-inch drain line. A globe valve shall be provided on each drain connection. Contractor shall also tap the blind flange on tee in steam connection to condenser well and provide a $\frac{3}{4}$-inch drain connection with trap and discharge connection to the feed-water heater. A by-pass connection with three valves shall be provided at trap. A $\frac{1}{2}$-inch drain shall also be provided from lowest point of steam connection in condenser well to drain sump.

Contractor shall run a $\frac{3}{4}$-inch drain with valve from the steam casing of the three auxiliary turbines driving the boiler-feed pumps and the turbine driving the exciter and connect them into a 1-inch line and run to the hot water reservoir. Drain from casing of service pump turbine to be run to drain sump in condenser well with a valve at turbine.

Contractor shall also provide such other drip and drain connections as may be necessary to properly drain the entire system of steam connections, these to be connected as may be directed by the Engineers.

Blow-off Connections.

Boiler Blow-off Connections. — Each of the eight boilers will be provided with six (6) $2\frac{1}{2}$-inch blow-off fittings on mud drums, which this Contractor shall connect up to a special fitting on each side of each boiler and from which $2\frac{1}{2}$-inch connections shall be made to the blow-off header under each row of boilers. Eight (8) $2\frac{1}{2}$-inch blow-off valves shall be provided on the blow-off connection from each of the eight boilers, all arranged as indicated on Drawings.

Contractor shall also provide the 4-inch blow-off header under each row of boilers and run 4-inch connections from same to the steel blow-off tank in boiler-room basement. This tank will be furnished and installed by Contractor for steel tanks, but this Contractor shall provide the overflow and drain connections to discharge well and vent connections to atmosphere, all of sizes and arranged as indicated on the Drawings.

Superheater Blow-off Connections. — This Contractor shall furnish and install the superheater blow-off connections from each of the eight boilers to the blow-off header in basement, as indicated on Drawings. Each boiler will be provided with two (2) 2-inch elbows and two (2) 2-inch valves, one on each end of each drum and two elbows and two valves on superheater, which this Contractor must connect to the headers. Six (6) 2-inch valves must be provided for these connections on each boiler, all arranged as indicated on Drawings.

Blow-off from Economizers. — Each of the eight (8) economizers will be provided with eight (8) $2\frac{1}{2}$-inch blow-off outlets, provided with angle valves. This Contractor shall connect these together to a 4-inch header, providing a $2\frac{1}{2}$-inch valve on each of the two divisions on each of the eight economizers. Headers shall be run along just below economizer floor, and 4-inch connection shall be run to hot water reservoir and 4-inch to discharge line from blow-off tank. A globe valve with extended stem shall be provided on each of these connections. A check valve shall also be provided where connection is made to discharge from blow-off tank. On the economizer side of these globe valves tee shall be tapped for $\frac{1}{2}$-inch pipe and connection run to pet cock above boiler-room floor, which shall drain into a funnel connected to discharge well.

Exhaust Connections.

Exhaust Connections from Turbines. — This Contractor shall furnish and install the 42-inch free air exhaust connections from each of the two (2) turbines, as indicated on Drawing No. —, made up of cast-iron pipe and fittings and riveted steel pipe with forged steel riveted flanges, as made by the American Spiral Pipe Works. The steel pipe shall be close riveted and thoroughly calked so as to be air and water tight. Copper expansion joint shall be provided between main turbine exhaust and relief valve on each unit. The vertical risers shall be of $\frac{1}{8}$-inch plate and shall terminate above roof, with hoods over same, as per detail on Drawings. Horizontal pipe between relief valve and base elbow shall be of $\frac{3}{16}$-inch steel plate. There is to be no longitudinal seam on bottom of this pipe. The exhaust relief valves in these lines shall be as hereinafter specified under "Material and Workmanship."

Exhaust Connections from Auxiliaries. — This Contractor shall connect up the exhaust outlet on the three (3) turbine-driven feed pumps, auxiliary oil pumps, service pump and exciter together, and make connection to each of the two feed-water heaters, with gate valve at each pump, each heater and sectionalizing valve between heaters, all of sizes and arranged as indicated on Drawings. A 10-inch riser to atmosphere with combination back pressure and relief valve near heater and ——— exhaust head above roof shall be provided on connections to each of the two heaters. Exhaust heads shall be of No. 16 galvanized iron and of most improved type. Each heater will also be provided with a 4-inch relief outlet, which this Contractor shall connect up with a back pressure valve to the 10-inch relief pipe to atmosphere on each unit, all arranged as indicated on Drawings.

Heating System for Switch House, Operating Room and Offices. — Contractor shall furnish and install for heating switch house, operating room, and offices, a complete two-pipe heating system, with overhead supply system and drain in basement. The switch house heating system shall have a total direct radiation of approximately 1912 square feet, divided into 17 radiators. The operating room, offices, bedrooms, stair hall, etc., at end of turbine room shall have a total radiation of approximately 3188 square feet, divided into 55 radiators, all of sizes and arranged as may be directed by the Engineers. A layout drawing showing size of radiators and sizes of branch connections will be provided later. All radiators to be "———————————————" two-column radiators, or other make that the Engineers may approve. All radiators to have top steam connections.

Steam for this system shall be taken from the auxiliary exhaust header in boiler room, with a 6-inch connection running up the stair hall to the bus chamber under switch house, with gate valve and 3-inch safety valve set at 5 pounds pressure in boiler room. A low-pressure header shall be run across the bus chamber and up to the overhead header in switch house, which shall be run along the south wall and connected to the radiators in switch house. An overhead line shall also be run around three sides of the office space over switchboard room with drop connections to the radiators on the different floors.

Drains from the radiators shall all be brought together and connected to a direct-connected, geared, motor-driven vacuum pump as made by the American Steam Pump Co. and of ample capacity for the service and to maintain a vacuum of 5 inches at the outlet of radiators. Motor to be similar to those hereafter specified and must be complete with starting equipment switches, fuses, etc. All wiring between motor and equipment to be provided.

Discharge from pump shall be connected to the feed-water heater by means of a float-controlled vent, as made by ——— Company.

A $\frac{1}{2}$-inch syphon trap shall be provided on outlet of each radiator, as made by ———, and a standard radiator valve provided on inlet of each radiator. All piping to be rigidly suspended in approved manner.

Safety Valve Vent Pipe. — This Contractor shall furnish and install the safety valve vent pipes on each of the eight (8) boilers, as shown on Drawings, Nos. ————. The Discharge openings of the six (6) $4\frac{1}{2}$-inch safety valves on drum of each boiler shall be connected together as indicated, and a 12-inch riser run through roof and terminating in a 12-inch tee. He shall also furnish and install the safety valve vent pipes from the discharge openings on each of the two (2) 4-inch superheater safety valves on each of the eight (8) boilers. The outlets of two valves shall be combined into a 6-inch pipe and run through roof terminating in a 6-inch tee. A $\frac{1}{2}$-inch drain pipe shall be provided on elbows at each safety valve, connecting into a $\frac{3}{4}$-inch pipe from each boiler, which shall be run to ash pit.

Exhaust Drips. — This Contractor shall install a $2\frac{1}{2}$-inch drip pipe from the 42-inch free exhaust from each turbine, providing a deep U-trap and discharging into hot water reservoir under boiler room basement floor.

The Turbine Contractor will connect up the drains from the carbon packing rings into a 3-inch pipe on each of the two (2) turbines. This Contractor shall connect each of these pipes to the hot water reservoir. Gate valves on vertical connections from auxiliaries shall be tapped above seats for $\frac{1}{2}$-inch bleeders, which shall be connected together into a 1-inch line and run to hot water reservoir. Drain from gate valve on service pump shall be run to drain sump in condenser well.

Support for Exhaust Piping. — Relief valves on turbine exhaust lines shall be provided with bases, which will be supported from floor under valves, and the vertical risers will be carried on the base elbows, but this Contractor shall provide and set angle iron braces for vertical risers, as per detail.

This Contractor shall provide all necessary anchors, hangers, and braces for properly supporting the auxiliary exhaust lines, as may be required by the Engineers.

Water Piping.

Circulating Water Connections. — Purchaser will provide and install the suction connection from intake crib to the suction inlet on each of the two circulating pumps.

Condenser Contractor will provide the discharge connection from circulating pump to condenser on each unit.

Purchaser will furnish and install the condenser discharge piping outside of consender well, including gate valves, elbows, and vertical pipe length in discharge well, but this Contractor shall provide the special fitting, pipe lengths, and expansion joints on condenser discharge connections inside of condenser well. One of the pipe lengths on discharge connection from Unit No. 1 in the condenser well will be provided on ground by Purchaser, but this Contractor shall install same, providing gaskets and bolts for making up joints, all arranged and of sizes as indicated on Drawing ——. Contractor shall also provide the 6-inch tail pipes from 54-inch gate valves in discharge well.

Hot-well Pump Connections. — Contractor shall connect up the two hot-well pump discharge outlets on each unit to the inlet on primary heater in upper section of condenser, providing check and gate valve at each pump. From outlet of primary heater, connection shall be run to inlet on top of heater of each unit. The primary heater is also to be by-passed with necessary valves, all of sizes and arranged as indicated on drawings, Nos. ————. Connections to heaters shall be cross connected with valves as indicated on Drawings.

Feed Pump Suction Connections. — Contractor shall furnish and install the suction connections to the two (2) feed pumps on each unit with connections from heater, filtered water header and unfiltered water system with valve on each connection, all of sizes and arranged as indicated on Drawings, Nos. ————. Suction connections from heaters shall be cross connected with valve as indicated.

Boiler-Feed Piping. — This Contractor shall furnish and install discharge connections from the feed pumps to the feed headers and from feed headers to economizers and boilers, all arranged as shown on Drawings. There are to be two separate feed-water systems for each unit with independent connections from pumps to boilers, as shown. The auxiliary feed header is to be run in the boiler room at rear end between boilers and in basement across firing room to boiler on north side of room, with connections from same to boilers. The main feeder header shall be suspended from the economizer floor framing over boilers with connections to each of the eight (8) economizers and from economizers to the boilers. Connections between the economizer divisions will be provided by Economizer Contractor.

Each boiler will have two (2) feed inlet connections and Boiler Contractor will provide a 4-inch automatic stop and check valve on each of these outlets, to which this Contractor shall connect.

Each economizer will be provided with a 4-inch inlet at bottom and a 4-inch outlet at top, which this Contractor shall connect up.

From the 7-inch auxiliary feed headers, this Contractor shall run a 4-inch connection up the front of boilers, with a 4-inch connection to the inlet at each end of drum, providing a gate valve at header connection and a globe and check valve in horizontal run at front of boiler.

From 7-inch main feed headers, Contractor shall make a 4-inch connection to each economizer with two gate valves on each connection. He shall also make a 4-inch connection from outlet of each economizer to the feed line connecting to each of the boilers, providing a gate and check valve at economizer outlet and an angle globe valve with extended stem all arranged as indicated on Drawings.

Contractor shall provide two air chambers on each of the two main feed headers, and one air chamber on each of the two auxiliary headers, with gate valve on headers and with compressed air connections with extra-heavy stop and check valves.

Contractor shall provide a 6-inch cross connection between the two (2) 7-inch main feed lines and auxiliary feed lines, with gate valve on each connection, as indicated. Connections at pumps shall be arranged with special two-way check valves and gate valve, all of sizes and arranged as indicated on Drawings.

Water Connections to Hydraulically Operated Valves. — This Contractor shall provide and connect up a four-way cock for the hydraulically operated valve on the steam lead to turbine; the two 14-inch valves on steam lead from boilers; the 12-inch valve on steam loop on each unit and the 12-inch valve for future steam loop. The four 4-way cocks on each unit are to be located in a box set in the division wall between boiler and turbine rooms, all as indicated on Drawings. Boxes shall also be provided by this Contractor. Water supply for the four-way cocks is to be taken from both the feed headers, with gate and check valves arranged as indicated on Drawing. Drain connections with troughs and drain pipes connected to hot water well are to be provided as indicated.

The following items included in the complete specifications have been omitted:

High-pressure Boiler Washing System.
Service Water Piping.
Make-up Water Connections.
Water Drains.
Miscellaneous Drains and Vents.
Oil Connection to Turbines.
Pipe and Fittings for Oiling Systems.
Compressed-air System.
Air Washer Circulating Pump Suction.
Floor and Wall Thimbles.
Hose.
Thermometers and Gauges.

Material and Workmanship.

General Instructions. — All material and workmanship supplied under these Specifications shall be the best of their respective kinds.

All material shall be such as specified herein and free from defects or flaws of any kind, and subject to such tests and requirements as may be herein described or as may be necessary to prove the effectiveness of the material or workmanship. All labor is to be performed by men skilled in their particular line of work, and to the full satisfaction of the Supervising Engineers or their representatives. The Specifications contemplate the very best quality of material and the most mechanical character of workmanship.

All of the work shall be erected, ready for practical use, to the satisfaction of the Engineers, and all bolts, gaskets, and necessary adjuncts shall be furnished by this Contractor.

This Contractor shall satisfy himself as to the accuracy of the Drawings, and must take such measurements and allow for such make-up lengths or pieces as may be necessary to make his work come accurately together. The piping must be erected so as to preserve accurate alignment and no iron gaskets or fillers will be allowed between flanges.

Where the work of this Contractor connects to that of another, the connections shall be made by this Contractor, and he must see that all flanges for connection to the other work are properly drilled to fit the latter, irrespective of drilling dimensions on the Drawings or herein given.

The work contemplated herein shall be carried on so as to harmonize and not interfere with the work of other contractors or with the operation of the Station or any of the machinery that may be contained therein. Where connections are made to the old work, they shall be done at such time as shall meet the approval of the Chief Engineer of the Station. The work shall be installed as expeditiously as possible and subject to the general direction of the authorized Engineers.

The following items pertaining to material and construction details are included in the complete specifications but have been omitted from this copy.

Steel Pipe.
Welded Flanges.
Threaded Flanges and Unions.
Fittings.
Valves.
Hydraulically Operated Valves.
Relief Valves.
Special Valves and Appliances.
Traps.
Flanged Joints.
Cast-iron Pipe.
Supports and Hangers.
Testing.
Pipe Covering.
Painting.

437. Government Specification and Proposal for Supplying Coal.

U. S. TREASURY DEPARTMENT.

United States......................................

................................, 190..

PROPOSAL.

Sealed proposals will be received at this office until 2 o'clock p. m., ..., 190.., for supplying coal to the United States ....building at ....as follows:

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....

The quantity of coal stated above is based upon the previous annual consumption, and proposals must be made upon the basis of a delivery of 10 per cent more or less than this amount, subject to the actual requirements of the service.

Proposals must be made on this form, and include all expenses incident to the delivery and stowage of the coal, which must be delivered in such quantities, and at such times within the fiscal year ending June 30, 190 , as may be required.

Proposals must be accompanied by a deposit (certified check, when practicable, in favor of..) amounting to 10 per cent of the aggregate amount of the bid submitted, as a guaranty that it is bona fide. Deposits will be returned to unsuccessful bidders immediately after award has been made, but the deposit of the successful bidder will be retained until after the coal shall have been delivered, and final settlement made therefor, as security for the faithful performance of the terms of the contract, with the understanding that the whole or a part thereof may be used to liquidate the value of any deficiencies in quality or delivery that may arise under the terms of the contract.

When the amount of the contract exceeds $10,000, a bond may be executed in the sum of 25 per cent of the contract amount, and in this case, the deposit or certified check submitted with the proposal will be returned after approval of the bond.

The bids will be opened in the presence of the bidders, their representatives, or such of them as may attend, at the time and place above specified.

In determining the award of the contract, consideration will be given to the quality of the coal offered by the bidder, as well as the price per ton, and should it appear to be to the best interests of the Government to award the contract for supplying coal at a price higher than that named in lower bid or bids received, the award will be so made.

The right to reject any or all bids and to waive defects is expressly reserved by the Government.

DESCRIPTION OF COAL DESIRED.*

Bids are desired on coal described as follows:

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Coals containing more than the following percentages, based upon dry coal, will not be considered:

Ash.. ....per cent.
Volatile matter.............................. ....per cent.
Sulphur...................................... ....per cent.
† Dust and fine coal as delivered at point of consumption.. ....per cent.

DELIVERY.

The coal shall be delivered in such quantities and at such times as the Government may direct.

In this connection, it may be stated that all the available storage capacity of the coal bunkers will be placed at the disposal of the contractor to facilitate delivery of coal under favorable conditions.

After verbal or written notice has been given to deliver coal under this contract, a further notice may be served in writing upon the contractor to

* Note. — This information will be given by the Government as may be determined by boiler and furnace equipment, operating conditions, and the local market.

† Note. — All coal which will pass through a $\frac{1}{8}$-inch round-hole screen.

make delivery of the coal so ordered within twenty-four hours after receipt of said second notice.

Should the contractor, for any reason, fail to comply with the second request the Government will be at liberty to buy coal in the open market, and to charge against the contractor any excess in price of coal so purchased over the contract price.

SAMPLING.

Samples of the coal delivered will be taken by a representative of the Government.

In all cases where it is practicable, the coal will be sampled at the time it is being delivered to the building. In case of small deliveries, it may be necessary to take these samples from the yards or bins. The sample taken will in no case be less than the total of one hundred (100) pounds, to be selected proportionally from the lumps and fine coal, in order that it will in every respect truly represent the quality of coal under consideration.

In order to minimize the loss in the original moisture content the gross sample will be pulverized as rapidly as possible until none of the fragments exceed ½ inch in diameter. The fine coal will then be mixed thoroughly and divided into four equal parts. Opposite quarters will be thrown out, and the remaining portions thoroughly mixed and again quartered, throwing out opposite quarters as before. This process will be continued as rapidly as possible until the final sample is reduced to such amount that all of the final sample thus obtained will be contained in the shipping can or jar and sealed air-tight.

The sample will then be forwarded to the Chief Clerk of the Treasury Department, care of the storekeeper.

If desired by the coal contractor, permission will be given to him, or his representative, to be present and witness the quartering and preparation of the final sample to be forwarded to the Government laboratories.

Immediately on receipt of the sample, it will be analyzed and tested by the Government, following the method adopted by the American Chemical Society, and using a bomb calorimeter. A copy of the result will be mailed to the contractor upon the completion thereof.

CAUSES FOR REJECTION.

A contract entered into under the terms of this specification shall not be binding if, as the result of a practical service test of reasonable duration, the coal fails to give satisfactory results due to excessive clinkering, or to a prohibitive amount of smoke.

It is understood that the coal delivered during the year will be of the same character as that specified by the contractor. It should, therefore, be supplied, as nearly as possible, from the same mine or group of mines.

Coal containing percentages of volatile matter, sulphur, and dust higher than the limits indicated on line 54, and coal containing a percentage of ash in excess of the maximum limits indicated in the following table, will be subject to rejection.

In the case of coal which has been delivered and used for trial, or which has been consumed or remains on the premises at the time of the determination of its quality, payment will be made therefor at a reduced price computed under the terms of this specification.

Occasional deliveries containing ash up to the percentage indicated in the column of "Maximum limits for ash," on page 912, may be accepted.

Frequent or continued failure to maintain the standard established by the contractor, however, will be considered sufficient cause for cancellation of the contract.

Payment will be made on the basis of the price named in the proposal for the coal specified therein, corrected for variations in heating value and ash, as shown by analysis, above and below the standard established by contractor in this proposal. For example, if the coal contains two (2) per cent, more or less, British thermal units than the established standard, the price will be increased or decreased two (2) per cent accordingly.

The price will also be further corrected for the percentages of ash. For all coal which by analysis contains less ash than that established in this proposal a premium of 1 cent per ton for each whole per cent less ash will be paid. An increase in the ash content of two (2) per cent over the standard established by contractor will be tolerated without exacting a penalty for the excess of ash. When such excess exceeds two (2) per cent above the standard established, deductions will be made from price paid per ton in accordance with following table:

* PRICE AND PAYMENT.

Ash as established in proposal.	No deduction for limits below.	Cents per ton to be deducted.							Maximum limits for ash.
		2	4	7	12	18	25	35	
		Percentages of ash in dry coal.							
Per cent.									
5.........	7	7– 8	8– 9	9–10	10–11	11–12	12–13	13–14	12
6.........	8	8– 9	9–10	10–11	11–12	12–13	13–14	14–15	13
7.........	9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	14
8.........	10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	14
9.........	11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	15
10.........	12	12–13	13–14	14–15	15–16	16–17	17–18	....	16
11.........	13	13–14	14–15	15–16	16–17	17–18	18–19	....	16
12.........	14	14–15	15–16	16–17	17–18	18–19	19–20	....	17
13.........	15	15–16	16–17	17–18	18–19	19–20	20–21	....	18
14.........	16	16–17	17–18	18–19	19–20	20–21	21–22	....	19
15.........	17	17–18	18–19	19–20	20–21	21–22	....	....	19
16.........	18	18–19	19–20	20–21	21–22	22–23	....	....	20
17.........	19	19–20	20–21	21–22	22–23	....	....	....	21
18.........	20	20–21	21–22	22–23	....	....	....	....	22

* NOTE. — The economic value of a fuel is affected by the actual amount of combustible matter it contains, as determined by its heating value shown in British thermal units per pound of fuel, and also by other factors, among which is its ash content. The ash content not only lowers the heating value and decreases the capacity of the furnace, but also materially increases the cost of handling the coal, the labor of firing, and the cost of the removal of ashes, etc.

Proposals to receive consideration must be submitted upon this form and contain all of the information requested.

....

............................., 190

The undersigned hereby agree to furnish to the U. S...................... building at......................................, the coal described, in tons of 2240 pounds each and in quantity, 10 per cent more or less than that stated on page 912, as may be required during the fiscal year ending June 30, 190 , in strict accordance with this specification; the coal to be delivered in such quantities and at such times as the Government may direct.

Price per ton (2240 pounds)........................$..............

Commercial name of the coal.......................................

Name of the mine or mines...

Location of the mine or mines.....................................

Name or other designation of the coal bed or vein.......................

Size (indicate information which will apply) —

Unsized............ Lump........... Run of mine.................

Screened, through.......inch and over......inch, {Round, Square} Openings. Bar screen.

Data to establish a basis for payment:

British thermal units in coal as delivered...............................

Ash in dry coal (Method of American Chemical Society)...........per cent.

It is important that the above information does not establish a higher standard than can be actually maintained under the terms of the contract; and in this connection it should be noted that the small samples taken from the mine are invariably of higher quality than the coal actually delivered therefrom. It is evident, therefore, that it will be to the best interests of the contractor to furnish a correct description with average values of the coal offered, as a failure to maintain the standard established by contractor will result in deductions from the contract price, and may cause a cancellation of the contract, while deliveries of a coal of higher grade than quoted will be paid for at an increased price.

Signature: ....

Address: ....

Name of corporation, ...

Name of president,...

Name of secretary,...

Under what law (State) corporation is organized: ....

CHAPTER XX

TYPICAL CENTRAL STATIONS

438. The advancements that are being made in the design of large central stations and central station machinery are so rapid that it is futile to apply the term "modern" to any installation with the assurance that the plant thus designated will be representative of current practice for even a brief period of time. To-day it is possible to install in a given space approximately five times the capacity that could be installed a few years ago, with the cost per unit capacity only about one fifth and with very little increase in cost per square foot of floor space occupied. That the limit has not been reached is evidenced by the fact that boiler pressures of 350 lb. per sq. in. are to be employed in several plants in course of construction and even higher pressures have been considered for future designs. A few years ago boiler capacities during peak loads of 250 per cent rating were considered exceptional; to-day 400 per cent and even 500 per cent rating has been obtained with high overall efficiency. Improvement has not been limited to boilers and prime movers, but has been extended to all parts of the equipment.

The Essex Station of the Public Service Electric Co. of New Jersey, the Niagara River Station of the Buffalo General Electric Co., Buffalo, N. Y., the Northwest Station of the Commonwealth Edison Co., Chicago, may be considered the latest (1917) achievements in power plant design; every detail necessary to promote efficient operation and continuity of service has been incorporated.

Essex Power Station. — The plant is built on the unit system in what may be considered four separate structures: Switchhouse, turbine room, boiler house, coal bunkers and coal bridge. The four buildings occupy a total frontage of 401 ft.

The top of the coal tower is 215 ft. above high water and it has a lift of 156 ft. The tower with the bunkers is on the east side of the boiler room and is equipped with a 600-hp. hoisting engine of the two-drum type, and has a capacity of 240 tons per hr. when using a 2-ton clamshell bucket. The hoisting speed is 1300 ft. per min. When the bucket is dropping, it is driving the motor as an induction generator and pumping back into the line. The hoisting engine is driven by a 600-hp. induction motor.

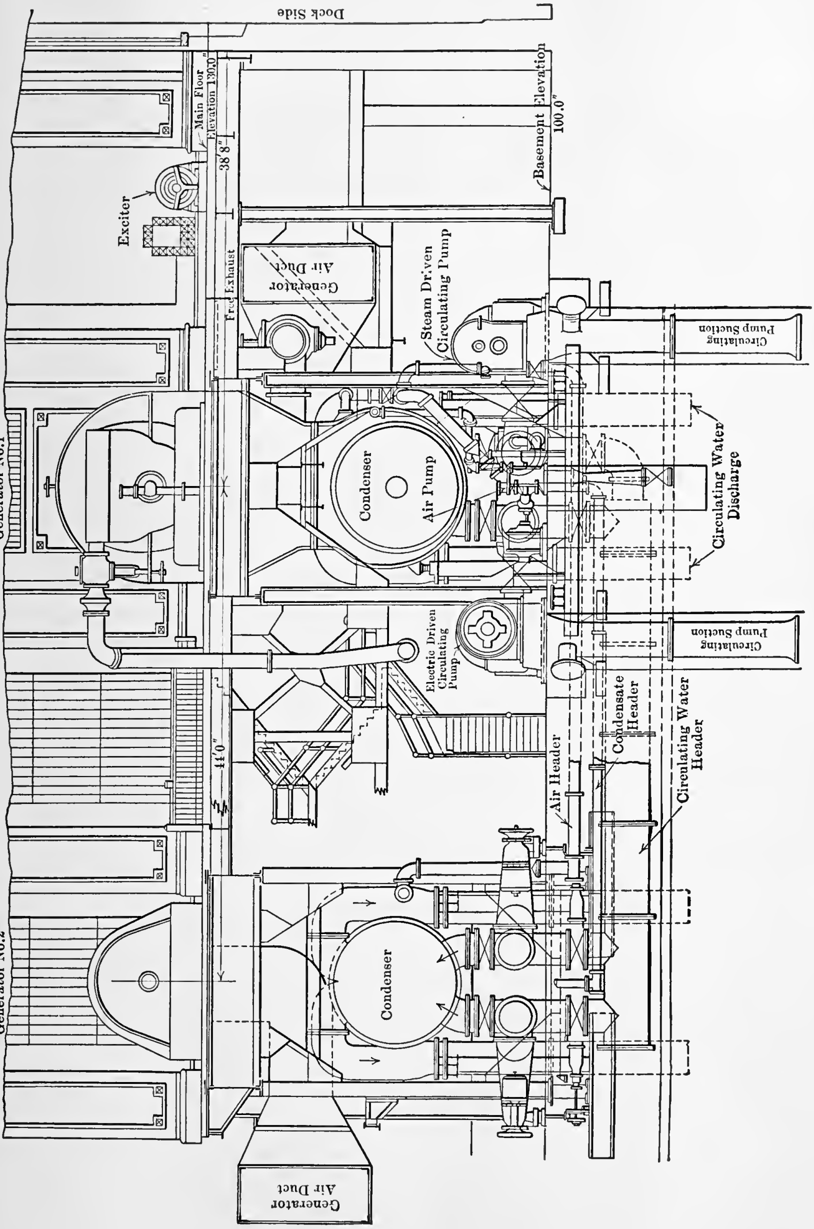


Fig. 607. Essex Station. — Front Elevation, Turbine Equipment.

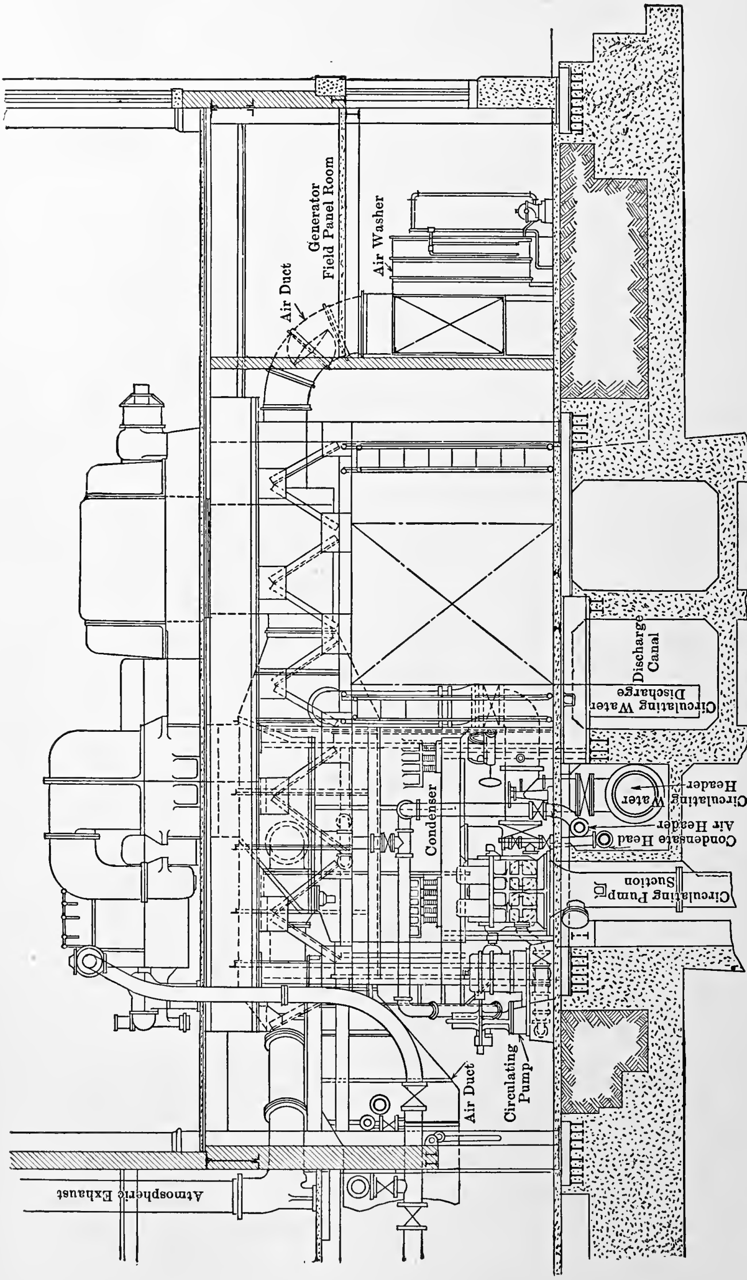


FIG. 608. Essex Station. — Side Elevation, Turbine Equipment.

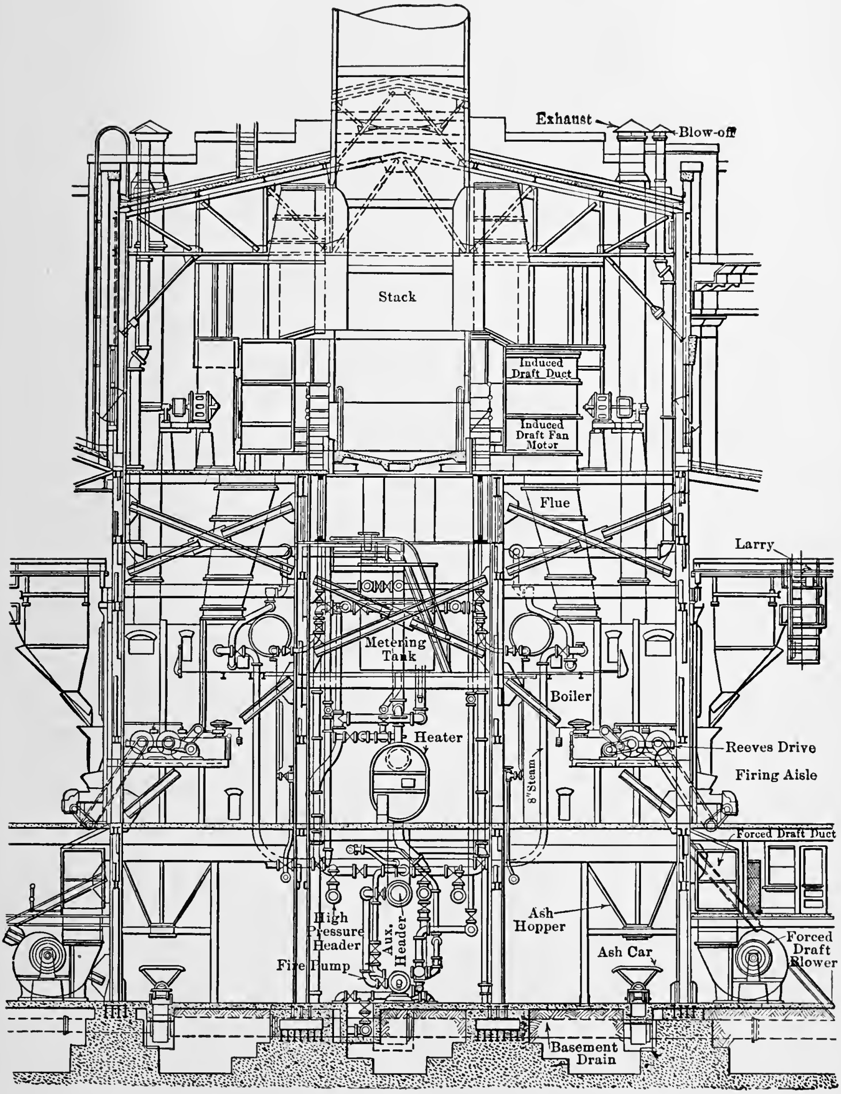


Fig. 609. Essex Station. — Front Elevation, Boiler Equipment.

Coal, after being hoisted to the tower, goes into a hopper from which it passes through a feeder into the crusher and is then distributed by a belt conveyor to the bunkers. The conveyors are driven by induction motors and are automatically cut out by push-buttons placed in convenient locations along the conveyor runway. Provisions are also made for unloading coal directly from cars or barges to the storage yard and for reclaiming the coal from the storage yard. All the coal-handling machinery is driven by induction motors.

The bunkers have a capacity of 2000 tons and are built of reinforced concrete supported on the steel structure of the building. From the outside bunkers the coal is brought by means of 15-ton motor-driven weighing larries, one in each firing aisle, and distributed to the stoker hoppers. Each hopper will hold seven tons of coal which is weighed automatically as it is distributed from the larry.

The boiler room contains eight 1373-hp. cross-drum marine-type Babcock & Wilcox water-tube boilers working under a steam pressure of 225 pounds. They contain 672 tubes 4 inches by 18 feet, arranged 42 tubes wide by 16 tubes high, giving a heating surface of 13,723 square feet. The boilers are guaranteed to evaporate 41,200 pounds of water from and at 212 deg. fahr. per hour and will give 300 per cent rating with clean heating surfaces. There are also two rows of circulating tubes which connect the upper ends of the front headers to the steam drum, as indicated in Fig. 609. Each boiler is equipped with six 4.5-inch Crosby safety valves arranged in three pairs so as to blow into a common header, which is piped through the roof. They are also equipped with 2 steam-flow meters, 2 steam gauges, 2 water columns, 2 feed-water regulators and 2 feed-water inlets.

The firing is done with 16-retort underfeed Sanford Riley stokers. The drive equipment for each firing aisle consists of four 12-hp. four-speed motors, two driving the mainshaft through a jack-shaft and two driving through Reeves conical variable-speed transmissions, giving a mainshaft speed of 32 to 290 r.p.m. This is equivalent to a coal feed of from 1600 to 15,000 lb. per hr. for each boiler. The furnace has an active grate area of 200 square feet. This gives a ratio of grate area to heating surface of 1: 63.5. The tubes are 7 feet 10 inches above the grate at the curtain wall and 9 feet at the back wall. Each boiler is equipped for forced, natural or induced draft, or all three may be used at the same time. Forced draft for each boiler is obtained by a 60,000 cubic feet multivane fan, driven by a 150-hp. motor, which can maintain a 6-inch water pressure under the grates. The air supply to the furnaces is controlled by a Mason regulator. The air pressure under the grates acts upon a flexible diaphragm, which through gears shifts the screws on the Reeves drive, consequently regulating the rate at which coal is fed to the furnace.

Induced draft is obtained by a 100,000-cubic-feet-per-minute multi-vane fan located at the economizer outlet and driven by a 100-hp. motor. The fan gives a 2-inch suction in the uptake of the boilers. After the gases have passed from the boiler, they may go directly to the stack, or, by closing dampers in the breechings, can be made to pass through the economizer and then to the stack; by closing a second damper, the gases will pass through the induced-draft fans before going to the stack. This makes it possible to operate the boilers under the most economical conditions at all times.

The economizers contain 480 four-inch tubes 12 feet long, giving a heating surface of 7750 square feet, which is 56 per cent of the boiler-heating surface. They are built to stand a working pressure of 300 pounds per square inch.

The stacks are two in number, 250 feet high above the grates and 16 feet inside diameter. They are built of steel plates and are of the self-supporting type. They have a 4-inch stack-brick lining with from one to two inches of grout between the brick and the shell.

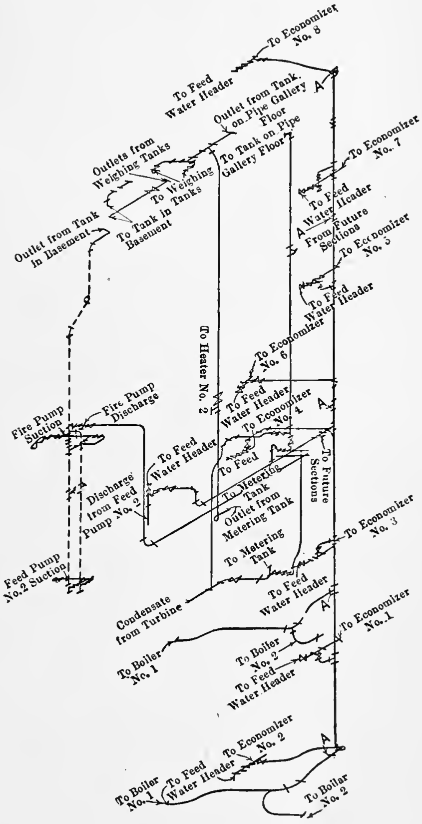


FIG. 610. Essex Station. — Steam Piping.

Drips and drain lines from all sources, as well as the leaders from the roof, discharge into concrete storage tanks beneath the boiler-room floor, and the water so collected is used for boiler make-up. The balance of the make-up water is taken from the city mains. The condensate from each main unit passes through a metering tank provided with a V-notch recording meter having a capacity of 4,000,000 pounds per hour and drops by gravity to a 77,000-pound storage tank. The make-up water passes through another V-notch recorder, and also goes to the storage tank with the condensate, whence it passes to two 10,000-hp. open feed-water metering heaters and then to the boiler-feed pumps at a temperature of 164 deg. fahr. and is pumped through the economizers into the boilers at a temperature of 244 deg. fahr. The feed water to each boiler is controlled by two Copes feed-water regulators, which maintain the water level in the boilers constant. A regulator maintains constant pressure difference between feed pressure and steam pressure to turbines on feed pumps. The layout of the feed-water-piping system and economizers is given in Fig. 611. The system is self-explanatory; it will be seen that the layout is very simple for what may be accomplished with it and the last word in flexibility. There are three boiler-feed pumps. Each pump is 3-stage

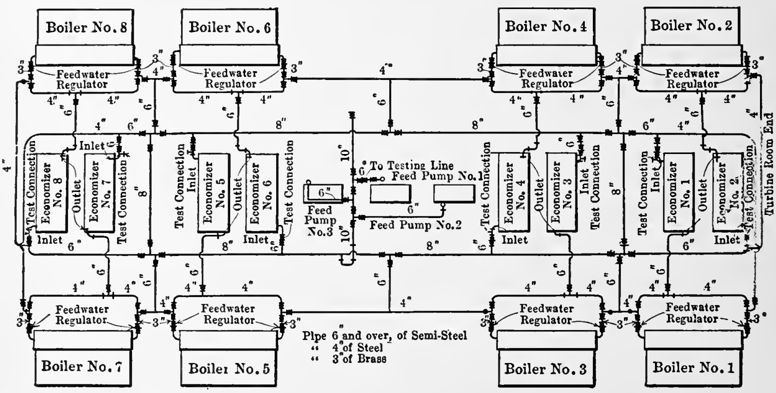


FIG. 611. Essex Station. — Feed-water Piping.

double suction having a capacity of 1000 gallons per minute against a total dynamic head of 700 feet and is driven by a 250-hp. Westinghouse turbine running 2200 r.p.m.

