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DESIGN AND SHOP PRACTICE

NUMBER 72

PUMPS AND CONDENSERS STEAM AND WATER PIPING

CONTENTS

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CHAPTER I

PUMPS

Pumps are used for various purposes in connection with power and heating plants, the most important use being for the feeding of boilers, for the return of condensation from heating systems, for tank and fire service, and as part of the condensing outfit. The action of a pump is best described by reference to one of the simplest types, known as the direct-acting steam pump. This pump is shown in elevation in Fig. 1 and in section in Fig. 2. The "water-end" is at the right, and consists of a cylinder H, a piston G, valves V and W, and airchamber A. The water piston is actuated by a steam piston P, attached to a common piston rod R, as shown. The steam piston moves in cylinder C, and is forced backward and forward by steam pressure, which is controlled by the slide-valve E, the same as in a steam engine. Motion is transmitted to the valve E by means of the piston rod through the bell-crank F. The inlet connection is at S, and communicates with the space I just below the lower valves V. The discharge is at D.

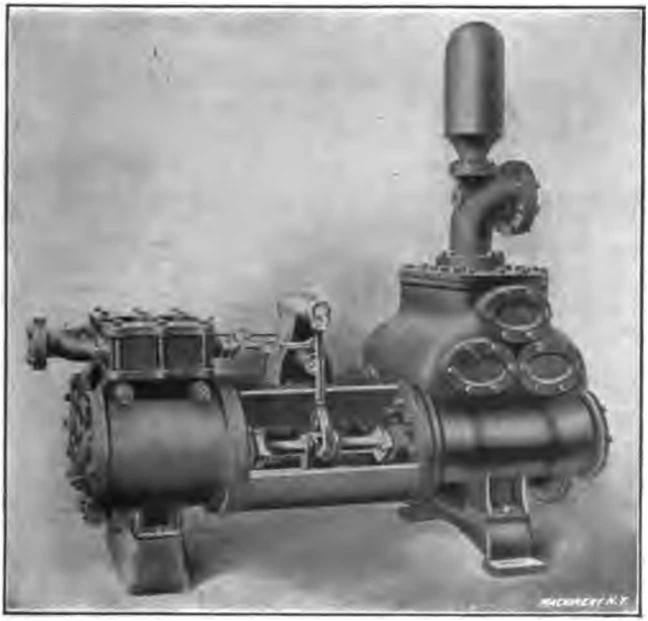

Fig. 1. Direct-acting Steam Pump

The action of the pump is as follows: Assume that the piston is moving toward the left, as indicated by the arrow; this causes a pressure in the left-hand end of the cylinder which raises valve *W* and allows the water to flow into the upper chamber, and thus outward through the delivery pipe *D*. In the meantime a partial vacuum is formed in the other end of the cylinder, which causes the valve *V* to lift, due to the greater pressure in the space *I*, and the cylinder space at the right of the piston is thus constantly kept full of water when the piston moves toward the left. At the end of the stroke the steam valve reverses, and the piston moves toward the right, forcing out the water in front of it into the delivery chamber, and drawing in a supply behind it for delivery at the next stroke, as already described.

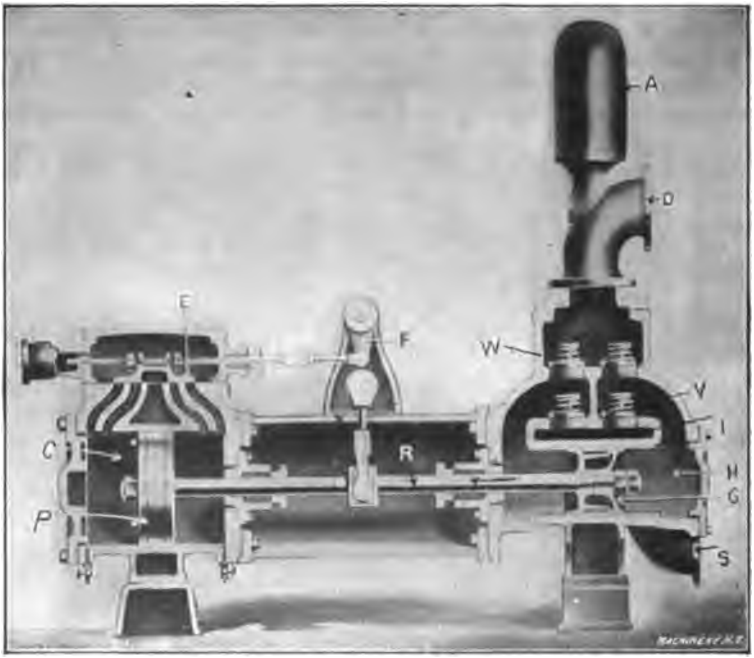

Fig. 2. Section of Direct-acting Steam Pump

The air-chamber *A* tends to equalize the pressure at the instant the piston changes its direction of travel, thus causing a steadier flow of water through the discharge pipe. This is brought about by the cushion of air which is compressed in the upper part of the chamber, and which exerts a momentary pressure after the piston stops. The water end of the pump above described is typical of the cylinder pump, whether driven by a steam piston or by other means. Centrifugal and rotary pumps differ in principle from the above, and will be described in detail later.

Suction is the term commonly employed to denote a lowering of the pressure within the inlet pipe below that acting upon the surface of the water with which the "suction pipe" connects. For example, let the suction pipe of a pump connect with a reservoir of water open to the atmosphere. If now the piston moves forward, a vacuum will be formed back of it which will at once be filled by the water which is being forced in through the suction pipe and inlet valves, due to the pressure of the atmosphere acting upon the surface of the reservoir.

A column of water, 1 inch square and 1 foot high, weighs 0.433 pound when at a temperature of 60 degrees F. The normal pressure of the atmosphere at sea level is 14.7 pounds per square inch. Hence, atmospheric pressure will raise water to the height of $14.7 \div 0.433 = 34$ feet. in round numbers, in a pipe connected with a perfect vacuum. From the foregoing it is evident that what is commonly known as suction is in reality the forcing of water under a higher pressure (atmospheric in most cases) into the cylinder, which is under a partial vacuum due to the forward movement of the piston.

In actual practice it is not possible to produce a perfect vacuum back of the piston owing to imperfections which allow the leakage of air into the cylinder. With the very best construction it is possible to reduce the pressure within the cylinder to about 2.5 pounds per square inch, leaving an unbalanced pressure of 12.2 pounds, which will raise a column of water in the suction pipe to $12.2 \div 0.433 = 28$ feet. The average pump in good working order will lift water by suction only about from 25 to 26 feet, with a fair degree of economy.

In cases where a pump takes its supply directly from the mains of a town or city water system, some means should be provided for equalizing the pressure, as this is constantly changing due to the water being drawn off at different points. When the water comes to the pump under varying pressure, an increase of pressure in the suction pipe is equivalent to a decrease in the pressure pumped against, and this condition is, therefore, likely to vary the speed of the pump, even when the head pumped against remains constant. This is especially noticeable in boiler feeding, where the pumps are set to run at a uniform speed against a constant pressure, thus maintaining the water line at a given point. If the speed of the pump varies, due to changes of pressure in the suction or supply pipe, the water level in the boilers is liable to fluctuate rapidly and must be carefully watched.