The boilers blow down through 6 blowdown pipes equipped with a Babcock & Wilcox and Everlasting valve in series into a tank, thence through a V-notch meter into the sewer.

Very elaborate arrangements have been made in the piping system whereby any turbine or boiler may be tested while in regular service.

All the soot from the economizers, stacks, etc., is taken out by steam ejectors. The ashes drop from the grates into hoppers, where they fall into side-dump pivot cars and are hauled out into the yard by a 5-ton electric locomotive and are used for filling-in purposes. Provisions have been made so that when the ashes are no longer required for filling purposes, they will be raised by a skip hoist to a bunker in the coal tower and from there discharged into barges.

A switchboard is located in front of each boiler, from which are controlled the stoker drive, forced- and induced-draft fans. Indicating and recording meters are also mounted on these panels for draft, temperature, motor current, and stoker speed.

All boiler stop valves, sectionalizing valves, and turbine stop valves are operated by 125-volt, direct-current motors and are controlled from a switchboard, in the boiler-room engineer's office on the main floor of the boiler room. The turbine stop valves can also be operated from a remote-control station on the main floor of the turbine room.

The steam is taken from a double-ended superheater through Edwards stop and check valves into two 8-inch pipes, one at each end of the boiler, down through the boiler-room floor to 12-inch to 18-inch double headers as shown in Fig. 610. These headers are cross-connected at each connection from the boilers. A 16-inch steam line goes to each turbine and an 8-inch to the auxiliaries. Each end of the superheater is furnished with a 4.5-inch safety valve.

The steam headers are anchored at the center so that one-half of the expansion is in each direction from that point. The headers are also anchored in the turbine-room basement before they connect to the turbine inlet through expansion bends and risers. There are no expansion joints in the headers; they are installed under a tension between anchorages, which causes an elongation equal to about one half of the expansion of the section normal temperature to that of the steam. Therefore, when the headers are at the temperature of the surrounding air they are in tension, and when at the temperature of the steam they are in about the same amount of compression. By this scheme it has been possible to do away with expansion joints in the headers, and so far they have worked very satisfactorily. The headers are carried on sling rods with stirrups resting on springs to allow for come and go.

The turbine room contains two 25,000-kilovolt-ampere General Electric main units, only one being operated at a time. Three boilers are required to supply steam for one unit, which gives a ratio of boiler to engine horsepower of 1:8. The main turbines are 12-stage, tandem-compound, with 8 stages for the high pressure and 4 for the low pressure, and exhaust into a surface condenser of the two-pass type containing 6434 one-inch tubes 19 feet active length. This gives 1.28 square feet of cooling surface per kilowatt and provides for the condensation of 7.5 pounds of steam per square foot of cooling surface per hour. An average vacuum of 28.73 inch is maintained with 70-degree circulating water. The condensers are of the radial-flow type and are rigidly connected to the turbines; the expansion and contraction is taken care of by supporting the condensers on springs.

The circulating water is supplied by two 24,000-gallon centrifugal pumps for each main unit, one motor-driven and the other turbine-driven. In the winter the turbine-driven pump usually has capacity enough to maintain the vacuum, making it unnecessary to run the motor-driven pump except during the summer months. The electrically-driven pump is so arranged that when the temperature of the discharge water rises above a certain value a thermostat closes, automatically starting the motor, and puts the second pump into service.

The vacuum pumps are of the Westinghouse Le Blanc type and are motor-driven. A turbine-driven hot-well pump takes care of the condensate, which is pumped back into a tank whence it passes through a V-notch meter to the feed-water heater. The exhaust of all auxiliaries goes to the open feed-water heaters.

The circulating water is taken from the river through three intake tunnels 9 feet 4 inches by 8 feet, equipped with motor-driven revolving screens. The discharge tunnels are two in number, 12 feet 6 inches by 9 feet 4 inches, and rest on top of the intake tunnels.

The main generator units are 25,000-kilovolt-ampere (continuous rating), 60-cycle, three-phase, 13,200-volt machines running 1800 r.p.m. They are equipped with 100-kilowatt, 250-volt, direct-connected exciters. These are the only generators of this speed and capacity that have direct-connected exciters.

The air for ventilating the generators is taken from outside the buildings and is washed by water sprays, one on each generator, directed into the incoming air in several directions. This not only cleans the air, but also cools and humidifies it, the drop in temperature in some cases being as much as 15 deg. fahr. The air is forced through the ventilating ducts of the generator by fans on the rotor. The heated air from the generator is carried back to the forced-draft fan-room in the boiler house and supplies part of the air for the furnaces, thus recovering some of the losses in the generator.

The exciter system is designed to secure maximum reliability together with independent excitation for each generator, and consists of a regular, emergency, and spare. Regular excitation is supplied by a 250-volt shunt generator directly connected to each alternator shaft. In case of trouble on the regular exciter, a low-voltage relay instantly closes the emergency-exciter circuit, which consists of a 900-ampere (30-minute rating) storage battery equipped with four 14-point end-cell switches, after which the direct-connected exciter is cut out automatically by a reverse current relay. The spare exciter, which is a 75-kilowatt, motor-driven shunt generator, may then be started by the operator from the main switchboard and cut in parallel with the battery on the field of the generator and the battery cut out.

The power for the station is supplied from a 3000-kilovolt-ampere, 13,200- to 440-volt, 60-cycle, water-cooled, oil-insulated transformer. The transformer is equipped with water-flow indicators and thermometers, which operate an alarm in case the transformer has no cooling water or becomes overheated.

All the motors used throughout the plant except those which operate valves are 440-volt machines. A 440-volt system was selected on account of the greater safety to the attendants over a 2300-volt system and also on account of the great saving in cable and bus capacity compared with 220-volt. Where variable speed is required of the alternating-current motors, it is obtained either by changing the number of poles in the stator winding or by rotor resistance.

TABLE 163.

ESSEX STATION — GENERAL DATA.

Coal Handling.

No.	Equipment.	Kind.	Size.	Operating Condition.
1	Hoisting engine	Two-drum	24-in. drums, 200 r.p.m.	180 tons per hour
1	H. E. motor	Induction	410 hp., 200 r.p.m.	Direct connected to engine
1	Traversing engine	Single-drum	30-in. drum	Gear-driven motor
1	T. E. motor	Induction	75 hp.	Constant speed
1	Feeder	Apron	5 ft. wide, 16 ft. long	Gear-driven motor
1	Feeder motor	Induction	7.5 hp.	Constant speed
2	Crushers	2-roll	36 in. by 36 in. rolls	Gear-driven motor
2	Crusher motors	Induction	35 hp.	Constant speed
1	Conveyor	Belt	30 in. by 161 ft. long	300 tons per hour
1	Conveyor	Belt	30 in. by 120 ft. long	200 tons per hour
1	Conveyor motor	Induction	20 hp.	Constant speed
1	Conveyor motor	Induction	10 hp.	Constant speed
1	Coal bucket	Clamshell	$1\frac{1}{2}$ ton	....
1	Automatic skip hoist	Balanced	2 ton buckets — 100 tons per hr.	Full automatic
1	Skip hoist motor	Induction	25 hp.	Constant speed
1	Elevator	Electric freight	$7\frac{1}{2}$ ton	50 ft. per minute
1	Elevator motor	Induction	40 hp.	Constant speed

Turbine House.

No.	Equipment.	Kind.	Size.	Operating Condition.
2	Generators	General electric	25,000 kv.a.	132,000 v., 3 hp., 60 cy., 1800 r.p.m.
2	Exciters	Direct-connected	100 kw., 250 v., compound wound	1800 r.p.m.
1	Exciter	Shunt wound	150 kw., 250 v.	Reserve equipment
1	Exciter motor	Induction	220 hp. — 1175 r.p.m.	Reserve equipment
2	Turbines	Curtis — horizontal	25,000 kw. — 12 stage	190 lb. steam, 150 deg. superheat
2	Condensers	Two pass surface	32,000 sq. ft. — 255,000 lb. per hr.	....
4	Circulating pumps	Horizontal-centrifugal	24,000 gal. per min. 43 ft. max. hd	Turbine driven
2	C. P.turbines	Horizontally split	350 hp.	190 lb. steam, 150 deg. superheat
2	C. P. motors	Induction	350 hp.	Constant speed
2	Air pumps	Le Blanc	37.5 cu. ft. free air per min.	Motor driven
2	Air pump motors	Induction	100 hp.	Constant speed
2	Condensate pumps	Centrifugal	500 gal. per minute	Turbine driven
2	C. P. turbines	Horizontally split	20 hp.	190 lb. steam, 150 deg. superheat
1	Crane	Electric traveling	100 ton — 94 ft. span	2-50 ton, 1-10 ton, hooks
2	Air washers	Spray system	60,000 cu. ft. air per minute	Motor-driven pumps — 10 hp.
2	Oil pumps	Centrifugal	3-in. — 150 gal. per minute	Motor driven
2	Oil pump motors	Induction	7.5 hp.	Constant speed
1	Oil filter	....	....	For new & make-up oil only

TABLE 163. — *Continued.*

Boiler House.

8	Boilers	B. & W. cross drum	1373 hp. on 10 lb. per sq. ft. basis	225 lb. press., 150 deg. superheat
8	Superheaters	Flash type	1711 sq. ft. heating surface	150 deg. superheat
8	Soot cleaners	Steam blow	....	Live steam
8	Stokers	Underfeed (Riley)	16 retort — 15,000 lb. coal per hr.	Continuous dumping
4	Reeves transmissions	....	Class F. — No. 6½	Variable speed
8	Stoker motors	Induction	12 hp.	Four speed
8	Forced-draft blowers	Turbovane (Sturtevant)	60,000 cu. ft. min. at 7 in. st. pres.	2 speed, motor driven
8	F. D. B. Motors	Induction	150 hp.	2 speed
8	Economizers	C. I. tube (Sturtevant)	7750 sq. ft. 40 by 12 by 12 ft.	1 economizer per boiler
8	Induced-draft fans	Multivane (Sturtevant)	106,000 cu. ft. gases 400 deg. fahr.	Constant speed motor driven
8	I. D. F. motors	Induction	100 hp.	Constant speed
2	Heaters	Cochrane metering	10,000 hp., 500,000 lb. per hour	....
1	Metering tank	Blow-off	200,000 lb. per hour	....
1	Metering tank	Feed-water	1,000,000 lb. per hour	....
3	Boiler-feed pumps	3-stage centrifugal	1000 gal. per minute	700 ft. total head
3	B. F. P. turbines	Horizontal	250 hp.	190 lb. steam, 150 deg. superheat
1	Fire pump	2-stage centrifugal	1000 gal. per minute	110 lb. total head
1	Fire pump motor	Induction	100 hp.	Constant speed
1	Air compressor	Straight line	225 cu. ft. per minute	100 lb. pressure
1	A. C. motor	Induction	....	Constant speed
2	Service pumps	Single stage	500 gal. per minute	110 lb. total head
2	S. P. turbine	Horizontal	22 hp.	190 lb. gauge, 150 deg. superheat
1	Air compressor	Straight line	100 cu. ft. per minute	100 lb. pressure
1	A. C. motor	Induction	15 hp.	Constant speed
2	Coal larries	Weighing	15 ton	Motor driven
2	Stacks	Steel-brick lined	15 ft. 4 in. by 250 ft.	....
2	Locomotives	Electric storage battery	....	Storage battery
16	Meters	Steam flow	....	....
1	Charging equipment	Motor generator	15 kw.	For elec. locomotives

TABLE 164.

BUFFALO GENERAL ELECTRIC CO. — GENERAL DATA.

Boiler Room

Type of boilers	Babcock and Wilcox, cross-drum water tube
Number now installed	5
Anticipated station load, immediate, kw.	40,000
Heating surface, each, sq. ft.	11,400
Superheater surface, each boiler, sq. ft.	3,815
Grate surface per boiler, total, sq. ft.	418
Heating surface per sq. ft. grate surface, sq. ft.	27.27
Heating surface per sq. ft. superheater surface, sq. ft.	3
Superheater surface per sq. ft. grate surface, sq. ft.	9.1
Heating surface per kw. (95,000 kw., five 11,400 sq. ft. boilers)	0.6
Working pressure, lb. per sq. in.	275
Superheat, deg. fahr.	275
Total temperature steam, deg. fahr.	689.4
Mud drum material	Forged steel
Stokers, Riley underfeed, 2 per boiler; retorts per boiler	30
Maximum capacity each retort, lb. coal per hr.	1,000
Capacity each retort at approx. 400 per cent. rating, lb. per hr.	700
Boiler rating on peaks, per cent, 350; between peaks, per cent.	100 to 250
Rating on peaks, anticipated maximum, lb. water per hr. from and at 212 deg. fahr.	160,300
Water evaporated per sq. ft. heating surface at approx. 400 per cent rating, lb. per hr.	14.4
Capacity each stoker on peaks, kw.	5,000 to 10,000
Coal burned per sq. ft. grate at 250 per cent rating, lb. per hr., 26; at approx. 400 per cent rating, lb. per hr.	50.25

TABLE 164. — *Continued.*

Economizers, green, type H, maximum pressure, lb. per sq. in......	400
Economizer material	Cast iron
Economizer heating surface per boiler, sq. ft.	9,435
Economizer heating surface per sq. ft. boiler-heating surface, sq. ft.	1.208
Economizer surface per boiler horsepower (34.5 lb. water per hr.) rated capacity, sq. ft.	0.228

Economizer — Guarantees for each unit economizer:

Feed Water, Lb. per Hr.	Temperature Leaving Economizer when Entering at 180 Deg. Fahr.	Gas Temperature Entering, Deg. Fahr.	Gas Temperature Leaving, Deg. Fahr.	Gas Temperature Difference, Deg. Fahr.
53,000	263	535	294	241
86,000	281	633	366	267
103,000	289	670	396	274
120,000	288	705	443	262

Coal	Bituminous run of mine
Coal bunker, type	Non-suspended, steel frame, concrete lined
Coal conveyors: Two bucket conveyors, capacity each per hr., tons	200
Two belt conveyors over bunker, each 36 in. wide, capacity each per hr., tons	200
Feed pumps, 3 Jeansville, centrifugal, turbine-driven, all-bronze casings.	
Feed pump capacity per sq. ft. heating surface, gal. per min	0.053
Make-up water evaporator system capacity, lb. per hr.	30,000
Present make-up water evaporator capacity, per cent of hot-well supply (based on 60,000 kw. turbine capacity)	5
Main open feed-water heaters: Cochran horizontal cylindrical; capacity each, boiler horsepower	10,000
Heater capacity per sq. ft. boiler-heating surface, boiler horsepower (34.5 lb. water per hr.)	0.53
Heater capacity per rated horsepower capacity of boilers, boiler horsepower	5.3
Heater capacity per lb. main-unit steam consumption (95,000 kw. @ 10.25 lb. per kw-hr.), boiler horsepower	0.0308
Chimneys: Two steel-lined.	
Contractors, Lackawanna Steel Co. Builders, Merchants Iron Works, Chicago.	
Height above lower grate, ft. 192; height above upper grate, ft.	185
Diameter at flue entrance, ft.	19
Diameter at top, ft.	19
Boilers per chimney	4
Coal burned per sq. ft. chimney cross-sectional area at approx. 400 per cent rating, lb. per hr.	185.1
Forced-draft fans, Green, radial flow; number of	3
Capacity of each at 6 in. static pressure, 550 r.p.m., cu. ft. per min. (hp. 308)	210,000
Capacity of each at $4\frac{1}{2}$ in. static pressure, 430 r.p.m., cu. ft. per min. (hp. 153)	150,000
Capacity of each at 3 in. static pressure, 336 r.p.m., cu. ft. per min. (hp. 67)	100.000
Induced-draft fans: Buffalo Forge Co., number of	6
Induced-draft fan capacity, each, with gas at 496 deg. fahr., 482 r.p.m. cu. ft. per min. (hp. 130)	120,500
Forced-draft fan capacity per sq. ft. grate, cu. ft. per min	251
Induced-draft fan capacity per sq. ft. grate, cu. ft. per min	288
Bunker coal storage over present 8 boilers, maximum tons	3,000
Yard coal storage at plant, tons	50,000

TABLE 164. — *Continued.*

Turbines.

Three 20,000-kw. at 90 per cent power factor, installed; one 35,000-kw. on order.

General Electric Co., single-cylinder, horizontal, speed, r.p.m.	1,500
Operating pressure, lb. abs.	290
Operating superheat, deg. fahr.	275

Performance guarantees: Operating conditions — 265 lb. abs. 250 deg. fahr. superheat. 1-in. absolute pressure, 30-in. barometer, in condenser:

Net Kw. Load of Generator.	Lb. of Steam per Kw-hr.
7,500	11.85
10,000	11.05
15,000	10.25
20,000	10.60

Note. For higher pressures and temperatures the following factors are used: 1 per cent for each 15 lb. pressure for range of 25 lb. above or below normal; 1 per cent for each 11 deg. fahr. superheat for range of 25 deg. fahr. above or below normal.

Blading material: First 2 and last 3 rows, nickel steel; intermediate rows, monel metal and nickel bronze.

Peripheral speed last rows of low-pressure blading, ft. per sec.	717
Total weight each 20,000-kw. machine, lb.	540,000
Weight of turbine per rated kw. capacity, lb.	27
Heaviest piece to be lifted by crane, tons	70
Floor space occupied by each turbine, outside measurements, sq. ft.	830
Turbine rated capacity per sq. ft. floor covered by turbine, kw.	24
Steam consumption of auxiliaries:	
At most economical load, lb. per hr., 6100; with 70-deg. fahr. circ. water, lb. per hr.	10,500
At full load, lb. per hr., 9100; with 70 deg. fahr. circ. water, lb. per hr.	13,500
Exciters: Three, capacity each, kw.	300
Type	Combination turbine and induction motor drive
Builders	Terry Turbine Co.
Main unit capacity per kw. capacity of exciters, installed, kw.	105.5

Condensers.

Builder	Westinghouse Electric and Mfg. Co.
Total tube surface per condenser, sq. ft.	33,000
Tube area per kw. turbine capacity served by condenser, sq. ft.	1.65
Tubes, 1 in. O. D., composition, Muntz metal (60 per cent copper, 40 per cent zinc)	
Chief guarantee: With 70 deg. circulating water, pressure in condenser, lb. abs.	1.33
Circulating pumps, capacity each, gal. per min.	25,000
Builder, Westinghouse Electric and Mfg. Co., type, double suction, centrifugal.	
Diameter discharge pipe, in.	42
Circulating-water pumping capacity per lb. steam condensed at consumption of 10.6 lb. per kw-hr., lb.	118
Circulating pumps (two per condenser), capacity each, gal. per min.	25,000
Cross-sectional area each intake and each discharge tunnel for each unit, sq. ft.	30
Intake, tunnel area per 1000 gal. per min. circulating-water pumping capacity, sq. ft.	0.6
Screens at circulating water intake	Wire mesh, stationary
Dry-vacuum pump, type	Le Blanc
Hot-well pump: Centrifugal; builder, Worthington; size in.	4

TABLE 165.

COMMONWEALTH EDISON CO., NORTHWEST, UNIT No. 3—*GENERAL DATA.*

Turbine.

Maker	General Electric Co.
Type	Horizontal compound
Capacity, hp	45,000
Number single stages, h-p. element	10
Number of double stages, l-p. element	2
Speed, r.p.m	1,500

Condenser.

Maker	Wheeler Condenser and Engineering Co.
Number of tubes	11,000
Size of tubes, in	1
Surface in condenser, sq. ft	50,000
Surface per kilowatt of generator rating, sq. ft	1.67
Capacity, lb. of steam per hr	360,000
Steam condensed per square foot of surface, lb	7.2
Weight of condenser, empty, tons	176
Weight of cooling water in condenser, tons	66.5
Circulating-pump capacity, gal. per min	52,000
Circulating-pump capacity, lb. per hr	26,000,000
Cooling water per pound of steam, lb	72
Condensate pump, gal. per min	1,200

Generator.

Maker	General Electric Co.
Capacity, rated	30,000
Voltage	9,000
Frequency, cycles	25
Speed, r.p.m	1,500
Number field poles	2
Length complete unit, overall ft	59.5
Width, ft	18.33
Floor area cover, sq. ft	1,091
Area per kilowatt of generator rating, sq. ft	0.036
Exciter voltage	220

Boilers.

Maker	Babcock & Wilcox Co.
Type	Cross-drum, water-tube
Pressure, lb. per sq. in. gauge	230
Superheat, deg. fahr	200
Temperature of steam, deg. fahr	600
Number of boilers in unit	5
Number of tubes per boiler	588
Diameter of tubes, in	4
Length of tubes, ft	18
Steam-making surface in boiler, sq. ft	12,200
Stokers per boiler	2
Type of stoker	B. & W. chain-grate
Active area of two stokers, sq. ft	273
Ratio grate area to boiler-heating surface	1 to 45
Per 1000 sq. ft. of boiler-heating surface:	
Connected grate area, sq. ft	22.3
Stack area, sq. ft	4.17
Economizer surface, sq. ft	538.2
Capacity of each boiler, lb. steam per hr	85,000
Evaporation per sq. ft. of heating surface, lb	7

TABLE 165. — *Continued.*

Coal capacity of each boiler, lb. per hr.	12,600
Coal per square foot of grate, lb.	46
Size of steam main to turbine, in.	20
Boiler-feed pumps:	
Maker	Henry R. Worthington
Type	Turbine-driven, three-stage, double-suction impeller
Capacity, lb. per hr.	450,000
Speed, r.p.m.	2,500
Water temperatures:	
In feed-water heater, deg. fahr.	100–120
Leaving economizer and entering boiler, deg. fahr.	270

Economizers.

Maker	B. F. Sturtevant Co.
Type	High-pressure
Number of economizers	5
Number of tubes in each	456
Length of tubes, ft.	12
Heating surface in tubes	6,566
Maker	B. F. Sturtevant Co.
Type	Multivane
Capacity, cubic feet hot gases per min.	90,000
Horsepower of motor	100
Draft in boiler uptake, in. of water	2.4
Height of stack above boiler-room floor, ft.	250
Diameter of stack, inside, ft.	18

CHAPTER XXI

A TYPICAL MODERN ISOLATED STATION *

Bleeder Turbines and Condenser System

439. The new power plant of the W. H. McElwain Company at Manchester, N. H., is an excellent example of current practice in generation of power by steam for industrial purposes.

General Arrangement. — General arrangement of the boiler and engine rooms is shown in plan in Fig. 613. At the present time there have been installed three 300-horsepower water-tube boilers and one 1000-kilowatt turbo-generator outfit. The boiler room contains sufficient space for a fourth 300-horsepower unit, as indicated by dotted lines. The completed plant will include duplicates of the two batteries shown, making a total of 2400 horsepower. The future boilers will face those already installed, the building being extended for this purpose, and the firing space shown will be common to both sections.

The chimney, which is 176 feet in height, with a flue 9 feet in diameter, is designed with reference to the final capacity of the plant. In the engine room, at the right, is shown space for two additional generating units, which provide for an ultimate capacity of 3000 kilowatts. Sectional elevations, showing the boilers, turbines, and the various auxiliary equipment and their connections, are illustrated in Figs. 612, 613, and 614.

Boilers. — Present equipment consists of three Babcock and Wilcox water-tube boilers, each containing 2972 square feet of heating surface and about 50 square feet of grate surface. The heating surface is made up of two steam drums, tubes, and mud drum, and a superheater of the form shown in Fig. 614.

Each boiler contains 144 4-inch tubes, 18 feet in length, made up in 12 sections of 12 tubes each, and 2 steam drums, 3 feet in diameter by 20 feet 4 inches in length. The superheaters each contain approximately 372 square feet of surface, which is $12\frac{1}{2}$ per cent of the heating surface of the boiler, and are designed to give 100 degrees superheat when the boilers are operated at their normal rating of 300 horsepower. The proportions of all parts are designed for a working pressure of 160 pounds per square inch and the safety valves are set at that point.

* From the Practical Engineer, Chicago, July 1, 1912.

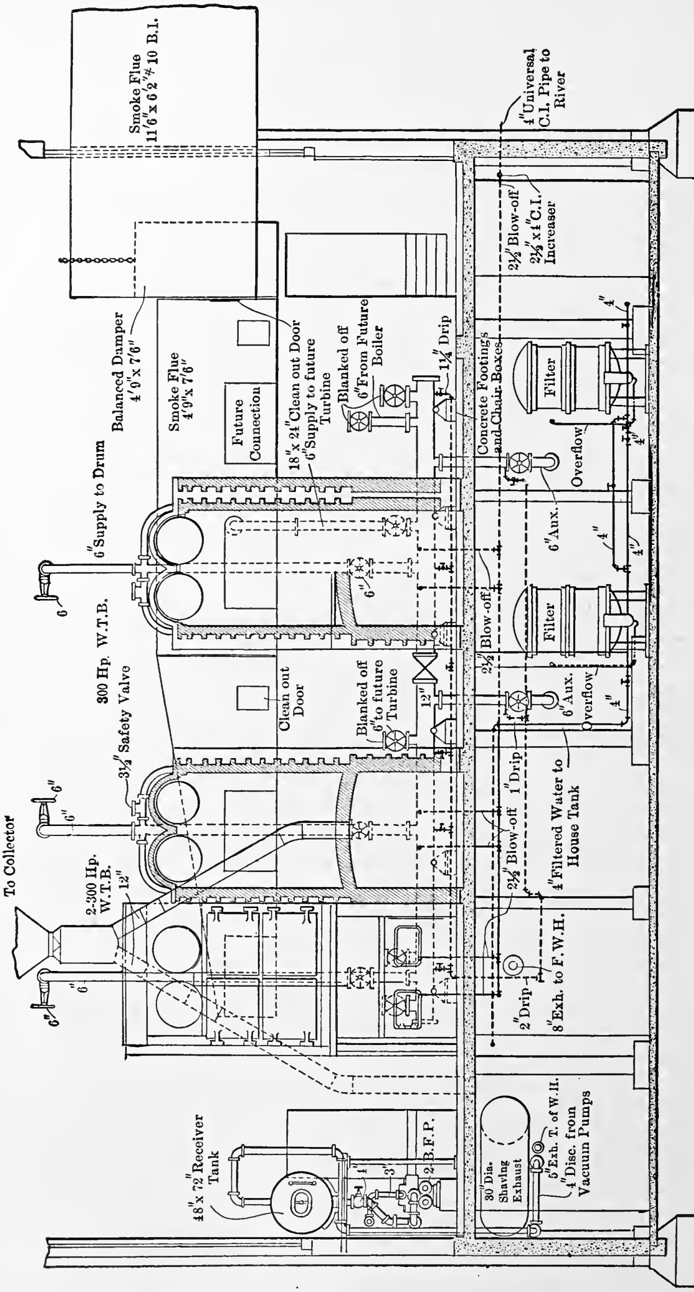


Fig. 612. Sectional Elevation through Boiler Room and Basement.

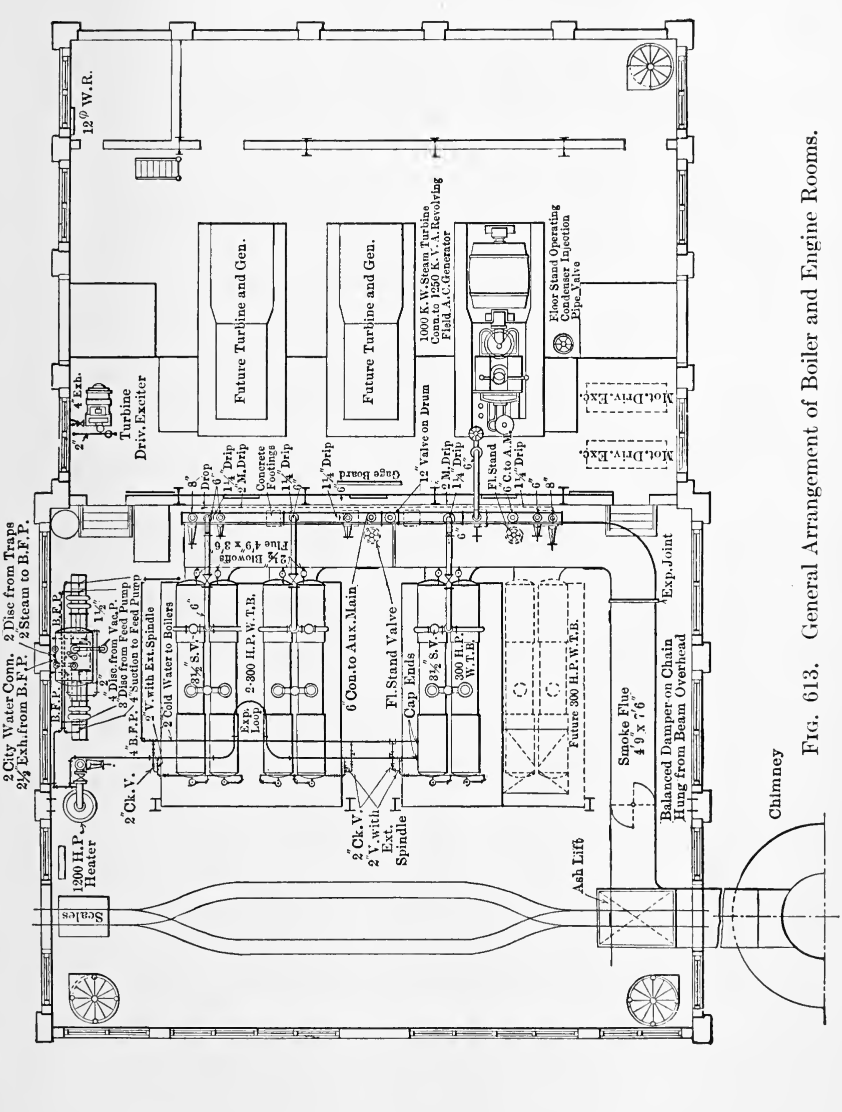


FIG. 613. General Arrangement of Boiler and Engine Rooms.

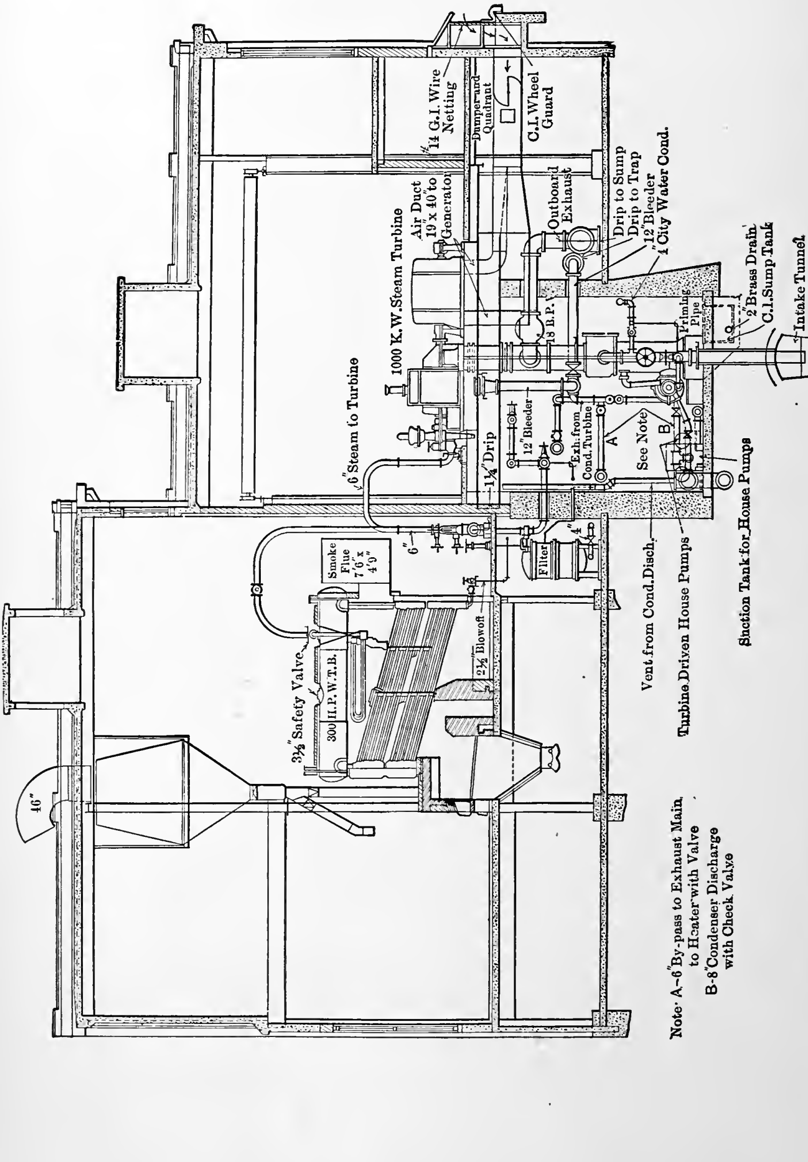


FIG. 614. Sectional Elevation through Boiler and Engine Room, showing Steam Piping and Condenser Connections.

Each boiler is provided with a water column fitted with high and low water alarm, try cocks, and gauge glass with special device for shutting off in case of breakage. Also $3\frac{1}{2}$-inch lock pop-safety valve, and 12-inch steam gauge reading to 300 pounds pressure. The feed pipes are 2 inches in diameter, provided with both check and gate valves, the latter having special extension handles. The blow-off connections are of $2\frac{1}{2}$-inch extra heavy pipe, and are each provided with two blow-off valves of special design.

Boiler settings are of hard-burned brick, laid in cement mortar, consisting of 1 part cement to 3 of sand, up to the level of the grates, and in lime mortar above that point. All parts of the furnaces and setting exposed to the fire are lined with firebrick laid in fire clay. The furnaces are of the "Dutch oven" type as shown in Fig. 614.

Smoke Connections. — Location of the main smoke flue is best shown in Fig. 613. It is 4 feet 9 inches by 7 feet 6 inches in size and constructed of No. 10 black iron. It is stiffened with angle-iron braces and supported from the roof. The uptake from each boiler is provided with an adjusting damper for hand manipulation from the floor level.

A balanced damper is located in the main flue at the point indicated, and operated by an automatic regulator of the hydraulic type. An interesting detail in connection with this work is the method of attaching the covering to the lower side of the flue so that it will not sag or peel off. This consists of cross pieces of 1-inch tee-bars placed 24 inches apart and riveted to the flue. The projecting flanges of these bars are drilled at frequent intervals and wires strung through, to which the covering is attached.

Handling of Fuel and Ash. — Coal is brought to the fire room by cars running on a special track, as shown in Fig. 613. This track passes over platform scales just inside the building, where each load may be weighed as it is brought in. The track is double within the fire room so that the cars may pass, and also to furnish storage space for both coal and ash cars when so desired.

The arrangement for the removal of ash is best shown in Figs. 613 and 614. A dumping chute is provided in the bottom of each ashpit and at such an elevation that a car may be run underneath it as indicated. When filled, they are pushed to the ash lift (see Fig. 613) where they are raised to the boiler-room level and run out on the coal track for disposal. Combustible waste from the factory is brought through a 36-inch pipe to a collector placed in the upper part of the boiler room, as shown in Fig. 614, and fed into the furnaces as there indicated.

Turbine and Generator. — The turbo-generator unit is one of the Westinghouse make, of 1000-kilowatt capacity, and equipped with an automatic bleeder connection and constant-pressure valve. It is 6 feet 6 inches in width by 24 feet 8 inches in length and weighs approximately 79,000 pounds. It is of the regular Westinghouse-Parsons type, the most interesting feature being the bleeder attachment which adapts it for use in combined power and heating plants. An important requirement for the economical operation of the ordinary steam turbine is the maintenance of a high vacuum at the exhaust end, which, of course, prevents the utilization of exhaust steam for heating purposes.

The capacity of the turbine under different conditions is as follows: With a throttle pressure of 150 pounds per square inch (gauge), a vacuum of 28 inches, 100 degrees superheat, and a speed of 3600 r.p.m., the normal capacity when condensing is 1500 b.hp. and the maximum 2250 b.hp. When running non-condensing with a back pressure not exceeding that of the atmosphere, the maximum capacity is 1500 b.hp.

It is interesting to note the probable steam economy of a turbine of this type when operating under varying loads, as expressed in the guarantee placed upon this machine, which is as follows: When operating under the above conditions, in connection with the generator attached, the steam consumption per hour, including all leakage and loss with the turbine, shall not exceed the quantities given below:

Load, Per Cent.	Power Factor, Per Cent.	Kilowatts.	Pounds Steam per Kilowatt-Hour.
150	80	1500	18.8
125	80	1250	18.3
100	80	1000	17.9
75	80	750	18.8
50	80	500	20.7

When operating under the same general conditions, with 3 pounds gauge pressure at the bleeder connection, the steam consumption per hour shall not exceed the following at the loads indicated, when withdrawing the following amounts of steam through the bleeder connection:

Load. Per Cent.	Kilowatts.	Pounds of Steam		Steam to Condenser.	
		To Throttle.	To Bleeder.	Total.	Kilowatts.
150	1500	38,000	18,600	19,400	12.9
		31,000	10,000	21,000	14.0
125	1265	38,000	22,000	16,000	12.7
		36,300	20,000	16,300	12.9
		29,200	10,000	19,200	15.2
100	1000	37,200	30,000	7,200	7.2
		30,000	20,000	10,000	10.0
		24,400	10,000	14,400	14.4
75	716	30,500	30,000	500	0.7
		25,600	20,000	5,600	7.8
		20,200	10,000	10,200	14.2
50	469	21,700	21,700	0	0.0
		20,600	20,000	600	1.3
		16,000	10,000	6,000	12.8

Generator. — The generator is of the revolving-field type with inclosed frame, generating a 3-phase, 60-cycle, alternating current of 600 volts. The efficiency rating, with a power factor of 100 per cent, is as follows:

Load, Per Cent.	Efficiency, Per Cent.	Load, Per Cent.	Efficiency, Per Cent.
50	90.10	125	95.50
75	93.00	150	95.75
100	94.50		

Temperature rise based on its normal rating and a power factor of 80 per cent, for periods of different length and for various loads, is given below:

Load, Per Cent.	Length of Run, Hours.	Temperature Rise, Armature.	Degs. F., Field.
100	24	72	72
125	24	90	90
150	1	108	108

The maximum conditions of continuous operation with a power factor of 80 per cent and for a room temperature of 77 deg. fahr. are as follows: Output, 1250 kilowatts (25 per cent overload). Rise in temperature:

Armature, 90 deg. fahr.
Field, 90 deg. fahr.

Maximum temperature to which insulation can be subjected without injury:

Armature, 194 deg. fahr.
Field, 302 deg. fahr.

There are two exciters provided, one being turbine driven and having a normal capacity of 25 kilowatts; the other motor driven, with a capacity of 40 kilowatts. The turbine is of the Westinghouse make, horizontal type, with a normal capacity of 38 b.hp. at a speed of 3500 r.p.m. when running non-condensing, and a continuous overload capacity of 25 per cent. The steam requirements for this machine as regards temperature and pressure are the same as for the main turbine.

The exciter is a direct-current machine with shunt winding, generating a current of 125 volts at full load.

Condensing Apparatus. — In connection with the main turbine a Westinghouse-Le Blanc jet condenser is used, and is shown in elevation in Figs. 614 and 615. This is designed to operate under a normal lift of 18 feet and takes its water supply from the intake tunnel as shown. When using injection water at a temperature of 70 deg. fahr. the following results are guaranteed, with a water consumption not exceeding 724,000 pounds per hour:

Steam Condensed per Hour, Pounds.	Vacuum Maintained, Inches (Barometer, 30 Ins.)	Steam Condensed per Hour, Pounds.	Vacuum Maintained, Inches (Barometer, 30 Ins.)
10,350	28.65	19,950	28.00
14,100	28.44	22,900	27.80
18,000	28.17	30,000	27.11

The vacuum air pump is of the turbine type and is mounted upon the same shaft with the centrifugal ejector pump, both being driven by a steam turbine of 41 b.hp. running at 1500 r.p.m. under an atmospheric exhaust pressure. This piece of apparatus is shown at the base of the condenser in Figs. 614 and 615.