This condition is commonly overcome in two ways, one of which is to use a pressure reducing valve in the suction pipe of the pump, which maintains a constant pressure at this point; the other method is to employ a pump governor which maintains a constant speed under widely varying pressures. The effect of a fluctuating suction pressure is sometimes overcome to a considerable extent by throttling the supply by partially closing the valve and causing the pump to "draw" the water through it. The most satisfactory method, however, is the employment of a governor, which makes the action of the pump independent of the pressures either upon the suction or forcing side of the piston.

When pumping hot water, it is usually necessary to place the pump low enough for the water to flow into it by gravity. This is because water at high temperatures will break into steam under low pressures. Theoretically, water at a temperature of 200 degrees can be raised 8 feet by suction, but in practice it has been found safer to bring the water to the pump by gravity when the temperature approximates 190 degrees. A pump built for hot-water service also requires special packing for the valves and piston in the water end.

Pressure Head

The pressure against which a pump forces the water is usually expressed in "feet head." For example, a pump feeding a boiler against a pressure of 100 pounds per square inch is operating under a head of 100 ÷ 0.433 = 231 feet, that is, each pound pressure per square inch against which the water is forced is equivalent to lifting a column of water 1 inch square and 2.31 feet high. From the above, it is evident that

pressure per square inch in pounds ÷ 0.433 = head in feet, and
head in feet × 0.433 = pressure per square inch in pounds.

In determining the pressure head or total height to which the water must be raised, the distance must be taken from the surface of the water in the reservoir from which it is drawn to the point of discharge. The same power is required to raise water by suction as to force it, and the height of the pump above the water does not enter separately into the calculation at all, provided it is not more than 28 feet. This is made plain by a practical example. Assume that a pump is raising water by suction 18 feet, and discharging it at this elevation without forcing it at all, all the work being done on the suction side of the piston. When water is raised to this height by suction, the air pressure in the suction pipe is reduced to 14.7 — (18 × 0.433) = 6.9 pounds per square inch. This leaves an unbalanced pressure upon the other side of the piston equal to 14.7 — 6.9 = 7.8 pounds per square inch. The effect is therefore, the same as if the pump were forcing the water against this pressure with the water flowing into the cylinder by gravity. To illustrate this, take a case where the water flows to the pump by gravity, and is raised to a height of 18 feet. Here the pressure per square inch against which the piston must work is 18 × 0.433 = 7.8 pounds, the same as in the case above. Hence it is evident that the work done by the pump is the same whether the water is raised a given distance by suction or forced to the same height by the pressure of the piston.

Friction Head

In what has been said regarding the pressure head required for raising water to a given height, or forcing it against a pressure, as in boiler feeding, no reference has been made to the resistance due to the friction of the water against the sides of the pipes. In computing the required power for operating a pump, and the pipe sizes in a boiler plant where the distances are short, no account is taken of this, but when water is moved long distances through pipes, this must be taken into consideration. For convenience in making computations, tables have been prepared giving the frictional resistance for pipes of different diameters and different velocities of flow of water. A portion of such a table is given herewith (see Table I) for purposes of illustration. More complete tables can be found in any engineer's handbook on hydraulics.

Table I gives the velocity in feet per second, and the friction head in pounds per square inch for pipes from 1 to 8 inches in diameter and 100 feet in length. As the frictional resistance is proportional to the length, the friction head for any other length is easily found as follows: From Table I the friction head for a 4-inch pipe discharging 300 gallons per minute is 2.7 pounds per square inch. For a pipe 800 feet long this would be 8 × 2.7 = 21.6 pounds, and for a pipe 50 feet long, 0.5 × 2.7 = 1.35 pounds per square inch.

TABLE I. PIPE SIZES, CAPACITIES, VELOCITY AND FRICTION HEAD
Velocity in feet per second. Friction head in pounds per square inch per 100 feet

Gallons per Minute	1-inch		2-inch		3-inch		4-inch		6-inch		8-inch	
	Velocity	Friction Head	Velocity	Friction Head	Velocity	Friction Head	Velocity	Friction Head	Velocity	Friction Head	Velocity	Friction Head
25	10.2	19.0	....	....	....	....	....	....	....	....	....	....
30	12.3	27.5	....	....	....	....	....	....	....	....	....	....
35	14.3	37.0	....	....	....	....	....	....	....	....	....	....
40	16.3	48.0	4.1	1.6	....	....	....	....	....	....	....	....
45	....	....	4.6	2.0	....	....	....	....	....	....	....	....
50	....	....	5.1	2.4	....	....	....	....	....	....	....	....
75	....	....	7.7	5.3	....	....	....	....	....	....	....	....
100	....	....	10.2	9.5	4.5	1.3	....	....	....	....	....	....
125	....	....	15.8	14.9	5.7	2.0	....	....	....	....	....	....
150	....	....	16.3	21.2	6.8	2.8	....	....	....	....	....	....
150	....	....	17.1	28.1	7.9	3.8	....	....	....	....	....	....
175	....	....	....	....	9.1	5.0	4.5	1.0	....	....	....	....
200	....	....	....	....	11.3	7.8	5.1	1.2	....	....	....	....
250	....	....	....	....	13.6	11.2	6.4	1.9	....	....	....	....
300	....	....	....	....	15.9	15.2	7.7	2.7	....	....	....	....
350	....	....	....	....	....	....	8.9	3.6	4.0	0.5	....	....
400	....	....	....	....	....	....	10.2	4.7	4.5	0.6	....	....
450	....	....	....	....	....	....	11.5	6.0	5.1	0.8	....	....
500	....	....	....	....	....	....	....	....	5.7	1.0	3.2	0.2
750	....	....	....	....	....	....	....	....	8.5	2.2	4.8	0 5
1000	....	....	....	....	....	....	....	....	11.3	3.9	6.4	0.9
1500	....	....	....	....	....	....	....	....	....	....	9.6	2.1

The friction heads given in Table I are for straight runs of pipe; when elbows and valves are introduced, the resistance is increased. In computing the friction head under these conditions it is sufficiently accurate to assume the resistance due to each elbow as increasing the length of the pipe 60 diameters, and the resistance due to each globe valve as increasing the length 90 diameters.

"Slip" is the term used to denote the difference between the theoretical capacity of a pump and the actual, and is usually expressed as a percentage of the theoretical or calculated discharge. Slip is due partly to leakage around the piston and valves, but more especially to the results of too high speed. When a pump runs too fast, the piston speed is so high that the water cannot flow through the valves fast enough to completely fill the cylinder; hence the actual discharge is less than the theoretical. Another effect of high speed is its action upon the seating of the valves. These do not act instantaneously, but require a certain length of time to reach their seats when the piston reverses its direction. When a pump runs at high speed the piston will move a considerable distance while the valves are descending to their seats, and water will flow back into the cylinder from the discharge chamber, thus reducing the volume actually pumped at each stroke. The average slip in pumps of different kinds at medium speeds is given in Table II.

Pump Valves

The valves in the usual type of pump are carried by two plates or decks, the inlet valves being below the discharge valves, as shown in Fig. 2. The valves used in practically all pumps, except pumping engines, are of the flat disk type shown in Fig. 3, and consist of a ring of special material pressed into a metal casing or plate, which slides upon a bolt screwed into a bridge across the port opening, as shown. A conical spring is employed to hold the valve firmly to its seat, the spring being held in position by the head of the bolt.

TABLE II. PERCENTAGE OF SLIP IN PUMPS

Type of Pump	Percentage of Slip	Actual Discharge expressed as a Percentage of the Theoretical Value
Boiler feed pumps..............	20	80
Water works pumps............	5	95
Small centrifugal pumps........	65	35
Medium centrifugal pumps.....	45	55
Large centrifugal pumps........	20	80

In order to reduce the slip of a pump, it is customary to use several small valves instead of a single large one of equivalent area, and to secure a full port opening it is necessary for the valve disk to rise a distance equal to one-quarter of its diameter from the seat. From this it is evident that the travel of the valve and the consequent wear and jar is much less with disks of small size. For a quick-running pump for ordinary service the valves should not exceed 4 to 4½ inches in diameter, and their combined area should not be less than 35 per cent of the area of the water piston or plunger.

Size of Suction and Delivery Pipes

The area of the suction pipe is based upon the velocity of flow through it, and may be found by means of the following formula:

$$b = \frac{a \times S}{V} \tag{1}$$

in which

b = area of suction pipe in square inches,
a = area of piston or plunger in square inches,
S = piston speed in feet per minute,
V = velocity of flow through suction pipe in feet per minute. This velocity equals:

200 for 25 feet in length,
180 for 50 feet in length,
150 for 100 feet in length,
125 for 125 feet in length.

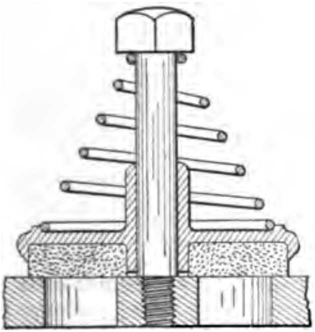

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Fig. 3. Disk-type Pump Valve

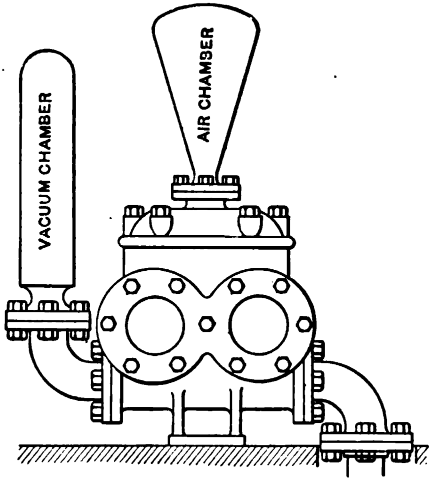

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Fig. 4. Pump with Vacuum Chamber

These velocities allow for two or three elbows, a stop valve and a foot valve.

The area of the delivery pipe may be found by the same formula by substituting 300 for the value of V in all cases. This is because in the suction pipe the pressure acting to force the water through it is practically constant and never exceeds that of the atmosphere, hence any increase in frictional resistance due to a greater length of pipe must be allowed for by assuming a lower velocity of flow through it. In the case of the delivery pipe a constant rate of flow may be maintained by increasing the steam pressure or by using a larger steam cylinder.

Speed

For pumps of usual construction, having water cylinders less than 10 inches in diameter, the best results are obtained with piston speeds of from 30 to 40 feet per minute for continuous operation. When the pump is only used occasionally for short lengths of time, the speed may be increased to 60 or 80 feet per minute without undesirable results. The speed of a pump is often given in strokes per minute instead of in feet, as the reversal of the piston at the end of the stroke is what is detrimental to the pump at high speeds. For elevator service and similar purposes, the maximum speed of a direct-acting pump should not exceed 60 strokes per minute, and for boiler feeding, it should be kept down to from 25 to 35 strokes.

Area of Steam and Water Cylinders

The steam cylinders of nearly all direct-acting pumps are of larger diameter than the water cylinders; as a rule they are from 25 to 50 per cent greater in diameter. In pumps employed for boiler feeding, the ratio of diameter of steam to water cylinder is usually about 1.25. In pumps used exclusively for low-pressure work, that is, for moving large volumes of water under low pressures, the ratio is less, and for high-pressure work it is considerably more. The equations given below will be found useful for proportioning the steam and water cylinders to meet different conditions.

$$P = \frac{p}{A \div a} \quad (2); \qquad p = \frac{A \times P}{a} \quad (3)$$

$$A = \frac{a \times p}{P} \quad (4); \qquad a = \frac{A \times P}{p} \quad (5)$$

in which

P = steam pressure per square inch,
p = water pressure per square inch (total head),
A = area of steam piston in square inches,
a = area of water piston in square inches.

The theoretical capacity of a pump in cubic feet per minute may be found by multiplying the area of the water piston in square feet by the piston speed in feet per minute, or by multiplying the piston displacement in cubic feet by the number of strokes per minute. To obtain the actual capacity these results must be corrected for slip. While the above results are obtained in cubic feet, they may be changed to other denominations by the use of the factors given below:

Cubic feet × 7.5 = gallons,
Cubic feet × 62 = pounds,
Gallons × 8.3 = pounds.

The power required for operating a pump may be found by either of the following equations, depending upon the data at hand.

$$\text{H. P.} = \frac{W \times H}{33{,}000} \qquad (6)$$

$$\text{H. P.} = \frac{a \times p \times S}{33{,}000} \qquad (7)$$

$$\text{H. P.} = \frac{A \times P \times S}{33{,}000} \qquad (8)$$

in which

H. P. = delivered horsepower,
W = pounds of water pumped per minute,
H = vertical height to which it is raised, in feet,
A = area of steam piston, in square inches,
a = area of water piston, in square inches,
P = steam pressure, in pounds per square inch,
p = water pressure, in pounds per square inch,
S = piston speed, in feet per minute.

Equation (6) applies to any form of pump, as the power is based entirely upon the weight of water and the height to which it is raised; the piston areas and the pressures do not enter into the computation. Equations (6) and (7) are for short pipe connections and do not take into account the friction head. If the discharge is of considerable length, the power required to overcome the friction should be computed from data given in Table 1, and added to the results given by equations (6) and (7). Equation (9) may be used for determining the horsepower due to friction in the discharge pipe:

$$\text{H. P.} = \frac{F \times c \times V}{33{,}000} \qquad (9)$$

in which

F = friction head in pounds per square inch for the given length of pipe and velocity of flow through it,
c = area of pipe, in square inches,
V = velocity of flow through pipe, in feet per minute.

Air and Vacuum Chambers

Air chambers, as already stated, are used on pumps for the purpose of causing a steady discharge of water and allowing the pump to run at a higher rate of speed. The location and general form of an air-chamber is shown at A in Fig. 2. The air which it contains is compressed during each stroke. When the piston stops momentarily at the end of the stroke, the air expands to a certain extent, and tends to produce a gradual stopping of the flow of water, thus permitting the valves to seat easily without shock or jar.