High-pressure Piping System. — This includes all high-pressure piping in the boiler and engine rooms for the supply of turbines, pumps, etc., and for the supplementary supply to the heating system as may be needed. Pipe used for this purpose is full weight, wrought iron being used for sizes below 6 inches and open-hearth steel for larger sizes. The main drum at the rear of the boilers is of gun metal with nozzles cast in place. Expansion is provided for, so far as possible, by the use of sweep pipe bends and fittings of the long-turn pattern, all $2\frac{1}{2}$-inch and larger fittings being of this design with flange joints. The high-pressure connections are shown in Figs. 613, 614, and 616. Starting at the boilers (Fig. 614), 6-inch leads are carried to a 12-inch drum supported on lower piers and rolls at the rear of the boilers. From here a 6-inch branch leads to the main turbine, and two branches of the same size to a 6-inch auxiliary main, running beneath the engine-room

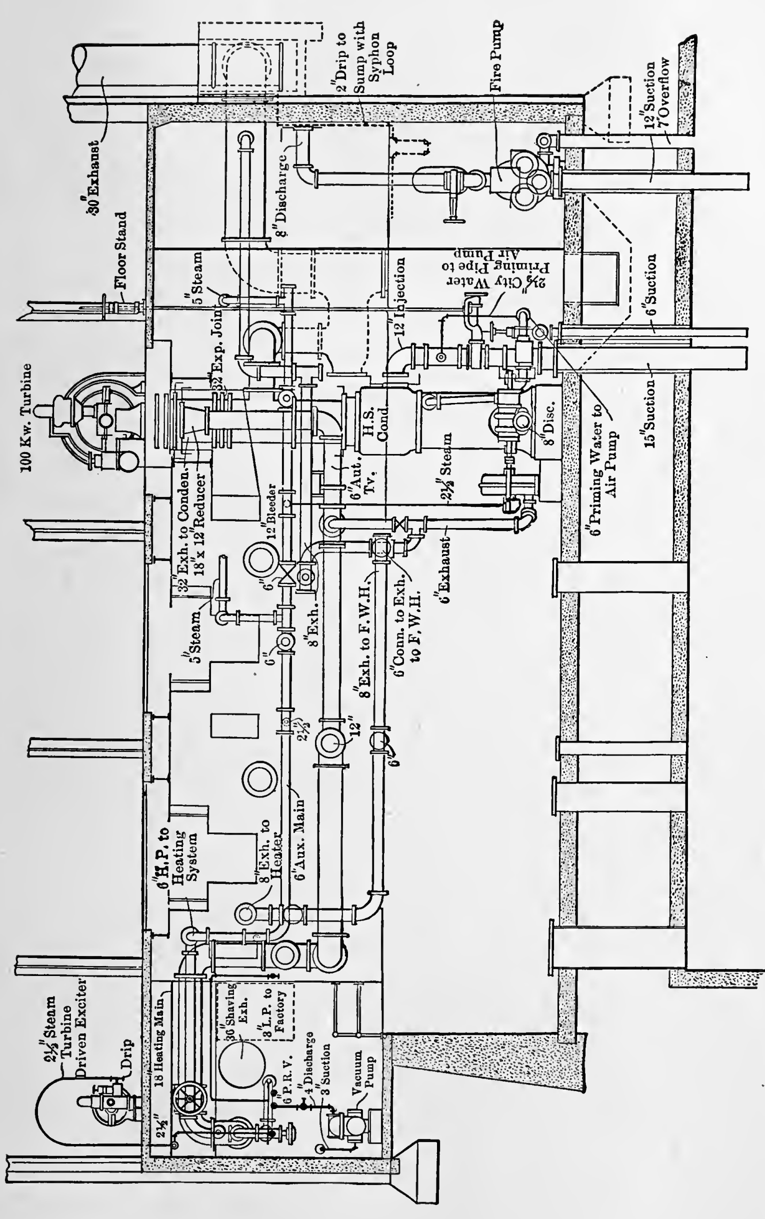


FIG. 615. Longitudinal Section of Engine Room Basement, showing Condenser Connections and Piping to Heating System.

floor, near to, and parallel with, the boiler-room wall. From this auxiliary main are taken the supplies to the various minor turbines and pumps, and also the branches leading to the low and intermediate-pressure system through reducing valves. The main drum is divided into two sections by means of a valve at the center, and each of these sections is connected with the auxiliary drum as shown in Figs. 612 and 616. The supplies to the various pumps are easily traced from Fig. 616, also the connections with the 18-inch heating main and the intermediate-pressure line, leading to the factory through the tunnel leaving the building as indicated in the upper right-hand corner of the drawing.

Exhaust System. — All low and intermediate pressure piping is full weight, sizes up to, and including, 12-inch being of wrought iron, while open-hearth steel is employed for the larger sizes. Standard-weight fittings are used for this work, those 6 inches and over being of the long-turn pattern. Flange joints are provided on all piping $2\frac{1}{2}$ inches and larger in diameter, the same as for high-pressure work. The exhaust piping is most clearly shown in Figs. 614, 615, and 616. Referring to Fig. 616 the 18-inch exhaust from the main turbine is shown as leading through a back-pressure valve into a 30-inch outboard line designed for the completed plant. This is clearly shown in elevation in Fig. 165. An 8-inch auxiliary exhaust connecting with the various pumps is shown in Fig. 615, parallel with, and below, the auxiliary high-pressure main already described. Steam from this enters the heating system through an oil separator. The 12-inch bleeder connection from the turbine leads to the 18-inch heating main and is shown in the same drawing, although more clearly in Fig. 616.

Drainage. — The blow-off main from the boilers is carried directly to the river through a 4-inch cast-iron pipe. Drips from high-pressure piping are trapped to the main receiving tank and pumped back to the boilers. Exhaust drips, and all condensation containing oil, are trapped to a cast-iron sump tank located in the condenser pit, and, together with other drainage, are discharged by means of a water ejector.

Water Supply, Feed Piping, etc. — Water for condensing and fire purposes is brought from the river through a cement conduit, a section of this, together with the 15-inch suction to the condenser, being shown in Figs. 614 and 615. The discharge from the condenser pump is into an 18-inch pipe leading to the river and shown in section in Fig. 614. Water pressure for fire protection is furnished by an 18 by 10 by 12-inch Underwriters' fire pump of 1000 gallons capacity, placed in the condenser pit; this is shown in elevation in Fig. 615 and in plan in Fig. 616 and takes its supply from the intake tunnel as there shown.

The house tank and boilers have two sources of supply, one directly

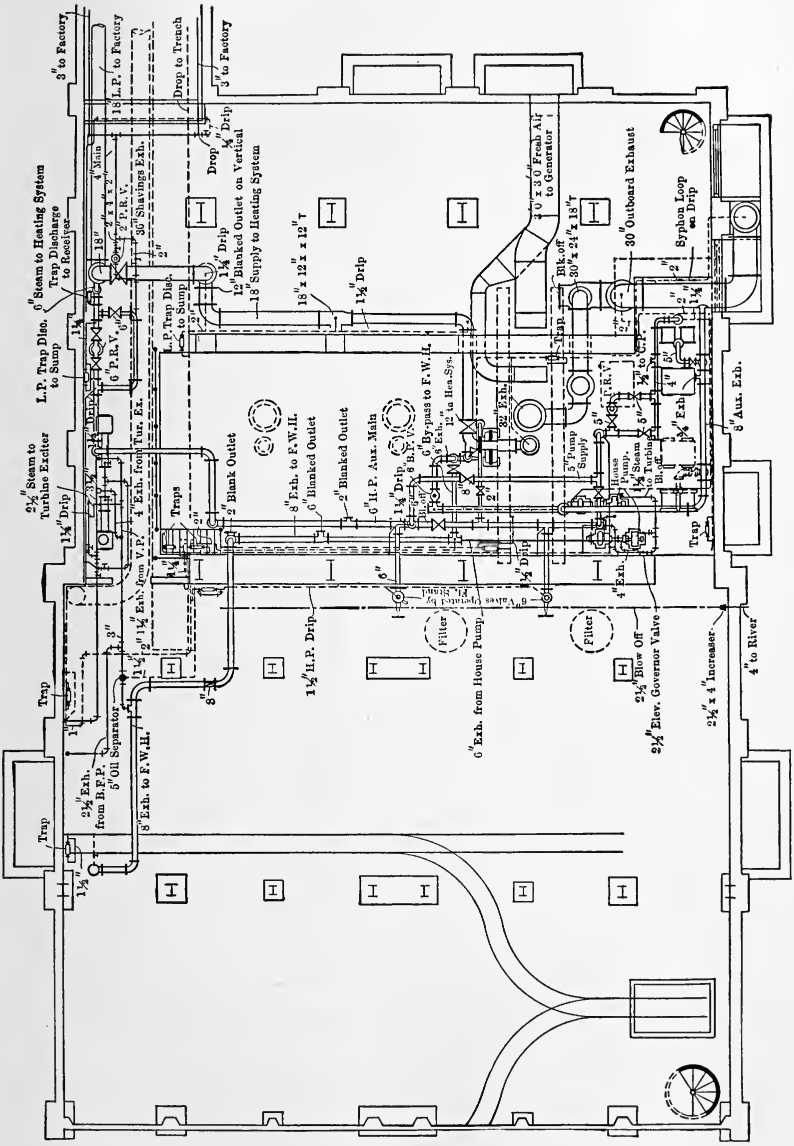


Fig. 616. Plan of Steam and Exhaust Piping.

from the city mains and the other from the intake tunnel. There is also a tank arrangement whereby water may be drawn from the discharge pipe of the condenser pump.

These various lines are shown in Fig. 617. A 6-inch connection from the city main enters as shown at the upper part of the drawing, toward the left, and, after passing through a meter, branches are carried to the

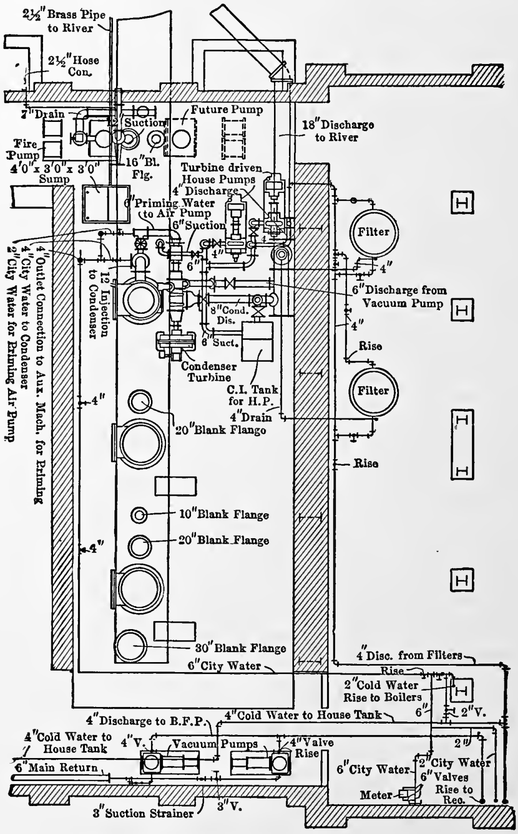


FIG. 617. Plan of Condenser Piping.

house tank, the receiving tank, the boilers, and to the priming pipes of the condenser and vacuum pumps.

The second source of supply, that from the intake tunnel, requires the use of two turbine-driven house pumps of the one-stage turbine type, located in the condenser pit as shown in Fig. 617. These pumps each have a capacity of 200 gallons per minute against a head of 150 feet, and discharge into a line of piping having branches connecting with the house tank, receiving tank, and boiler-feed pipe. A filtering equipment is also provided, as shown in Fig. 617, and so connected that the water from this source may be purified if desired.

Boiler Feed. — Feed lines connecting with the boilers are shown in Fig. 613. One of these supplies water either directly by city pressure or from the turbine house pumps. The other supply is from a pair of boiler-feed pumps connecting with a receiving tank located in the boiler room as shown. The feed pumps, two in number, are of the duplex, outside packed, pot-valve type, 8 by 5 by 10 inches in size. The tank is 4 feet in diameter by 6 feet in length, of $\frac{3}{8}$-inch iron plate, and is connected with both city pressure and the house pumps. Under ordinary working conditions the feed supply is first discharged into the tank and then pumped to the boilers through a heater of 1000-horsepower capacity located as shown in the drawing.

Heating System. — Factory buildings are heated by direct radiation in the form of coils and cast-iron radiators as best suited to local conditions. The Webster system of circulation is employed, a pair of 6 by 10 by 12-inch single-piston vacuum pumps being connected with the main return as shown in Fig. 617. These discharge into the receiving tank in the boiler room, and the condensation is pumped back to the boilers with the fresh feed.

Steam supply for the radiation has already been mentioned, coming principally through the bleeder connection from the main turbine, supplemented, when necessary, by live steam through a reducing valve.

Insulation. — In general, tanks, heaters, etc., are covered with 85 per cent magnesia blocks, finished with a plastic coat of the same material, the total thickness of the covering, when finished, being 2 inches. In addition to this, tanks and heater are provided with a covering of 7-ounce canvas. The insulation on that portion of the smoke pipe which comes outside of the building is protected by a covering of heavy sheet iron. Steam piping, both high and low pressure, is insulated with 85 per cent magnesia sectional covering. All cold-water piping, with the exception of the connections to the condenser, are covered with wool felt, having a lining of tarred paper. Pipe covering of all kinds is finished with a heavy canvas jacket and painted.

CHAPTER XXII. — Supplementary

PROPERTIES OF SATURATED AND SUPERHEATED STEAM

440. General. — The thermal and physical properties of water vapor though based on experimental data permit of accurate mathematical formulation, but the equations involved are too complex and unwieldy for everyday use. Tables and graphical charts calculated and plotted from these laws offer a simple and accurate means of solving practically all steam problems and recourse to thermodynamic analysis is seldom necessary.

Several tables and graphical charts of the properties of saturated and superheated steam have been published and though the values given by the various authorities differ somewhat from each other the variation is negligible for most engineering purposes. The recent tables of Peabody,* Marks and Davis,† and of Goodenough‡ embody the latest and most accurate researches and are most commonly used in engineering practice. These tables give the simultaneous physical and thermal properties of saturated and superheated steam for various pressures and temperatures. All three tables are practically identical in arrangement as far as saturated steam is concerned but differ somewhat in the treatment of superheated steam.

441. Notations. — It is to be regretted that there is no accepted standard set of symbols for designating the various properties of steam. The use of different notations for the same property as in the case with the tables under consideration leads to much confusion. In the following discussion an attempt has been made to follow general practice rather than that of any particular author.

442. Standard Units. — The mean B.t.u. or $\frac{1}{180}$ of the heat required to raise one pound of water from 32 deg. to 212 deg. fahr. is the accepted standard heat unit in all recent works on thermodynamics.

The mechanical equivalent of heat J may be taken for all engineering purposes as

$$1 \text{ mean B.t.u.} = 778 \text{ standard ft. lb.}$$

$$(\text{Goodenough, } J = 777.64; \text{ Marks \& Davis, } J = 777.54.)$$

The reciprocal of J or $\frac{1}{778}$ is generally designated by the letter A.

* Steam and Entropy Tables, Peabody, John Wiley & Sons, 1909.

† Steam Tables and Diagrams, Marks & Davis, Longmans Green & Co., 1912.

‡ Properties of Steam and Ammonia, Goodenough, John Wiley & Sons, 1915.

The value of the absolute zero has been variously given as ranging from 459.2 to 460.66 deg. fahr. below zero. The most generally accepted value is 459.6. For all engineering purposes, the value 460 degrees is sufficiently accurate. Temperatures referred to zero deg. fahr. are generally designated by t and absolute temperature by T.

The normal pressure of the atmosphere or one standard atmosphere is taken as 29.921 inch of mercury at 32 deg. fahr., or 14.6963 pounds per square inch. For most purposes these values may be taken as 30 inches of mercury at ordinary room temperature and 14.7 pounds per square inch, respectively. Steam pressure should always be stated in absolute terms and not "gauge" since the atmospheric pressure varies within wide limits. Notations p and P are commonly used to designate pressure but because of the various methods of measuring this property they should be qualified to this effect. In the following discussion p represents pounds per square inch absolute and P pounds per square foot absolute.

443. Quality. — This term applies strictly to the per cent of vapor in a mixture of vapor and water or *wet* steam and is usually designated by x; thus a quality of 95 signifies that 95 per cent of the total weight of the mixture is vapor. For saturated steam $x = 1$. The quality of superheated steam is designated by the temperature of the vapor or the degree of superheat. The latter term refers to the difference between the actual temperature and that of saturated vapor of the same pressure.

444. Temperature-Pressure Relation. *Saturated Steam.* — All properties of saturated steam depend on temperature only. For any temperature there is a corresponding pressure, the relationship being determined from formulas based upon experimental data. A large number of formulas have been proposed to represent this relationship but the more exact equations are too cumbersome for everyday use. In Marks & Davis' steam tables the pressure-temperature relationship is based upon the following law:

$$\log p = 10.51535 - 4873.71\, T^{-1} - 0.00405096\, T + 0.00000139296\, T^2. \quad (306)$$

Wet Steam. — The relation between pressure and temperature is the same for wet steam as for saturated since the quality does not affect the temperature.

Superheated Steam. — The temperature of superheated steam is not dependent solely upon the pressure and some additional property is necessary to fix the relationship.

445. Specific Volume. *Saturated Steam.* — The specific volume s of saturated steam or the number of cubic feet occupied by one pound, varies with the pressure and is equal to the sum of the original volume of one pound of water σ, and u the increase in volume during vaporization, thus:

$$s = u + \sigma. \tag{307}$$

Goodenough's modification of Linde's equation is

$$u = 0.59465 \frac{T}{p} - (1 + 0.0513\, p^{\frac{1}{2}}) \frac{m}{T^4}, \tag{308}$$

$$\log m = 10.825.$$

Wet Steam. — The specific volume v of wet steam may be calculated as follows:

$$v = xs + (1 - x)\,\sigma \tag{309}$$

$$= xu + \sigma. \tag{310}$$

s is given in all saturated steam tables. σ varies from 0.0161 cu. ft. per lb. at a pressure of 1 lb. per sq. in., absolute, to 0.02 cu. ft. at 300 lb. σ is so small compared with s that it may be neglected for most purposes and the specific volume becomes $v = xs$. v may be taken directly from the volume-entropy chart.

Superheated Steam. — The specific volume of superheated steam v' is given in all superheated tables. The values in Goodenough's tables were calculated from equation (308) by substituting $u = v' - \sigma$.

Wm. J. Goudie (Engineering, July 1, 1901) gives the following simple rule for determining the specific volume which gives satisfactory results for moderate degrees of superheat.

$$v' = s\,(1 + 0.0016\,t'), \tag{311}$$

in which

s = specific volume of saturated steam, pound per cubic foot,
t' = degree of superheat.

Tumlirz's formula is a simple and fairly accurate abridgment of equation (308) for moderate degrees of superheat but at higher temperatures gives results too low.

$$v' = 0.5962 \frac{T_{\text{sup.}}}{p} - 0.256. \tag{312}$$

446. Heat of the Liquid. — The heat of the liquid q, B.t.u. per pound above 32 deg. fahr., is the amount added to water at 32 deg. fahr. in order to bring it to the temperature of vaporization, thus:

$$q = \int_{492}^{T} c\,dT, \tag{313}$$

in which c = specific heat at constant pressure.

c varies with the temperature, but the relationship does not permit of simple formulation. If c_m = mean specific heat for the temperature range,

$$q = c_m (t - 32). \tag{314}$$

For many purposes it is sufficiently accurate to assume $c_m = 1$, then $q = t - 32$. The relationship between t, c, and c_m is shown in Fig. 618 for a wide range in temperatures.

The heat of the liquid is manifestly constant for a given temperature whatever may be the condition of the steam.

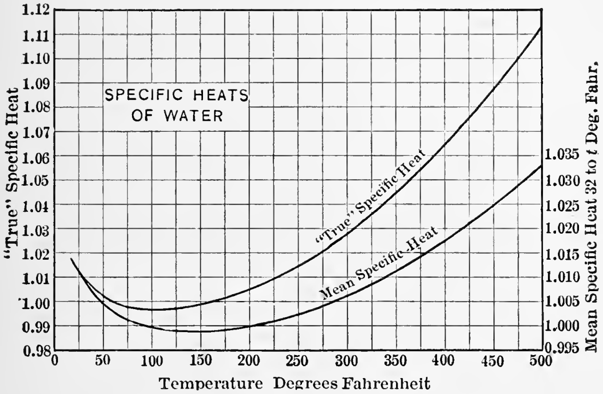


FIG. 618. Specific Heats of Water.

447. Latent Heat of Vaporization. — The latent heat of vaporization r, B.t.u. per pound above 32 deg. fahr., is the amount of heat required to change the fluid from a liquid to vapor at the same temperature. The latent heat has been accurately determined by direct experiment from 32 degrees to 356 deg. fahr. and numerous formulas have been based upon the experiments for calculating this quantity. Goodenough's values are calculated from the Clapeyron relation:

$$r = A (s - u) T \frac{dp}{dT}. \tag{315}$$

A simple formula which gives accurate results from 32 degrees to 400 deg. fahr. is

$$r = 970.4 - 0.655 (t - 212) - 0.00045 (t - 212)^2. \tag{316}$$

At higher temperatures Hennings' exponential formula as modified by Dr. Davis is perhaps more accurate than equation (316),

$$r = 139 (689 - t)^{0.315}. \tag{317}$$

The latent heat decreases with the increase in temperature until a temperature of approximately 706 deg. fahr. (corresponding pressure

3200 lb. per sq. in.) is reached when its value becomes 0. This is called the *critical* temperature.

Values of r are given in all saturated steam tables.

Special interest attaches to the values of r at 212 deg. fahr. because of its common use in engine and boiler tests. The following values have been assigned to this quantity.

Regnault	966.0	Marks and Davis	970.4
Peabody	969.7	Smith	972.0
Heck	971.2	Goodenough	971.6

The correct value is probably quite close to 972.0.

External Latent Heat. — During the heating of the liquid the change in volume is very small and may be neglected, hence the external work done is negligible and also practically all of the heat goes to increase the energy of the liquid. During vaporization, however, the volume changes from σ to s. Since the pressure remains constant, the external work that must be done to provide for increase in volume is

$$P(s - \sigma) = Pu \tag{318}$$

and the corresponding heat or external latent heat is

$$AP(s - \sigma) = APu. \tag{319}$$

Internal Latent Heat. — The heat r added during vaporization is used in increasing the energy and is doing external work. Hence the difference, or internal latent heat ρ, B.t.u. per pound above 32 deg. fahr.,

$$\rho = r - APu, \tag{320}$$

is the heat required to do disgregation work.

448. Total Heat or Heat Content. — *The total heat of saturated steam* λ, B.t.u. per pound above 32 deg. fahr., is evidently the sum of the heat of the liquid and the heat of vaporization, or

$$\lambda = r + q \tag{321}$$

$$= \rho + APu + q. \tag{322}$$

The total heat of saturated steam may be calculated by means of the Davis formula:

$$\lambda = 1046.187 + 0.6077\,t - 0.00055\,t^2. \tag{323}$$

The quantity $(\rho + q)\frac{1}{A}$ gives the increase in energy of the saturated vapor over that of the liquid at 32 deg. fahr. and is called the *intrinsic energy.*

Wet Steam. — If vaporization is not complete the heat content H_w B.t.u. per pound above 32 deg. fahr. may be expressed:

$$H_w = xr + q \tag{324}$$

$$= x\rho + AP\,xu + q. \tag{325}$$

Superheated Steam. — If heat is added at constant pressure after vaporization is completed, the vapor will be superheated, and the heat content H_s is

$$H_s = r + q + C_m t' \qquad (326)$$

$$= \lambda + C_m t', \qquad (327)$$

in which

C_m = mean specific heat of the superheated vapor at constant pressure,

t' = degree of superheat = $t_{\text{sup.}} - t_{\text{sat.}}$.

Goodenough gives the following formula for calculating the total heat of superheated steam, absolute temperature of the steam T_s deg. fahr.

$$H_s = 0.320\ T_s + 0.000063\ T_s^3 - \frac{23{,}583}{T_s} - \frac{C_3 p\left(1 + 0.0342\ p^{\frac{1}{2}}\right)}{T_s^4} + 0.00333\ p + 948.7, \qquad (328)$$

$\log C_3 = 10.791155.$

449. Specific Heat of Steam. *Saturated Steam.* — If the amount of heat required to raise the temperature of saturated steam one degree and still maintain a saturated condition is construed as the specific heat of saturated steam, then the quantity is negative, since heat must be abstracted to effect this result.

$$C_{\text{sat.}} = 0.35 - 0.000666\ (t - 212) - \frac{r}{T}. \qquad (329)$$

Superheated Steam. — The *true* or *instantaneous* specific heat C' of superheated steam at constant pressure is the amount required to increase the temperature of one pound one degree fahr. Goodenough's equation based on the experiment of Knoblauch and Jakob is

$$C' = 0.320 + 0.000126\ T_s + \frac{23{,}583}{T_s^2} + \frac{C_2 p\left(1 + 0.0342\ p^{\frac{1}{2}}\right)}{T_s^5}, \qquad (330)$$

$\log C_2 = 11.3936.$

The *mean* specific heat may be calculated from superheated steam tables as follows:

$$C_m = \frac{H_{\text{sup.}} - \lambda}{t'}. \qquad (331)$$

The true and mean specific heat of superheated steam at constant pressure for a wide range in pressures and temperatures are shown in Figs. 619 and 620. The curves are taken from Goodenough's "Principles of Thermodynamics."

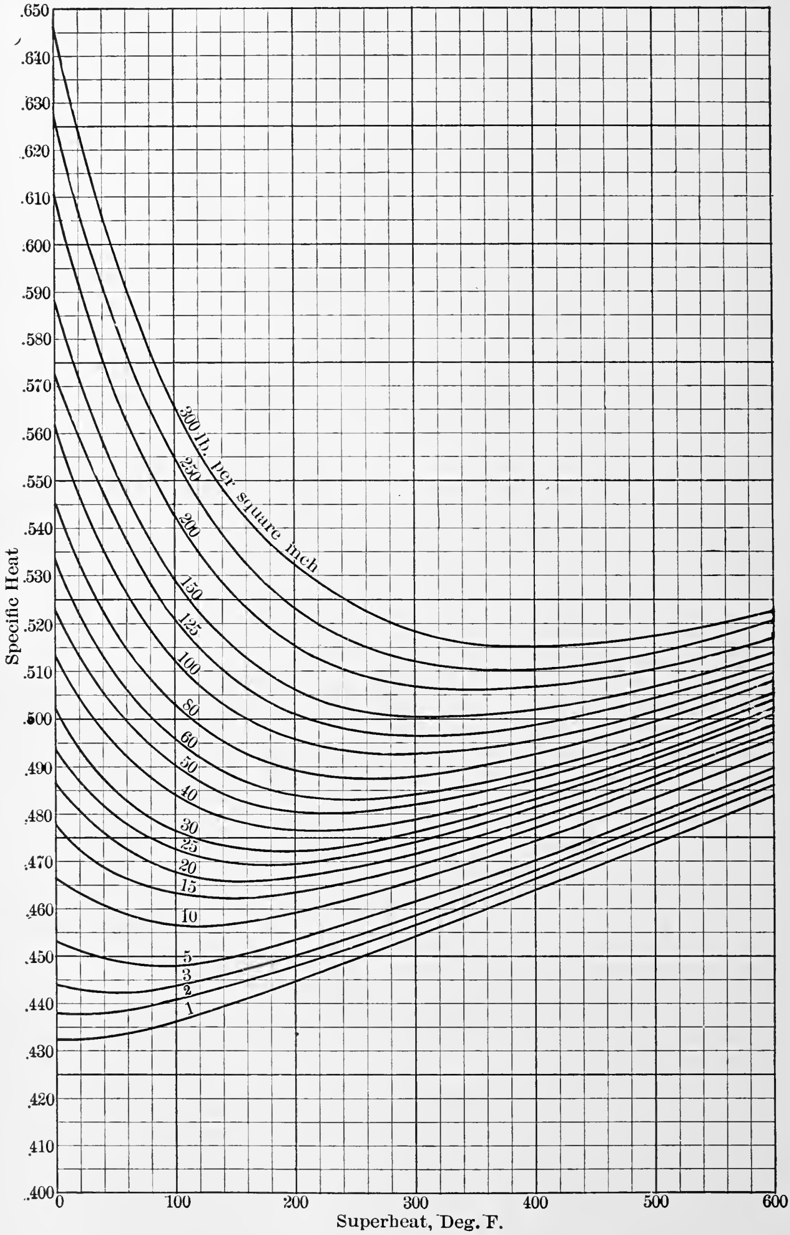


FIG. 619. True Specific Heat of Superheated Steam.

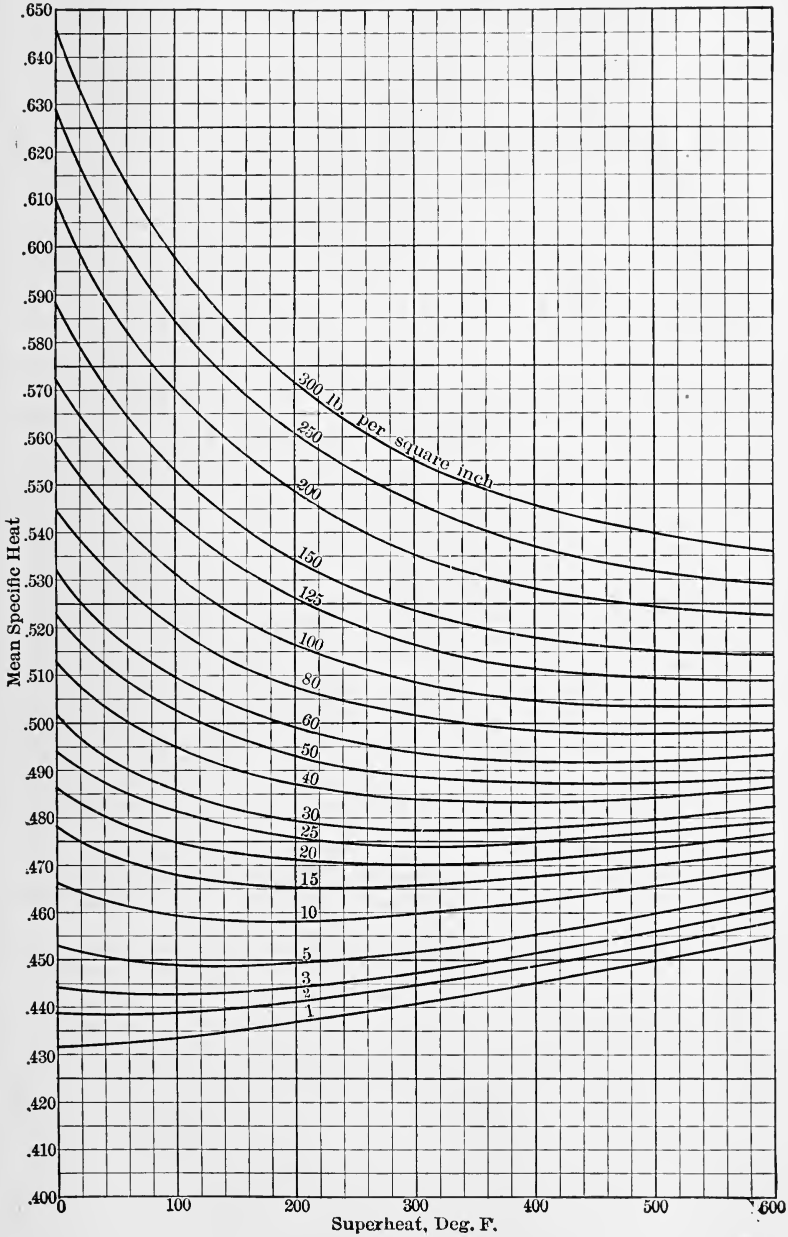


FIG. 620. Mean Specific Heat of Superheated Steam.

TABLE 166.

MEAN SPECIFIC HEAT OF SUPERHEATED STEAM.

(Computed from Marks and Davis' Steam Tables.)

Absolute Pressure, Pounds per Square Inch.	Degrees of Superheat, Fahr.																			
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
1	.452	.452	.453	.453	.454	.454	.454	.455	.455	.455	.455	.455	.455	.456	.456	.456	.456	.456	.456	.457
5	.460	.460	.460	.460	.460	.460	.459	.459	.459	.459	.459	.459	.459	.459	.459	.459	.460	.460	.460	.460
10	.465	.465	.465	.465	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.464	.465	.465	.465	.465
15	.470	.470	.470	.470	.470	.470	.469	.469	.469	.469	.469	.469	.469	.469	.469	.468	.468	.468	.468	.468
20	.475	.475	.475	.475	.474	.474	.474	.474	.473	.473	.473	.473	.473	.472	.472	.472	.472	.472	.471	.471
25	.480	.480	.480	.479	.479	.479	.478	.478	.478	.477	.477	.477	.477	.477	.476	.476	.476	.475	.475	.475
30	.485	.485	.484	.484	.484	.483	.483	.483	.483	.482	.482	.481	.481	.481	.480	.480	.479	.479	.478	.477
35	.490	.490	.489	.489	.488	.488	.487	.487	.486	.486	.486	.485	.485	.484	.484	.483	.482	.482	.481	.480
40	.495	.494	.494	.493	.492	.492	.491	.491	.490	.490	.489	.489	.488	.488	.487	.486	.485	.484	.483	.482
50	.509	.508	.507	.506	.504	.503	.501	.500	.499	.498	.497	.496	.496	.495	.494	.493	.491	.490	.489	.487
75	.540	.535	.533	.530	.528	.525	.523	.521	.519	.517	.515	.513	.512	.511	.509	.506	.504	.502	.500	.497
100	.570	.560	.556	.552	.550	.546	.542	.540	.537	.534	.531	.528	.526	.524	.522	.518	.515	.512	.509	.505
125	.590	.585	.580	.575	.570	.565	.560	.556	.552	.548	.545	.542	.539	.537	.534	.532	.524	.520	.517	.510
150	.620	.615	.606	.597	.592	.585	.579	.572	.566	.562	.558	.554	.550	.547	.544	.539	.533	.528	.524	.518
175	.660	.645	.633	.622	.614	.605	.597	.589	.582	.577	.572	.566	.562	.558	.555	.548	.541	.535	.531	.524
200	.690	.675	.657	.645	.634	.623	.614	.605	.596	.590	.584	.578	.572	.568	.563	.556	.548	.542	.537	.529
225	.730	.710	.686	.672	.656	.643	.631	.620	.611	.603	.595	.589	.583	.578	.573	.562	.553	.547	.543	.535
250	.770	.740	.712	.695	.678	.663	.648	.635	.624	.615	.606	.599	.593	.587	.582	.570	.562	.555	.549	.540
275	.800	.770	.746	.722	.700	.681	.664	.650	.638	.629	.620	.610	.602	.596	.590	.578	.569	.561	.555	.545
300	.850	.805	.773	.740	.724	.702	.683	.666	.652	.641	.630	.621	.612	.605	.599	586	.576	.568	.561	.550

450. Entropy. *General.* — No change in a system of bodies that takes place of itself can increase the available energy of the system. As a matter of fact the actual physical process is accompanied by frictional effects and the quantity of energy available for transformation into work is decreased. This decrease in available energy or increase in unavailable energy is given the name *increase of entropy.* Although the solution of all engineering problems involving thermodynamic changes can be obtained without employing entropy, still its use simplifies the calculation in much the same manner that logarithms facilitate complex numerical computations. Increase of entropy between the absolute temperatures T_2 and T_1 may be expressed mathematically

$$\text{Increase of entropy} = \int_{T_2}^{T_1} \frac{dQ}{T}, \tag{332}$$

in which dQ represents an infinitesimal amount of heat and T the absolute temperature at which it is added.

Entropy of the Liquid. — The increase in entropy θ due to heating one pound of liquid from 32 deg. fahr. to temperature T is

$$\theta = \int_{492}^{T_1} \frac{dq}{T} = \int_{492}^{T_1} \frac{c\,dT}{T}, \tag{333}$$

in which

T_1 = absolute temperature of the liquid = $t_1 + 460$,
q = heat of the liquid above 32 deg. fahr., B.t.u. per pound,
c = specific heat of water at temperature T.

Since c varies with the temperature according to a rather complex law, the integration in equation (333) does not reduce to a simple form. For example, Goodenough's equation for the range 32 – 212 deg. fahr. assumes the form

$$\theta = 2.3623 \log T + 0.0045775 \log (t + 4) - 0.00022609\, T + 0.00000012867\, T^2 - 6.28787. \tag{334}$$

If the value of the mean specific c_m is known for the given temperature range equation (333) reduces to the simple form

$$\theta = C_m \log_e \frac{T}{492}. \tag{335}$$

Values of θ are found in all unabridged steam tables.

Entropy of Vaporization. — Since the temperature at which vaporization takes place is constant the change of entropy experienced by the fluid during vaporization is

$$n = \frac{Q}{T} = \frac{r}{T}. \tag{336}$$

If vaporization is incomplete as in case of wet steam

$$n_w = xn = \frac{xr}{T}. \tag{337}$$

Entropy of Superheat. — The entropy change during superheating may be expressed

$$n_s = \int_{T_v}^{T_s} \frac{C'\,dT}{T}, \tag{338}$$

T_v = temperature of the vapor.

If the value of the mean specific heat C_m for the temperature range T_v to T_s is known the integration of equation (338) reduces to the simple form

$$n_s = C_m \log_e \frac{T_s}{T_v}. \tag{339}$$

Total Entropy of Saturated Steam. — The increase in entropy from liquid at 32 deg. fahr. to saturated vapor at temperature T is

$$N = n + \theta = \frac{r}{T} + \theta. \tag{340}$$

Total Entropy of Wet Steam.

$$N_w = xn + \theta = \frac{xr}{T} + \theta. \tag{341}$$

Total Entropy of Superheated Steam.

$$N = n + n_s + \theta = \frac{r}{T} + C_m \log_e \frac{T_s}{T_v} + \theta. \tag{342}$$

Using Knoblauch and Jakob's values for the specific heat of superheated steam, Goodenough gives the following rule for calculating the total entropy of superheated steam

$$N_s = 0.73683 \log T_s + 0.000126\, T_s - \frac{11791.5}{T_s^2} - 0.2535 \log p$$
$$- \frac{C_4\, p\,(1 + 0.0342\, p)}{T_s^5} - 0.08085. \tag{343}$$

$\log C_4 = 10.69464.$

Tables 167 and 168 are abridged from Marks and Davis' "Steam Tables and Diagrams."

TABLE 167.

PROPERTIES OF SATURATED STEAM.* (Marks and Davis.)

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
P	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
† 0.1	35.03	3.05	1071.7	1074.7	1017.3	54.4	0.0062	2.1666	2.1728	2935.0	0.000340
† 0.2	53.15	21.23	1061.6	1082.8	1005.2	56.5	0.0423	2.0704	2.1127	1524.0	0.000656
† 0.3	64.49	32.57	1055.3	1087.9	997.7	57.6	0.0640	2.0135	2.0775	1041.0	0.000961
0.4	72.91	40.95	1050.6	1091.6	992.4	58.5	0.0800	1.9730	2.0530	794.0	0.001259
0.5	79.68	47.71	1047.0	1094.6	987.6	59.3	0.0926	1.9413	2.0332	642.0	0.001555
0.6	85.32	53.34	1043.8	1097.1	983.9	59.9	0.1029	1.9155	2.0184	541.0	0.001850
0.7	90.18	58.18	1041.1	1099.3	980.7	60.4	0.1117	1.8936	2.0053	467.0	0.002143
0.8	94.46	62.45	1038.7	1101.2	977.8	61.0	0.1195	1.8747	1.9942	412.0	0.002431
0.9	98.33	66.31	1036.6	1102.9	975.2	61.4	0.1265	1.8578	1.9843	367.9	0.002719
1	101.83	69.8	1034.6	1104.4	972.9	61.7	0.1327	1.8427	1.9754	333.0	0.00300
2	126.15	94.0	1021.0	1115.0	956.7	64.3	0.1749	1.7431	1.9180	173.5	0.00576
3	141.52	109.4	1012.3	1121.6	946.4	65.8	0.2008	1.6840	1.8848	118.5	0.00845
4	153.01	120.9	1005.7	1126.5	938.6	67.0	0.2198	1.6416	1.8614	90.5	0.01107
5	162.28	130.1	1000.3	1130.5	932.4	68.0	0.2348	1.6084	1.8432	73.33	0.01364
6	170.06	137.9	995.8	1133.7	927.0	68.8	0.2571	1.5814	1.8285	71.89	0.01616
7	176.85	144.7	991.8	1136.5	922.4	69.4	0.2579	1.5582	1.8161	53.56	0.01867
8	182.86	150.8	988.2	1139.0	918.2	70.0	0.2673	1.5380	1.8053	47.27	0.02115
9	188.27	156.2	985.0	1141.1	914.4	70.6	0.2756	1.5202	1.7958	42.36	0.02361
10	193.22	161.1	982.0	1143.1	910.9	71.1	0.2832	1.5042	1.7874	38.38	0.02606
11	197.75	165.7	979.2	1144.9	907.8	71.5	0.2902	1.4895	1.7797	35.10	0.02849
12	201.96	169.9	976.6	1146.5	904.8	71.8	0.2967	1.4760	1.7727	32.36	0.03090

* Courtesy of the Publishers, Longmans, Green & Co. † Interpolated.