In the case of single-cylinder boiler feed pumps, and those employed for elevator service, the volume of the air chamber should be at least three times that of the piston displacement. For duplex pumps it should not be less than twice the piston displacement of one of the pumps. In the case of high-speed pumps, this ratio should be increased to 5 or 6.

The action of a vacuum chamber is the reverse of that of an air chamber. When the column of water in the suction pipe is once set in motion, it is important to keep it in full motion, and when it is stopped, it should be done gradually. This is accomplished by placing a vacuum chamber on the end of the suction pipe as shown in Fig. 4. The moving column of water compresses the air in the chamber at the end of the stroke, and when the piston starts again, the air expands and thus aids in setting the column of water in motion once more. The vacuum chamber is usually made the same size as the suction pipe and of considerable length, rather than of large diameter and short.

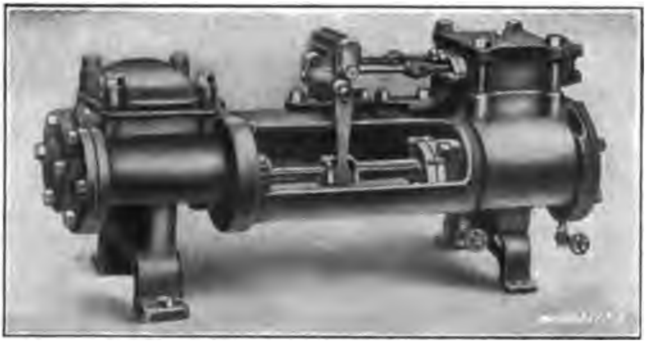

Fig. 5. Low-pressure Steam Pump

Types of Pumps

Fig. 1 shows a duplex piston pump designed for water pressures up to 150 pounds. The two pumps are placed side by side, and so combined that one piston actuates the steam supply for the other, after which it finishes its own stroke and waits for its valve to be acted upon by the other pump before it can renew its motion. This pause allows the water valves to seat quietly, and prevents any harshness of

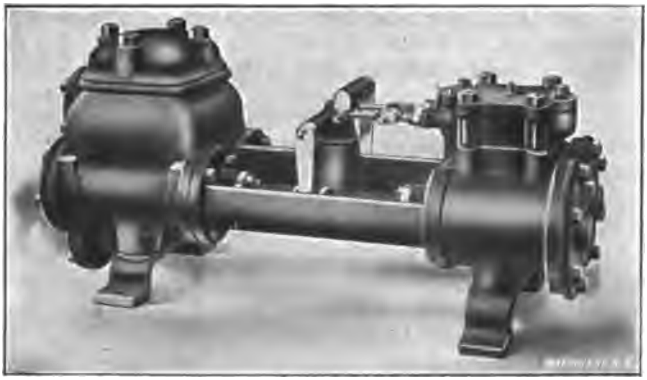

Fig. 6. Heavy Pattern Duplex Piston Pump

motion. This particular pump is made in sizes from 2 x 1¼ x 2¾ inches up to 16 x 9¼ x 10 inches, in which the first dimension is the diameter of the steam cylinder, the second, the diameter of the water cylinder, and the third, the length of the stroke. The maximum capacity varies from 4.4 to 614 gallons per minute.

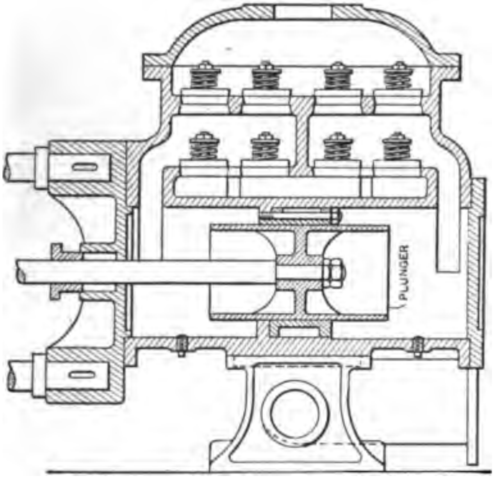

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Fig. 7. Section of Plunger-type Pump

The pump shown in Fig. 5 is designed especially for use in apartment houses and private buildings, where low-pressure steam heating systems are in use, and where pumps are required to run with a low steam pressure. This condition requires a larger steam piston for a given size of water piston than is furnished in pumps of regular pattern. This pump is made in sizes ranging from 3 x ¾ x 3 inches to 9 x 3¼ x 10 inches with corresponding capacities of 1.5 to 70 gallons per minute.

Fig. 8. Outside Packed Plunger Pump

Fig. 6 shows a heavy pattern duplex piston pump especially adapted to boiler feeding, although the larger sizes are used for fire purposes and general service work. The water ends are made to carry a working pressure of 150 pounds, and the pumps are made in sizes ranging from 2 x 1¼ x 2¾ inches to 10 x 6 x 10 inches with corresponding capacities of 3 to 200 gallons per minute.

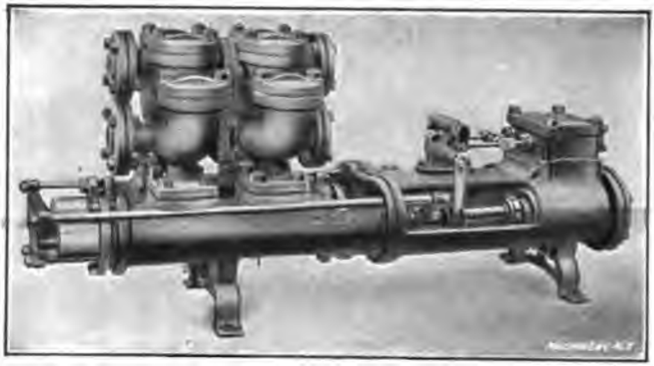

Fig. 9. Pot Valve Pressure Pump

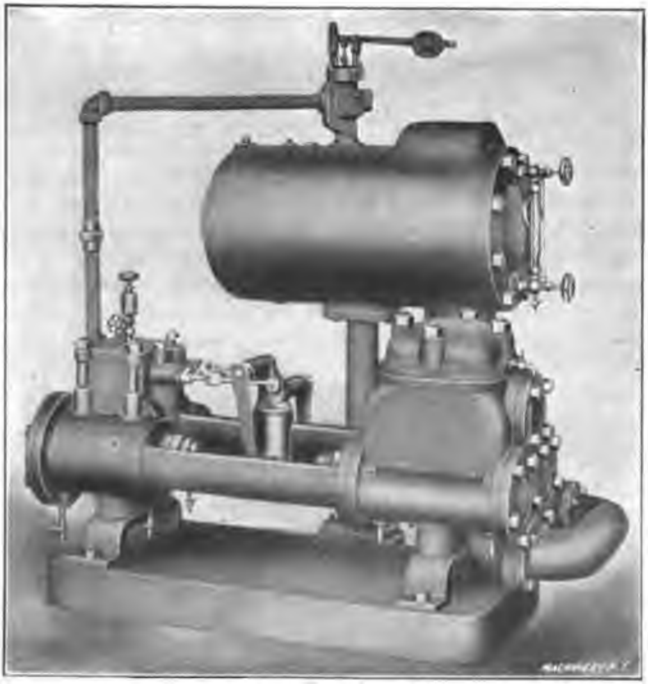

Fig. 10. Automatic Pump and Receiver

The pumps thus far described have been of the piston type shown in section in Fig. 2. A different design, known as a plunger pump, is illustrated in Fig. 7. In this case the piston is replaced by a plunger working in a renewable bushing, which is more easily replaced than a cylinder lining. For this reason plunger pumps are often preferred when the water is gritty, or when it for any other reason cuts out the packing rapidly.