PROPERTIES OF SATURATED STEAM — (*Continued*).

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
p	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
13	205.87	173.8	974.2	1148.0	902.0	72.2	0.3025	1.4639	1.7664	30.03	0.03330
14	209.55	177.5	971.9	1149.4	899.3	72.6	0.3081	1.4523	1.7604	28.02	0.03569
14.7	212.00	180.0	970.4	1150.4	897.6	72.9	0.3118	1.4447	1.7565	26.79	0.03732
15	213.0	181.0	969.7	1150.7	896.8	72.9	0.3133	1.4416	1.7549	26.27	0.03806
20	228.0	196.1	960.0	1156.2	885.8	74.3	0.3355	1.3965	1.7320	20.08	0.04980
25	240.1	208.4	952.0	1160.4	876.8	75.3	0.3532	1.3604	1.7136	16.30	0.0614
30	250.3	218.8	945.1	1163.9	869.0	76.2	0.3680	1.3311	1.6991	13.74	0.0728
35	259.3	227.9	938.9	1166.8	862.1	76.9	0.3868	1.3060	1.6868	11.89	0.0841
40	267.3	236.1	933.3	1169.4	855.9	77.6	0.3920	1.2841	1.6761	10.49	0.0953
45	274.5	243.4	928.2	1171.6	850.3	78.1	0.4021	1.2644	1.6665	9.39	0.1065
50	281.0	250.1	923.5	1173.6	845.0	78.6	0.4113	1.2468	1.6581	8.51	0.1175
55	287.1	256.3	919.0	1175.4	840.2	78.9	0.4196	1.2309	1.6505	7.78	0.1285
60	292.7	262.1	914.9	1177.0	835.6	79.7	0.4272	1.2160	1.6432	7.17	0.1394
65	298.0	265.7	911.0	1178.5	831.4	79.8	0.4344	1.2034	1.6368	6.65	0.1503
70	302.9	272.6	907.2	1179.8	827.3	80.1	0.4411	1.1896	1.6307	6.20	0.1612
75	307.6	277.4	903.7	1181.8	823.5	80.5	0.4474	1.1778	1.6252	5.81	0.1721
80	312.0	282.0	900.3	1182.3	819.8	80.7	0.4535	1.1665	1.6200	5.47	0.1829
85	316.3	286.3	897.1	1183.4	816.3	81.0	0.4590	1.1561	1.6151	5.16	0.1937
90	320.3	290.5	893.9	1184.4	813.0	81.2	0.4644	1.1461	1.6105	4.89	0.2044
95	324.0	294.5	890.9	1185.4	809.7	81.5	0.4694	1.1367	1.6061	4.65	0.2151
100	327.8	298.3	888.0	1186.3	806.6	81.7	0.4743	1.1277	1.6020	4.429	0.2258

105	331.4	302.0	885.2	1187.2	803.6	81.9	0.4789	1.1191	1.5980	4.230	0.2365
110	334.8	305.5	882.5	1188.0	800.7	82.1	0.4834	1.1108	1.5942	4.047	0.2472
115	338.1	309.0	879.8	1188.8	797.9	82.3	0.4877	1.1030	1.5907	3.880	0.2577
120	341.3	312.3	877.2	1189.6	795.2	82.5	0.4919	1.0954	1.5873	3.726	0.2683
125	344.4	315.5	874.7	1190.3	792.6	82.6	0.4959	1.0880	1.5839	3.583	0.2791
130	347.4	318.6	872.3	1191.0	790.0	82.8	0.4998	1.0809	1.5807	3.452	0.2897
135	350.3	321.7	869.9	1191.6	787.5	82.9	0.5035	1.0742	1.5777	3.331	0.3002
140	353.1	324.6	867.6	1192.2	785.0	83.0	0.5072	1.0675	1.5747	3.219	0.3107
145	355.8	327.4	865.4	1192.8	782.7	83.2	0.5107	1.0612	1.5719	3.112	0.3213
150	358.5	330.2	863.2	1193.4	780.4	83.3	0.5142	1.0550	1.5692	3.012	0.3320
155	361.0	332.9	861.0	1194.0	778.1	83.5	0.5175	1.0489	1.5664	2.920	0.3425
160	363.6	335.6	858.8	1194.5	775.8	83.6	0.5208	1.0431	1.5639	2.834	0.3529
165	366.0	338.2	856.8	1195.0	773.6	83.7	0.5239	1.0376	1.5615	2.753	0.3633
170	368.5	340.7	854.7	1195.4	771.5	83.8	0.5269	1.0321	1.5590	2.675	0.3738
175	370.8	343.2	852.7	1195.9	769.4	83.9	0.5299	1.0268	1.5567	2.602	0.3843
180	373.1	345.6	850.8	1196.4	767.4	84.0	0.5328	1.0215	1.5543	2.533	0.3948
185	375.4	348.0	848.8	1196.8	765.4	84.1	0.5356	1.0164	1.5520	2.468	0.4052
190	377.6	350.4	846.9	1197.3	763.4	84.2	0.5384	1.0114	1.5498	2.406	0.4157
195	379.8	352.7	845.0	1197.7	761.4	84.3	0.5410	1.0066	1.5476	2.346	0.4262
200	381.9	354.9	843.2	1198.1	759.5	84.4	0.5437	1.0019	1.5456	2.290	0.437
205	384.0	357.1	841.4	1198.5	757.6	84.5	0.5463	0.9973	1.5436	2.237	0.447
210	386.0	359.2	839.6	1198.8	755.8	84.5	0.5488	0.9928	1.5416	2.187	0.457
215	388.0	361.4	837.9	1199.2	754.0	84.6	0.5513	0.9885	1.5398	2.138	0.468
220	389.9	363.4	836.2	1199.6	752.3	84.7	0.5538	0.9841	1.5379	2.091	0.478
225	391.9	365.5	834.4	1199.9	750.5	84.7	0.5562	0.9799	1.5361	2.046	0.489
230	393.8	367.5	832.8	1200.2	748.8	84.8	0.5586	0.9758	1.5341	2.004	0.499
240	397.4	371.4	829.5	1200.9	745.4	85.0	0.5633	0.9676	1.5309	1.924	0.520
250	401.1	375.2	826.3	1201.5	742.0	85.1	0.5676	0.9600	1.5276	1.850	0.541
275	409.5	384.2	818.6	1202.8	734.2	85.3	0.5780	0.9419	1.5199	1.686	0.593
300	417.5	392.7	811.3	1204.1	726.8	85.6	0.5878	0.9251	1.5129	1.551	0.645

451. Mollier Diagram. — Steam tables are often accompanied by graphical charts that may be used to great advantage in the solution of thermodynamic problems. Fig. 621 gives a skeleton outline of the total heat-entropy diagram and Fig. 622 a reduced copy of the complete chart. The first conception of the heat-entropy chart is due to Dr. R. Mollier of Dresden, hence the name, Mollier Diagram.

Referring to Fig. 621 abscissas represent total entropy and ordinates represent B.t.u. per pound. Vertical lines then indicate constant entropy and horizontal lines constant heat content. P_1P_1 and P_2P_2 represent lines of constant pressure and X_1X_1 and X_2X_2 lines of constant quality. Evidently any point in the chart represents a fixed

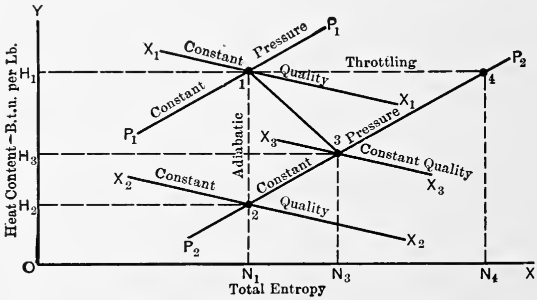


Fig. 621. Mollier Diagram — Skeleton Outline.

condition of heat content, pressure, quality, and entropy as determined by its location with respect to the different lines. Thus, point 1 represents a pressure P_1 as determined by the numerical value of line P_1P_1, quality X_1 by its location on line X_1X_1, entropy N_1 by its projection on the X axis and heat content H_1 by its projection on the Y axis.

In addition to the Mollier diagram the Marks and Davis tables include a total heat-pressure diagram which is of great assistance in the solution of problems involving ratios of expansion.

The Ellenwood Charts (John Wiley & Sons, Publishers) have a much wider field of application than the diagrams mentioned above and afford a simple and accurate means of solving practically all thermodynamic problems involving the use of the properties of steam.

TABLE 168.

PROPERTIES OF SUPERHEATED STEAM.

Reproduced by Permission from Marks and Davis' "Steam Tables and Diagrams."

Pressure, Pounds Absolute.		Saturated Steam.	Degrees of Superheat.							Pressure, Pounds Absolute.
			50	100	150	200	250	300		
	t	162.3	212.3	262.3	312.3	362.3	412.3	462.3	*t*	
5	*v*	73.3	79.7	85.7	91.8	97.8	103.8	109.8	*v*	5
	h	1130.5	1153.5	1176.4	1199.5	1222.5	1245.6	1268.7	*h*	
	t	193.2	243.2	293.2	343.2	393.2	443.2	493.2	*t*	
10	*v*	38.4	41.5	44.6	47.7	50.7	53.7	56.7	*v*	10
	h	1143.1	1166.3	1189.5	1212.7	1236.0	1259.3	1282.5	*h*	
	t	213.0	263.0	313.0	363.0	413.0	463.0	513.0	*t*	
15	*v*	26.27	28.40	30.46	32.50	34.53	36.56	38.58	*v*	15
	h	1150.7	1174.2	1197.6	1221.0	1244.4	1267.7	1291.1	*h*	
	t	228.0	278.0	328.0	378.0	428.0	478.0	528.0	*t*	
20	*v*	20.08	21.69	23.25	24.80	26.33	27.85	29.37	*v*	20
	h	1156.2	1179.9	1203.5	1227.1	1250.6	1274.1	1297.6	*h*	
	t	240.1	290.1	340.1	390.1	440.1	490.1	540.1	*t*	
25	*v*	16.30	17.60	18.86	20.10	21.32	22.55	23.77	*v*	25
	h	1160.4	1184.4	1208.2	1231.9	1255.6	1279.2	1302.8	*h*	
	t	250.4	300.4	350.4	400.4	450.4	500.4	550.4	*t*	
30	*v*	13.74	14.83	15.89	16.93	17.97	18.99	20.00	*v*	30
	h	1163.9	1188.1	1212.1	1236.0	1259.7	1283.4	1307.1	*h*	
	t	259.3	309.3	359.3	409.3	459.3	509.3	559.3	*t*	
35	*v*	11.89	12.85	13.75	14.65	15.54	16.42	17.30	*v*	35
	h	1166.8	1191.3	1215.4	1239.4	1263.3	1287.1	1310.8	*h*	
	t	267.3	317.3	367.2	417.3	467.3	517.3	567.3	*t*	
40	*v*	10.49	11.33	12.13	12.93	13.70	14.48	15.25	*v*	40
	h	1169.4	1194.0	1218.4	1242.4	1266.4	1290.3	1314.1	*h*	
	t	274.5	324.5	374.5	424.5	474.5	524.5	574.5	*t*	
45	*v*	9.39	10.14	10.86	11.57	12.27	12.96	13.65	*v*	45
	h	1171.6	1196.6	1221.0	1245.2	1269.3	1293.2	1317.0	*h*	
	t	281.0	331.0	381.0	431.0	481.0	531.0	581.0	*t*	
50	*v*	8.51	9.19	9.84	10.48	11.11	11.74	12.36	*v*	50
	h	1173.6	1198.8	1223.4	1247.7	1271.8	1295.8	1319.7	*h*	
	t	287.1	337.1	387.1	437.1	487.1	537.1	587.1	*t*	
55	*v*	7.78	8.40	9.00	9.59	10.16	10.73	11.30	*v*	55
	h	1175.4	1200.8	1225.6	1250.0	1274.2	1298.1	1322.0	*h*	
	t	292.7	342.7	392.7	442.7	492.7	542.7	592.7	*t*	
60	*v*	7.17	7.75	8.30	8.84	9.36	9.89	10.41	*v*	60
	h	1177.0	1202.6	1227.6	1252.1	1276.4	1300.4	1324.3	*h*	
	t	298.0	348.0	398.0	448.0	498.0	548.0	598.0	*t*	
65	*v*	6.65	7.20	7.70	8.20	8.69	9.17	9.65	*v*	65
	h	1178.5	1204.4	1229.5	1254.0	1278.4	1302.4	1326.4	*h*	
	t	302.9	352.9	402.9	452.9	502.9	552.9	602.9	*t*	
70	*v*	6.20	6.71	7.18	7.65	8.11	8.56	9.01	*v*	70
	h	1179.8	1205.9	1231.2	1255.8	1280.2	1304.3	1328.3	*h*	
	t	307.6	357.6	407.6	457.6	507.6	557.6	607.6	*t*	
75	*v*	5.81	6.28	6.73	7.17	7.60	8.02	8.44	*v*	75
	h	1181.1	1207.5	1232.8	1257.5	1282.0	1306.1	1330.1	*h*	
	t	312.0	362.0	412.0	462.0	512.0	562.0	612.0	*t*	
80	*v*	5.47	5.92	6.34	6.75	7.17	7.56	7.95	*v*	80
	h	1182.3	1208.8	1234.3	1259.0	1283.6	1307.8	1331.9	*h*	
	t	316.3	366.3	416.3	466.3	516.3	566.3	616.3	*t*	
85	*v*	5.16	5.59	6.99	6.38	6.76	7.14	7.51	*v*	85
	h	1183.4	1210.2	1235.8	1260.6	1285.2	1309.4	1333.5	*h*	

t = Temperature, deg. fahr.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees, B.t.u.

TABLE 168. — *Continued.*

Pressure, Pounds Absolute.		Saturated Steam.	Degrees of Superheat.							Pressure, Pounds Absolute.
			50	100	150	200	250	300		
	t	320.3	370.3	420.3	470.3	520.3	570.3	620.3	t	
90	v	4.89	5.29	5.67	6.04	6.40	6.76	7.11	v	90
	h	1184.4	1211.4	1237.2	1262.0	1286.6	1310.8	1334.9	h	
	t	324.1	374.1	424.1	474.1	524.1	574.1	624.1	t	
95	v	4.65	5.03	5.39	5.74	6.09	6.43	6.76	v	95
	h	1185.4	1212.6	1238.4	1263.4	1288.1	1312.3	1336.4	h	
	t	327.8	377.8	427.8	477.8	527.8	577.8	627.8	t	
100	v	4.43	4.79	5.14	5.47	5.80	6.12	6.44	v	100
	h	1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8	h	
	t	331.4	381.4	431.4	481.4	531.4	581.4	631.4	t	
105	v	4.23	4.58	4.91	5.23	5.54	5.85	6.15	v	105
	h	1187.2	1214.9	1240.8	1265.9	1290.6	1314.9	1339.1	h	
	t	334.8	384.8	434.8	484.8	534.8	584.8	634.8	t	
110	v	4.05	4.38	4.70	5.01	5.31	5.61	5.90	v	110
	h	1188.0	1215.9	1242.0	1267.1	1291.9	1316.2	1340.4	h	
	t	338.1	388.1	438.1	488.1	538.1	588.1	638.1	t	
115	v	3.88	4.20	4.51	4.81	5.09	5.38	5.66	v	115
	h	1188.8	1216.9	1243.1	1268.2	1293.0	1317.3	1341.5	h	
	t	341.3	391.3	441.3	491.3	541.3	591.3	641.3	t	
120	v	3.73	4.04	4.33	4.62	4.89	5.17	5.44	v	120
	h	1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7	h	
	t	344.4	394.4	444.4	494.4	544.4	594.4	644.4	t	
125	v	3.58	3.88	4.17	4.45	4.71	4.97	5.23	v	125
	h	1190.3	1218.8	1245.1	1270.4	1295.2	1319.5	1343.8	h	
	t	347.4	397.4	447.4	497.4	547.4	597.4	647.4	t	
130	v	3.45	3.74	4.02	4.28	4.54	4.80	5.05	v	130
	h	1191.0	1219.7	1246.1	1271.4	1296.2	1320.6	1344.9	h	
	t	350.3	400.3	450.3	500.3	550.3	600.3	650.3	t	
135	v	3.33	3.61	3.88	4.14	4.38	4.63	4.87	v	135
	h	1191.6	1220.6	1247.0	1272.3	1297.2	1321.6	1345.9	h	
	t	353.1	403.1	453.1	503.1	553.1	603.1	653.1	t	
140	v	3.22	3.49	3.75	4.00	4.24	4.48	4.71	v	140
	h	1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9	h	
	t	355.8	405.8	455.8	505.8	555.8	605.8	655.8	t	
145	v	3.12	3.38	3.63	3.87	4.10	4.33	4.56	v	145
	h	1192.8	1222.2	1248.8	1274.2	1299.1	1323.6	1347.9	h	
	t	358.5	408.5	458.5	508.5	558.5	608.5	658.5	t	
150	v	3.01	3.27	3.51	3.75	3.97	4.19	4.41	v	150
	h	1193.4	1223.0	1249.6	1275.1	1300.0	1324.5	1348.8	h	
	t	361.0	411.0	461.0	511.0	561.0	611.0	661.0	t	
155	v	2.92	3.17	3.41	3.63	3.85	4.06	4.28	v	155
	h	1194.0	1223.6	1250.5	1276.0	1300.8	1325.3	1349.7	h	
	t	363.6	413.6	463.6	513.6	563.6	613.6	663.6	t	
160	v	2.83	3.07	3.30	3.53	3.74	3.95	4.15	v	160
	h	1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6	h	
	t	366.0	416.0	466.0	516.0	566.0	616.0	666.0	t	
165	v	2.75	2.99	3.21	3.43	3.64	3.84	4.04	v	165
	h	1195.0	1225.2	1252.0	1277.6	1302.5	1327.1	1351.5	h	
	t	368.5	418.5	468.5	518.5	568.5	618.5	668.5	t	
170	v	2.68	2.91	3.12	3.34	3.54	3.73	3.92	v	170
	h	1195.4	1225.9	1252.8	1278.4	1303.3	1327.9	1352.3	h	

t = Temperature, deg. fahr.

v = Specific volume, in cubic feet, per pound.

h = Total heat from water at 32 degrees, B.t.u.

TABLE 168. — *Continued.*

Pressure, Pounds Absolute.		Saturated Steam.	Degrees of Superheat.							Pressure, Pounds Absolute.
			50	100	150	200	250	300		
	t	370.8	420.8	470.8	520.8	570.8	620.8	670.8	t	
175	v	2.60	2.83	3.04	3.24	3.44	3.63	3.82	v	175
	h	1195.9	1226.6	1253.6	1279.1	1304.1	1328.7	1353.2	h	
	t	373.1	423.1	473.1	523.1	573.1	623.1	673.1	t	
180	v	2.53	2.75	2.96	3.16	3.35	3.54	3.72	v	180
	h	1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9	h	
	t	375.4	425.4	475.4	525.4	575.4	625.4	675.4	t	
185	v	2.47	2.68	2.89	3.08	3.27	3.45	3.63	v	185
	h	1196.8	1227.9	1255.0	1280.6	1305.6	1330.2	1354.7	h	
	t	377.6	427.6	477.6	527.6	577.6	627.6	677.6	t	
190	v	2.41	2.62	2.81	3.00	3.19	3.37	3.55	v	190
	h	1197.3	1228.6	1255.7	1281.3	1306.3	1330.9	1355.5	h	
	t	379.8	429.8	479.8	529.8	579.8	629.8	679.8	t	
195	v	2.35	2.55	2.75	2.93	3.11	3.29	3.46	v	195
	h	1197.7	1229.2	1256.4	1282.0	1307.0	1331.6	1356.2	h	
	t	381.9	431.9	481.9	531.9	581.9	631.9	681.9	t	
200	v	2.29	2.49	2.68	2.86	3.04	3.21	3.38	v	200
	h	1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0	h	
	t	384.0	434.0	484.0	534.0	584.0	634.0	684.0	t	
205	v	2.24	2.44	2.62	2.80	2.97	3.14	3.30	v	205
	h	1198.5	1230.4	1257.7	1283.3	1308.3	1333.0	1357.7	h	
	t	386.0	436.0	486.0	536.0	586.0	636.0	686.0	t	
210	v	2.19	2.38	2.56	2.74	2.91	3.07	3.23	v	210
	h	1198.8	1231.0	1258.4	1284.0	1309.0	1333.7	1358.4	h	
	t	388.0	438.0	488.0	538.0	588.0	638.0	688.0	t	
215	v	2.14	2.33	2.51	2.68	2.84	3.00	3.16	v	215
	h	1199.2	1231.6	1259.0	1284.6	1309.7	1334.4	1359.1	h	
	t	389.9	439.9	489.9	539.9	589.9	639.9	689.9	t	
220	v	2.09	2.28	2.45	2.62	2.78	2.94	3.10	v	220
	h	1199.6	1232.2	1259.6	1285.2	1310.3	1335.1	1359.8	h	
	t	391.9	441.9	491.9	541.9	591.9	641.9	691.9	t	
225	v	2.05	2.23	2.40	2.57	2.72	2.88	3.03	v	225
	h	1199.9	1232.7	1260.2	1285.9	1310.9	1335.7	1360.3	h	
	t	393.8	443.8	493.8	543.8	593.8	643.8	693.8	t	
230	v	2.00	2.18	2.35	2.51	2.67	2.82	2.97	v	230
	h	1200.2	1233.2	1260.7	1286.5	1311.6	1336.3	1361.0	h	
	t	395.6	445.6	495.6	545.6	595.6	645.6	695.6	t	
235	v	1.96	2.14	2.30	2.46	2.62	2.77	2.91	v	235
	h	1200.6	1233.8	1261.4	1287.1	1312.2	1337.0	1361.7	h	
	t	397.4	447.4	497.4	547.4	597.4	647.4	697.4	t	
240	v	1.92	2.09	2.26	2.42	2.57	2.71	2.85	v	240
	h	1200.9	1234.3	1261.9	1287.6	1312.8	1337.6	1362.3	h	
	t	399.3	449.3	499.3	549.3	599.3	649.3	699.3	t	
245	v	1.89	2.05	2.22	2.37	2.52	2.66	2.80	v	245
	h	1201.2	1234.8	1262.5	1288.2	1313.3	1338.2	1362.9	h	
	t	401.0	451.0	501.0	551.0	601.0	651.0	701.0	t	
250	v	1.85	2.02	2.17	2.33	2.47	2.61	2.75	v	250
	h	1201.5	1235.4	1263.0	1288.8	1313.9	1338.8	1363.5	h	
	t	402.8	452.8	502.8	552.8	602.8	652.8	702.8	t	
255	v	1.81	1.98	2.14	2.28	2.43	2.56	2.70	v	255
	h	1201.8	1235.9	1263.6	1289.3	1314.5	1339.3	1364.1	h	

t = Temperature, deg. fahr.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees, B.t.u.

CHAPTER XXIII — SUPPLEMENTARY

ELEMENTARY THERMODYNAMICS — CHANGE OF STATE

452. General. — The laws governing the transformation of steam from one state to another form the basis of practically all thermodynamic analyses of the steam engine and turbine. The more common and important changes are

(1) Isobaric or equal pressure.
(2) Isovolumic or equal volume.
(3) Isothermal or equal temperature.
(4) Constant heat content.
(5) Adiabatic or no external heat exchange.
(6) Polytropic.

453. Isobaric or Equal Pressure Change. *Saturated Vapor.* — Since the temperature of wet or saturated steam is dependent on the pressures only, a constant pressure change of such material must also be a constant temperature one. Denoting the initial and final properties by subscripts 1 and 2 respectively:

$$\text{Initial volume } v_1 = x_1 s_1 + (1 - x_1)\,\sigma_1 = x_1 u_1 + \sigma_1. \tag{344}$$

$$\text{Final volume } v_2 = x_2 s_1 + (1 - x_2)\,\sigma_1 = x_2 u_1 + \sigma_1. \tag{345}$$

$$\text{Change of volume } v_2 - v_1 = u_1\,(x_2 - x_1). \tag{346}$$

$$\text{External work } W = P_1\,(v_2 - v_1) = P_1 u_1\,(x_2 - x_1). \tag{347}$$

$$\text{Change of energy} = \frac{\rho_1}{A}\,(x_2 - x_1). \tag{348}$$

$$\text{Heat absorbed} = r_1\,(x_2 - x_1). \tag{349}$$

Notations:

$A = \frac{1}{778}$. p = lb. per sq. in. abs.
P = lb. per sq. ft. abs. x = quality of wet steam.
s = specific volume of dry steam, lb. per cu. ft.
v = specific volume of vapor, lb. per cu. ft.
σ = specific volume of water, lb. per cu. ft.
u = increase in volume during evaporation, cu. ft.
t = deg. fahr. above zero. T = deg. fahr. abs.
c_m = mean specific heat of water.
C = mean specific heat of superheated steam.
H = heat content above 32 deg. fahr., B.t.u. per lb.
λ = total heat of dry steam, B.t.u. per lb.
r = latent heat of vaporization, B.t.u. per lb.
ρ = internal latent heat, B.t.u. per lb.
q = heat of liquid, B.t.u. per lb.
θ = entropy of the liquid.
n = entropy of the vapor.
N = total entropy.

Prime marks indicate superheat.
Subscripts *1*, *2*, *w*, *s* indicate, respectively, initial condition, final condition, wet steam, and superheated steam.

Example 77. At a pressure of 115 lb. per sq. in. absolute the volume of one pound of vapor and liquid is increased 1 cu. ft. Required the change of quality, external work, increase of energy and heat absorbed.

From steam tables $s_1 = 3.88$; $\sigma_1 = 0.0179$; $\rho_1 = 797.9$; $r_1 = 879.8$.

$$\text{Change of quality} = x_2 - x_1 = \frac{v_2 - v_1}{s_1 - \sigma_1} = \frac{1}{3.88 - 0.0179} = 0.259.$$

$$\text{External work} = P_1 (v_2 - v_1) = 144 \times 115 \times 1 = 16{,}560 \text{ ft. lb.}$$

$$\text{Change of energy} = \frac{\rho}{A}(x_2 - x_1) = 797.9 \times 778 \times 0.259$$
$$= 160{,}778 \text{ ft. lb.}$$

$$\text{Heat absorbed} = r_1 (x_2 - x_1) = 879.8 \times 0.259 = 227.79 \text{ B.t.u.}$$

Superheated Steam. — Let superheated steam change state at constant pressure p_1 from an initial temperature t_1 to a final temperature t_2.

Change of volume $= v_2' - v_1'$. The values of v' corresponding to pressure p_1 and temperatures t_1 and t_2 may be taken directly from steam tables or they may be calculated from equation (308). They may be approximated from equations (311) and (312).

$$\text{External work} = P_1 (v_2' - v_1'). \tag{350}$$

$$\text{Change of energy} = \left(\frac{H_2'}{A} - P_1 v_2'\right) - \left(\frac{H_1'}{A} - P_1 v_1'\right). \tag{351}$$

$$= \frac{H_2' - H_1'}{A} - P_1(v_2' - v_1'). \tag{352}$$

$$\text{Heat absorbed} = H_2' - H_1'. \tag{353}$$

$$\text{Change of entropy} = N_2' - N_1'. \tag{354}$$

Example 78. Using the data in the preceding example determine the various quantities, if the initial degree of superheat is 100 deg. fahr.

From superheated steam tables for $p_1 = 115$ and $t_1 = 438.1$ ($= 338.1 + 100$) we find: $v_1' = 4.51$; $H_1' = 1243.1$; $N_1' = 1.6549$.

For $p_2 = p_1 = 115$ and $v_2' = (4.51 + 1) = 5.51$ we find by interpolation $H_2' = 1328.5$; $N_2' = 1.7419$; $t_2' = 621.3$.

$$\text{Increase of superheat} = t_2' - t_1'$$
$$= 621.3 - 438.1 = 183.2 \text{ deg. fahr.}$$

$$\text{External work} = P_1 (v_2' - v_1')$$
$$= 144 \times 115 \times 1 = 16{,}560 \text{ ft. lb.}$$

$$\text{Increase of energy} = \frac{H_2' - H_1'}{A} - P_1 (v_2' - v_1')$$
$$= (1328.5 - 1243.1)778 - 16{,}560$$
$$= 49{,}881 \text{ ft. lb.}$$

$$\text{Increase of entropy} = N_1' - N_2'$$
$$= 1.7419 - 1.6549 = 0.087.$$

$$\text{Heat absorbed} = H_2' - H_1'$$
$$= 1328.5 - 1243.1 = 85.4 \text{ B.t.u.}$$

454. Isovolumic or Equal Volume Change. *Saturated Steam.* — Since the volumes s_1 and s_2 are equal

$$s_1 = s_2 \text{ or } x_1u_1 + \sigma_1 = x_2u_2 + \sigma_2. \tag{355}$$

$$\text{External work} = 0. \tag{356}$$

$$\text{Heat absorbed} = x_1\rho_1 + q_1 - (x_2\rho_2 + q_2). \tag{357}$$

Example 79. A pound of mixture of vapor and liquid at 115 lb. per sq. in. absolute and quality 0.9 is cooled at constant volume to a pressure of 1 lb. per sq. in. absolute. Required the various properties at the final condition and the heat taken from the mixture.

From steam tables:

$$p_1 = 115, s_1 = 3.88, \sigma_1 = 0.0179,$$
$$\rho_1 = 797.9, q_1 = 309, n_1 = 1.103, \theta_1 = 0.4877,$$
$$p_2 = 1, s_2 = 333, \sigma_2 = 0.0161, \rho_2 = 972.9,$$
$$q_2 = 69.8, n_2 = 1.8427, \theta_2 = 0.1327,$$

$$\begin{aligned}
\text{Final quality } x_2 &= \frac{x_1u_1 + \sigma_1 - \sigma_2}{u_2} \\
&= \frac{0.9(3.88 - 0.0179) + 0.0179 - 0.0161}{333 - 0.0161} \\
&= 0.0105. \\
\text{Heat removed} &= x_1\rho_1 + q_1 - (x_2\rho_2 + q_2) \\
&= 0.9 \times 797.9 + 309 - (0.0105 \times 972.9 + 69.8) \\
&= 947 \text{ B.t.u.} \\
\text{Initial entropy } N_1 &= x_1n_1 + \theta_1 \\
&= 0.9 \times 1.103 + 0.4877 = 1.4804. \\
\text{Final entropy } N_2 &= x_2n_2 + \theta_2 \\
&= 0.0105 \times 1.8427 + 0.1327 = 0.1520.
\end{aligned}$$

Superheated Steam. — Since the final volume is equal to the initial, and both pressures and the initial temperature are known, the final temperature may be calculated from equation (308) or it may be taken directly or interpolated from the steam tables.

Example 80. Using the data in the preceding problem determine the various factors if the initial degree of superheat is 100 deg. fahr.

From steam tables for $p_1 = 115$ and $t_1 = 338.1 + 100 = 438.1$ we find: $v_1' = 4.51, H_1' = 1243.1, N_1' = 1.6549$.

s for 1 lb. per sq. in. absolute pressure = 333 cu. ft. but the given volume is 4.51 cu. ft. Therefore the steam is wet at the final condition.

From steam tables for $p_2 = 1$ we find:

$$\rho_2 = 972.9, q_2 = 69.8, n_2 = 1.8427, \theta_2 = 0.1327.$$

Since the volumes are equal

$$\begin{aligned}
v_1' &= v_2 \\
&= x_2u_2 + \sigma_2. \\
\text{Final quality } x_2 &= \frac{v_1' - \sigma_2}{u_2} \\
&= \frac{4.51 - 0.0161}{333 - 0.0161} = 0.0135.
\end{aligned}$$

$$\text{Heat removed} = H_1' - AP_1v_1' - (x_2\rho_2 + q_2)$$
$$= 1243.1 - \frac{144 \times 115}{778} \times 4.51 - (0.0135 \times 972.9 + 69.8)$$
$$= 1065 \text{ B.t.u.}$$

Initial entropy (from steam tables) $N_1' = 1.6549$.

Final entropy $N_2 = x_2n_2 + \theta_2$
$= 0.0135 \times 1.8427 + 0.1327 = 0.1575.$

455. Isothermal or Equal Temperature Change. *Saturated Vapor.* — Since the temperature of wet or saturated steam is dependent solely upon the pressure, an isothermal change is also isobaric, and the data in paragraph (458) is applicable to this change.

Superheated Steam. — The properties at initial and final conditions may be calculated from equations of the properties of superheated steam or they may be taken directly from steam tables or charts. If wet or saturated steam expands isothermally into the superheated state the pressure must drop in order to maintain constant temperature. The relation between pressure, volume, and temperature for the superheated state is given in equation (308).

Example 81. One pound of steam at initial pressure 115 lb. per sq. in. absolute and superheat 100 deg. fahr. is expanded isothermally to a pressure of 1 lb. per sq. in. absolute. Required the various properties at the final pressure, the heat absorbed during expansion and the external work done.

From superheated steam tables for $p_1 = 115$ and $t_1' = 338.1 + 100 = 438.1$ we find: $v_1' = 4.51$, $H_1' = 1243.1$, $N_1' = 1.6549$.

For $p_2 = 1$ and $t_2' = 438.1$, $v_2' = 535$, $H_2' = 1258.3$, $N_2' = 2.1888$.

Final quality $t_2' - t_2 = 438.1 - 101.8 = 336.3$ deg. superheat.

$$\text{Heat added during expansion} = T_2'(N_2' - N_1').$$
$$= 898\,(2.1888 - 1.6541)$$
$$= 1378 \text{ B.t.u.}$$

(Note that the heat added is not equal to the difference in total heats since the isothermal is not a constant pressure line.)

$$\text{External work} = \int_1^2 P\,dv. \tag{358}$$

Since the temperature is constant dv may be obtained by differentiating equation (308). Substituting this value of dv in equation (358) and integrating we have,

$$\text{External work} = 85.63 \log_e T_s \frac{p_1}{p_2} + 2.46\left(p_1^{\frac{3}{2}} - p_2^{\frac{3}{2}}\right)\frac{C}{T^4} \tag{359}$$
$$= 85.63 \log_e 898 \frac{115}{1} + 2.46\,(115^{\frac{3}{2}} - 1^{\frac{3}{2}})\,\frac{C}{897.7^4}$$
$$= 368{,}000 \text{ ft. lb. (approx.)}$$
$$(\log C = 10.8250.)$$

456. Constant Heat Content. — Expansion from one pressure to a lower one with constant heat content is exemplified in throttling or wire drawing. The energy utilized in imparting velocity to the fluid is all returned to the fluid at the lower pressure when the velocity is brought to zero and there are no radiation losses.

For steam wet throughout expansion

$$x_1 r_1 + q_1 = x_2 r_2 + q_2. \tag{360}$$

For steam initially wet but dry at the lower pressure

$$x_1 r_1 + q_1 = \lambda_2. \tag{361}$$

For steam initially wet but superheated at the lower pressure

$$x_1 r_1 + q_1 = \lambda_2 + C_m t_2' = H_2'. \tag{362}$$

For steam initially dry

$$\lambda_1 = \lambda_2 + C_m t_2' = H_2'. \tag{363}$$

For steam initially superheated

$$H_1' = H_2'. \tag{364}$$

Loss of available energy due to throttling or wire drawing

$$\text{Loss B.t.u. per lb.} = T_2 (N_2 - N_1). \tag{365}$$

Example 82. One pound of steam at an initial pressure of 115 lb per sq. in. absolute is expanded through a throttling calorimeter to a pressure of 16 lb. per sq. in. absolute. If the temperature of the steam at the lower pressure is 256.3 deg. fahr. required the initial quality of the steam.

From saturated steam tables:

$$p_1 = 115, r_1 = 879.8, q_1 = 309, N_1 = 1.5907.$$

From superheated steam tables for $p_2 = 16$ and $t_2' = 256.3$ we find:

$$H_2 = 1170.8, N_2 = 1.7765, t_2 \text{ (sat.)} = 216.3,$$
$$x_1 r_1 + q_1 = H_2,$$
$$879.8 x_1 + 309 = 1170.8, \qquad x_1 = 0.98.$$

Mollier diagram analysis, Fig. 622. From intersection of constant superheat line $t_2' = 40$ (= 256.3 − 216.3) and constant pressure line $p_2 = 16$ trace horizontally to constant pressure line $p_1 = 115$ and read from its intersection with the constant quality line, $x_1 = 0.98$.

$$\begin{aligned} \text{Decrease of available energy} &= T_2 (N_2 - N_1) \\ &= (216.3 + 460)(1.7765 - 1.5907) \\ &= 125.6 \text{ B.t.u.} \end{aligned} \tag{366}$$

457. Adiabatic Change of State. — Since in an adiabatic change there is no heat added to or abstracted from the fluid the entropy remains constant.

Steam wet throughout change of state

$$N_1 = N_2. \tag{366a}$$

$$x_1 n_1 + \theta_1 = x_2 n_2 + \theta_2. \tag{367}$$

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2. \tag{368}$$

For water only $x = 0$; for dry steam $x = 1$.
Steam initially superheated but finally wet

$$N_1' = N_2. \tag{369}$$

$$N_1 + n_s = x_2 n_2 + \theta_2. \tag{370}$$

Steam superheated throughout change of state

$$N_1' = N_2', \tag{371}$$

$$N_1 + n_s = N_2 + n_{s(2)}, \tag{372}$$

$$\frac{r_1}{T_1} + \theta_1 + C_m \log_e \frac{T_s}{T_v} = \frac{r_2}{T_2} + \theta_2 + \left[C_m \log_e \frac{T_s}{T_v}\right]_2 + \theta_2. \tag{373}$$

Final Quality. Saturated Steam. — This quantity may be calculated directly from equations (366a) and (367).

$$x_2 = \frac{N_1 - \theta_2}{n_2} \tag{374}$$

$$= \left(\frac{x_1 r_1}{T_1} + \theta_1 - \theta_2\right)\frac{r_2}{T_2}. \tag{375}$$

If water only is present at the beginning of expansion substitute $N_1 = \theta_1$ in equation (374).

For initial qualities of $x_1 = 0.50$ (approx.) or greater the final quality x_2 decreases as the expansion progresses, and for initial qualities of $x_1 = 0.50$ (approx.) or less the final quality increases. For initial quality $x_1 = 0.50$ the final quality x_2 remains practically constant.

The final volume may be calculated as follows:

Wet steam, $v_2 = x_2 u_2 + \sigma_2,$ (376)

x_2 as calculated from equations (367) and (370),

Dry steam, $v_2 = s_2.$ (377)

Superheated Steam. — For superheat at the end of expansion the calculations involved in equation (373) are too cumbersome and unwieldy and the Mollier diagram may be used to advantage.

Volume Change. — Superheated steam: the final volume v_2' may be calculated from equation (3) by substituting for p the final pressure, and for T_s the final temperature as calculated from equation (373). The final volume, however, may be taken directly from the pressure-entropy chart.

External Work. — Since the heat added or subtracted is zero, the external work is equal to the change of intrinsic energy, or in general

$$W = \frac{1}{A}\left[(H_1 - AP_1v_1) - (H_2 - AP_2v_2)\right]. \qquad (378)$$

Steam initially wet

$$W = \frac{1}{A}\left[(x_1\rho_1 + q_1) - (x_2\rho_2 + q_2)\right]. \qquad (379)$$

Steam initially dry, substitute $x_1 = 1$.

Steam initially superheated but wet at end of expansion

$$W = \frac{1}{A}\left[(H_1' - AP_1v_1') - (x_2\rho_2 + q_2)\right]. \qquad (380)$$

Steam initially superheated but dry at end of expansion substitute $x_2 = 1$.