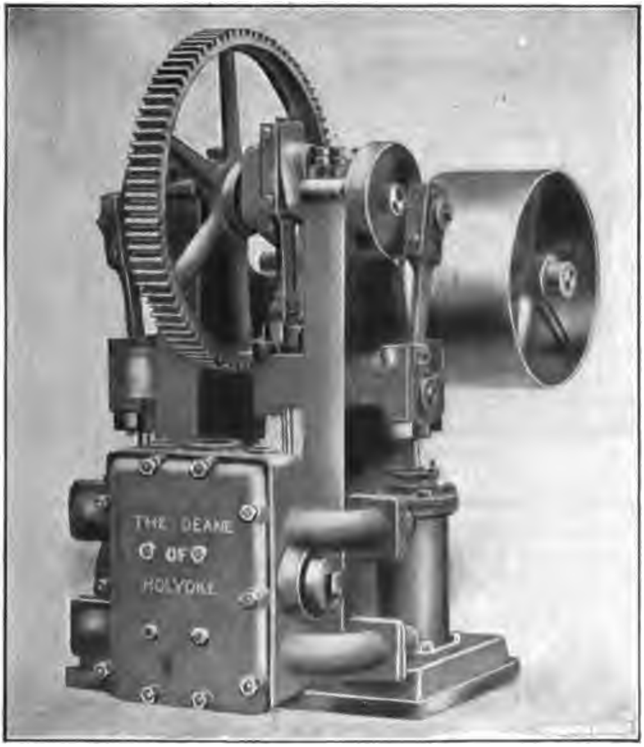

Fig. 11. Triplex Power Pump

The outside packed plunger pump, shown in Fig. 8, is designed especially for heavy service, and for use where the water carries large quantities of grit or other foreign matter of a similar nature. It is of heavy construction, being built for water pressures up to 250 pounds per square inch. One important feature of this design is the arrange-

TABLE III. CYLINDER DIAMETERS OF COMPOUND STEAM PUMPS

Diameter, High-pressure Cylinder, Inches	Diameter, Low-pressure Cylinder, Inches	Resulting Ratio of Expansion	Diameter, High-pressure Cylinder, Inches	Diameter. Low-pressure Cylinder, Inches	Resulting Ratio of Expansion
6	10	2 8	12	18	2.3
7	12	2.9	14	20	2.0
8	12	2.3	16	24	2.3
9	14	2.4	18	30	2.8
10	16	2.6	..	..	...

ment for packing the plungers from the outside, as the name implies. These pumps are especially adapted to boiler feeding, and are built in sizes ranging from 95 to 4500 boiler horsepower capacity.

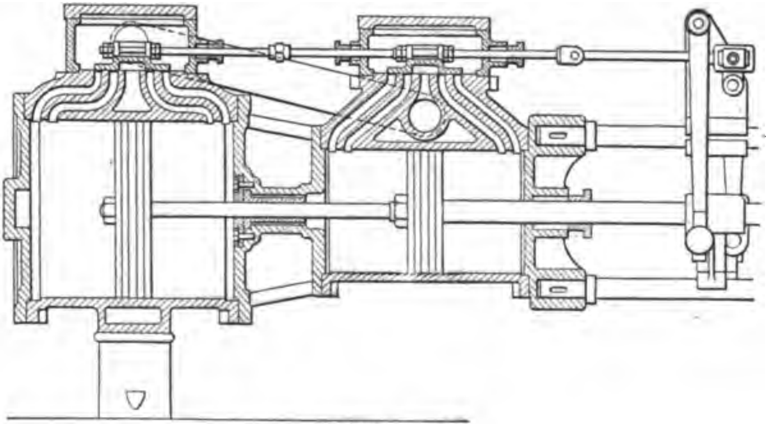
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Fig. 12. Section of Compound Steam Pump

The pot valve pressure pump shown in Fig. 9 is extensively used for boiler feeding, it being designed for a working pressure of 300 pounds. It has four water cylinders and four single-acting plungers which are packed from the outside. The larger sizes are extensively used for elevators and hydraulic presses.

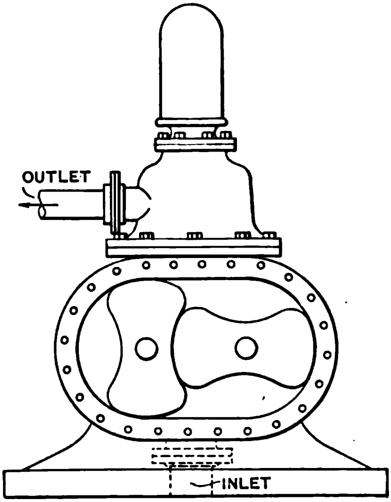

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Fig. 13. General Construction of Rotary Pump

Fig. 14. Impeller of Centrifugal Pump

The outfit illustrated in Fig. 10 is used for automatically returning to the boilers the condensation from low-pressure heating systems, drying cylinders,, steam jackets, etc. A float within the receiver maintains a constant water level, starting and stopping the pump as required by means of lever connections between the float and a balanced valve in the steam supply pipe.

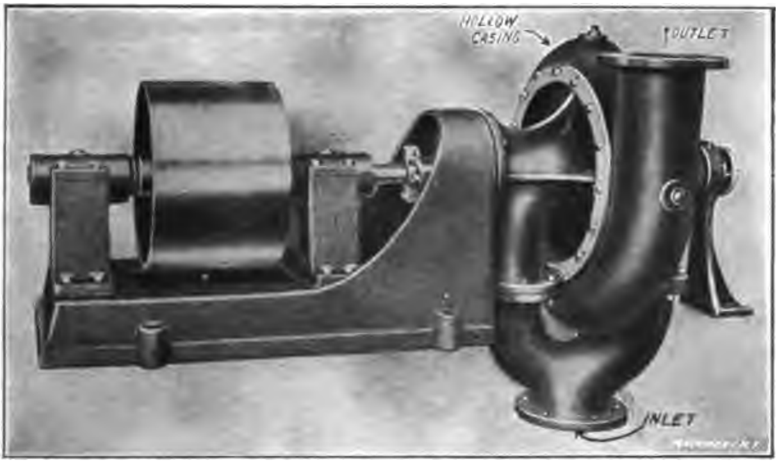

Fig. 15. Double Inlet Centrifugal Pump

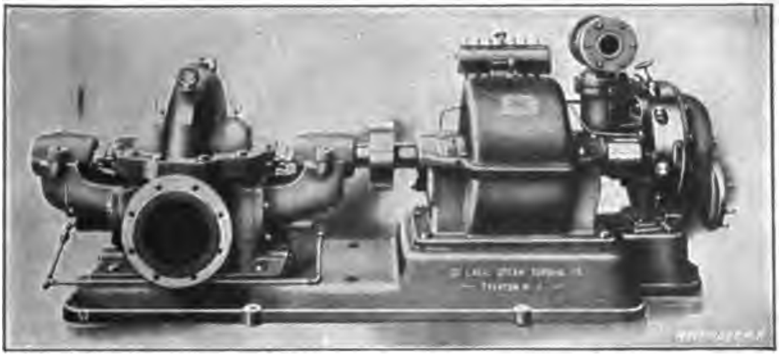

Fig. 16. Steam-turbine Driven Centrifugal Pump

Fig. 17. Impellers of Turbine Pump

Triplex power pumps, (Fig. 11) are used in place of the direct-acting type where it is desired to operate them by belting from a line of shafting or from an engine shaft. Power pumps are not so well adapted to boiler feeding as steam pumps, because they run at a constant speed, and any variation in the amount of water required must be cared for by a relief valve in the discharge pipe, while the speed of a direct-acting steam pump may be adjusted to meet varying requirements.