Steam superheated throughout expansion

$$W = \frac{1}{A}\left[(H_1' - AP_1v_1') - (H_2' - AP_2v_2')\right]. \qquad (381)$$

Heat Absorbed $= H_1 - H_2$.

Steam initially wet

$$H_1 - H_2 = (x_1r_1 + q_1) - (x_2r_2 + q_2). \qquad (382)$$

x_2 as calculated from equation (374).

Steam initially dry, substitute $x_1 = 1$.

Steam initially superheated but wet at end of expansion

$$H_1' - H_2 = H_1' - (x_2r_2 + q_2). \qquad (383)$$

Steam superheated throughout expansion, heat absorbed =

$$H_1' - H_2'. \qquad (384)$$

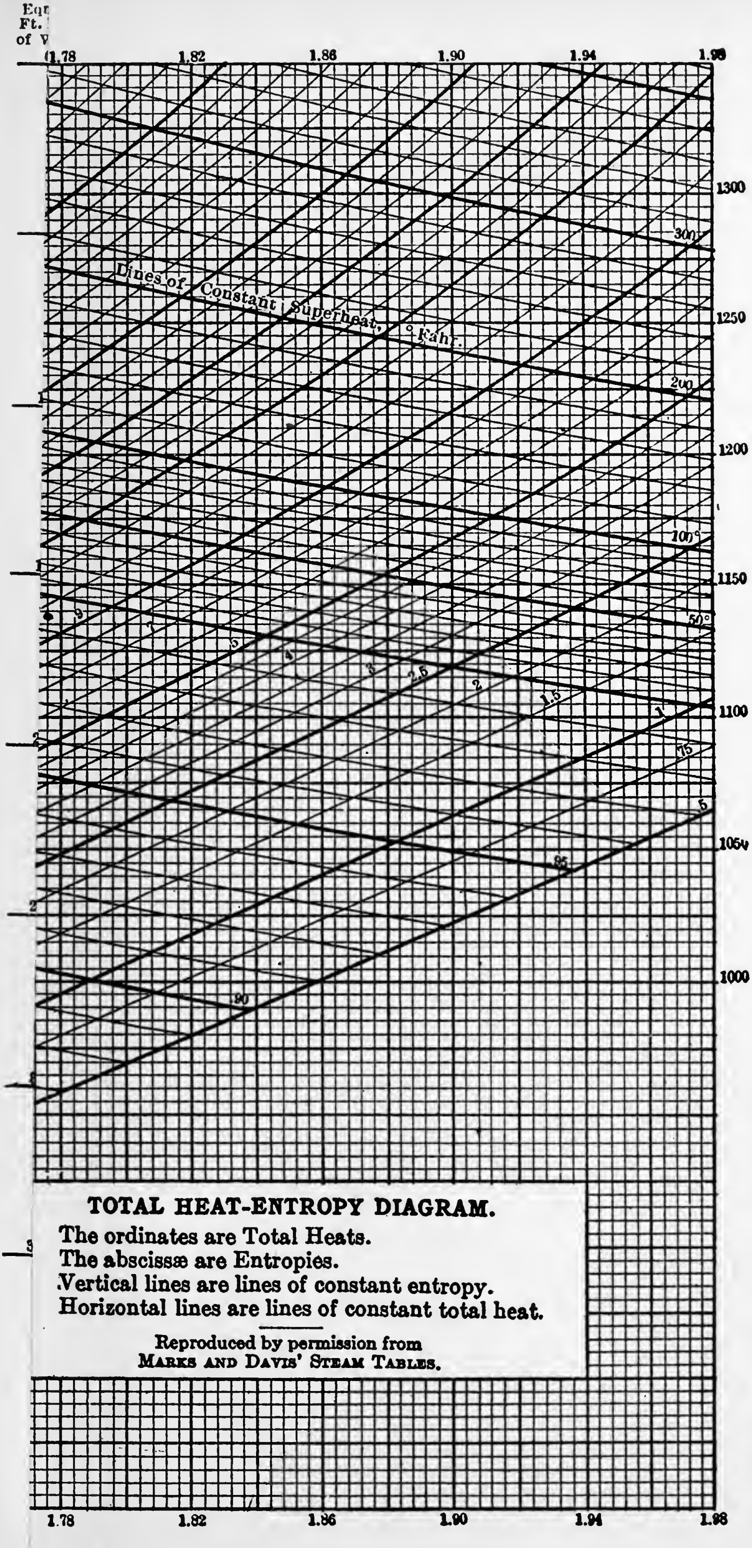
Example 83. One pound of steam at initial pressure 115 pounds per square inch absolute and superheat 100 deg. fahr. expands adiabatically to 1 pound per square inch absolute. Required the various quantities at the final condition.

From superheated steam tables for $p_1 = 115$ and $t_1' = 438.1 = (338.1 + 100)$ we find: $H_1' = 1243$, $v_1' = 4.51$, $N_1' = 1.6549$.

From saturated steam tables: $p_2 = 1$, $s = 333$, $q_2 = 69.8$, $H_2 = 1104.4$, $r_2 = 1034.6$, $\rho_2 = 972.9$, $n_2 = 1.8427$, $\theta_2 = 0.1327$, $\sigma_2 = 0.016$.

Final quality: $$x_2 = \frac{N_1' - \theta_2}{n_2}$$

$$= \frac{1.6549 - 0.1327}{1.8427} = 0.826.$$



TOTAL HEAT-ENTROPY DIAGRAM.

The ordinates are Total Heats.
The abscissæ are Entropies.
Vertical lines are lines of constant entropy.
Horizontal lines are lines of constant total heat.

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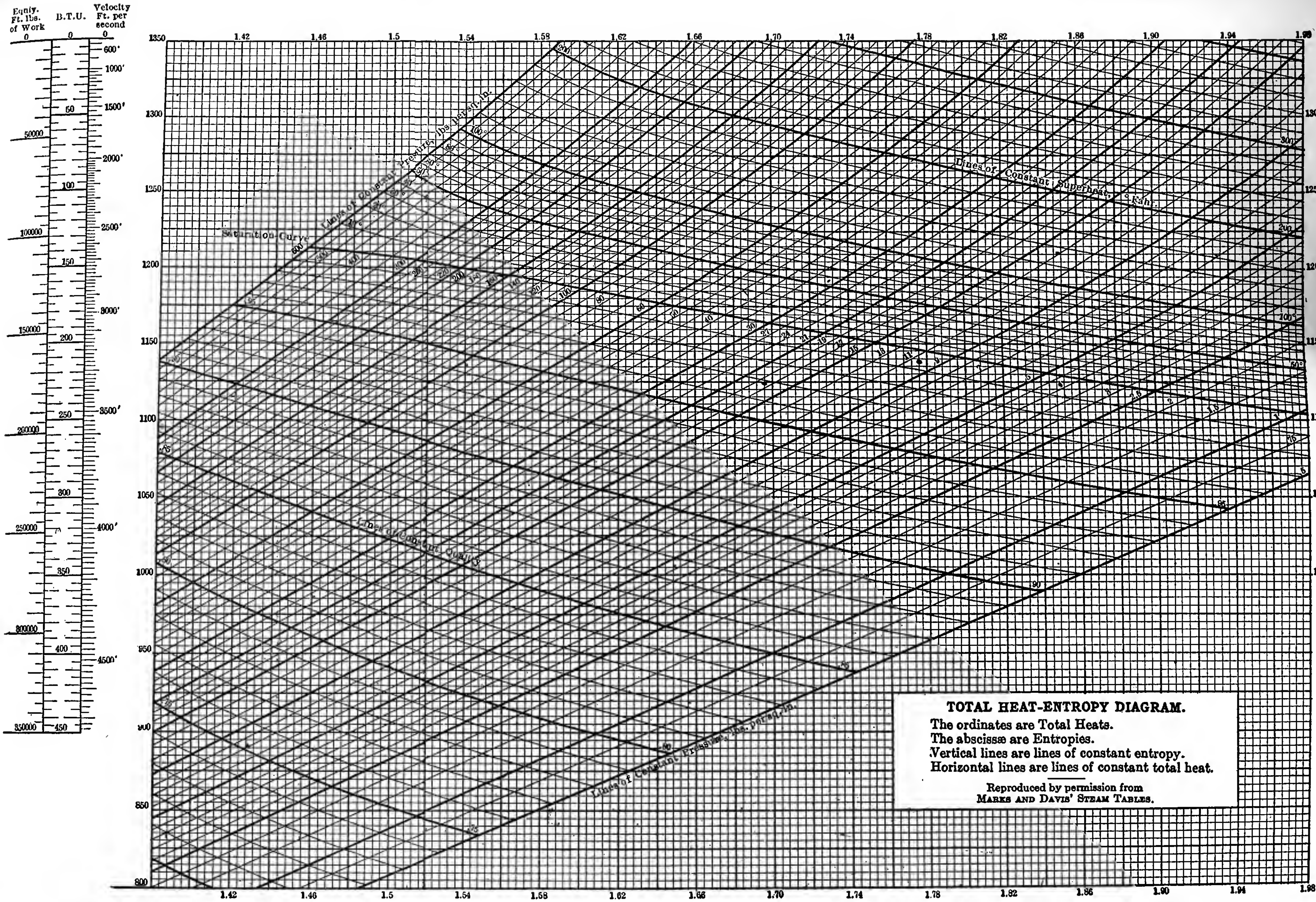
Equiv. Ft. lbs. of Work
B.T.U.
Velocity Ft. per second
Saturation Curve
Lines of Constant Pressure, lbs. per sq. in.
Lines of Constant Superheat, °Fahr.
Lines of Constant Quality
Lines of Constant Pressure, lbs. per sq. in.
TOTAL HEAT-ENTROPY DIAGRAM.
The ordinates are Total Heats.
The abscissæ are Entropies.
Vertical lines are lines of constant entropy.
Horizontal lines are lines of constant total heat.
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FIG. 622.

Mollier diagram analysis, Fig. 622: Trace the intersection of $p_1 = 115$ and $t_1' = 438.1$ vertically downward (constant entropy) to the line $p_2 = 1$ and read 0.826 at the intersection of this line with the constant quality line (interpolated in this case).

Final volume:
$$\begin{aligned} v_2 &= x_2u_2 + \sigma_2 \\ &= 0.826 \times 333 + 0.016 \\ &= 275 \text{ cubic feet.} \end{aligned}$$

(This quantity may be taken directly from the total heat pressure diagram.)

External work: $W = \frac{1}{A}[(H_1' - AP_1v_1') - (x_2\rho_2 + q_2)],$

$$= 778\left[\left(1243.1 - \frac{144 \times 115}{778}4.51\right) - (0.826 \times 972.9 + 69.8)\right]$$

$= 213{,}938$ foot pounds.

Heat absorbed from the fluid

$$\begin{aligned} &= H_1 - (x_2r_2 + q_2) \\ &= 1243.1 - (0.826 \times 1034.6 + 69.8) = 318.8 \text{ B.t.u.} \end{aligned}$$

Mollier diagram, Fig. 622: Project the intersection of $p_1 = 115$ and $t_1' = 438.1$ upon the Y axis and read $H_1' = 1243$. Similarly the projection of the intersection of $p_2 = 1$ and $x_2 = 0.826$ gives $H_2 = 924.3$, $H_1' - H_2 = 1243 - 924.3 = 318.7$ B.t.u.

458. Polytropic Change of State. — A general law for the expansion of any vapor (wet, dry, or superheated) is

$$pv^n = \text{constant}, \tag{385}$$

$$p_1v_1^n = p_2v_2^n, \tag{386}$$

$$v_2 = v_1\left(\frac{p_2}{p_2}\right)^{\frac{1}{n}}. \tag{387}$$

By giving n special values we are able to obtain the various changes of state for constant volume, constant pressure, isothermal and adiabatic.

The work done by expansion for all values of n, except $n = 1$, may be expressed

$$W = \int_1^2 P\,dv^n \tag{388}$$

$$= \frac{P_1v_1 - P_2v_2}{n - 1}. \tag{389}$$

For $n = 1$,

$$W = \int_1^2 P\,dv \tag{390}$$

$$= P_1v_1 \log_e \frac{v_2}{v_1}. \tag{391}$$

Saturated Steam. — Since with wet or saturated steam there can be no change of pressure without a change of temperature the value of n will vary with every change of state and for this reason the use of equations (385) and (388) are more troublesome than the preceding thermal analysis. An exception is that of "saturated expansion" in which steam remains saturated throughout change of state. A study of the actual volume occupied by a pound of dry steam at various pressures will show that n has an approximately constant value of 1.0646 or,

$$p_1 u_1^{1.0646} = \text{constant}, \tag{392}$$

$u = s - \sigma$. (Except for high pressures the influence of σ is negligible and $u = s$ may be safely assumed.)

This condition of constant saturation during expansion seldom occurs in steam engine practice but equation (392) offers the only simple solution of problems involving work done by such a change of state.

Example 84. One pound of steam at an initial pressure of 115 pounds per square inch absolute expands to a pressure of 2 pounds absolute and maintains a saturated condition throughout expansion. Required the final volume and the work done during expansion.

From equations (386) and (392)

$$u_2 = u_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= (3.88 - 0.018) \left(\frac{115}{2}\right)^{\frac{1}{1.0646}}$$

$$= 173.6 \text{ cubic feet.}$$

This value checks with that obtained from steam tables.

Work done $$W = \frac{P_1 u_1 - P_2 u_2}{n - 1}$$

$$= \frac{144\ (115 \times 3.862 - 2 \times 173.6)}{1.0646 - 1} = 216{,}000.$$

Wet Steam. Actual Expansion. — The values of n for the expansion and compression curves of indicator diagrams from actual engines are subject to a wide variation. A study of several types and sizes of engines by J. Paul Clayton* gave values of n varying from 0.7 for wet steam to 1.34 for highly superheated steam. The *average* value of n is, however, not far from 1. That $n = 1$ for isothermal gas expansion and the average actual steam cylinder expansion is a mere coincidence and does not signify that the expansion in the latter is isothermal. See Conventional Diagram, par. 464.

* University of Illinois Bulletin, Vol. 9, No. 26, 1915.

Example. — One pound of saturated steam at an initial pressure of 115 pounds per square inch absolute expands so that its volume has been increased 5 times. Required the work done during expansion.*

$$\begin{aligned} W &= P_1 v_1 \log_e \frac{v_2}{v_1} \\ &= 144 \times 115 \times 3.88 \log_e \tfrac{5}{1}, \\ &= 103{,}200 \text{ foot pounds.} \end{aligned}$$

Wet Steam. Adiabatic Expansion. — The ease with which problems involving adiabatic expansion of vapor or moderately superheated steam can be solved by exact thermal analysis precludes the use of the more troublesome polytropic expansion law. A number of attempts have been made to derive laws which will give the value of n for adiabatic expansion of saturated or wet steam but their accuracy is limited to a comparatively narrow range of pressures and quality. A rule formulated by H. E. Stone † and often used in this connection is:

$$n = 1.059 - 0.000315\, p + (0.0706 + 0.000376\, p)\, x. \qquad (393)$$

Example 85. One pound of steam expands adiabatically from an initial pressure of 115 pounds per square inch and quality 0.9 to a pressure of 1 pound absolute. Required the final volume and the work done during expansion by exact thermal methods and by the polytropic law using equation (393) for determining the value of n.

From steam tables:

$p_1 = 115,\ q_1 = 309,\ \rho_1 = 797.9,\ \theta_1 = 0.4877,\ n_1 = 1.103,\ v_1 = 3.880,$
$p_2 = 1,\ q_2 = 69.8,\ \rho_2 = 972.9,\ \theta_2 = 0.1327,\ n_2 = 1.9754,\ v_2 = 333.$

Exact thermal methods:

$$\begin{aligned} x_1 &= \frac{x_1 u_1 + \theta_1 - \theta_2}{n_1} \\ &= \frac{0.9 \times 1.103 + 0.4877 - 0.1327}{1.8427} \\ &= 0.785. \\ v_2 &= x_2 u_2 + \sigma_2 \\ &= 0.785\,(333 - 0.016) + 0.016 \\ &= 261.4 \text{ cubic feet.} \\ W &= \frac{1}{A}\,[(x_1\rho_1 + q_1) - (x_2\rho_2 + q_2)] \\ &= 778\,[(0.9 \times 797.9 + 309) - (0.785 \times 972.9 + 69.8)] \\ &= 149{,}843 \text{ foot pounds.} \end{aligned}$$

Polytropic law:

$$\begin{aligned} n &= 1.059 - 0.000315 \times 115 + (0.0706 + 0.000376 \times 115)\, 0.9 \\ &= 1.125. \\ v_1 &= x_1 u_1 + \sigma_1 = 0.9 \times (3.88 - 0.016) + 0.016 \\ &= 3.5 \text{ cubic feet.} \end{aligned}$$

$$\begin{aligned} p_1 v_1^{\,n} &= p_2 v_2^{\,n}. \\ 115 \times 3.5^{1.125} &= 1 \times v_2^{1.125}, \\ v_2 &= 235.6 \text{ cubic feet.} \end{aligned}$$

* Assuming $n = 1$.

† University of Illinois Bulletin, Vol. 9, No. 26, p. 79.

$$W = \frac{P_1v_1 - P_2v_2}{n - 1}$$
$$= \frac{144\ (115 \times 3.5 - 1 \times 236.5)}{1.125 - 1}$$
$$= 181{,}232 \text{ foot pounds.}$$

The value of n which will give the same work during expansion according to the polytropic law as the exact thermal analysis for the conditions specified in the problem may be determined as follows:

$$W = \frac{P_1v_1 - P_2v_2}{n - 1},$$
$$149{,}843 = \frac{144\ (115 \times 3.5 - 1 \times 261.4)}{n - 1},$$
$$n = 1.135.$$

This value of n is an *average* only since the true value varies at different points along the expansion line. This may be shown by plotting the true adiabatic expansion line on logarithmic cross-section paper. See par. 465.

Superheated Steam. Isothermal Expansion. — For steam so highly superheated that it does not approach the wet state at any point during the change of state, $n = 1$, and the exponential law offers the only simple solution for the work done during expansion. This case has been treated in par. 455.

Superheated Steam. Adiabatic Expansion. — The work done during adiabatic expansion may be approximated from the polytropic law by making $n = 1.3$. Goodenough gives the following as more accurate than the simple law pv^n = constant.

$$p\ (v' + 0.088)^{1.31} = \text{constant.} \tag{394}$$

Example 86. Steam at 60 pounds per square inch absolute pressure and initially superheated to 300 deg. fahr. expands to a pressure of 15 pounds absolute. Required the final volume and work done according to the polytropic law.

From superheated steam tables for $p_1 = 60$ and superheat of 300 deg. fahr.

$$v_1' = 10.41,$$
$$60\ (10.41 + 0.088)^{1.31} = 15\ (v_2' + 0.088)^{1.31},$$
$$v_2' = 30.2.$$

Thermal analysis gives $v_2' = 30$.

$$W = \frac{P_1\ (v_1' + 0.088) + P_2\ (v_2' + 0.088)}{n - 1}$$
$$= \frac{144\ (60 \times 10.5 + 15 \times 30.1)}{1.31 - 1}$$
$$= 83{,}000 \text{ approx.}$$

Thermal analysis gives $W = 78{,}800$.

CHAPTER XXIV. — SUPPLEMENTARY

ELEMENTARY THERMODYNAMICS OF THE STEAM ENGINE

459. General. — The recent marked improvement in the heat economy of piston engine is largely due to a better understanding of the thermodynamic principles involved in its operation. Once constructed no amount of attention or mechanical adjustment will appreciably affect the economy since the heat efficiency is primarily a function of the design. It is not the object of this chapter to analyze the various thermodynamic laws underlying the design and operation of the piston engine but rather to show their application to the existing types of steam prime movers. In developing an engine with a view of bettering the performance a knowledge is necessary of the theoretical limitations of the particular type under consideration. With this limit as a guide the degree of perfection of the actual mechanism is readily ascertained by comparing test results with those theoretically obtainable. Complete conversion of the heat supplied into useful work is impossible for even the perfect or ideal engine, hence some other standard than the heat supplied is desirable for comparison. There are several ideal cycles which simulate to a certain extent the action of steam in the real engine. The more important of these will be treated in detail.

460. Carnot Cycle. — The Carnot cycle gives the highest possible efficiency for any type of heat and it would seem to be the most desirable cycle for the steam engine, but, as will be shown later, there

Notations:

$A = \frac{1}{778}$. p = lb. per sq. in. abs.
P = lb. per sq. ft. abs. x = quality of wet steam.
s = specific volume of dry steam, lb. per cu. ft.
v = specific volume of vapor, lb. per cu. ft.
σ = specific volume of water, lb. per cu. ft.
u = increase in volume during evaporation, cu. ft.
t = deg. fahr. above zero. T = deg. fahr. abs.
c_m = mean specific heat of water.
C = mean specific heat of superheated steam.
H = heat content above 32 deg. fahr., B.t.u. per lb.
λ = total heat of dry steam, B.t.u. per lb.
r = latent heat of vaporization, B.t.u. per lb.
ρ = internal latent heat, B.t.u. per lb.
q = heat of liquid, B.t.u. per lb.
θ = entropy of the liquid.
n = entropy of the vapor.
N = total entropy.
Prime marks indicate superheat.
Subscripts *1*, *2*, *w*, *s* indicate, respectively, initial condition, final condition, wet steam, and superheated steam.

are practical limitations which more than offset the thermodynamic advantage. Nevertheless a study of this cycle is of importance in showing the absolute degree of perfection which can be realized theoretically.

The diagram in Fig. 623 represents the pressure-volume action or indicator card of an ideal steam engine cylinder operating in the Carnot cycle. For simplicity assume the cylinder to be one square foot in area, to contain unit weight of water and to have a piston displacement equivalent to one pound of saturated steam at the existing back pressure. At the beginning of the stroke *0* the nonconducting cylinder contains water at temperature T_1 corresponding to pressure P_1. Heat is added to the liquid until vaporization is complete, the movement of the frictionless piston being such that the pressure and therefore the temperature is constant, that is, expansion from *0* to *1* is *isothermal.* The source of heat is now removed and the piston is forced from *1* to *2* by the expansion of the steam. Since the cylinder is nonconducting and there is no reception or rejection of heat the expansion from *1* to *2* is *adiabatic.* From *2* to *3* heat is abstracted from the steam at such a rate that the temperature and hence the pressure remain constant, that is, the steam is compressed *isothermally.* At *3* the heat abstraction is terminated and the mixture of vapor and liquid is compressed *adiabatically* to the initial temperature and pressure T_1. The location of point *3* is such that water only at temperature T_1 will be present at the end of compression. This assumption that there is only water at *0* and saturated steam at *1* is not necessary and any degree of wetness or superheat may be assumed since it in no way affects the efficiency.

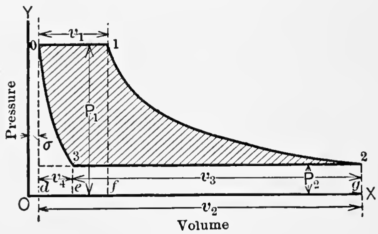

FIG. 623. Indicator Card for Perfect Engine Operating in the Carnot Cycle.

The net work per cycle is represented by the shaded area *0123.*

$$\text{Area } 0123 = \text{area } 01fd + \text{area } 12gf - \text{area } 32ge - \text{area } d03e \qquad (395)$$

$$\text{Area } 01fd = P_1v_1 = P_1(s_1 - \sigma_1) = P_1u_1. \quad \text{See equation (347).}$$

Since no heat is added during expansion from *1* to *2* the internal work is equal to the difference in intrinsic energy. See equation (379), hence:

$$\text{Area } 12gf = [(\rho_1 + q_1) - (x_2\rho_2 + q_2)]\,\frac{1}{A}. \qquad (396)$$

$$\text{Area } 32ge = P_2v_3 - P_2v_2 = P_2v_4. \qquad (397)$$

But $\qquad v_2 = x_2u_2 + \sigma_2$ (see equation (398))

and $\qquad v_4 = x_3u_2 + \sigma_2.$

Substituting these values in equation (397)

$$\begin{aligned}\text{Area } 32ge &= P_2x_2u_2 - P_2x_3u_2 \\ &= P_2u_2\,(x_2 - x_3).\end{aligned}$$

Since no heat is added during compression from *3* to *0* and there is only liquid at *0* the external work done on the steam is equal to the increase in intrinsic energy, or

$$\text{Area } d03e = [q_1 - (x_3\rho_2 + q_2)]\,\frac{1}{A}.$$

All of these factors with the exception of x_2 and x_3 may be obtained directly from the steam tables. x_2 and x_3 may be calculated from equation (374) or they may be taken directly from the temperature-entropy diagram.

From the above data the *PV* diagram or indicator card may be readily plotted to scale. In order to obtain the true contour of the expansion and compression lines several intermediate points should be calculated and located on the diagram.

The area *0123* when correctly drawn should check with the calculated work. Substituting the values of the different areas in equation (395) we have

$$\begin{aligned}\text{Net work per cycle} &= P_1u_1 + [(\rho_1 + q_1) - (x_2\rho_2 + q_2)]\,\frac{1}{A} - P_2u_2\,(x_2 - x_3) \\ &\quad - [q_1 - (x_3\rho_2 + q_2)]\,\frac{1}{A} \\ &= P_1u_1 + \frac{\rho_1}{A} - x_2\left(P_2u_2 + \frac{\rho_2}{A}\right) + x_3\left(P_2u_2 + \frac{\rho_2}{A}\right). \qquad (398)\end{aligned}$$

Heat absorbed in doing work

$$\begin{aligned}&= AP_1u_1 + \rho_1 - x_2\,(AP_2u_2 + \rho_2) + x_3\,(AP_2u_2 + \rho_2), \\ &= AP_1u_1 + \rho_1 - (x_2 - x_3)\,(AP_2u_2 + \rho_2). \qquad (399)\end{aligned}$$

From equation (325) $AP_1u_1 + \rho_1 = r_1$ and $AP_2u_2 + \rho_2 = r_2$.
Therefore heat absorbed

$$= r_1 - r_2\,(x_2 - x_3). \qquad (400)$$

The water rate or steam consumption per hp-hr. of the ideal engine working in this cycle is

$$W = \frac{\text{Heat equivalent of 1 hp-hr.}}{\text{Heat absorbed per lb. of fluid}} \qquad (401)$$

$$= \frac{2546}{r_1 - r_2\,(x_2 - x_3)}. \qquad (402)$$

Efficiency:

$$E = \frac{\text{Heat absorbed}}{\text{Heat supplied}} \tag{403}$$

$$= \frac{r_1 - r_2 (x_2 - x_3)}{r_1}. \tag{404}$$

But $$r_2 (x_2 - x_3) = \frac{T_2}{T_1} r_1, \text{ see equation (368).} \tag{405}$$

Therefore $$E = \frac{r_1 - \frac{T_1}{T_2} r_1}{r_1} = \frac{T_1 - T_2}{T_1}, \tag{406}$$

which is independent of the nature of the working substance and dependent only on the range of temperature.

The shaded area *0123*, Fig. 624, represents the indicator card of Fig. 623 plotted in the temperature-entropy diagram in which ordinates are absolute temperatures and abscissas increase of entropy. This diagram is useful in visualizing the thermal changes per stroke or cycle. Line *ww* represents the increase of entropy of the liquid above 32 deg. fahr. and *ss* the increase of entropy of the vapor. Both of these lines are readily constructed by plotting several values of θ and N as abscissas for corresponding values of T as ordinates. These quantities may be taken directly from steam tables. *0–1* therefore represents the isothermal expansion of the fluid from water at temperature T_1 to dry steam at the same temperature. Since the entropy is constant for adiabatic expansion *1–2* represents the expansion of the saturated fluid from temperature T_1 to temperature T_2. Similarly *2–3* represents isothermal compression at temperature T_2 and *3–0* adiabatic compression from temperature T_2 to the initial condition. If the various lines are drawn to scale

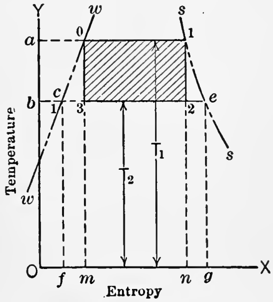

FIG. 624. Temperature-entropy Diagram; Perfect Engine, Carnot Cycle.

Heat supplied above 32 deg. fahr. = area *m01n*.

$$\text{Area } m01n = 0\text{–}1 \times T_1 = n_1 T_1 = r_1.$$

Heat rejected above 32 deg. fahr. = area *m32n*.

$$\text{Area } m32n = 3\text{–}2 \times T_2 = n_1 T_2.$$

$$\begin{aligned}\text{Heat absorbed} &= \text{area } 0123 = \text{area } m01n - \text{area } m32n \\ &= r_1 - n_1 T_2. \\ &= r_1 - r_2 (x_2 - x_3).\end{aligned}$$

Quality at end of expansion $x_2 = \frac{c2}{ce} = \frac{a0 + 01 - bc}{ce} = \frac{n_1 + \theta_1 - \theta_2}{n_2}.$

Quality at beginning of compression $x_3 = \frac{c3}{ce} = \frac{a0 - bc}{ce} = \frac{\theta_1 - \theta_2}{n_2}.$

For any degree of wetness at the beginning and end of isothermal expansion the point *0* will lie to the right of the intersection of *ww* and T_1, and the point *1* will lie to the left of the intersection of *ss* and T_1. The figure *0123*, however, will always be a rectangle.

If isothermal application of heat is continued during admission until the fluid is superheated the point *1* will still lie on the line *a01* but to the right of the vapor line *ss*. In order to maintain a constant temperature of T_1 in the superheated zone, the pressure must be lowered according to the law expressed by equation (308). Since superheat is supplied in practice with gradually increasing temperature and not isothermally the Carnot cycle is not a satisfactory standard for comparing engines using superheated steam and hence this case will not be considered.

Example 87. Determine the heat absorbed, water rate and efficiency of a perfect engine working in the Carnot cycle if the cylinder contains only water at the beginning of the cycle and saturated steam at cut off. Initial pressure 215 lb. per sq. in. absolute; back pressure, 2 lb. absolute. Assume one pound of fluid per cycle.

From steam tables:

$$p_1 = 215, t_1 = 388, s_1 = 2.138, q_1 = 361.4, r_1 = 837.9, \rho_1 = 754,$$
$$\theta_1 = 0.5513, n_1 = 0.9885, \sigma_1 = 0.0185, N_1 = 2.138,$$
$$p_2 = 2, t_2 = 126.15, s_2 = 173.5, q_2 = 94, r_2 = 1021, \rho_2 = 956.7,$$
$$\theta_2 = 0.1749, n_2 = 1.7431, \sigma_2 = 0.0162.$$

Qualities:

$x_0 =$ zero. $\quad x_1 =$ unity.

$$x_2 = \frac{N_1 - \theta_2}{n_2} = \frac{2.138 - 0.1749}{1.74321} = 0.7833. \quad \text{(See equation (374).)}$$

$$x_3 = \frac{\theta_1 - \theta_2}{n_2} = \frac{0.5513 - 0.1749}{1.7421} = 0.216.$$

Specific volumes:

$$v_0 = \sigma_1 = 0.0185.$$
$$v_1 = s_1 - \sigma_1 = 2.138 - 0.0185 = 2.12.$$
$$v_2 = x_2u_2 + \sigma_2 = 0.7833 \times 173.5 = 135.9. \quad \text{(See note, equation (310).)}$$
$$v_3 = v_2 - v_4 = 135.9 - 37.53 = 98.37.$$
$$v_3 = x_3u_2 + \sigma_2 = 0.216 \times 173.5 = 37.53. \quad \text{(See note, equation (310).)}$$

Work:

$$
\begin{aligned}
\text{Admission: } P_1v_1 &= 144 \times 215 \times 2.12 \\
&= 65{,}625 \text{ ft. lb.} \\
\text{Expansion} &= \frac{1}{A}\left[(\rho_1 + q_1) - (x_2\rho_2 + q_2)\right] \\
&= 778\left[(754 + 361.4) - (0.7833 \times 956.7 + 94)\right] \\
&= 211{,}616 \text{ ft. lb.} \\
\text{Exhaust: } P_2v_3 &= 144 \times 2 \times 98.37 \\
&= 28{,}350 \text{ ft. lb.} \\
\text{Compression} &= \frac{1}{A}\left[(q_1 - (x_3\rho_2 + q_2)\right] \\
&= 778\left[361.4 - (0.216 \times 956.7 + 94)\right], \\
&= 47{,}302 \text{ ft. lb.} \\
\text{Net work} &= (65{,}635 + 211{,}616) - (28{,}350 + 47{,}302), \\
&= 201{,}599 \text{ ft. lb.}
\end{aligned}
$$

Heat:

Equivalent of work done = 201,599 ÷ 778 = 259.1 B.t.u.
Supplied = r_1 = 837.8 B.t.u.

$$\text{Efficiency:} \quad E_r = \frac{259.1}{837.9} = 0.309 = 30.9 \text{ per cent.}$$

$$\text{Water rate:} \quad W_r = \frac{2546}{259.1} = 9.83 \text{ lb. per hp-hr.}$$

Temperature-Entropy Diagram.

$$
\begin{aligned}
\text{Heat equivalent of work done} &= n_1(T_1 - T_2) = n_1(t_1 - t_2) \\
&= 0.9885\,(388 - 126.15) \\
&= 259.0 \text{ B.t.u.}
\end{aligned}
$$

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1} = \frac{261.85}{848} = 0.309 = 30.9 \text{ per cent.}$$

While it is conceivable to build an engine which will simulate the true Carnot cycle it would be practically impossible to do so without introducing evils which would more than counterbalance the thermodynamic advantage. The compression in the actual engine must not be confused with the adiabatic compression of the Carnot cycle since the cushion steam involved in the operation of the former is but a fraction of the total fed to the cylinder and has but little influence on the thermodynamic action of the engine.

A modification of the Carnot cycle, known as the *regenerative steam-engine cycle* and which has the same efficiency as the former, has been simulated by a special type of Nordberg pumping engine. The engine is quadruple expansion with four cylinders, three receivers and five feed-water heaters in series *a*, *b*, *c*, *d*, and *e*. The feed water is taken from the hot well and passed in succession through the various heaters: *a* receives its heat from the exhaust steam on its passage to the condenser;

b receives its heat from the low-pressure cylinder jacket; and c, d, and e, respectively, from the third, second, and first receivers. Referring to Fig. 625, if $1–c'$ is drawn parallel to the water line ww the area $01c'c$ will equal the area of the Carnot cycle 0123. The Nordberg engine approximates this cycle as indicated by the broken lines. The expansion in the first stage corresponds to $1–a_1$, that in the second to $a_1–a_2$, and so on for each of the other stages. Heat represented by the area below $a_1–a_1'$ is abstracted from the first stage and is used to raise the condition of the water from b_2' to b_1; heat corresponding to the area below a_2a_2' is withdrawn from the second stage and is used to raise the condition of the water from b_3 to b_2; and so on for each stage. Thus heat is abstracted by steps from the expanding steam and is used for progressively heating the feed water. By increasing the number of steps the nearer will the actual cycle approach that of the ideal. The Nordberg compressor, Table 82, attained 73.7 per cent of the efficiency of the Carnot cycle for the same temperature limits and its heat economy has not yet been excelled.

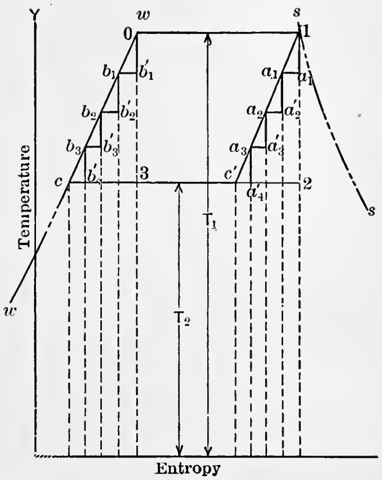

FIG. 625. Regenerative Steam Engine Cycle.

461. Rankine Cycle. Complete Expansion.* — This cycle has been adopted by the American Society of Mechanical Engineers and the British Institution of Civil Engineers as the standard for comparing the performance of all steam prime movers. It is of value not only in comparing the performances of steam engines with each other but also in comparing engines with turbines. In an engine working according to the Rankine cycle, steam is admitted at constant pressure,

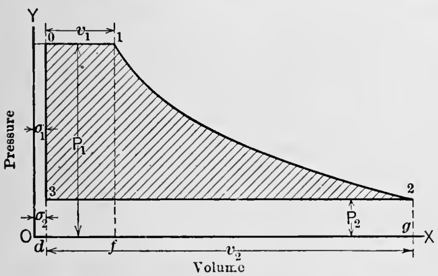

FIG. 626. Indicator Card for Perfect Engine Working in the Rankine Cycle with Complete Expansion.

* This is often called the Clausius cycle since it was published simultaneously but independently by both Clausius and Rankine.

expanded adiabatically to the back pressure and exhausted at that pressure. The engine has no clearance and there are no heat losses from friction, imperfect expansion, or otherwise, all the energy taken from the steam being converted into work. The diagram *0123*, Fig. 62ſ, represents the familiar indicator card or pressure-volume diagram of the working fluid operating in this cycle. *0–1* represents the admission of steam from the boilers at constant pressure P_1; *1–2* is an adiabatic expansion to exhaust pressure P_2; *2–3* exhaust at constant pressure P_2; and *3–0* a practically constant volume pressure rise.

For all conditions of steam:

$$\begin{aligned} \text{Work done during admission} &= \text{area } 01fd \\ \text{Work done during expansion} &= \text{area } 12gf \\ \text{Work done during exhaust} &= \text{area } 32gd \\ \text{Net work} &= \text{area } 01fd - \text{area } 12gf - \text{area } 32gd \\ &= \text{area } 0123 \end{aligned}$$

Per pound of wet or saturated steam:

$$\begin{aligned} \text{Work done during admission} &= P_1 (x_1u_1 + \sigma_1) \text{ ft. lb.} \\ \text{Work done during expansion} &= \frac{1}{A}[(x_1\rho_1 + q_1) - (x_2\rho_2 + q_2)] \text{ ft. lb.} \\ \text{Work done during exhaust} &= P_2 (x_2u_2 + \sigma_2) \text{ ft. lb.} \\ \text{Net work} &= P_1 (x_1u_1 + \sigma_1) + \frac{1}{A}[(x_1\rho_1 + q_1) \\ &\quad - (x_2\rho_2 + q_2)] - P_2 (x_2u_2 + \sigma_2) \text{ ft. lb.} \qquad (407) \\ &= x_1r_1 + q_1 - (x_2r_2 + q_2)^* \text{ B.t.u.} \qquad (408) \\ &= H_1 - H_2 \text{ B.t.u.} \qquad (409) \end{aligned}$$

Per pound of steam superheated at admission but wet or saturated at end of expansion:

$$\begin{aligned} \text{Work done during admission} &= P_1v_1' \text{ ft. lb.} \\ \text{Work done during expansion} &= \left(\frac{1}{A}H_1' - P_1v_1'\right) - \frac{1}{A}(x_2\rho_2 + q_2) \text{ ft. lb.} \\ \text{Work done during exhaust} &= P_2 (x_2u_2 + \sigma_2) \text{ ft. lb.} \\ \text{Net work} &= P_1v_1' - \left[\left(\frac{1}{A}H_1' - P_1v_1'\right) - (x_2\rho_2 + q_2)\right] \\ &\quad - P_2 (x_2u_2 + \sigma_2) \text{ ft. lb.} \\ &= H_1' - (x_2\rho_2 + q_2) - AP_2(x_2u_2 + \sigma_2) \text{ B.t.u.} \\ &= H_1' - (x_2r_2 + q_2)^* \text{ B.t.u.} \qquad (410) \\ &= H_1' - H_2 \text{ B.t.u.} \qquad (411) \end{aligned}$$

* The quantities $P_1\sigma_1$ and $P_2\sigma_2$ found by reducing equation are negligible and have been omitted.