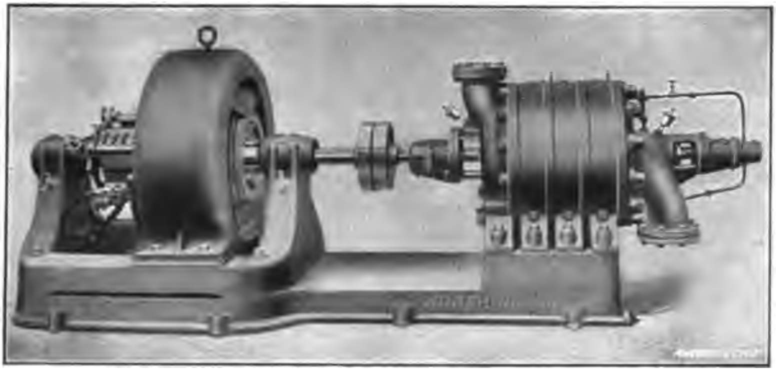

Fig. 18. Turbine Pump

The ordinary form of direct-acting steam pump does not use steam like a steam engine, expansively, but takes it at boiler pressure for the full stroke. This makes it wasteful in the use of steam, so that with the larger sizes it is customary in many cases, where the steam pressure runs from 75 to 100 pounds, to use a compound pump. By doing this a saving of from 20 to 35 per cent is made with non-condensing pumps, and of from 25 to 40 per cent with condensing pumps. A common arrangement of the steam cylinders is shown in Fig. 12, the high-pressure cylinder, being at the right, and the low-pressure at the left. Steam is first admitted to the high-pressure cylinder, and from here it is exhausted into the steam chest of the low-pressure cylinder through a side pipe. The effect of this is to use the steam expansively, the ratio of expansion commonly running from 2 to 4. Table III gives the sizes of cylinders commonly used for compound pumps and the resulting ratios of expansion.

Fig. 19. Diffusion Vanes of Turbine Pump

Rotary Pumps

Pumps of the rotary type are employed for lifting and forcing water under low heads, and are somewhat more efficient than the direct-acting piston pump. Rotary pumps are driven by means of belts from line shafting, by gearing, and by direct-connected motors of various kinds. There are several designs of this type of pump in common use, one of which is shown in section in Fig. 13. This contains two *impellers*, each carried on a steel shaft running in bearings outside of the cylinder. These shafts are connected by gearing, and power is applied by means of belting to one of the shafts. The suction pipe is at the bottom, and the delivery at the top, the water being carried by the impellers which are always in contact as they revolve.

Centrifugal Pumps

Centrifugal pumps are used largely for circulating the water through condensers in turbine power plants, for forced hot-water circulation in certain forms of heating, and for many other purposes where large volumes of water are to be handled quickly. A common form of double-inlet centrifugal pump is shown in Fig. 15, and consists of a hollow casing inside of which is a revolving fan or impeller of the general form shown in Fig. 14. When in action, the water enters the opening at the center of the impeller, and is thrown outward into the casing, partly by the pressure of the blades, and partly by centrifugal force.

Pumps of this form are most efficient when working under pressure heads of from 20 to 30 feet, but may be so designed that lifts up to 500 feet or more may be obtained with a good degree of efficiency. The capacity of a centrifugal pump depends upon the size and speed of the impeller and height of lift. A centrifugal pump driven by a direct-connected steam turbine is shown in Fig. 16.

Turbine Pumps

A turbine pump is a centrifugal pump of slightly different design from the one just described. The turbine pump shown in Fig. 18 is driven by an electric motor, and contains four impellers mounted on a single shaft as shown in Fig. 17, these running between diffusion vanes of the form shown in Fig. 19. The principle of operation is practically the same as in the centrifugal pump already described. Turbine pumps are successfully employed for water works, elevator service, boiler feeding, hydraulic mining, fire service, etc.

CHAPTER II

CONDENSERS

The purpose of attaching a condenser to a steam engine is to obtain a reduction in the back-pressure, due to the formation of a partial vacuum in the chamber into which the engine exhausts. The effect of a condenser is either to increase the power of an engine at a given steam consumption, or to reduce the steam consumption at a given power; this matter has been taken up in detail in MACHINERY'S Reference Series No. 70, "Steam Engines."

There are four general types of apparatus commonly employed for condensing the exhaust steam and producing a vacuum in the engine cylinder or exhaust pipe, known as *surface*, *jet*, *barometric* or *siphon*, and *atmospheric* condensers. These, in turn, each comprise two or three sub-divisions. For example, the surface condenser is made both vertical and horizontal in form; the jet condenser is made according to three different designs, the horizontal double-acting, the vertical single-acting, and the duplex; and the barometric condenser is made in two forms, the nozzle and the spray type.

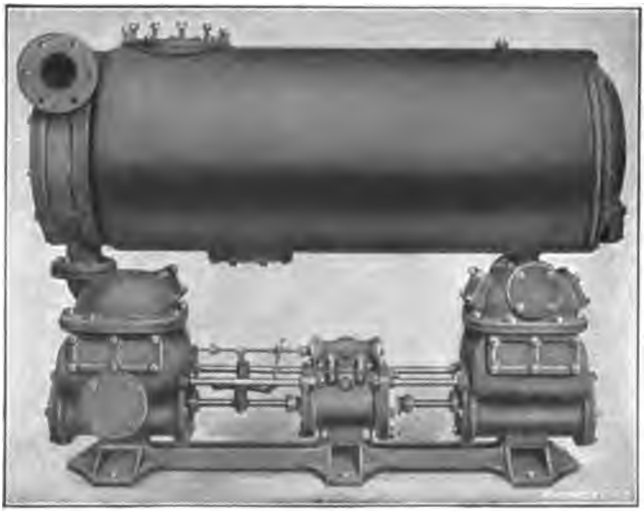

Fig. 20. Common Form of Surface Condenser

Surface Condensers

A common form of surface condenser, together with its air and circulating pumps, is shown in elevation in Fig. 20, and in section in Fig. 21. The essential parts of this apparatus are a condensing cham-