Per pound of steam superheated throughout admission and expansion:

$$\text{Work done during admission} = P_1v_1' \text{ ft. lb.}$$

$$\text{Work done during expansion} = \frac{1}{A}H_1' - P_1v_1' - \left(\frac{1}{A}H_2' - P_2v_2'\right) \text{ft. lb.}$$

$$\text{Work done during exhaust} = P_2v_2' \text{ ft. lb.}$$

$$\text{Net work} = P_1v_1' + \frac{1}{A}(H_1' - H_2') - P_1v_1' + P_2v_2' - P_2v_2' \text{ ft. lb.} \quad (412)$$

$$= H_1' - H_2' \text{ B.t.u.} \quad (413)$$

Calling H_i and H_n the initial and final heat content for all conditions of steam, a general expression for the heat converted into work H_w is

$$H_w = H_i - H_n. \quad (414)$$

Heat supplied H_t above exhaust temperature t is

$$H_t = H_i - q_n. \quad (415)$$

$$\text{Efficiency } E_r = \frac{H_i - H_n}{H_i - q_n}. \quad (416)$$

Steam consumption or water rate, lb. per hp-hr., is

$$W_r - \frac{2546}{H_i - H_n}. \quad (417)$$

The temperature-entropy diagrams for the conditions discussed above are shown in Figs. 627 to 629. For saturated or wet steam it will be

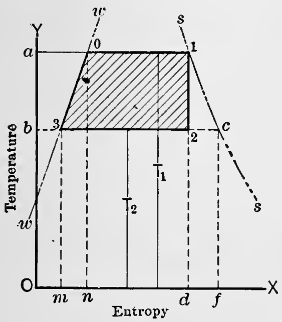

FIG. 627. Temperature-entropy Diagram; Perfect Engine, Rankine Cycle with Complete Expansion. Steam Dry at Cut-off.

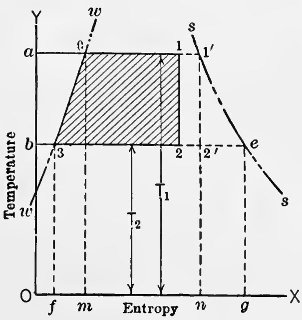

FIG. 628. Temperature-entropy Diagram; Perfect Engine, Rankine Cycle for Wet Steam at Cut-off.

noted that the admission line is an isothermal since a constant pressure expansion for saturated steam is also a constant temperature one. For superheated steam, however, the temperature increases with the degree of superheat, the pressure remaining constant, and the relation between pressure and volume varies according to the law expressed in equation (308), that is, the location of point 1′, Fig. 629, is fixed by determining the entropy corresponding to pressure P_1 and temperature T_1'. This may be calculated from equation (343) or it may be taken directly from superheated steam tables.

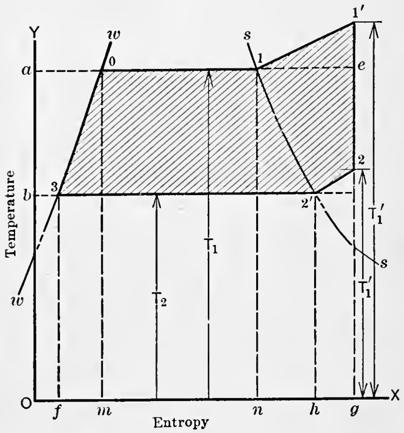

Fig. 629. Temperature-entropy Diagram; Perfect Engine, Rankine Cycle for Steam Superheated throughout Expansion.

A study of equation (416) in connection with the Mollier diagram will show that

(1) The Rankine cycle when using superheated steam has a lower theoretical efficiency than that of the same cycle with saturated vapor having the same maximum temperature.

(2) The theoretical efficiency increases but slightly with the increase in superheat, the maximum pressure remaining constant; see Table 76.

(3) The theoretical efficiency increases rapidly with the increase in pressure range; see Table 71.

The behavior of the actual engine under these conditions is discussed in paragraphs 179 and 182.

A comparison of the Carnot and Rankine cycle shows a lower efficiency for the latter for the same operating conditions, as would be expected. The water rate for the Carnot cycle, however, is higher. This apparent anomaly is due to the fact that the heat supplied per pound of fluid is much larger in the Rankine than in the Carnot. Thus less weight of steam is used per hp-hr., but each pound receives more heat and this is used less efficiently.

Example 88. A perfect engine operating in the Rankine cycle with complete expansion takes steam at 115 lb. per sq. in. absolute pressure, quality 98, and exhausts against a back pressure of 1 lb. absolute. Required the condition of the steam at end of expansion, the work done, efficiency, and water rate.

From steam tables:

$$p_1 = 115,\ t_1 = 338.1,\ r_1 = 879.8,\ q_1 = 309,\ H_1 = 1188.8,\ \theta_1 = 0.4877,\quad n_1 = 1.103,$$

$$p_2 = 1,\quad t_2 = 101.8,\ r_2 = 1034.6,\ q_2 = 69.8,\qquad \theta_2 = 0.1327,\quad n_2 = 1.8427,$$

$$x_2 = \frac{x_1 n_1 + \theta_1 - \theta_2}{n_2} \quad \text{(See equation (374).)}$$
$$= \frac{0.98 \times 1.103 + 0.4877 - 0.1327}{1.8427}$$
$$= 0.779.$$

Heat converted into work

$$= x_1 r_1 + q_1 - (x_2 r_2 + q_2)$$
$$= 0.98 \times 879.8 + 309 - (0.779 \times 1034.6 + 69.8)$$
$$= 1171.2 - 875.7 = 295.5 \text{ B.t.u. per lb.}$$

$$\text{Efficiency} = \frac{H_i - H_2}{H_i - q_2}$$
$$= \frac{295.5}{1171.2 - 69.8} = 0.268 = 26.8 \text{ per cent.}$$

$$\text{Water rate} = \frac{2546}{H_i - H_2}$$
$$= \frac{2546}{295.5} = 8.62 \text{ lb. per hp-hr.}$$

The initial and final heat content may be taken directly from the Mollier diagram; as a matter of fact it is customary in practice to use the diagram except where extreme accuracy is necessary or when the given conditions are beyond the range of the charts.

462. Rankine Cycle with Incomplete Expansion. — If expansion after cut-off is not carried far enough to reduce the pressure to that of the back pressure line as shown in Fig. 630 the Rankine cycle more nearly simulates the cycle of the actual engine. This cutting the "toe" off the diagram decreases the efficiency, but permits of the use of a smaller cylinder. A comparison of the diagram in Fig. 626 with that in Fig. 630 will show that the area *012′3′* is of the same outline as area *0123*, consequently the work done would be that corresponding to complete expansion to pressure P_c plus that represented by area *3′2′23*.

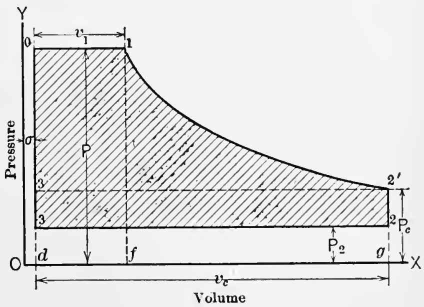

FIG. 630. Indicator Card for Perfect Engine Working in the Rankine Cycle with Incomplete Expansion.

If H_c represents the heat content corresponding to complete expansion, to pressure P_c the heat equivalent of the work done (area *012′3′*) is $H_i - H_c$ B.t.u. per lb.

Work corresponding to area $3'2'23 = (P_c - P_2)\, v_c$ ft. lb. per lb.

Hence, heat converted into work $= H_i - H_c + A\,(P_c - P_2)\, v_c$.

Heat supplied is the same as for complete expansion $= H_i - q_2$.

$$\text{Therefore efficiency } E_r' = \frac{H_i - H_c + A\,(P_c - P_2)\,v_c}{H_i - q_2}. \tag{418}$$

$$\text{Water rate } W = \frac{2546}{H_i - H_c + A\,(P_c - P_2)\,v_c}. \tag{419}$$

For wet steam, $v_c = x_c u_2 + \sigma_2 = x_c s_2$ (for all practical purposes).
For dry steam, $v_c = s_2 - \sigma_2$.
For superheated steam, $v_c = v_2' - \sigma_2$.

The temperature-entropy diagram differs from that for complete expansion in the curtailment of lines *1–3′* and *3′–3* by constant-volume pressure drop *2′–2*, Fig. 631.

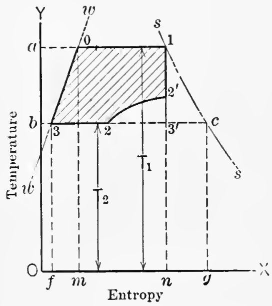

FIG. 631. Temperature-entropy Diagram; Perfect Engine, Rankine Cycle with Incomplete Expansion. Steam Dry at Cut-off.

Example 89. Same data and requirements as in preceding example except that release occurs at a pressure of 4 lb. absolute.

From steam tables: p_1 and p_2 as in preceding example,

$$p_c = 4.\; r_c = 1005.7,\; q_c = 120.9,\; \theta_c = 0.2198,$$
$$n_c = 1.6416,\; s_2 = 90.5,$$

$$x_c = \frac{x_1 n_1 + \theta_1 - \theta_c}{n_c}$$
$$= \frac{0.98 \times 1.103 + 0.4877 - 0.2198}{1.6416}$$
$$= 0.822.$$
$$v_c = x_c s_2 = 0.822 \times 90.5$$
$$= 74.4.$$

$$H_c = x_c r_c + q_c$$
$$= 0.822 \times 1005.7 + 120.9$$
$$= 947.6.$$
$H_i = 1171.2$ (same as in preceding example).

$$\text{Efficiency} = \frac{H_i - H_c + A\,(P_c - P_2)\,v_c}{H_i - q_2}$$
$$= \frac{1171.2 - 947.6 + \frac{144}{778}\,(4 - 1)\,74.4}{1171.2 - 69.8}$$
$$= \frac{1171.2 - 947.6 + 41}{1171.2 - 69.8} = \frac{264.6}{1101.4}$$
$$= 0.24 = 24 \text{ per cent.}$$
$$\text{Water rate} = \frac{2546}{264.6} = 9.62 \text{ lb. per hp-hr.}$$

463. Rankine Cycle with Rectangular PV-Diagram. — This cycle is the least efficient of all vapor cycles in practical use but represents the action of the fluid in direct-acting steam pumps, direct-acting air compressors and engines taking steam full stroke. It may be looked upon as a limiting case of the Rankine cycle. From Fig. 632 it is apparent that

$$\text{Work done} = A\ (P_1 - P_2)\ v \text{ B.t.u.} \qquad (420)$$

For wet steam, $v = x_1u_1 + \sigma_1 = x_1s_1$ (for most purposes).

For dry steam, $v = s_1 - \sigma_1$.

For superhetead steam, $v = v_1' - \sigma_1$.

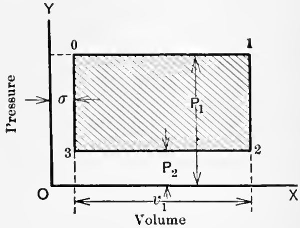

FIG. 632.

Heat received is the same as that in the Rankine cycle

$$= H_i - q_n.$$

$$\text{Efficiency} = \frac{A\ (P_1 - P_2)\ v}{H_i - q_n}. \qquad (421)$$

$$\text{Water rate} = \frac{2546}{A\ (P_1 - P_2)\ v}. \qquad (422)$$

Example 90. A perfect direct-acting steam pump operating in the rectangular PV cycle takes steam at initial pressure 115 lb. per sq. in. absolute, quality 98 per cent and exhaust against a back pressure of 15 lb. absolute. Required the work done per lb. of fluid, efficiency and the water rate.

From steam tables: $p_1 = 115,\ s_1 = 3.88,\ H_1 = 1188.8,$

$p_2 = 15,\ q_n = q_2 = 181.0.$

$$\text{Heat converted into work} = A\ (P_1 - P_2)\ x_1s_1$$
$$= \tfrac{144}{778}\ (115 - 15)\ 0.98 \times 3.88$$
$$= 70.4 \text{ B.t.u.}$$

$$\text{Efficiency} = \frac{70.4}{1188.8 - 180} = 0.07 \text{ approx.} = 7 \text{ per cent.}$$

$$\text{Water rate} = \frac{2546}{70.4} = 36 \text{ lb. per hp-hr.}$$

464. Conventional Diagram.—In designing an engine it is customary to assume as a basis of reference an ideal cycle which considers only the kinetic action of the steam in the cylinder. This permits of analysis without the use of steam tables. The expansion is assumed to be hyperbolic because the equilateral hyperbola is readily constructed and because expansion in the actual engine conforms approximately to the law $Pv^n = C$ (see paragraph 458). According to the 1915 A.S.M.E. Code the ideal engine is assumed to have no clearance and no losses through wire-drawing during admission or release. The initial pressure is that of the boiler and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for

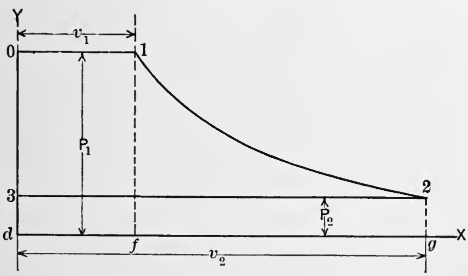

FIG. 633.

a condensing engine. Such a diagram for a simple non-condensing engine is illustrated in Fig. 633. *0–1* represents admission at constant pressure P_1, *1–2* represents hyperbolic expansion from cut-off *1* to release at *2* and *2–3* represents exhaust at atmospheric pressure P_2.

The work done is represented by the

$$\text{area } 0123 = \text{area } 01fd + \text{area } 12gf - \text{area } 32gd,$$

$$\text{area } 01fd = P_1 v_1,$$

$$\text{area } 12gf = P_1 v_1 \log_e \frac{v_2}{v_1} \text{ (see paragraph 458)},$$

$$\text{area } 32gd = P_2 v_2.$$

Therefore net work done

$$W = P_1 v_1 \left(1 + \log_e \frac{v_2}{v_1}\right) - P_2 v_2, \tag{423}$$

letting

$$\frac{v_2}{v_1} = r = \text{ratio of expansion},$$

$$W = P_1 v_1 (1 + \log_e r) - P_2 v_2. \tag{424}$$

$$\text{Mean effective pressure } P_m = \frac{\text{area } 0123}{v_2}$$

$$= \frac{P_1}{r}(1 + \log_e r) - P_2. \tag{425}$$

As the m.e.p. is generally used in pounds per square inch, dividing both members of the equation by 144 gives

$$p_m = \frac{p_1}{r}(1 + \log_e r) - p_2. \tag{426}$$

$$\text{Theoretical maximum horsepower} = \frac{plan}{33{,}000}, \tag{427}$$

in which

l = length of stroke, feet,
a = area of cylinder, sq. in.,
n = number of working strokes.

The ratio of the m.e.p. of the actual engine to that of the ideal diagram as determined above is called the diagram factor. This factor is determined by experiment and ranges as follows (Heat Power Engineering, Hirshfeld and Barnard, 1915, p. 325):

Simple slide-valve engine	55 to 90 per cent
Simple Corliss engine	85 to 90 " "
Compound slide-valve engine	55 to 80 " "
Compound Corliss engine	75 to 85 " "
Triple expansion engine	55 to 70 " "

The probable mean effective pressure for the engine under consideration is

$$\text{M.e.p.} = p_m \times \text{diagram factor.} \tag{428}$$

Example 91. Determine the probable horsepower of a 12 inch × 12 inch simple engine, 250 r.p.m., initial pressure 120 lb. per sq. in. absolute, cut off $\frac{1}{4}$ stroke, diagram factor 0.75.

$$\begin{aligned} \text{Theoretical m.e.p.} &= \tfrac{120}{4}(1 + \log_e 4) - 15, \\ &= 56.53. \\ \text{Probable actual m.e.p} &= 56.53 \times 0.75 = 42.4. \\ \text{Probable i.hp.} &= \frac{42.4 \times 1 \times 113 \times 500}{33{,}000} \\ &= 72.4. \end{aligned}$$

465. Logarithmic Diagram.—It is a well-known fact that the equation of the polytropic curve $Pv^n = C$ becomes a straight line when plotted on logarithmic cross-section paper and the slope of the line is the value of n. Conversely, when the expansion or compression curve of an indicator becomes a straight line in the logarithmic diagram it shows that the change of state is in accordance with the law $Pv^n = C$. The logarithmic diagram derived from the indicator card is useful in analyzing cylinder performance and gives valuable information which cannot be readily obtained otherwise. Thus it has been demonstrated * that the logarithmic diagram is of great assistance in

(1) Approximating clearance volume.
(2) Locating the stroke positions of cyclic events.
(3) Detecting leakage.
(4) Approximating steam consumption.

Construction of the Logarithmic Diagram. — If the clearance volume is given the construction of the diagram is very simple. Draw the clearance line OY and the absolute pressure line OX on the indicator diagram as illustrated in Fig. 634. Locate points *1, 2, 3,* etc. on the expansion line and tabulate the corresponding absolute pressures and volumes. For example, the pressure corresponding to point *1* is P_1 and its value is the length of the line P_1 multiplied by the scale of the indicator spring. Similarly the volume corresponding to point *1* is v_1 and its value is the length of the line v_1 multiplied by constant

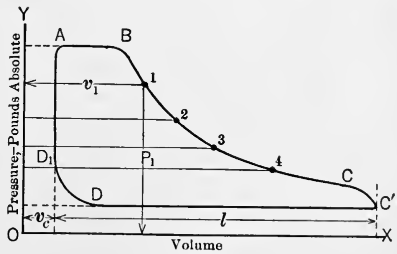

FIG. 634.

* A New Analysis of the Cylinder Performance of Reciprocating Engines. J. Paul Clayton, Univ. of Ill. Bull. No. 26, Vol. 9, May 6, 1912.

m (= piston displacement per stroke in cu. ft. divided by the length of the card l measured in inches). Transfer these points to logarithmic cross-section paper as illustrated in Fig. 635, using absolute pressures in lb. per sq. in. as ordinates and cu. ft. as abscissas. Repeat the operation for the compression curve and draw a smooth line through the various points. The ratio $\frac{ab}{bc}$ (measured in inches) will be the value of n for the expansion line and $\frac{de}{df} = n$ for compression.

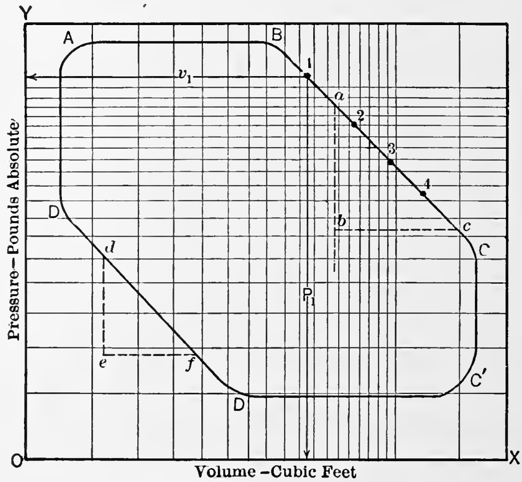

FIG. 635. Indicator Card — Logarithmic Diagram.

Approximating Clearance Volume. — If expansion and compression vary substantially according to the law $Pv^n = C$ the clearance volume may be approximated by trial and error. All that is necessary is to assume different values of clearance and to plot the logarithmic diagram for each assumed value until the expansion or compression curve is a straight line.

Locating the Stroke Position of Cyclic Events. — Except with a few types of four-valve engines it is difficult and oftentimes impossible to locate the points of cut-off, release, and compression from the indicator diagram. If there is no leakage the true points may be located on the logarithmic diagram by noting when the expansion and compression curves become straight; see Fig. 193, Chapter IX.

Detecting Leakage. — The law $Pv^n = C$ is applicable only to cases where the weight of steam remains practically constant during change of state. When the weight changes materially as by leakage, the resulting expansion and compression lines on the logarithmic diagram depart from straight lines. This is clearly shown in Fig. 195.

Approximating Steam Consumption. — According to Clayton (1) there is a definite relation existing between x_c (quality at cut-off) and n in any one cylinder which is practically independent of cut-off position. (2) This relation is practically independent of cylinder size and of engine speed; it is therefore applicable to other cylinders of the same type. (3) By means of the experimentally determined relations of x_c and n, the value of x_c may be approximated from the average value of n obtained from the expansion curves of one set of indicator diagrams taken simultaneously; therefore the actual weight of steam present in one revolution may be approximated. (4) The actual steam consumption may be obtained by this method from the indicator diagram to within an average of 4 per cent of test measurements. These statements apply strictly to non-jacketed steam cylinders in good condition, exhausting at or near atmospheric pressure. In applying this method it is only necessary to determine n as previously outlined and find from the curve in Fig. 194 the corresponding value of x_c. Knowing the quality of steam at cut-off the weight of fluid per stroke can be readily calculated. It will be noted that the curve in Fig. 194 is only an average approximation and that there is a considerable range in the values of x_c for a given value of n. By separating the points into groups of similar pressures and speeds, several lines coördinating n and x_c may be obtained and a greater accuracy is possible. For a complete discussion of this important subject consult Clayton's paper.

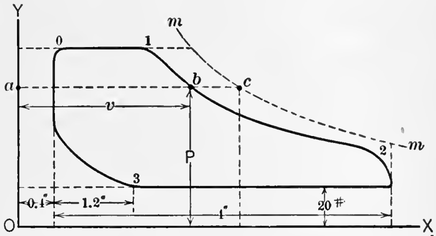

FIG. 636.

466. Temperature-Entropy Diagram. — If the actual indicator card is transferred to the temperature-entropy chart the various heat exchanges during expansion and compression may be seen at a glance. The area represented by the actual diagram, however, does not give the heat utilized in doing work since the weight of steam is not constant throughout the cycle. From cut-off to release the weight is constant if there is no leakage, as is the case from beginning of compression to admission, but the weights involved in each case are not the same. Therefore, only the expansion line shows the true behavior of all the steam used per cycle and the rest of the diagram is more or less conventional. The transfer of the pressure-volume to the temperature-entropy diagram is best illustrated by a specific example.

Example 92. Curve *0123*, Fig. 636, is an average indicator card taken from a 12 × 12 engine running at 300 r.p.m.; clearance volume 10 per cent; steam consumption by tests 2700 lb. per hr. Transfer the indicator card to temperature-entropy chart.

Locate the zero clearance line OY and zero pressure lines OX, and measure the diagram as indicated. The cylinder displacement per stroke $= \dfrac{3.14 \times 12^2}{144 \times 4} = 0.785$ cu. ft.

Compression volume $= 0.785 \left(\dfrac{1.2 + 0.4}{4}\right) = 0.314$ cu. ft.

Weight of "cushion steam" on the assumption that the steam is dry at the beginning of compression

$$= 0.314 \times 0.0498 = 0.0156 \text{ lb.}$$

(0.0498 = wt. of 1 cu. ft. of steam at 20 lb. abs. pressure.)

Weight of steam used per stroke or "cylinder feed"

$$= \frac{2700}{600 \times 60} = 0.075 \text{ lb.}$$

Total weight of steam expanding = 0.075 + 0.0156 = 0.09 lb.

Lay off saturation line *mm*. This line represents the volume of 0.09 lb. of saturated steam for the various pressures within the range of the diagram.

Draw several pressure lines such as *abc* and tabulate the ratio $\dfrac{ab}{ac}$. This ratio gives the quality of the steam at point b in the expansion curve $\left(\dfrac{ab}{ac} = \dfrac{xs}{s} = x\right)$. Note that $\dfrac{ab}{ac}$ represents quality only during expansion after cut-off and that it is simply a ratio for other parts of the cycle.

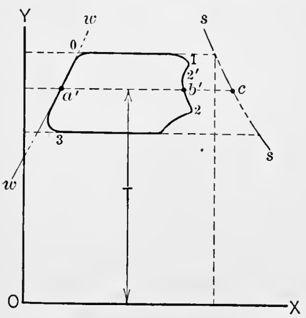

Fig. 637.

Tabulate also the absolute temperature corresponding to the pressure under consideration. Next construct the water and saturation curves *ww* and *ss*, respectively, as illustrated in Fig. 637. This may be done conveniently by using absolute temperatures and entropies of water and vapor given in steam tables, the entropies being multiplied by 0.09. Locate point b' on the corresponding temperature line in such a position that rates $\dfrac{a'b'}{a'c} = \dfrac{ab}{ac}$ obtained from the indicator card. The locus of the point b' will be the desired diagram. The thermal action during actual expansion is apparent from the diagram; thus it will be seen by inspection that the steam is wet at cut-off, that condensation takes place from 1 to 2′ more rapidly than if expansion were adiabatic, and that reëvaporation takes place from b' to 2.

The foregoing analysis applies only to saturated or wet steam. In case of superheat the actual expansion will lie beyond the saturation curve as illustrated in Fig. 638 and the ratio $\frac{ab}{ac} = \frac{v'}{s}$ will not give the quality. To find the temperature corresponding to v' multiply s, the specific volume of one pound of saturated steam at pressure P by the ratio $\frac{ab}{ac}$ as measured from the diagram. From superheated steam tables or by means of equation (311) determine the temperature corresponding to volume $s \times \frac{v'}{s}$ and pressure P. To transfer the point b to the temperature-entropy diagram draw the temperature line T' corresponding to that just determined and locate point b' on this line such that cb' = total entropy for pressure

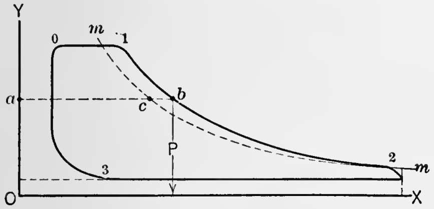

FIG. 638.

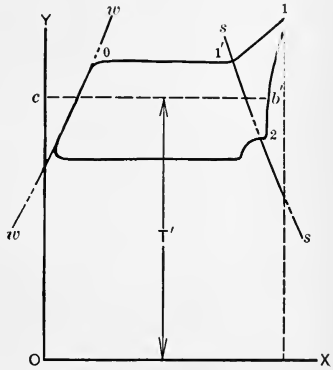

FIG. 639.

P and temperature T'. The total entropy may be taken from superheated steam tables or it may be calculated from equation (342). For the problem under consideration the entropy thus obtained must be multiplied by 0.09, the weight of fluid expanding per cycle. The locus of the point b' will be the desired diagram.

466a. Steam Accounted for by Indicator Diagrams at Points near Cut-off and Release. — The steam accounted for, expressed in pounds per i.hp. per hour, may readily be found by using the equation

$$\frac{13{,}750}{\text{m.e.p.}} [(C + E)\, W_c - (H + E)\, W_h], \tag{429}$$

in which

m.e.p. = mean effective pressure,

C = proportion of direct stroke completed at points on expansion line near cut-off or release,

E = proportion of clearance,

H = proportion of return stroke uncompleted at point on compression line just after exhaust closure,

W_c = weight of 1 cu. ft. steam at pressure shown at cut-off or release point,

W_h = weight of 1 cu. ft. steam at pressure shown at compression point.

The points near cut-off release and compression referred to are indicated in Fig. 640.

In multiple expansion engines the mean effective pressure to be used in the above formula is the aggregate m.e.p. referred to the cylinder under consideration. In a compound engine the aggregate m.e.p. for the h-p. cylinder is the sum of the actual m.e.p. of the h-p. cylinder and that of the l-p. cylinder multiplied by the cylinder ratio. Likewise the aggregate m.e.p. for the l-p. cylinder is the sum of the actual m.e.p. of the l-p. cylinder and the m.e.p. of the h-p. cylinder divided by the cylinder ratio.

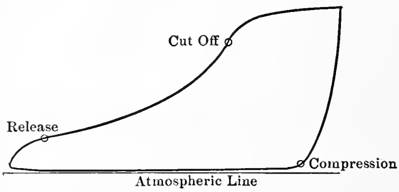

FIG. 640. Points where "Steam Accounted for by Indicator" is Computed.

The relation between the weight of steam shown by the indicator at any point in the expansion line and the weight of the mixture of steam and water in the cylinder may be represented graphically by plotting on the diagram a saturated steam curve showing the total consumption per stroke (including steam retained at compression) and comparing the abscissas of the curve with the abscissas of the expansion line, both measured from the line of no clearance.

CHAPTER XXV. — SUPPLEMENTARY

PROPERTIES OF AIR. — DRY, SATURATED, AND PARTIALLY SATURATED

467. General. — Tables and charts giving the simultaneous physical and thermal properties of dry and saturated air for various temperatures are of great assistance in solving problems relative to the design and performance of evaporative surface condensers, water-cooling apparatus and air-conditioning devices. Table 169 gives the properties of dry and saturated air for various temperatures ranging from 0 to 212 deg. fahr. and Figs. 461 and 462 give a complete psychrometric chart for all conditions of dry, saturated, and partially saturated air within a temperature range of 20 to 350 deg. fahr. These charts are extremely useful in avoiding laborious calculations.

468. Dry Air. — The physical and thermal properties of dry air as used in these tables and charts are based on the following laws established by the latest experiments with gases and vapors:

$$\frac{P_a V_a}{T_a} = \text{constant} = 0.755, \tag{430}$$

$$C_{pa} = 0.2411 + 0.0000045\,(t_1 + t_2), \tag{431}$$

$$H_a = C_{pa}\,(t_2 - t_1), \tag{432}$$

in which

P_a = absolute pressure of the dry air, in. of mercury,
V_a = volume of 1 lb. of dry air, cu. ft.,
T_a = absolute temperature of the air, deg. fahr.,
C_{pa} = mean specific heat of air at constant pressure between temperatures t_1 and t_2,
H_a = heat content, B.t.u. per lb. of air above temperature t_1,
t_1 = initial temperature, deg. fahr.,
t_2 = final temperature, deg. fahr.

A sample calculation of the properties of dry air as listed in Table 169 is given in Example 93.

Example 93. Required the specific volume and density of dry air at 100 deg fahr. under standard atmospheric pressure (= 29.92 in.). Required also the heat content per lb. above 0 deg. fahr.

TABLE 169.

PROPERTIES OF SATURATED AIR. (Barometer 29.921.)

Mixture of Air Saturated with Water Vapor.

Temperature, Degrees Fahr.	Weight of 1000 Cu. Ft. of Dry Air, Pounds.	Volume of One Lb. of Dry Air, Cu. Ft.	Elastic Force of Vapor, In. of Mercury.*	Elastic Force of the Dry Air in the Mixture, In. of Mercury.	Weight of 1000 Cu. Ft., Lb.		
					Weight of the Dry Air, Content.	Weight of the Vapor, Content.*	Total Weight of the Mixture.
1	2	3	4	5	6	7	8
0	86.35	11.58	0.037	29.88	86.23	0.067	86.90
10	84.53	11.83	0.063	29.85	84.31	0.110	84.42
20	82.71	12.09	0.103	29.81	82.44	0.177	82.62
30	81.04	12.34	0.165	29.76	80.62	0.278	80.90
32	80.71	12.39	0.181	29.74	80.24	0.303	80.54
35	80.19	12.47	0.203	29.72	79.70	0.340	80.04
40	79.43	12.59	0.248	29.67	78.77	0.410	79.18
45	78.61	12.72	0.300	29.62	77.86	0.492	78.35
50	77.88	12.84	0.362	29.56	76.94	0.588	77.53
55	77.10	12.97	0.436	29.48	75.98	0.699	76.68
60	76.33	13.10	0.521	29.40	75.05	0.823	75.88
62	76.04	13.15	0.560	29.36	74.66	0.887	75.54
65	75.64	13.22	0.622	29.30	74.08	0.979	75.06
70	74.91	13.35	0.739	29.18	73.08	1.153	74.23
72	74.63	13.40	0.790	29.13	72.68	1.229	73.90
75	74.24	13.48	0.874	29.05	72.08	1.352	73.42
80	73.53	13.60	1.031	28.89	71.01	1.580	72.59
85	72.83	13.73	1.212	28.71	69.92	1.841	71.76
90	72.15	13.86	1.421	28.50	68.78	2.137	70.92
95	71.53	13.98	1.659	28.26	67.59	2.474	70.06
100	70.87	14.11	1.931	27.99	66.34	2.855	69.19
105	70.22	14.24	2.241	27.69	65.05	3.285	68.33
110	69.64	14.36	2.594	27.33	63.64	3.769	67.41
115	69.01	14.49	2.993	26.93	62.16	4.312	66.47
120	68.40	14.62	3.444	26.48	60.60	4.920	65.52
125	67.80	14.75	3.952	25.97	58.92	5.599	64.52
130	67.20	14.88	4.523	25.40	57.14	6.356	63.50
135	66.67	15.00	5.163	24.76	55.23	7.187	62.43
140	66.09	15.13	5.878	24.04	53.18	8.130	61.31
145	65.53	15.26	6.677	23.25	51.01	9.160	60.17
150	64.98	15.39	7.566	22.35	48.63	10.30	58.93
155	64.43	15.52	8.554	21.37	46.12	11.56	57.68
160	63.94	15.64	9.649	20.27	43.39	12.94	56.33
165	63.41	15.77	10.86	19.06	40.47	14.45	54.92
170	62.89	15.90	12.20	17.72	37.33	16.11	53.44
175	62.46	16.03	13.67	16.25	33.96	17.93	51.89
180	61.88	16.16	15.29	14.63	30.34	19.91	50.25
185	61.42	16.28	17.07	12.85	26.44	22.06	48.50
190	60.94	16.41	19.01	10.91	22.26	24.41	46.67
195	60.61	16.50	21.14	8.78	17.17	26.96	44.13
200	59.98	16.67	23.46	6.46	12.97	29.72	42.69
205	59.74	16.74	26.00	3.92	7.82	32.71	40.53
210	59.31	16.86	28.75	1.17	2.30	35.94	38.24
212	59.10	16.92	29.92	0	0	37.32	37.32

* Goodenough.

TABLE 169. — *Continued.*

Temperature, Degrees Fahr.	Weight of Water Necessary to Saturate 100 Lb. of Dry Air.	Volume of One Pound of Dry Air + Vapor to Saturate it, Cubic Feet.	Heat Content per Pound of Dry Air, B.t.u.	Latent Heat of Vapor in One Lb. of Dry Air Saturated with Vapor, B.t.u.	Heat Content of One Lb. of Dry Air Saturated with Vapor, B.t.u.
0	0.078	11.59	0.000	0.964	0.964
10	0.131	11.86	2.411	1.608	4.019
20	0.214	12.13	4.823	2.623	7.446
30	0.344	12.41	7.234	4.195	11.429
32	0.378	12.47	7.716	4.058	11.783
35	0.427	12.55	8.44	4.57	13.02
40	0.520	12.70	9.65	5.56	15.21
45	0.632	12.85	10.86	6.73	17.59
50	0.764	13.00	12.07	8.12	20.19
55	0.920	13.16	13.28	9.76	23.04
60	1.105	13.33	14.48	11.69	26.18
62	1.188	13.40	14.97	12.12	26.84
65	1.323	13.50	15.69	13.96	29.65
70	1.578	13.69	16.90	16.61	33.51
72	1.692	13.76	17.38	17.79	35.17
75	1.877	13.88	18.11	19.71	37.81
80	2.226	14.09	19.32	23.31	42.64
85	2.634	14.31	20.53	27.51	48.04
90	3.109	14.55	21.74	32.39	54.13
95	3.662	14.80	22.95	38.06	61.01
100	4.305	15.08	24.16	44.63	68.79
105	5.05	15.39	25.37	52.26	77.63
110	5.93	15.73	26.58	61.11	87.69
115	6.94	16.10	27.79	71.40	99.10
120	8.13	16.52	29.00	83.37	112.37
125	9.53	16.99	30.21	97.33	127.54
130	11.14	17.53	31.42	113.64	145.06
135	13.05	18.13	32.63	132.71	165.34
140	15.32	18.84	33.85	155.37	189.22
145	18.00	19.64	35.06	182.05	217.10
150	21.22	20.60	36.27	214.03	250.30
155	25.11	21.73	37.48	252.61	290.10
160	29.87	23.09	38.69	299.55	338.20
165	35.77	24.75	39.91	357.75	397.70
170	43.24	26.84	41.12	431.20	472.30
175	52.90	29.51	42.33	526.0	568.30
180	65.77	33.04	43.55	651.9	695.50
185	83.59	37.89	44.76	826.1	870.90
190	109.80	45.00	45.97	....	....
195	191.00	56.20	47.20	....	....
200	229.50	77.24	48.40	....	....
205	419.00	....	49.62	....	....
210	....	....	50.83	....	....
212	....	....	51.39	....	....

From equation (430),

$$\frac{29.92 \times V_a}{100 + 459.6} = 0.755,$$

$$V_a = 14.11 \text{ cu. ft. per lb.}$$

$$\text{Density} = \frac{1}{14.11} = 0.071 \text{ lb. per cu. ft.}$$

From equation (431),

$$C_{pa} = 0.2411 + 0.0000045\,(0 + 100) = 0.2416,$$

and from equation (432),

$$H_a = 0.2416\,(100 - 0) = 24.16 \text{ B.t.u. per lb.}$$

469. Saturated Air. — Water, if placed in a vacuum chamber, will evaporate until the pressure in the chamber has reached that of vapor corresponding to the temperature of the water. If the water is introduced into a chamber containing dry air the evaporation will proceed precisely the same as in the vacuum until the pressure has risen by an amount corresponding to the vapor pressure for the temperature. In this case, according to Dalton's law (paragraph 226) each substance will exert the pressure it would if alone occupying the volume, and the final pressure will be the sum of that of the vapor and that of the air. Air is said to be saturated with moisture when it contains the saturated vapor of water. It might be better to say that the space is saturated since the presence of air has no effect on the vapor (the temperatures being the same) other than that the air retards the diffusion of water particles. Perfectly dry air does not exist in nature since evaporation of water from the earth's surface causes the atmosphere to be more or less diluted with vapor.

The weight of saturated water vapor per cubic foot depends only on the temperature and not on the presence of air.

The various properties for air completely saturated with water vapor may be calculated by means of equations (430) to (432), and Dalton's law which may be expressed

$$P_a + P_v = P, \tag{433}$$

in which

P_a = absolute pressure of the dry air in the mixture, inches of mercury,

P_v = absolute pressure of saturated steam at the temperature of the mixture, in.,

P = total pressure, which for atmospheric conditions = 29.921.

Therefore,

$$P_a = P - P_v. \tag{434}$$

P_v may be taken directly from steam tables.

From equation (430),

$$V = V_a = \frac{0.755\, T_a}{P - P_v}, \tag{435}$$

in which

V_a = volume of 1 lb. of dry air (plus vapor to saturate) at pressure P_a and absolute temperature T_a,

V = volume of vapor in 1 lb. of dry air when saturated, cu. ft.

Evidently

$$w_a = \frac{1}{V_a},$$

in which

w_a = weight of dry air in 1 cu. ft. of saturated mixture.

The weight, w_v, of vapor in 1 cu. ft. of saturated mixture is the density of saturated vapor at pressure P_v and temperature T_a. This may be taken directly from steam tables.

Total weight of mixture per cu. ft. = $w_a + w_v$.

The weight, w_v', of vapor necessary to saturate 1 lb. of dry air,

$$w_v' = Vw_v = V_a w_v. \tag{436}$$

Heat content H', or total heat in a mixture of 1 lb. of dry air saturated with water vapor, measured above 0 deg. fahr., and not including the heat of liquid, is

$$H' = C_{pa}t_a + r_v w_v', \tag{437}$$

in which

t_a = temperature of the mixture, deg. fahr.,

r_v = latent heat of saturated vapor at temperature t_a and pressure P_v.

An application of these formulas to the calculation of the various quantities in Table 169 for a temperature of 100 deg. fahr. is given in Example 94.

Example 94. Required the following properties of atmospheric air completely saturated with water vapor when the temperature of the mixture is 100 deg. fahr.: Elastic force or pressure of the vapor and of the dry air in the mixture, volume of 1 lb. of dry air plus vapor to saturate it, weight of dry air and vapor in 1000 cu. ft. of mixture, weight of water necessary to saturate 100 lb. of dry air, latent heat of the vapor content of 1 lb. of mixture and the heat content of 1 lb. of dry air saturated with vapor.