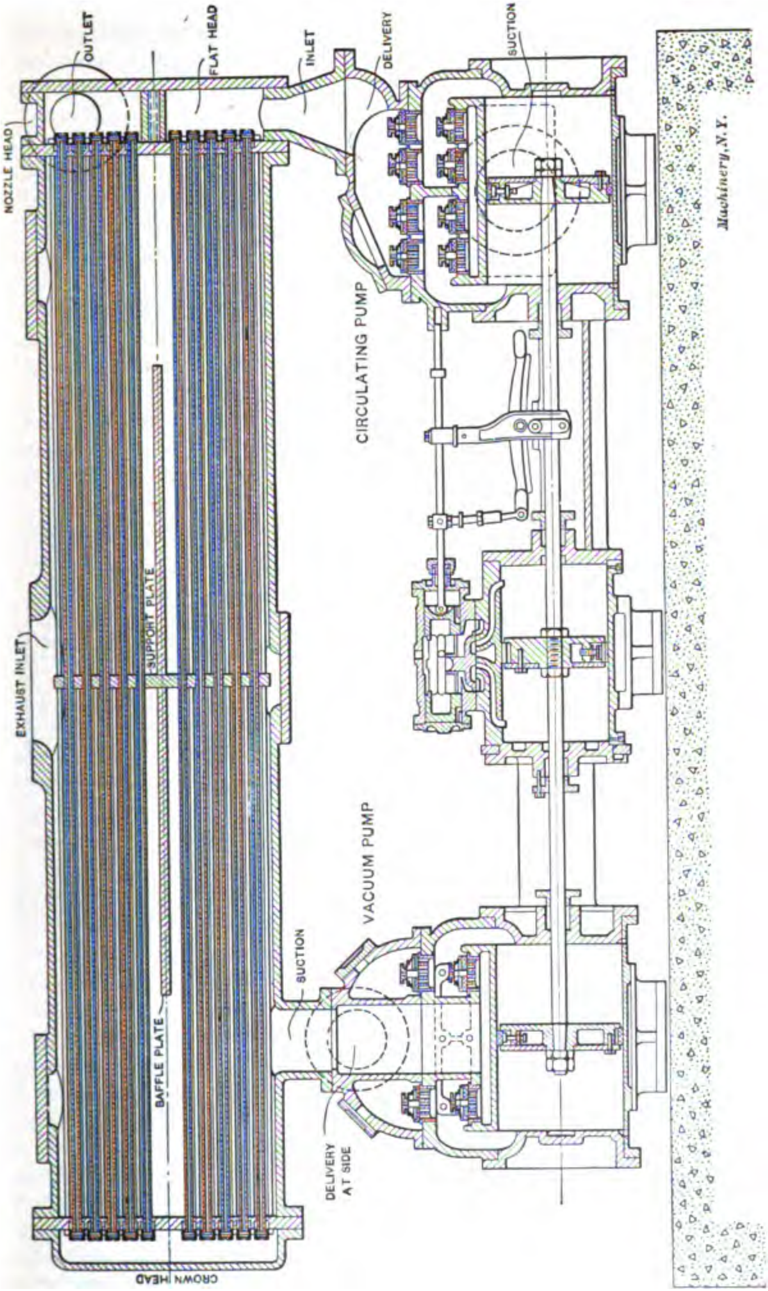

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Fig. 21. Section of Surface Condenser

ber, nearly filled with horizontal brass or copper tubes connecting with special chambers at each end, separated from the main body of the condenser by inner heads or tube sheets. Beneath the condensing chamber are the vacuum and circulating pumps, which in this case are driven by a single steam cylinder placed between them.

In action, the exhaust steam from the engine enters the shell at the top and fills the condensing chamber, flowing around and among the tubes, while the cooling water is made to pass through them by means of the circulating pump. The steam is condensed by contact with the cold surface of the tubes, and drops to the bottom of the shell where it flows to one end and enters the air or vacuum pump and is discharged into the hot-well. On entering the condensing chamber the steam strikes the baffle plate and is thrown in both directions towards the ends where it passes downward to the lower portion of the chamber, thus distributing itself over the entire tube surface. The cooling water is delivered from the circulating pump into the chamber directly above it, from which it passes through the lower group of tubes into the chamber at the left, and from here through the upper group of tubes to the outlet at the right.

The tubes are commonly made of drawn brass or copper, tinned on both sides. The diameter usually varies from ½ to 1 inch, depending upon the length. The thickness of metal depends upon the diameter, averaging about 0.05 inch for a tube ¾ inch in diameter. The pitch of the tubes commonly varies from 1.5 to 1.7 of the diameter. This results in a certain number of tubes per square foot. (See Table IV.)

TABLE IV. PITCH OF TUBES AND NUMBER OF TUBES PER SQUARE FOOT

Pitch of Tubes, Inches	Number per Square Foot of Space	Pitch of Tubes, Inches	Number per Square Foot of Space	Pitch of Tubes, Inches	Number per Square Foot of Space
1	172	$1\frac{5}{32}$	128	$1\frac{1}{4}$	110
$1\frac{1}{16}$	150	$1\frac{3}{16}$	121	$1\frac{9}{32}$	106
$1\frac{1}{8}$	137	$1\frac{7}{32}$	116	$1\frac{5}{16}$	99

The condenser shown in Fig. 21 is known as the single-tube type, and is one of the simplest arrangements. The tubes are connected to the heads in various ways. In some cases they are fastened rigidly at one end and allowed to move at the other, in order to take care of the expansion and contraction. Tubes of this kind are made steam- and water-tight by the use of a stuffing box and gland. In other makes this arrangement is provided at both ends, as shown in Fig. 22.

Another tube construction is shown in diagrammatical form in Fig. 23, and is known as the double-tube pattern. In this design the water enters and leaves the condenser at the same end, as in Fig. 21, first passing through the inner tube and then to the outlet by way of the outer tube, as indicated by the arrows. Since but one end of the tube is fixed, the opposite end merely projecting into the steam or condensing space, it is free to expand and contract independently of the shell and of the other tubes. In other arrangements both ends of the tubes are expanded rigidly into tube sheets or inner heads, and the expansion it taken care of by a special construction of the shell.

The cooling surface depends upon the temperature and weight of the steam to be condensed, and the initial and final temperatures of the cooling water. The formula commonly used for determining the tube or cooling surface in any given case is:

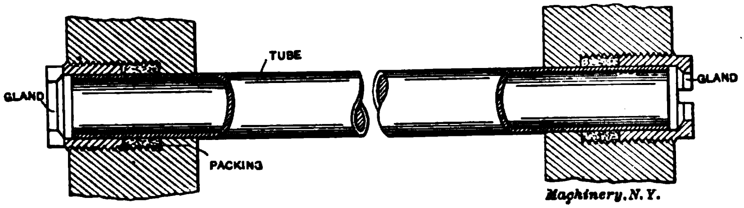

Fig. 22. Condenser Tube with Stuffing Box and Gland at both Ends

$$S = \frac{W\,L}{180\,(T - t)} \tag{10}$$

in which

S = cooling surface in square feet,
W = weight of steam to be condensed per hour, in pounds,
T = temperature of the steam at condenser pressure, in degrees F.,
t = average temperature of the circulating or cooling water, in degrees F.,
L = latent heat of steam at temperature T.