Pressure of vapor in the mixture:

$$P_v = 1.931 \text{ in. (from steam tables).}$$

Pressure of dry air in the mixture:

$$\begin{aligned} P_a &= P - P_v \\ &= 29.921 - 1.931 = 27.99 \text{ in.} \end{aligned}$$

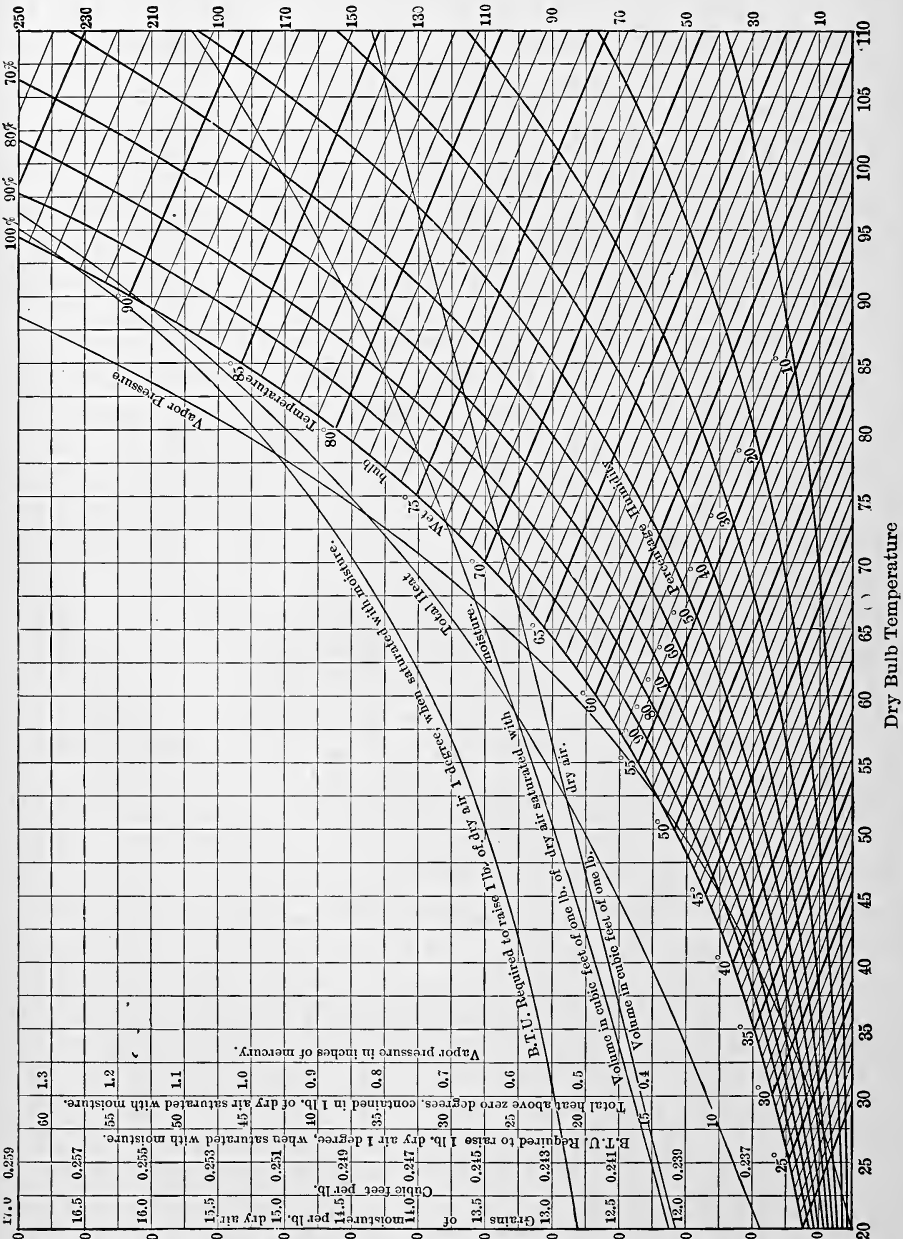

FIG. 641. Psychrometric Chart (W. H. Carrier).

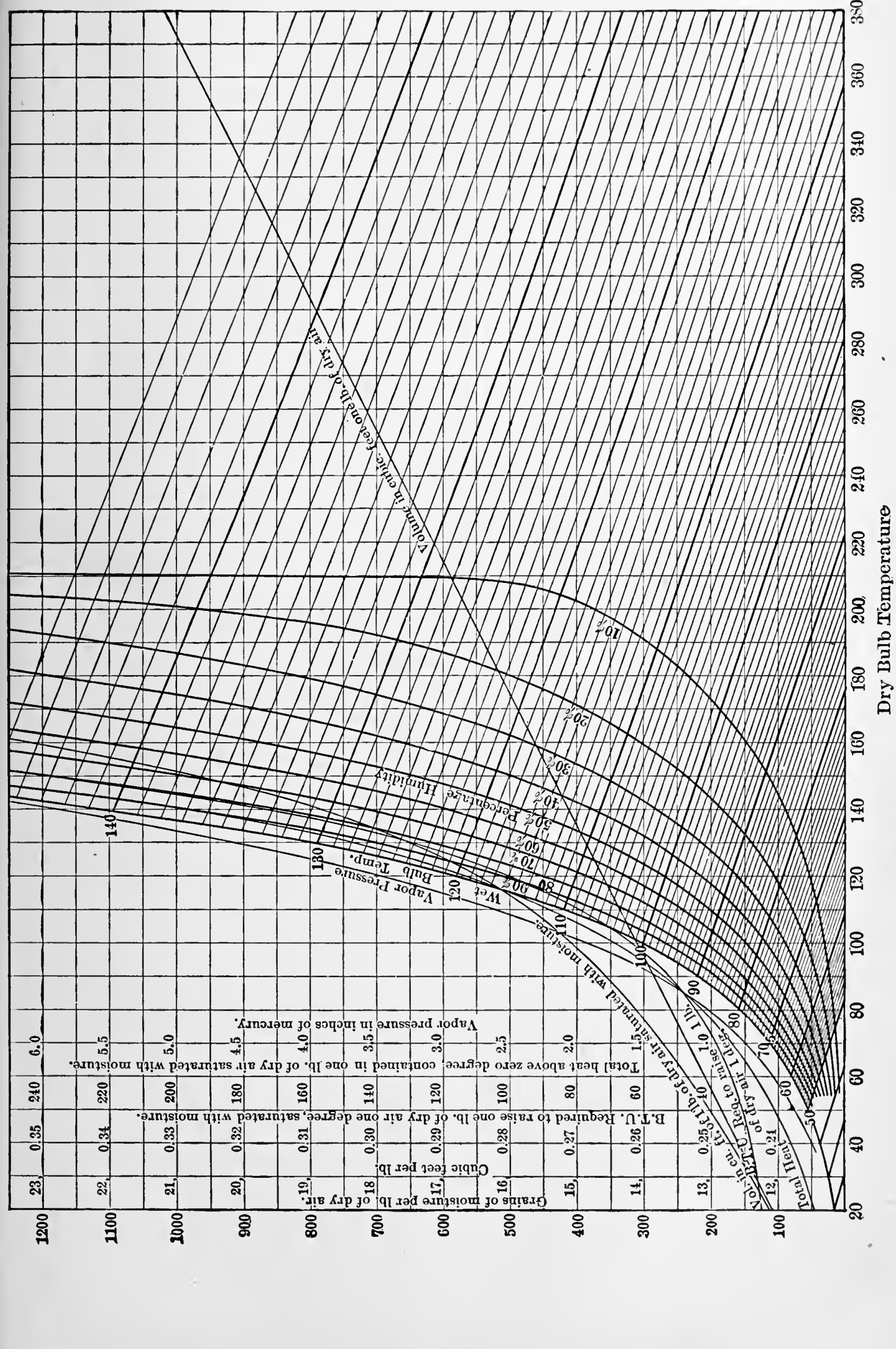

FIG. 642. Psychometric Chart (W. H. Carrier).

Volume of 1 lb. of dry air saturated with vapor:

$$V_a = \frac{0.755\ T_a}{P - P_v}$$
$$= \frac{0.755\ (100 + 459.6)}{29.921 - 1.931} = 15.08 \text{ cu. ft.}$$

Weight of dry air in 1000 cu. ft. of saturated mixture:

$$w_a = \frac{1}{V_a} = \frac{1}{15.08} = 0.06634 \text{ lb. per cu. ft.}$$
$$1000\ w_a = 1000 \times 0.06634 = 66.34 \text{ lb.}$$

Weight of water vapor in 1000 cu. ft. of mixture:

$$w_v = 0.002855 \text{ lb. per cu. ft. (from steam tables),}$$
$$1000\ w_v = 1000 \times 0.002855 = 2.855 \text{ lb.}$$

Total weight of 1000 cu. ft. of mixture:

$$= 66.34 + 2.855 = 69.19+ \text{ lb.}$$

Weight of vapor necessary to saturate 100 lb. of dry air:

$$w_v' = Vw_v = V_a w_v$$
$$= 15.08 \times 0.002855 = 0.04305 \text{ lb. per lb. of dry air}$$
$$= 0.04305 \times 100 = 4.305 \text{ lb. per 100 lb. of dry air.}$$

Total heat of the dry air content, above 0 deg. fahr.:

$$H_a = C_{pa}\,(100 - 0)$$
$$= 0.2416 \times 100 = 24.16 \text{ B.t.u. per lb.}$$

Latent heat of the vapor content:

$$r_v w_v' = 1036.6 \times 0.04305 = 44.63 \text{ B.t.u.}$$

Total heat of 1 lb. of dry air saturated with vapor:

$$H_1 = H_a + r_v w_v',$$
$$= 24.16 + 44.63 = 68.79 \text{ B.t.u.}$$

470. Partially Saturated Air. — As previously stated air is said to be saturated with moisture when it contains the saturated vapor of water. In this condition the weight of vapor per cu. ft. corresponds to the density of saturated steam at the temperature of the mixture. If the body of air contains only a fraction of the weight of vapor corresponding to saturation it is said to be partially saturated and the fraction is called the *relative humidity*. Partially saturated air in reality contains superheated vapor since the temperature of the mixture, which is also that of the vapor is higher than that of saturated vapor corresponding to the actual pressure of the vapor in the mixture. The water vapor in the atmosphere is usually superheated. If a partially saturated mixture of air and water vapor is cooled at constant pressure the mixture tends to become more and more saturated until at a certain temperature called the *dew point,* condensation begins to take place. The pressure of saturated vapor corresponding to the dew point is substantially the same as the partial pressure of the superheated vapor in the original mixture.

The relative humidity or degree of saturation is ordinarily determined by an instrument called the *psychrometer,* which consists of two thermometers suitably mounted, the bulb of one thermometer being covered by a close-fitting wick, which is kept moist, and the other being exposed directly to the air. There are two types in general use, the "stationary," and the "sling." In the former the two thermometers are suitably mounted and hung in the shade, and in the latter they are whirled at a rate of about 200 r.p.m. The sling psychrometer gives more reliable results than the stationary device. The "aspiration" psychrometer is used when more accurate results are required.

If the air is saturated, no evaporation takes place from the wet bulb and the two thermometers read alike, but if it is only partially saturated evaporation occurs and the readings of the wet bulb thermometer are lower than those of the dry. Experiment * has shown (1) that when an isolated body of water is permitted to evaporate freely in the air it assumes the true wet bulb temperature, (2) that the heat content of the air and vapor mixture is a constant for a given wet bulb temperature irrespective of the initial temperature and humidity, and (3) that the heat given up by the water and absorbed by the air-vapor mixture may be expressed

$$r_w(w_w - w) = C_{pa}\,(t_d - t_w) + wC_{ps}\,(t_d - t_w), \qquad (438)$$

in which

r_w = latent heat of vaporization at wet bulb temperature, t_w, B.t.u. per lb.,

w_w = weight of vapor in 1 lb. of dry air when saturated at wet bulb temperature, t_w, lb.,

w = actual weight of vapor contained in 1 lb. of dry air at dry bulb temperature, t_d.

C_{pa} and C_{ps} = mean specific heats, respectively, of the dry air and vapor between temperatures t_w and t_d.

Transposing equation (438) and reducing,

$$w = \frac{r_w w_w - C_{pa}\,(t_d - t_w)}{r_w - C_{ps}\,(t_d - t_w)}. \qquad (439)$$

For low pressures,

$$C_{ps} = 0.42 + 0.00005\,(t_d - t_w).$$

* Willis H. Carrier, Trans. A.S.M.E., Vol. 33, 1911, p. 1014.

At the low pressures under consideration Dalton's law may be assumed to hold good for vapors, thus

$$P' = hP_d, \tag{440}$$

$$h = \frac{P'}{P_d} = \frac{D'}{D_d}, \tag{441}$$

in which

h = relative humidity at temperature t_d,
D' = actual density of the vapor at temperature t_d, lb. per cu. ft.
D_d = density of saturated vapor at temperature t_d, lb. per cu. ft.
$P' = hP_d$ = actual pressure of the vapor in the mixture at temperature t_d,
P_d = pressure of saturated vapor at temperature t_d.

Other notations as previously defined.

By combining equations (438) to (441) and solving for h (omitting a number of negligible factors), Carrier (Trans. A.S.M.E., Vol. 33, 1911, p. 1023) has deduced the following expression:

$$h = \left[P_w - \frac{(P - P_w)\,d}{2800 - 1.3\,t_w}\right]\frac{1}{P_d}, \tag{442}^*$$

in which

P_w = pressure of saturated vapor at wet bulb temperature, in.,
P = barometric pressure, in.,
d = temperature difference between the wet and dry bulb thermometers.

Other notations as previously defined.

Since, according to statement (2), the heat content, H', of 1 lb. of dry air at temperature t_d with relative humidity h is the same as that, H, of 1 lb. of dry air at wet bulb temperature, t_w, when completely saturated, then

$$H' = H = r_w w_w + C_{pa} t_w. \tag{443}$$

For any atmospheric pressure P_1, other than P the relative humidity will be

$$h_1 = \frac{P_1 h}{P}, \tag{444}$$

in which

h_1 = relative humidity at pressure P_1,
h = relative humidity at pressure P.

The values in Figs. 641 and 642 are based on the foregoing analysis. An application of the equations formulas is given in Example 95.

* This expression is for use in connection with the aspiration psychrometer. For the sling psychrometer substitute $(2755 - 1.28\,t_w)$ for $(2800 - 1.3\,t_w)$.

Example 95. Determine the following quantities for partially saturated atmospheric air if the wet and dry bulb temperatures are 80 and 100 deg. respectively: relative humidity, pressure of the vapor in the mixture, pressure of dry air and vapor content of the mixture, weight of 1000 cu. ft. of mixture, actual weight of vapor in 1 lb. of dry air, dew point, and the heat content of the mixture.

Relative humidity:

$$h = \left[1.029 - \frac{(29.921 - 1.029)\ 20}{2800 - 1.3 \times 80}\right]\frac{1}{1.931}$$
$$= 0.42 \text{ or } 42 \text{ per cent.}$$

Vapor pressure in mixture:

$$P' = hP_d = 0.42 \times 1.931 = 0.811 \text{ in.}$$

Dry air pressure in mixture:

$$P_a = P - P' = 29.92 - 0.811 = 29.11 \text{ in.}$$

Volume of 1 lb. of dry air plus vapor content; see equation (435):

$$V = V_a = \frac{0.755\ T_a}{P_a}$$
$$= \frac{0.755\ (100 + 459.6)}{29.11} = 14.5 \text{ cu. ft.}$$

Weight of dry air in 1000 cu. ft. of mixture

$$= 1000\ \frac{1}{V} = \frac{1000}{14.5} = 68.96 \text{ lb.}$$

Weight of water vapor in 1000 cu. ft. of mixture

$$= 1000 \times h \times \text{density of saturated steam at 100 deg. fahr.}$$
$$= 1000 \times 0.42 \times 0.00285 = 1.19 \text{ lb.}$$

Total weight of mixture

$$= 68.96 + 1.19 = 70.15.$$

Weight of vapor in 1 lb. of dry air:
From equation (439)

$$w = \frac{1047.4 \times 0.02226 - 0.2416 \times 20}{1047.4 + 0.429 \times 20} = 0.0175 \text{ lb.}$$

w may also be closely approximated as follows:

$$w = hV \times \text{density of saturated vapor at 100 deg. fahr.}$$
$$= 0.42 \times 14.5 \times 0.002855 = 0.0174 \text{ lb.}$$

Total heat in 1 lb. of dry air containing w lb. of vapor at temperature t_d:
From equation (443)

$$H' = 1047.4 \times 0.02226 + 0.2415 \times 80 = 42.46 \text{ B.t.u.}$$

H' may also be approximated from the values in Table 169.

$$H' = \text{heat content of the dry air} + h \times \text{latent heat content of saturated vapor at temperature } t_d = 100$$
$$= 24.16 + 0.42 \times 44.63 = 42.9 \text{ B.t.u.}$$

An application of Table 169 and the psychrometric charts in Fig. 641 is given in Examples 96 and 97.

Example 96. Atmospheric air at 40 deg. fahr. and relative humidity 0.80 is to be conditioned to 70 deg. fahr. and relative humidity 0.50. Determine the amount of moisture and heat to be added, (1) by means of Table 169 and (2) by means of the curves in Fig. 641.

From Table 169:

Original moisture content = $0.52 \times 0.8 = 0.416$ lb. per 100 lb. of dry air.
Final moisture content = $1.578 \times 0.5 = 0.789$ lb. per 100 lb. of dry air.
Moisture to be added = $0.789 - 0.416 = 0.373$ lb. per 100 lb. of dry air.

From Fig. 641:

Initial moisture content (intersection of $t_d = 40$ and $h = 80$ per cent) = 29 grains per lb. of dry air.
Final moisture content (intersection of $t_d = 70$ and $h = 50$ per cent) = 55 grains per lb. of dry air.

Moisture to be added $= 100\left(\dfrac{55 - 29}{7000}\right) = 0.371$ lb. per 100 lb. of dry air. (7000 = grains per lb.)

From Table 169:

Initial heat content = $9.65 + 0.8 \times 5.56 = 14.1$ B.t.u. per lb.
Final heat content = $16.90 + 0.5 \times 13.96 = 23.88$ B.t.u. per lb.
Heat required = $23.88 - 14.1 = 19.78$ B.t.u. per lb. of dry air.

From Fig. 461:

Initial heat content (intersection of $t_d = 40$ and $h = 80$ per cent) gives wet bulb $t_w = 37.5$; follow constant temperature line $t_w = 37.5$ until it intersects saturation line $h = 100$ per cent; trace vertically upward to intersection of "total heat" line and read from marginal notation 14.1 B.t.u. per lb.

Final heat content (intersection of $t_d = 70$ and $h = 40$ per cent) gives $t_w = 55.8$; follow constant temperature line $t_w = 55.8$ until it intersects line $h = 100$ per cent; trace vertically upward to intersection of "total heat" line and read 23.88.

The charts in Figs. 641 and 642 are reproduced to a greatly reduced scale and the readings cannot be made with the accuracy indicated in the example. In the original charts the wet and dry bulb temperature can be read to an accuracy of 0.1 degree and the other quantities proportionately.

Example 97. Atmospheric air at 90 deg. fahr. and relative humidity of 80 per cent is to be conditioned to 70 deg. fahr. and 50 per cent relative humidity. Determine the temperature to which the original mixture must be reduced in order to have a relative humidity of 50 per cent when heated to 70 deg. fahr. Determine also the amount of heat to be abstracted to effect the initial cooling and that to be supplied to bring it to the final desired condition.

Moisture content at $t_d = 90$ and $h = 0.8 = 3.109 \times 0.8 = 2.487$ lb. per 100 lb. of dry air, corresponding dew point = 83 deg. fahr., that is, at 83 deg. fahr. condensation begins.

Moisture content at $t_d = 70$ and $h = 0.5 = 1.578 \times 0.5 = 0.789$ lb. per 100 lb. of dry air. Corresponding dew point = 51.8 deg. fahr. This is the temperature to which the air must be cooled in order to have the required humidity when reheated to 70 deg. fahr. Heat content at $t_d = 90$ and $h = 0.8 = 21.74 + 0.8 \times 32.39 = 47.65$ B.t.u. per lb.

Heat content at $t_d = 51.8$ and $h = 1.0 = 21.19$ B.t.u. per lb.

Heat to be removed from water condensed due to cooling from 83 to 51.8 deg. fahr. $= \dfrac{2.487 - 0.789}{100} \times \dfrac{83 - 51.8}{2} = 0.42$ B.t.u. per lb.

(This is comparatively small and may be omitted.)

Total heat to be removed in cooling from initial conditions to 51.8 deg. fahr. $= 47.65 - (21.19 + 0.42) = 26.04$ B.t.u. per lb.

Heat content at $t_d = 70$ and $h = 0.5 = 16.9 + 0.5 \times 13.96 = 23.88$ B.t.u.

Heat to be added to retemper from 51.8 to 70 deg. $= 23.88 - 21.19 = 2.69$ B.t.u. per lb.

These values, neglecting the heat of the liquid, may be taken directly from the curves in Fig. 641 as shown in the preceding example.

Example 98. *Evaporative Surface Condenser.* — How many cubic feet of air and how many pounds of water spray must be forced through an evaporative surface condenser of the fan type in order to condense 1000 pounds of steam per hour and maintain a vacuum of 25 inches, barometer 29? (Atmospheric air 80 deg. fahr., relative humidity 70 per cent.) The air and vapor issue from the discharge pipe under pressure of 4 inches of water, temperature 120 deg. fahr., relative humidity 98 per cent.

The absolute pressure in the condenser is $29.0 - 25.0 = 4$ inches of mercury.

The total heat to be withdrawn in order to cool and condense 1000 pounds of steam per hour at absolute pressure of 4 inches to 120 deg. fahr. is

$$1000\,[1114.8 - (120 - 32)] = 1{,}026{,}000 \text{ B.t.u.}$$

Neglecting radiation and leakage losses, this is the heat to be abstracted per hour by the air and water spray.

Air-vapor Mixture Entering Condenser.

Pressure P_1 of the dry air:

$$P_1 = 29.0 - 0.7 \times 1.0314 = 28.28 \text{ in.}$$

(1.0314 = pressure of saturated vapor at temperature t = 80. deg. fahr.)

Volume V_1 of 1 lb. of dry air plus its vapor content, equation (435):

$$V_1 = \frac{0.755\,(459.6 + 80)}{28.28} = 14.41 \text{ cu. ft.}$$

Weight w_1 of vapor in 1 lb. of dry air:

$$w_1 = 0.7 \times 14.41 \times 0.00158 = 0.0159 \text{ lb.}$$

(0.00158 = density of saturated vapor at t_1 = 80 deg. fahr.)

Heat content H_a of 1 lb. of dry air above 0 deg. fahr.:

$$H_a = C_{pa}t_1 = 0.2414 \times 80 = 19.32 \text{ B.t.u.}$$

Latent heat r_v of vapor content in 1 lb. of dry air:

$$r_v = 0.7\,(14.41 \times 0.00158 \times 1047.4) = 16.68 \text{ B.t.u.}$$

(1047.4 = latent heat of saturated vapor at temperature t = 80.)

Total heat H_1 of mixture in 1 lb. of dry air:

$$H_1 = 19.32 + 16.68 = 36.00 \text{ B.t.u.}$$

Air-Vapor Mixture Leaving Condenser.

Pressure P_2 of the dry air:

$$P_2 = (29.0 + 0.294) - 0.98 \times 3.444 = 25.92.$$

(0.294 = value in inches of mercury of 4 inches of water pressure.)

Volume V_2 of 1 lb. of dry air plus its vapor content:

$$V_2 = \frac{0.755\,(459.6 + 120)}{25.92} = 16.89 \text{ cu. ft.}$$

Weight w_2 of vapor in 1 lb. of dry air:

$$w_2 = 0.98 \times 16.89 \times 0.00492 = 0.08143.]$$

Heat content H_a' of the dry air in 1 lb. of mixture:

$$H_a' = C_pt = 0.2416 \times 120 = 29.00 \text{ B.t.u.}$$

Latent heat r_v' of vapor content in 1 lb. of dry air:

$$r_v' = 10.98\,(16.89 \times 0.00492 \times 1025.6) = 83.53 \text{ B.t.u.}$$

Total heat H_2 of the mixture in 1 lb. of dry air:

$$H_2 = 29.00 + 83.53 = 112.53 \text{ B.t.u.}$$

Heat taken up by 1 lb. of air plus water vapor in passing through the condenser

$$= H_2 - H_1 = 112.53 - 36.00 = 76.53 \text{ B.t.u.}$$

Total weight of dry air passing through condenser

$$= \frac{1{,}026{,}000}{76.53} = 13{,}400 \text{ lb. per hour.}$$

Total volume of air-vapor entering the condenser

$$= 13{,}400 \times 14.41 = 192{,}960 \text{ cu. ft.}$$

Water absorbed per lb. of dry air

$$= w_2 - w_1 = 0.08143 - 0.0159 = 0.06553 \text{ lb.}$$

Total moisture absorbed or weight of spray to be injected

$$= 13{,}400 \times 0.06553 = 878.0 \text{ lb. per hr.}$$

For purpose of design it is sufficiently accurate to disregard the actual barometric pressure and assume it to be 29.92 inches. With this assumption the problem may be readily solved by means of Table 169 or the curves in Figs. 461–2.

From Fig. 461 (for $t_1 = 80$ and $h_1 = 0.70$):

Wet bulb = 72.2, Dew point = 69.0.

$w_1 = 107$ grains = 0.0153 lb.

$H_1 = 35.5$ B.t.u.

From Fig. 462 (for $t_2 = 120$ and $h_2 = 0.98$):

Wet bulb = 119.4, Dew point = 119.2,

$w_2 = 555$ grains = 0.0793 lb.

$H_2 = 111$ B.t.u.

Moisture absorbed per lb. of dry air, and its vapor content,

$$w_2 - w_1 = 0.0793 - 0.0153 = 0.064 \text{ lb.}$$

Heat absorbed per lb. of dry air, and its vapor content,

$$H_2 - H_1 = 111 - 35.5 = 75.5 \text{ B.t.u.}$$

Since the moisture content per lb. of dry air at dew point is the same as that for all conditions of wet and dry bulb temperatures having that dew point temperature,

From Table 169:

w_1 for dew point 69.0 = 0.0152 lb.

w_2 for dew point 119.2 = 0.0793 lb.

Moisture absorbed per lb. of dry air = 0.0793 − 0.0152 = 0.0641 lb.

Since the heat content or total heat is constant for a given wet bulb temperature

H_1 for wet bulb 72.2 = 35.3 B.t.u.

H_2 for wet bulb 119.2 = 110.5.

Heat absorbed per lb. of dry air and its vapor content

$$H_2 - H = 110.5 - 35.3 = 75.2 \text{ B.t.u. per lb.}$$

These results check substantially with the calculated data.

Example 99. Determine the quantity of air passing through the cooling tower and the weight of circulating water lost by evaporation in a surface-condensing power plant operating under the following conditions: Turbines, average load 1000 kw.; average water rate 20 lb. per kw-hr.; initial steam pressure 150 lb. abs.; superheat 50 deg. fahr.; vacuum 26.92 in.; barometer 29.92 in.; temperature of injection water, discharge water and outside air, 70, 100, and 65 deg. fahr., respectively; temperature of air leaving tower 90 deg. fahr.; wet bulb temperature of outside air and air leaving cooling tower 57 and 89 deg. fahr. respectively.

Total heat to be abstracted from the steam =

$$1000 \times 20 \left(1223 - \frac{3412}{20} - 105^* + 32\right) = 19{,}580{,}000 \text{ B.t.u. per hr.}$$

* Assumed hot well temperature.

Atmospheric air entering tower:

From the curves in Fig. 461 (dry bulb temperature 65 deg. fahr. and wet bulb temperature 57 deg. fahr.).
Moisture content of 1 lb. of dry air, $w_1 = 56$ grains.
Total heat of 1 lb. of dry air, with its vapor content,

$$H_1 = 24.3 \text{ B.t.u.}$$

Air-vapor mixture leaving tower:

From the curves in Fig. 461 (dry bulb 90 and wet bulb 98).
Moisture content of 1 lb. of dry air, $w_2 = 209$ grains.
Total heat of 1 lb. of dry air, with its vapor content,

$$H_2 = 52.8 \text{ B.t.u.}$$

Moisture absorbed by 1 lb. of dry air in passing through the tower

$$= w_2 - w = 209 - 56 = 153 \text{ grains or } 0.02186 \text{ lb.}$$

Heat absorbed by 1 lb. of dry air (plus its initial vapor content) in passing through the tower

$$= H_2 - H = 52.8 - 24.3 = 28.5 \text{ B.t.u.}$$

Total weight of dry air required to abstract the heat from the circulating water

$$= \frac{19{,}580{,}000}{28.5} = 687{,}000 \text{ lb. per hr.}$$

Volume of 1 lb. of dry air and its vapor content entering tower

$$= 0.755\left(\frac{459.6 + 65}{29.54}\right) = 13.39 \text{ cu. ft.}$$

(29.54 = pressure of the dry air in the mixture $= 29.92 - 0.61 \times 0.6218$; 0.61 = relative humidity and 0.6218 = pressure of saturated vapor at 65 deg. fahr.)

Total volume of atmospheric air entering tower

$$= \frac{687{,}000 \times 13.39}{60} = 153{,}000 \text{ cu. ft. per min.}$$

APPENDIX A

DATA AND RESULTS OF EVAPORATIVE TEST

A.S.M.E. Code of 1915

(1) Test of .... boiler located at ....
To determine..
Test conducted by...

Dimensions.

(2) Number and kind of boilers..
(3) Kind of furnace...
(4) Grate surface (width —— length ——) *..................... sq. ft.
(*a*) Approximate width of air openings in grate..................... in.
(*b*) Percentage of area of air openings to grate surface.......... per cent

(5) Water heating surface.. sq. ft.
(6) Superheating surface... sq. ft.
(7) Total heating surface.. sq. ft.
(*a*) Ratio of water heating surface to grate surface............. (—) to 1
(*b*) Ratio of total heating surface to grate surface.............. (—) to 1
(*c*) Ratio of minimum draft area to grate surface.............. 1 to (—)
(*d*) Volume of combustion space between grate and heating surface, cu. ft.
(*e*) Distance from center of grate to nearest heating surface.......... ft.

Date, Duration, etc.

(8) Date..
(9) Duration.. hr.
(10) Kind and size of coal...

Average Pressures, Temperatures, etc.

(11) Steam pressure by gage............................... lb. per sq. in.
(*a*) Barometric pressure.............................. in. of mercury

(12) Temperature of steam, if superheated............................ deg.
(*a*) Normal temperature of saturated steam........................ deg.

(13) Temperature of feed water entering boiler....................... deg.
(*a*) Temperature of feed water entering economizer................ deg.
(*b*) Increase of temperature of water due to economizer............ deg.

* Unless otherwise designated this is the total area enclosed within the furnace walls projected horizontally.

(14) Temperature of escaping gases leaving boiler....................deg.

(*a*) Temperature of gases leaving economizer....................deg.
(*b*) Decrease of temperature of gases due to economizer............deg.
(*c*) Temperature of furnace....................................deg.

(15) Force of draft between damper and boiler................in. of water

(*a*) Draft in main flue near boiler........................in. of water
(*b*) Draft in main flue between economizer and chimney.....in. of water
(*c*) Draft in furnace.....................................in. of water
(*d*) Draft or blast in ash pit............................in. of water

(16) State of weather..

(*a*) Temperature of external air..............................deg.
(*b*) Temperature of air entering ash pit *....................deg.
(*c*) Relative humidity of air entering ash pit................per cent

Quality of Steam.

(17) Percentage of moisture in steam or number of degrees of superheating.................................per cent or deg.
(18) Factor of correction for quality of steam ....

Total Quantities.

(19) Total weight of coal as fired †...................................lb.
(20) Percentage of moisture in coal as fired ....per cent
(21) Total weight of dry coal $\left(\text{Item 19} \times \left[\frac{1 - \text{Item 20}}{100}\right]\right)$..............lb.
(22) Ash, clinkers, and refuse (dry)

(*A*) Withdrawn from furnace and ash pit.......................lb.
(*B*) Withdrawn from tubes, flues, and combustion chamber......lb.
(*C*) Blown away with gases....................................lb.
(*D*) Total..lb.
(*a*) Weight of clinkers contained in total ash...................lb.

(23) Total combustible burned (Item 21 – Item 22*D*) ‡................lb.
(24) Percentage of ash and refuse based on dry coal...............per cent
(25) Total weight of water fed to boiler §..............................lb.
(26) Total water evaporated, corrected for quality of steam (Item 25 × Item 18)...lb.
(27) Factor of evaporation based on temperature of water entering boiler....
(28) Total equivalent evaporation from and at 212 deg. (Item 26 × Item 27)...lb.

* Thermometer should be protected from direct radiation of boiler and furnace.

† The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.

‡ If either of the two items 22*B* and 22*C* is omitted, the fact should be so stated.

§ Corrected for inequality of water level and of steam pressure at beginning and end.

Hourly Quantities and Rates.

(29) Dry coal per hour..lb.
(30) Dry coal per sq. ft. of grate surface per hour.........................lb.
(31) Water evaporated per hour, corrected for quality of steam..........lb.
(32) Equivalent evaporation per hour from and at 212 deg.*.............lb.
(33) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water heating surface*......................................lb.

Capacity.

(34) Evaporation per hour from and at 212 deg. (same as Item 32) ....lb.
(a) Boiler horsepower developed (Item 34 ÷ $34\frac{1}{2}$) ....b.hp.

(35) Rated capacity per hour, from and at 212 deg.......................lb.
(a) Rated boiler horsepower ....b.hp.

(36) Percentage of rated capacity developed......................per cent

Economy.

(37) Water fed per lb. of coal as fired (Item 25 ÷ Item 19) ....lb.
(38) Water evaporated per lb. of dry coal (Item 26 ÷ Item 21)..........lb.
(39) Equivalent evaporation from and at 212 deg. per lb. of coal as fired (Item 28 ÷ Item 19) ....lb.
(40) Equivalent evaporation from and at 212 deg. per lb. of dry coal (Item 28 ÷ Item 21)..lb.
(41) Equivalent evaporation from and at 212 deg. per lb. of combustible (Item 28 ÷ Item 23)..lb.

Efficiency.

(42) Calorific value of 1 lb. of dry coal by calorimeter†...............B.t.u.
(a) Calorific value of 1 lb. dry coal by analysis ....B.t.u.

(43) Calorific value of 1 lb. of combustible by calorimeter.............B.t.u.
(a) Calorific value of 1 lb. combustible by analysis ....B.t.u.

(44) Efficiency of boiler, furnace, and grate

$$\left[100 \times \frac{\text{Item } 40 \times 970.4}{\text{Item } 42}\right] \text{.................per cent}$$

(45) Efficiency based on combustible

$$\left[100 \times \frac{\text{Item } 41 \times 970.4}{\text{Item } 43}\right] \text{.................per cent}$$

* The symbol "U. E.," meaning Units of Evaporation, may be substituted for the expression "Equivalent evaporation from and at 212°".

† If the calorific value is desired per lb. of coal "as fired," multiply Item 42 by

$$\frac{100 - \text{Item } 20}{100}.$$

Cost of Evaporation.

(46) Cost of coal per ton of —— lb. delivered in boiler room..........dollars

(47) Cost of coal required for evaporating 1000 lb. of water under observed conditions..........dollars

(48) Cost of coal required for evaporating 1000 lb. of water from and at 212 deg..........dollars

Smoke Data.

(49) Percentage of smoke as observed..........per cent

(*a*) Weight of soot per hour obtained from smoke meter..........per cent

Firing Data.

(50) Kind of firing, whether spreading, alternate, or coking..........

(*a*) Average thickness of fire..........in.

(*b*) Average intervals between firings for each furnace during time when fires are in normal condition..........min.

(*c*) Average interval between times of leveling or breaking up..........min.

(51) Analysis of dry gases by volume

(*a*) Carbon dioxide (CO_2)..........per cent

(*b*) Oxygen (O)..........per cent

(*c*) Carbon monoxide (CO)..........per cent

(*d*) Hydrogen and hydrocarbons..........per cent

(*e*) Nitrogen, by difference (N)..........per cent

(52) Proximate analysis of coal

	As fired.	Dry coal.	Combustible.
(*a*) Moisture..........	....	....	....
(*b*) Volatile matter..........	....	....	....
(*c*) Fixed carbon..........	....	....	....
(*d*) Ash..........	....	....	....
	100 per cent	100 per cent	100 per cent

(*e*) Sulphur, separately determined referred to dry coal..........per cent

(53) Ultimate analysis of dry coal

(*a*) Carbon (C)..........per cent

(*b*) Hydrogen (H)..........per cent

(*c*) Oxygen (O)..........per cent

(*d*) Nitrogen (N)..........per cent

(*e*) Sulphur (S)..........per cent

(*f*) Ash..........per cent

100 per cent

(54) Analysis of ash and refuse, etc.

(*a*) Volatile matter..........per cent

(*b*) Carbon..........per cent

(*c*) Earthy matter..........per cent

100 per cent

(*d*) Sulphur, separately determined..........per cent

(*e*) Fusing temperature of ash..........deg.

(55) Heat balance, based on dry coal

	Dry Coal.	
	B.t.u.	Per Cent.
(*a*) Heat absorbed by the boiler (Item 40 $\times$ 970.4)		
(*b*) Loss due to evaporation of moisture in coal		
(*c*) Loss due to heat carried away by steam formed by the burning of hydrogen		
(*d*) Loss due to heat carried away in the dry flue gases		
(*e*) Loss due to carbon monoxide		
(*f*) Loss due to combustible in ash and refuse		
(*g*) Loss due to heating moisture in air		
(*h*) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for		
(*i*) Total calorific value of 1 lb. of dry coal (Item 42)		100

If it is desired that the heat balance be based on coal "as fired" or on "combustible burned" the items in the first column are multiplied by the proportion $\frac{100 - \text{Item } 20}{100}$ for coal "as fired" or by the proportion $\frac{100 - \text{Item } 20}{100 - (\text{Item } 20 + \text{Item } 24)}$ for "combustible burned."

PRINCIPAL DATA AND RESULTS OF BOILER TEST.

(1) Grate surface (width —— length ——) . . . . . sq. ft.
(2) Total heating surface . . . . . sq. ft.
(3) Date . . . . .
(4) Duration . . . . . hr.
(5) Kind and size of coal . . . . .
(6) Steam pressure by gage . . . . . lb. per sq. in.
(7) Temperature of feed water entering boiler . . . . . deg.
(8) Percentage of moisture in steam or number of degrees of superheating . . . . . per cent or deg.
(9) Percentage of moisture in coal . . . . . per cent
(10) Dry coal per hour . . . . . lb.
(11) Dry coal per sq. ft. of grate surface per hour . . . . . 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. of dry coal . . . . . lb.
(17) Equivalent evaporation from and at 212 deg. per lb. of combustible . . . lb.
(18) Calorific value of 1 lb. of dry coal by calorimeter . . . . . B.t.u.
(19) Calorific value of 1 lb. of combustible by calorimeter . . . . . B.t.u.
(20) Efficiency of boiler, furnace, and grate . . . . . per cent
(21) Efficiency based on combustible . . . . . per cent

APPENDIX B

DATA AND RESULTS OF STEAM-ENGINE TEST

A.S.M.E. Code of 1915

(1) Test of.....................engine located at....................
To determine..
Test conducted by...

Dimensions, etc.

(2) Type of engine (simple or multiple expansion)......................
(3) Class of service (mill, marine, electric, etc.)....................
(4) Auxiliaries (steam or electric driven)

(*a*) Type and make of condenser equipment.........................
(*b*) Rated capacity of condenser equipment......................hp.
(*c*) Type of oil pump, jacket pump, and reheater pump (direct or independently driven)..

(5) Rated power of engine..

(*a*) Name of builders...
(*b*) Kind of valves...
(*c*) Type of governor*..

		1st	2d	3d
(6)	Diameter of cylinders...............in.	......	......	......
(7)	Stroke of pistons...................ft.	......	......	......
	(*a*) Diameter of piston-rod, each end..in.	......	......	......
(8)	Clearance (average) in per cent of piston displacement......................	......	......	......
(9)	Hp. constant 1 lb. 1 rev.............hp.	......	......	......
	(*a*) Cylinder ratio (based on net piston displacement)........ 1 to —	......	......	......
	(*b*) Area of interior steam surface..sq. ft.	......	......	......
	(*c*) Area of jacketed surfaces .....sq. ft.	......	......	......
(10)	Capacity of generator or other apparatus consuming power of engine ....hp.	......	......	......

Date and Duration.

(11) Date...
(12) Duration..hr.

* For other matters relating to the analysis of engine performance, see treatises on thermodynamics.

Average Pressures and Temperatures.

(13) Pressure in steam pipe near throttle, by gage............lb. per sq. in.
(14) Barometric pressure ....in. of mercury
  (a) Pressure at boiler, by gage.........................lb. per sq. in.
(15) Pressure in 1st receiver, by gage......................lb. per sq. in.
(16) Pressure in 2d receiver, by gage.......................lb. per sq. in.
(17) Pressure in exhaust pipe near engine by gage...........lb. per sq. in.
(18) Vacuum in condenser..................................in. of mercury
  (a) Corresponding absolute pressure.....................lb. per sq. in.
(19) Pressure in jackets and reheaters......................lb. per sq. in.
(20) Temperature of steam near throttle..............................deg.
  (a) Temperature of saturated steam at throttle pressure............deg.
  (b) Temperature of steam leaving 1st receiver, if superheated.......deg.
  (c) Temperature of steam leaving 2d receiver, if superheated.......deg.
(21) Temperature of steam in exhaust pipe near engine................deg.
  (a) Temperature of injection or circulating water entering condenser.deg.
  (b) Temperature of injection leaving condenser..................deg.
  (c) Temperature of air in engine room...........................deg.

Quality of Steam.

(22) Percentage of moisture in steam near throttle or number
  of degrees of superheating........................per cent or deg.

Total Quantities.

(23) Total water fed to boilers.......................................lb.
(24) Total condensed steam from surface condenser (corrected for condenser
  leakage)...lb.
(25) Total dry steam consumed (Item 23 or 24 less moisture in steam)......lb.

Hourly Quantities.

(26) Total water fed to boilers or drawn from surface condenser per hour..lb.
(27) Total dry steam consumed for all purposes per hour (Item 25 ÷
  Item 12)..lb.
(28) Steam consumed per hour for all purposes foreign to the main engine..lb.
(29) Dry steam consumed by engine per hour (Item 27 − Item 28).......lb.
  (a) Circulating water supplied to condenser per hour..............lb.

Hourly Heat Data.

(30) Heat units consumed by engine per hour [Item 29 × (total heat of
  steam per pound at pressure of Item 13 minus heat in 1 lb. of water
  at temperature of Item 21)]..................................B.t.u.

(*a*) Heat converted into work per hour........................B.t.u.
(*b*) Heat rejected to condenser per hour (Item 29*a* × [Item 21*b* − 21*a*]) (approximate)........................B.t.u.
(*c*) Heat rejected in form of uncondensed steam withdrawn from cylinders*........................B.t.u.
(*d*) Heat lost by radiation........................B.t.u.

Indicator Diagrams.

		1st Cyl.	2d Cyl.	3d Cyl.
(31)	Commercial cut-off in per cent of stroke.....per cent	....	....	....
(32)	Initial pressure above atmosphere......lb. per sq. in.	....	....	....
(33)	Back pressure at lowest point above or below atmosphere................lb. per sq. in.	....	....	....
	(*a*) Mean back pressure above atmosphere or zero.................lb. per sq. in.	....	....	....
(34)	Mean effective pressure..............lb. per sq. in.	....	....	....
	(*a*) Equivalent m.e.p. referred to 1st cylinder.....................lb. per sq. in.	....	....	....
	(*b*) Equivalent m.e.p. referred to 2d cylinder.....................lb. per sq. in.	....	....	....
	(*c*) Equivalent m.e.p. referred to 3d cylinder.....................lb. per sq. in.	....	....	....
(35)	Aggregate m.e.p. referred to each cylinder............................lb. per sq. in.	....	....	....
(36)	Steam accounted for per i.hp-hr. at point on expansion line shortly after cut-off..................................lb.	....	....	....
(37)	Steam accounted for per i.hp-hr. at point on expansion line just before release..................................lb.	....	....	....
	(*a*) Pressure at selected point near cut-off †....................lb. per sq. in.	....	....	....
	(*b*) Pressure at selected point near release .... lb. per sq. in.	....	....	....
	(*c*) Pressure at point on compression curve shortly after exhaust closure lb. per sq. in.	....	....	....
	(*d*) Proportion of direct stroke completed at selected point near cut-off................................	....	....	....
	(*e*) Proportion of direct stroke completed at selected point near release................................	....	....	....
	(*f*) Proportion of return stroke uncompleted at selected point on compression line..........................	....	....	....

* In multiple expansion engines. † Pressures all referred to zero.

(*g*) Ratio of expansion . . . .
(*h*) M.e.p. of hypothetical diagram (App. 27) . . . . lb. per sq. in. . . . .
(*i*) Diagram factor (App. 27) . . . .

Speed.

(38) Revolutions per minute . . . . r.p.m.
(39) Piston speed per minute . . . . ft.
(*a*) Variation of speed between no load and full load . . . . per cent
(*b*) Momentary fluctuation of speed on suddenly changing from full load to half-load . . . . per cent

Power.

(40) Indicated hp. developed, whole engine . . . . i.hp.
(*a*) I.hp. developed by 1st cylinder . . . . i.hp.
(*b*) I.hp. developed by 2d cylinder . . . . i.hp.
(*c*) I.hp. developed by 3d cylinder . . . . i.hp.

(41) Brake hp. . . . . br. hp.
(42) Friction of engine (Item 40 − Item 41) . . . . hp.
(*a*) Friction expressed in percentage of i.hp. (Item 42 ÷ Item 40 × 100) . . . . per cent
(*b*) Indicated hp. with no load, at normal speed . . . . i.hp.

Economy Results.

(43) Dry steam consumed by engine per i.hp. per hr. lb.
(44) Dry steam consumed by engine per brake hp-hr. lb.
(45) Percentage of steam consumed by engine accounted for by indicator at point near cut-off . . . . per cent
(46) Percentage of steam consumed near release . . . . per cent
(47) Heat units consumed by engine per i.hp-hr. (Item 30 ÷ Item 40) . . . . B.t.u.
(48) Heat units consumed by engine per br. hp-hr. (Item 30 ÷ Item 31) . . . . B.t.u.

Efficiency Results.

(49) Thermal efficiency of engine referred to i.hp. [(2546.5 ÷ Item 47) × 100] . . . . per cent
(50) Thermal efficiency of engine referred to br. hp. [(2546.5 ÷ Item 48) × 100] . . . . per cent
(51) Efficiency of Rankine cycle between temperatures of Items 20 and 21 . .
(52) Rankine cycle ratio referred to i.hp. (Item 49 ÷ Item 51) . . . .
(53) Rankine cycle ratio referred to br. hp. (Item 50 ÷ Item 51) . . . .

Work Done per Heat Unit.

(54) Net work per B.t.u. consumed by engine (1,980,000 ÷ Item 48) . . . . Ft-lb.

Sample Diagrams.

(55) Sample diagrams from each cylinder. . ...

(a) Steam pipe diagrams.

Note: — For an engine driving an electric generator the form should be enlarged to include the electrical data, embracing the average voltage, number of amperes each phase, number of watts, number of watt hours, average power factor, etc.; and the economy results based on the electric output embracing the heat units and steam consumed per electric hp-hr. and per kw-hr., together with the efficiency of the generator. (See table for Steam Turbine Code, Appendix C.)

Likewise, in a marine engine having a shaft dynamometer, the form should include the data obtained from this instrument, in which case the brake hp. becomes the shaft hp.

PRINCIPAL DATA AND RESULTS OF RECIPROCATING ENGINE TEST.

(1) Dimensions of cylinders...
(2) Date ....
(3) Duration ....hr.
(4) Pressure in steam pipe near throttle by gage.............lb. per sq. in.
(5) Pressure in receivers...............................lb. per sq. in.
(6) Vacuum in condenser...............................in. of mecury
(7) Percentage of moisture in steam near throttle or number of degrees of superheating ....per cent or deg.
(8) Net steam consumed per hour.................................lb.
(9) Mean effective pressure in each cylinder ....lb. per sq. in.
(10) Revolutions per minute..................................r.p.m.
(11) Indicated horsepower developed............................i.hp.
(12) Steam consumed per i.hp-hr..................................lb.
(13) Steam accounted for at cut-off each cylinder......................lb.
(14) Heat consumed per i.hp-hr...............................B.t.u.

APPENDIX C

DATA AND RESULTS OF STEAM TURBINE OR TURBO-GENERATOR TEST

A.S.M.E. Code of 1915

(1) Test of........................turbine located at................
To determine..
Test conducted by..

Dimensions, etc.

(2) Type of turbine (impulse, reaction, or combination).................
(a) Number of stages..
(b) Condensing or non-condensing...............................
(c) Diameter of rotors...
(d) Number and type of nozzles.................................
(e) Area of nozzles..
(f) Type of governor...

(3) Class of service (electric, pumping, compressor, etc.)................
(4) Auxiliaries (steam or electric driven)..............................
(a) Type and make of condensing equipment........................
(b) Rated capacity of condensing equipment.......................
(c) Type of oil pumps (direct or independently driven)..............
(d) Type of exciter (direct or independently driven)................
(e) Type of ventilating fan, if separately driven...................

(5) Rated capacity of turbine...
(a) Name of builders..

(6) Capacity of generator or other apparatus consuming power of turbine...

Date and Duration.

(7) Date...
(8) Duration...hr.

Average Pressures and Temperatures.

(9) Pressure in steam pipe near throttle by gage..............lb. per sq. in.
(10) Barometric pressure..................................in. of mercury
(a) Pressure at boiler by gage........................lb. per sq. in.
(b) Pressure in steam chest by gage....................lb. per sq. in.
(c) Pressure in various stages..........................lb. per sq. in.

(11) Pressure in exhaust pipe near turbine, by gage...........lb. per sq. in.

(12) Vacuum in condenser. . . . . in. of mercury
  (a) Corresponding absolute pressure. lb. per sq. in.
  (b) Absolute pressure in exhaust chamber of turbine. . . . . . . lb. per sq. in.

(13) Temperature of steam near throttle. . . . . deg.
  (a) Temperature of saturated steam at throttle pressure. . . . . deg.
  (b) Temperature of steam in various stages, if superheated. . . . . deg.

(14) Temperature of steam in exhaust pipe near turbine. . . . . deg.
  (a) Temperature of circulating water entering condenser. . . . . deg.
  (b) Temperature of circulating water leaving condenser. . . . . deg.
  (c) Temperature of air in turbine room. . . . . deg.

Quality of Steam.

(15) Percentage of moisture in steam near throttle, or number of degrees of superheating. . . . . per cent or deg.

Total Quantities.

(16) Total water fed to boilers. . . . . lb.

(17) Total condensate from surface condenser (corrected for condenser leakage and leakage of shaft and pump glands). . . . . lb.

(18) Total dry steam consumed (Item 16 or 17 less moisture in steam). . . . lb.

Hourly Quantities.

(19) Total water fed to boilers or drawn from surface condenser per hour. . lb.

(20) Total dry steam consumed for all purposes per hour (Item 18 ÷ Item 8). . . . . lb.

(21) Steam consumed per hour for all purposes foreign to the turbine (including drips and leakage of plant). . . . . lb.

(22) Dry steam consumed by turbine per hour (Item 20 − Item 21). . . . . . lb.
  (a) Circulating water supplied to condenser per hour. . . . . lb.

Hourly Heat Data.

(23) Heat units consumed by turbine per hour [Item 22 × (total heat of steam per pound at pressure of Item 9 less heat in 1 lb. of water at temperature of Item 14)]. . . . . B.t.u.
  (a) Heat converted into work per hour. . . . . B.t.u.
  (b) Heat rejected to condenser per hour (Item 22a × [Item 14b − Item 14a]) (approximate). . . . . B.t.u.
  (c) Heat rejected in the form of steam withdrawn from the turbine. . . B.t.u.
  (d) Heat lost by radiation from turbine, and unaccounted for. . . . . B.t.u.

Electrical Data.

(24) Average volts, each phase. . . . . volts
(25) Average amperes, each phase. . . . . amperes
(26) Average kilowatts, first meter. . . . . kw.
(27) Average kilowatts, second meter. . . . . kw.
(28) Total kilowatts output. . . . . kw.

(29) Power factor. . . . .
(30) Kilowatts used for excitation and for separately driven ventilating fan. . . . . .kw.
(31) Net kilowatt output. . . . . .kw.

SPEED.

(32) Revolutions per minute. . . . . .r.p.m.
(33) Variation of speed between no load and full load. . . . . .r.p.m.
(34) Momentary fluctuation of speed on suddenly changing from full load to half-load. . . . . .r.p.m.

POWER.

(35) Brake horsepower, if determined. . . . . .br. hp.
(36) Electrical horsepower. . . . . .e-hp.

ECONOMY RESULTS.

(37) Dry steam consumed by turbine per br. hp-hr. .lb.
(38) Dry steam consumed per net kw-hr. . . . . .lb.
(39) Heat units consumed by turbine per br. hp-hr. (Item 23 ÷ Item 35) . . . . .B.t.u.
(40) Heat units consumed per net kw-hr. . . . . .B.t.u.

EFFICIENCY RESULTS.

(41) Thermal efficiency of turbine (2546.5 ÷ Item 39) × 100. . . . . .per cent
(42) Efficiency of Rankine cycle between temperatures of Items 13 and 14 . . . . .per cent
(43) Rankine cycle ratio (Item 41 ÷ Item 42). . . . .

WORK DONE PER HEAT UNIT.

(44) Net work per B.t.u. consumed by turbine (1,980,000 ÷ Item 39). . . .ft.lb.

PRINCIPAL DATA AND RESULTS OF TURBINE TEST.

(1) Dimensions. . . . .
(2) Date. . . . .
(3) Duration. . . . . .hr.
(4) Pressure in steam pipe near throttle by gage. . . . . .lb. per sq. in.
(5) Vacuum in condenser. . . . . .in. of mecury
(6) Percentage of moisture in steam near throttle or number of degrees of superheating. . . . . .per cent or deg.
(7) Net steam consumed per hour. . . . . .lb.
(8) Revolutions per minute. . . . . .r.p.m.
(9) Brake horsepower developed. . . . . .br. hp.
(10) Kw. output. . . . . .kw.
(11) Steam consumed per brake hp-hr. . . . . .lb.
(12) Heat consumed per brake hp-hr. . . . . .B.t.u.
(13) Steam consumed per kw-hr. . . . . .lb.
(14) Heat consumed per kw-hr.. . . . . .B.t.u.

APPENDIX D

DATA AND RESULTS OF STEAM PUMPING MACHINERY TEST

A.S.M.E. Code of 1915

(1) Test of................pump located at........................
To determine..
Test conducted by...

Dimensions, etc.

(2) Type of machinery...
(3) Rated capacity in gallons per 24 hr...........................gal.
(4) Size of engine or turbine.....................................
(5) Size of pump..
(6) Auxiliaries (steam or electric driven)........................
(*a*) Type and make of condenser equipment.......................
(*b*) Rated capacity of condenser equipment......................
(*c*) Type of oil pump, jacket pump, and reheater pump (direct or independently driven)...............................

Date and Duration.

(7) Date..
(8) Duration..hr.

Average Pressures and Temperatures

(9) Pressure in steam pipe near throttle by gage.............lb. per sq. in.
(10) Barometric pressure.....................................in. of mercury
(*a*) Steam chest pressure.....................................lb. per sq. in.
(*b*) Pressure in receivers and reheaters by gage.............lb. per sq. in.
(*c*) Pressure in turbine stages by gage.......................lb. per sq. in.
(11) Pressure in exhaust pipe near engine or turbine by gage....lb. per sq. in.
(12) Vacuum in condenser.....................................in. of mercury
(*a*) Corresponding absolute pressure..........................lb. per sq. in.
(*b*) Absolute pressure in exhaust chamber.....................lb. per sq. in.
(13) Temperature of steam, if superheated, at throttle..............deg.
(*a*) Normal temperature of saturated steam at throttle pressure......deg.
(*b*) Temperature of steam leaving receivers, if superheated..........deg.
(14) Temperature of steam in exhaust pipe near engine or turbine.......deg.
(*a*) Temperature of circulating water entering condenser...........deg.
(*b*) Temperature of circulating water leaving condenser............deg.

(15) Pressure in force main by gage..........................lb. per sq. in.
(16) Vacuum or pressure in suction main by gage.
....................................in. of mercury or lb. per sq. in.
(*a*) Correction for difference in elevation of the two gages..lb. per sq. in.

(17) Total head expressed in lb. pressure per sq. in............lb. per sq. in.
(*a*) Total head expressed in ft.......................................ft.

Quality of Steam.

(18) Percentage of moisture in steam near throttle, or number of degrees of superheating.......................................per cent or deg.

Total Quantities.

(19) Total water fed to boilers..lb.
(20) Total condensed steam from surface condenser (corrected for condenser leakage)...lb.
(21) Total dry steam consumed (Item 19 or 20 less moisture in steam)....lb.
(22) Total water discharged, by measurement..........................gal.
(*a*) Total water discharged, by plunger displacement, uncorrected....gal.
(*b*) Percentage of slip $\left[\frac{\text{Item } 22a - \text{Item } 22}{\text{Item } 22a} \times 100\right]$
(*c*) Leakage of pump..gal.
(*d*) Total water discharged, by calculation from plunger displacement, corrected for leakage.......................................gal.
(*e*) Total weight of water discharged, as measured..................lb.
(*f*) Total weight of water discharged, by calculation from plunger displacement, corrected for leakage...........................lb.

Hourly Quantities.

(23) Total water fed to boilers or drawn from surface condenser per hour...lb.
(24) Total dry steam consumed for all purposes per hour (Item 21 ÷ Item 8)..lb.
(25) Steam consumed per hour for all purposes foreign to main engine.....lb.
(26) Dry steam consumed by engine or turbine per hour (Item 24 − Item 25)...lb.
(*a*) Circulating water supplied to condenser per hour.................lb.

(27) Weight of water discharged per hour, by measurement...............lb.
(*a*) Weight of water discharged per hour, calculated from plunger displacement, corrected...lb.

Hourly Heat Data.

(28) Heat units consumed by engine or turbine per hour [Item 26 × (total heat of one lb. of steam at pressure of Item 9, less heat in one lb. of water at temperature of Item 14)]...........B.t.u.

Indicator Diagrams.

(29) Mean effective pressure, each steam cylinder............lb. per sq. in.
(*a*) Mean effective pressure, each water cylinder, if any....lb. per sq. in.

Speed and Stroke.

(30) Revolutions per minute . . . . r.p.m.

(a) Number of single strokes per minute . . . . strokes
(b) Average length of stroke . . . . ft.

Power.

(31) Indicated horsepower developed . . . . i.hp.

(a) Brake horsepower consumed by pump . . . .

(32) Water horsepower . . . . hp.
(33) Friction horsepower (Item 31 − Item 32) . . . . hp.
(34) Percentage of i.hp. lost in friction . . . . per cent

Capacity.

(35) Water discharged in 24 hr., as measured . . . . gal.

(a) Water discharged in 24 hr., calculated from plunger displacement, corrected . . . . gal.
(b) Water discharged per minute, as measured . . . . gal.
(c) Water discharged per minute, calculated from plunger displacement, corrected . . . . gal.

Economy Results.

(36) Heat units consumed per i.hp-hr . . . . B.t.u.
(37) Heat units consumed per water hp-hr . . . . B.t.u.

(a) Dry steam consumed per i.hp-hr . . . . lb.
(b) Dry steam consumed per water hp-hr . . . . lb.

Efficiency Results.

(38) Thermal efficiency referred to i.hp. [(2546.5 ÷ Item 36) × 100] . . per cent

(a) Thermal efficiency referred to water hp. [(2546.5 ÷ Item 37) × 100] . . . . per cent
(b) Mechanical efficiency $\left[\frac{\text{Item 32}}{\text{Item 31}} \times 100\right]$ . . . . per cent
(c) Pump efficiency $\left[\frac{\text{Item 32}}{\text{Item 31}a} \times 100\right]$ . . . . per cent

Duty.

(39) Duty per 1,000,000 heat units . . . . ft-lb.

Work Done per Heat Unit.

(40) Work per B.t.u. (1,980,000 ÷ Item 37) . . . . ft-lb.

Sample Diagrams.

(41) Sample indicator diagrams from each steam and pump cylinder . . . .

Note: — The items relating to indicator diagrams and indicated horsepower are to be used only in the case of reciprocating machines.

APPENDIX E

DATA AND RESULTS OF STEAM POWER PLANT TEST

A.S.M.E. Code of 1915

(1) Test of....................plant located at....................
To determine..
Test conducted by..

Date, Duration, etc.

(2) Number and kind of boilers (superheaters, if any), engines, turbines, etc.
(3) Rated capacity of boilers in lb. of steam per hour from and at 212 deg. lb.

(a) Kind of furnace..
(b) Grate surface..sq. ft.
(c) Percentage of area of openings to area of grate............per cent
(d) Water heating surface......................................sq. ft.
(e) Superheating surface.......................................sq. ft.

(4) Rated power of engines or turbines..............................

(a) Dimensions of cylinders of engine..............................
(b) Dimensions of turbine..
(c) Type of engines or turbines and class of service...............
(d) Name of builders...

(5) Type of auxiliaries*...

(a) Dimensions of auxiliaries*.....................................

(6) Type and capacity of condenser.................................
(7) Capacity of generators, pumps, or other apparatus consuming power of engine or turbine...

Date, Duration, etc.

(8) Date...
(9) Duration. Length of time engine or turbine was in motion with throttle open..hr.

(a) Length of time engine or turbine was running at normal speed....hr.
(b) Elapsed time from start to finish..............................hr.

(10) Kind and size of coal..

* For full particulars see text of Report.

Average Pressures, Temperatures, etc.

(11) Boiler pressure by gage . . . . lb. per sq. in.
- (a) Steam pipe pressure near throttle by gage . . . . lb. per sq. in.
- (b) Barometric pressure . . . . in. of mercury
- (c) Steam chest pressure by gage . . . . lb. per sq. in.
- (d) Pressure in receivers and reheaters by gage . . . . lb. per sq. in.
- (e) Pressure in turbine stages by gage . . . . lb. per sq. in.
- (f) Pressure in exhaust pipe near engine or turbine . . . . lb. per sq. in.

(12) Vacuum in condenser . . . . in. of mercury
- (a) Corresponding absolute pressure . . . . lb. per sq. in.
- (b) Absolute pressure in exhaust chamber . . . . lb. per sq. in.

(13) Temperature of steam, if superheated (taken at boiler or superheater) . . . . deg.
- (a) Temperature of steam, if superheated (taken at throttle) . . . . deg.
- (b) Normal temperature of saturated steam at boiler pressure . . . . deg.
- (c) Normal temperature of saturated steam at throttle pressure . . . . . deg.
- (d) Temperature of steam leaving receivers, if superheated . . . . deg.
- (e) Temperature of steam in exhaust pipe near engine or turbine . . . . deg.
- (f) Temperature of condensed water in hot-well or feed tank . . . . deg.
- (g) Temperature of circulating water entering condenser . . . . deg.
- (h) Temperature of circulating water leaving condenser . . . . deg.
- (i) Temperature of air in boiler room . . . . deg.
- (j) Temperature of air in engine or turbine room . . . . deg.

(14) Temperature of feed water entering boilers (average) . . . . deg.
- (a) Temperature of each feed supply (if more than one) . . . . deg.
- (b) Temperature of feed water entering economizer, if any . . . . deg.
- (c) Increase in temperature of water due to economizer . . . . deg.

(15) Temperature of escaping gases leaving boiler . . . . deg.
- (a) Temperature of escaping gases leaving economizer . . . . deg.
- (b) Decrease in temperature of gases due to economizer . . . . deg.
- (c) Temperature of furnace . . . . deg.

(16) Force of draft in main boiler flue . . . . in. of water
- (a) Force of draft at base of chimney . . . . in. of water
- (b) Force of draft at each end of economizer . . . . in. of water
- (c) Force of draft at individual boiler dampers . . . . in. of water
- (d) Force of draft in individual furnaces . . . . in. of water
- (e) Force of draft or blast in individual ash pits* . . . . in. of water

(17) State of weather . . . .
- (a) Temperature of external air . . . . deg.

Quality of Steam.

(18) Percentage of moisture in steam, or number of degrees of superheating . . . . per cent or deg.
- (a) Factor of correction for quality of steam . . . .

* If artificial draft or blast is employed, the force of draft or blast at the fan should also be given.

Total Quantities of Coal and Water.

(19) Total weight of coal as fired....................................lb.

- (*a*) Percentage of moisture in coal........................per cent
- (*b*) Total weight of dry coal....................................lb.
- (*c*) Total ash, clinkers, and refuse (dry)............................lb.
- (*e*) Percentage of ash and refuse in dry coal..................per cent
- (*f*) Total combustible burned (Item 19*b* − 19*c*)......................lb.

(20) Total weight of water fed to boiler from all sources*...............lb.

- (*a*) Total water evaporated corrected for quality of steam (Item 20 × Item 18*a*)....................................lb.
- (*b*) Factor of evaporation based on average temperature of water entering boiler....................................
- (*c*) Total equivalent evaporation from and at 212 degrees (Item 20*a* × Item 20*b*)....................................lb.

(21) Coal, as fired, per hour (Item 19 ÷ Item 9)........................lb.

- (*a*) Dry coal per hour (Item 19*b* ÷ Item 9)........................lb.
- (*b*) Dry coal per sq. ft. of grate surface..........................lb.

(22) Water evaporated per hour (Item 20 ÷ Item 9)..................lb.

- (*a*) Equivalent evaporation per hour from and at 212 deg............lb.
- (*b*) Equivalent evaporation per sq. ft. of water heating surface........lb.

(23) Dry steam generated per hour (sum of sub-items *a* to *g*) (Item 20 less moisture in steam ÷ Item 9)..........................lb.

- (*a*) Moisture formed per hour between boiler and engine..............lb.
- (*b*) Dry steam consumed per hour by engine cylinders or turbine.....lb.
- (*c*) Dry steam consumed per hour by reheaters and jackets, if any....lb.
- (*d*) Dry steam consumed per hour by air and circulating pump of condenser.. lb.
- (*e*) Dry steam consumed per hour by boiler-feed pump..............lb.
- (*f*) Dry steam consumed per hour by other steam-driven auxiliaries...lb.
- (*g*) Dry steam consumed per hour to supply leakage of boilers and piping between boilers and engine (including steam supplied for foreign purposes, if any)....................................lb.
- (*h*) Live steam supplied for heating, or miscellaneous purposes........lb.
- (*i*) Injection or circulating water supplied condenser per hour........lb.

Calorific Value of Coal.

(24) Calorific value of 1 lb. of coal as fired, by calorimeter test........B.t.u.

- (*a*) Calorific value of 1 lb. of dry coal........................B.t.u.
- (*b*) Calorific value of 1 lb. of combustible....................B.t.u.

* If there are a number of supplies of feed water, the weight and temperature of each supply is to be given, and total weight and average temperature ascertained.

Hourly Heat Data.

(25) Heat units in coal as fired generated per hour (Item 21 × Item 24). B.t.u.

(26) Heat units consumed by engine and auxiliaries per hour (Item 22 × total heat of 1 lb. of steam at pressure of Item 11 less heat in 1 lb. of water at temperature of feed water supplied to boiler, or economizer, if any) .... B.t.u.

- (*a*) Heat converted into work per hour .... B.t.u.
- (*b*) Heat rejected to condenser per hour .... B.t.u.
- (*c*) Heat rejected in steam withdrawn from receivers or turbine-stages not used by feed water .... B.t.u.
- (*d*) Heat lost by radiation from engine and auxiliaries, including piping between boilers and condenser .... B.t.u.
- (*e*) Heat lost in operation of boiler, including economizer (if any) (Item 25 – Item 26) .... B.t.u.

Indicator Diagrams.

(27) Mean effective pressure, each cylinder .... lb.

- (*a*) Commercial cut-off (in per cent of stroke) each cylinder ...... per cent
- (*b*) Initial pressure, above atmosphere, each cylinder ...... lb. per sq. in.
- (*c*) Back pressure at lowest point above or below atmosphere, each cylinder .... lb. per sq. in.
- (*d*) Steam accounted for per i.hp. per hour at point near cut-off, each cylinder .... lb.
- (*e*) Steam accounted for per i.hp. per hour at point near release .... lb.

Electrical Data.

(28) Average kilowatt output, gross .... kw.

- (*a*) Volts each phase .... volts
- (*b*) Amperes each phase .... amperes
- (*c*) Kilovolt amperes .... kv-a.
- (*d*) Power factor ....

(29) Current used by exciter .... kw.

(30) Net kilowatt output (Item 28 – Item 29) .... kw.

(31) Revolutions per minute .... r.p.m.

- (*a*) Variation of speed between no load and full load .... r.p.m.

Power.

(32) Indicated horsepower .... i.hp.

(33) Brake horsepower .... br. hp.

Capacity.

(34) Water evaporated per hour from and at 212 degrees (same as Item 22*a*) .... lb.

- (*a*) Percentage of rated boiler capacity developed (Item 34 ÷ Item 3 × 100) .... per cent

(35) Percentage of rated engine or turbine capacity developed (Item 32 ÷ Item 4 × 100) . . . . per cent

Economy Results.

(36) Coal as fired per i.hp. of engine per hour . . . . lb.
(37) Coal as fired per brake hp. of engine or turbine per hour . . . . lb.
 (*a*) Dry coal per i.hp. per hr . . . . lb.
 (*b*) Dry coal per brake hp-hr . . . . lb.
 (*c*) Dry coal per kw-hr . . . . lb.

(38) Heat units in coal consumed per i.hp. of engine per hour . . . . B.t.u.
(39) Heat units in coal consumed per brake hp. of engine or turbine per hour (Item 37 × Item 24) . . . . B.t.u.
 (*a*) Heat units consumed by engine (including auxiliaries) per i.hp-hr . . . . B.t.u.
 (*b*) Heat units consumed by engine or turbine (including auxiliaries) per brake hp-hr. (Item 26 ÷ Item 33) . . . . B.t.u.
 (*c*) Heat units consumed by engine per kw-hr . . . . B.t.u.

(40) Heat units in coal consumed per kw-hr . . . . B.t.u.
(41) Water evaporated per lb. of coal as fired . . . . lb.
 (*a*) Water evaporated per lb. of dry coal . . . . lb.
 (*b*) Equivalent evaporation from and at 212 deg. per lb. of dry coal . . . lb.
 (*c*) Equivalent evaporation from and at 212 deg. per lb. of combustible . . lb.

(42) Dry steam consumed by engine alone per i.hp-hr . . . . lb.
 (*a*) Dry steam consumed by auxiliaries per i.hp-hr . . . . lb.
 (*b*) Dry steam consumed by combined engine and auxiliaries per i.hp-hr . lb.

(43) Dry steam consumed by engine or turbine alone per brake hp-hr . . . . . lb.
 (*a*) Dry steam consumed by auxiliaries per brake hp-hr . . . . lb.
 (*b*) Dry steam consumed by combined engine or turbine and auxiliaries per brake hp-hr . . . . lb.

Efficiency Results.

(44) Thermal efficiency of plant referred to i.hp. [(2546.5 ÷ Item 38) × 100] . . . .
(45) Thermal efficiency of plant referred to brake hp. [(2546.5 ÷ Item 39) × 100] . . . .
 (*a*) Efficiency of boilers (Item 41*b* × 970.4 × 100 ÷ Item 24*a*) . . . .
 (*b*) Efficiency of engine referred to i.hp. [(2546.5 ÷ Item 39*a*) × 100] . . . .
 (*c*) Efficiency of engine or turbine referred to brake hp. [(2546.5 ÷ 39*b*) × 100] . . . .

Fuel Cost of Power.

(46) Cost of coal per ton of —— lb. . . . . dollars
(47) Cost of coal per i.hp-hr . . . . cents
(48) Cost of coal per brake hp-hr . . . . cents

HEAT BALANCE OF STEAM POWER PLANT.

	Per Lb. Coal as Fired.	Per Cent.
(49) Heat units in coal (same as Item 24)		
(50) Boiler losses		
(*a*) Loss due to evaporation of moisture in coal		
(*b*) Loss due to heat carried away by steam formed by the burning of hydrogen		
(*c*) Loss due to heat carried away in the dry flue gases		
(*d*) Loss due to carbon monoxide		
(*e*) Loss due to combustible in ash and refuse		
(*f*) Loss due to heating moisture in air		
(*g*) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for		
(*h*) Heat supplied steam-driven appliances for operating boilers less that recovered by heating feed water		
(*i*) Total boiler losses		
(51) Engine consumption		
(*a*) Radiation from steam pipe		
(*b*) Radiation from engine or turbine		
(*c*) Heat rejected to condenser		
(*d*) Heat withdrawn from engine receivers or turbine stages or other use than heating feed water		
(*e*) Heat lost by leakage of steam piping		
(*f*) Heat converted into work		
(52) Heat in steam supplied for purposes foreign to engine or turbine		
Totals (same as Item 49)		

SAMPLE DIAGRAMS.

(53) Sample indicator diagrams from each cylinder of engine. Also sample steam pipe diagrams...........

The boiler output (Item 49 — Item 50*i*) may be divided into		
(*a*) Heat units absorbed by water in boiler		
(*b*) Heat units absorbed by water in economizer		
The quantity representing the sum of Items 51*b*, *c*, and *f* may be divided according to the steam distribution into		
(*c*) Heat consumed by engine cylinders or turbine alone (including reheaters or jackets, if any), i.e., total heat supplied to engine or turbine alone less heat recovered therefrom by heating feed water		
(*d*) Heat consumed by steam-driven auxiliaries, i.e., total heat supplied to auxiliaries less heat recovered therefrom by heating feed water		
The same quantity may be divided according to the distribution of work done by engine or turbine into		
(*e*) Heat consumed in supplying power lost in friction of engine or turbine		
(*f*) Heat consumed in supplying frictional, electrical, or other losses of power delivered by engine or turbine shaft		
(*g*) Heat consumed in supplying useful power delivered by engine or turbine, whether mechanical, electrical, or otherwise		

Note: — In the case of pumping and air machinery plants add lines under the various items as follows:

For Item (13)

(*k*) Pressure in delivery main by gage....................lb. per sq. in.
(*l*) Vacuum or pressure in suction main by gage........................lb. per sq. in. or in. of mercury
(*m*) Correction for difference in elevation of the two gages...lb. per sq. in.
(*n*) Total head expressed in lb. pressure per sq. in..........lb. per sq. in.
(*o*) Total head expressed in ft.......................................ft.

For Item (20)

(*d*) Temperature of delivery.......................................deg.
(*e*) Total weight of water discharged, by measurement..............lb.
(*f*) Total weight of water discharged, by calculation from plunger displacement, corrected......................................lb.
(*g*) Total volume of air delivered, by measurement..............cu. ft.
(*h*) Total volume of air delivered, reduced to atmospheric pressure and temperature..cu. ft.

For Item (23)

(*j*) Weight of water discharged per hour, by measurement...........lb.
(*k*) Weight of water discharged per hour, by plunger displacement, corrected..lb.
(*l*) Volume of water or air delivered per hour, by measurement......cu. ft.
(*m*) Volume of air delivered per hour, reduced to atmospheric pressure and temperature................................cu. ft.

For Item (31)

(*b*) Length of pump stroke.......................................ft.

For Item (33)

(*a*) Water (or air) hp...hp.

For Item (35)

(*a*) Gal. of water discharged in 24 hr. as measured................gal.
(*b*) Volume of air delivered per minute, reduced to atmospheric pressure and temperature.............................cu. ft.

For Item (36)

(*a*) Dry coal per water (or air) hp-hr.............................lb.

For Item (39)

(*a*) Duty per 1,000,000 B.t.u......................................

For Item (45)

(*a*) Thermal efficiency of plant referred to water (or air) hp...........

For Item (48)

(*a*) Cost of coal per water (or air) hp........................dollars.

PRINCIPAL DATA AND RESULTS OF STEAM POWER PLANT TEST.

(1) Dimensions of boilers

(2) Dimensions of engine or turbine

(3) Date

(4) Duration .... hr.

(5) Boiler pressure .... lb. per sq. in.

(6) Throttle pressure .... lb. per sq. in.

(7) Pressure in receiver or stages .... lb. per sq. in.

(8) Vacuum in condenser .... in. of mercury

(9) Percentage of moisture in steam near throttle or number of degrees of superheating .... per cent or deg.

(10) Temperature of feed water entering boilers .... deg.

(11) Temperature of escaping gases .... deg.

(12) Force of draft .... in. of water

(13) Coal, as fired, per hour .... lb.

(14) Percentage of moisture in coal .... per cent

(15) Percentage of ash in coal .... per cent

(16) Water evaporated per hour .... lb.

(17) Equivalent evaporation per hour from and at 212 deg. .... lb.

(*a*) Equivalent evaporation per hour from and at 212 deg. per sq. ft. water heating surface .... lb.

(18) Steam consumed per hour by engine .... lb.

(19) Steam consumed per hour by engine or turbine and auxiliaries .... lb.

(20) Mean effective pressure in each cylinder of engine .... lb. per sq. in.

(21) Revolutions per minute .... r.p.m.

(22) Indicated horsepower .... i.hp.

(23) Brake horsepower * .... brake hp.

(24) Coal as fired per i.hp-hr. .... lb.

(25) Coal as fired per brake hp-hr. * .... lb.

(26) Steam per i.hp-hr. .... lb.

(27) Steam per brake hp-hr.* .... lb.

(28) Heat consumed per i.hp-hr. .... B.t.u.

(29) Heat consumed per brake hp-hr.* .... B.t.u.

* For pumping engine (water or air) use Water or Air hp. in place of Brake hp.

APPENDIX F

MISCELLANEOUS CONVERSION FACTORS

1 POUND PER SQUARE INCH =

2.0355	inches of mercury at 32° F.
2.0416	inches of mercury at 62° F.
2.309	feet of water at 62° F.
0.07031	kilogram per square centimeter
0.06804	atmosphere
51.7	millimeters of mercury at 32° F.

1 FOOT OF WATER AT 62° F. =

0.433	pound per square inch
62.355	pounds per square foot
0.883	inch of mercury at 62° F.
821.2	feet of air at 62° F. and barometer 29.92

1 INCH OF WATER 62° F. =

0.0361	pound per square inch
5.196	pounds per square foot
0.5776	ounce per square inch
0.0736	inch of mercury at 62° F.
68.44	feet of air at 62° F. and barometer 29.92

1 FOOT OF AIR AT 32° F. AND BAROMETER 29.92 =

0.0761	pound per square foot
0.0146	inch of water at 62° F.

1 INCH OF MERCURY AT 62° F. =

0.4912	pound per square inch
1.132	feet of water at 62° F.
13.58	inches of water at 62° F.

1 ATMOSPHERE =

760.0	millimeters of mercury at 32° F.
14.7	pounds per square inch
29.921	inches of mercury at 32° F.
2116.0	pounds per square foot.
1.033	kilograms per square centimeter

1 MILLIMETER = 0.03937 inch

1 CENTIMETER = 0.3937 inch

1 METER = 39.37 inches

1 METER = 3.2808 feet

1 SQUARE METER = 10.764 square feet

1 LITER =

61.023	cubic inches
0.264	U. S. gallons

1 GRAM =

1	cubic centimeter of distilled water
15.43	grains troy
0.0353	ounce

1 KILOGRAM =

2.20462 pounds avoirdupois

APPENDIX G

EQUIVALENT VALUES OF ELECTRICAL AND MECHANICAL UNITS

1 Myriawatt =

10 kilowatts
10,000 watts
13.41 horsepower
13.597 cheval-vapeur
13.597 pferde-kraft
26,552,000 foot pounds per hour
8,605,000 gram calories per hour
3,670,000 kilogram meters per hour
34,150 B.t.u. per hour
1.02 boiler horsepower

1 Horsepower =

745.7 watts
0.7457 kilowatt
0.07457 myriawatt
1.0139 cheval-vapeur
1.0139 pferde-kraft
33,000 foot pounds per minute
641,700 gram calories per hour
273,743 kilogram meters per hour
2,547 B.t.u. per hour

1 Joule =

1 watt second
0.10197 kilogram meter
0.73756 foot pound
0.239 gram calorie
0.0009486 B.t.u.

1 B.t.u. =

1054 watt seconds
777.5 foot pounds
107.5 kilogram meters
0.0003927 horsepower hour

1 Kilowatt =

0.1 myriawatt
1000 watts
1.341 horsepower
1.3597 cheval-vapeur
1.3597 pferde-kraft
2,655,200 foot pounds per hour
860,500 gram calories per hour
367,000 kilogram meters per hour
3,415 B.t.u. per hour
0.102 boiler horsepower

1 Cheval-Vapeur or Pferde-Kraft =

75 kilogram meters per second
0.07354 myriawatt
0.7357 kilowatt
0.9863 horsepower
32,550 foot pounds per minute
632,900 gram calories per hour
2,512 B.t.u. per hour

1 Foot Pound =

1.3558 joules
0.13826 kilogram meter
0.001286 B.t.u.
0.03241 gram calorie
0.000000505 horsepower hour

1 Kilogram-Meter =

7.233 foot pounds
9.806 joules
2.344 gram calories
0.0093 B.t.u.

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