The condenser shell is made of cast iron, either circular or rectangular in section, that shown in Figs. 20 and 21 being of the former design. It is sometimes mounted on separate supports, but more commonly combined with the vacuum and circulating pumps as shown. It is customary to carry the tubes quite close to the shell, as it is found by experience that the steam will readily spread to all parts of the condensing chamber. Hence, the tube spacing arrangement becomes entirely a matter of mechanical construction. After computing the cooling surface required, the diameter of the tube may be assumed, the length not exceeding about 120 diameters, and the spacing may be taken from Table IV. This practically determines the size of the shell.

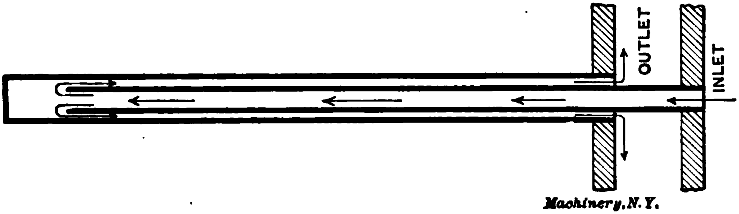

Fig. 23. Condenser Tube of the Double-tube Pattern

For marine work, surface condensers are used almost exclusively, and they are also employed in stationary power plants. They are more bulky for a given capacity than other types, but may be used with any kind of cooling water, which is a decided advantage if the condensed steam is to be fed into the boilers again. This is often of much importance, where the feed water supply is of poor quality, and can be done without injury if the oil is thoroughly removed from the exhaust steam. In such a case only from 10 to 12 per cent of the boiler supply is made up of fresh water.

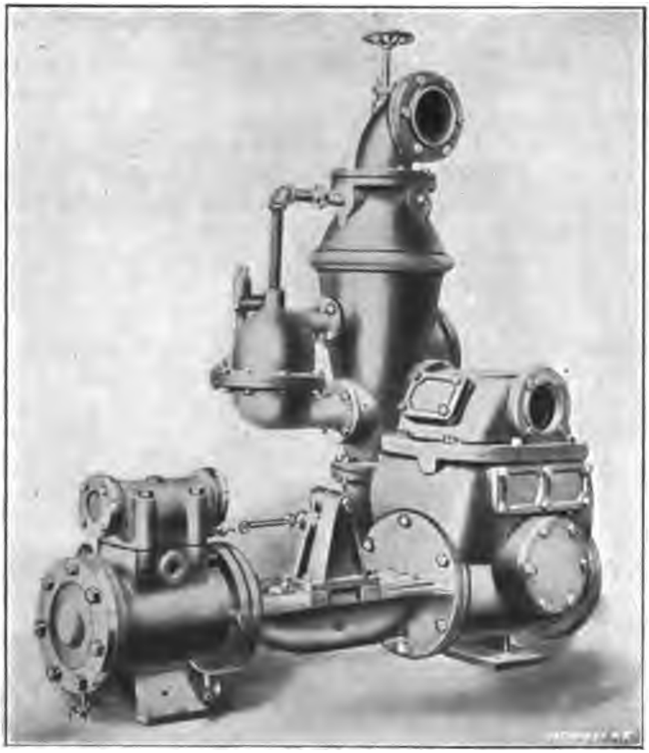

Fig. 24. External View of Jet Condenser

Jet Condensers

In a jet condenser the steam and condensing water mingle in the condensing cone, and the condensed steam is discharged with the water. As the condensing water acts directly upon the steam by actual contact, it will produce a greater drop in pressure for a given amount of water than when used in a surface condenser. When the condensed steam is fed back into the boilers it is evident that a portion of the cooling water goes with it; hence, in this case, the supply must be of a quality which is not detrimental to the boiler plates.

An external view of a jet condenser is shown in Fig. 24, and a section through the condensing cone in Fig. 25. By referring to the latter illustration, it is seen that the exhaust steam enters at the top of the condenser and then meets the injection or cooling water, which is drawn in by suction due to the partial vacuum and discharged in the form of a spray by means of the adjustable cone shown at the center. The result is an instantaneous and complete condensation of the steam, which mixes with the cooling water, and with it is drawn into the pump and discharged either to the sewer or hot-well as the case may be.

The vacuum breaker shown at the right prevents the water from rising any higher in the condenser than to its proper level. If by accident the air pump should stop, the water will rise in the condenser sufficiently to lift the float, thus admitting air and breaking the vacuum. As soon as the pump is again started, the float drops to its normal position, the air relief valve closes, and the work of condensation is again resumed.

Before determining the size of pump for a condenser, the volume of cooling water must be computed.

This may be found with sufficient exactness for a jet condenser by the formula:

$$Q = \frac{S - (D - 32)}{D - I} \qquad (11)$$

in which

Q = weight of cooling water per pound of steam condensed,

S = total heat in one pound of steam above 32 degrees, at terminal pressure,

I = initial temperature of cooling water,

D = final temperature of cooling water.

The terminal steam pressure will vary with the type of engine, the initial pressure, and the ratio of expansion. For average conditions, when no exact data is at hand, it may be assumed as 20 pounds per square inch, absolute. The total heat of evaporation corresponding to this pressure is 1151 heat units, which may be taken as the value of S in the formula. The initial temperature of the cooling water is commonly taken as 70 degrees F. in the summer time, and the final temperature as 110 degrees F.

When the condensing water is taken from ponds and streams, its temperature will be only slightly above the freezing point in the winter, so the quantity required will be considerably less than in the summer. When making an estimate for the entire year, it is customary to assume an average temperature of 50 degrees F.

When designing the pumps for a condenser it is necessary to make them of sufficient size to handle the maximum volume of condensing water, which of course increases with the initial temperature of the cooling water. In the case of surface condensers, the weight of water computed by Formula (11) should be increased about 20 per cent.

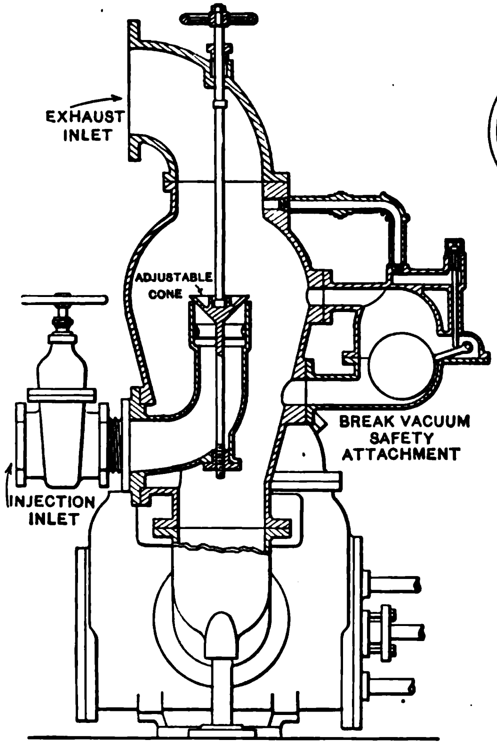

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Fig. 25. Section through Jet Condenser

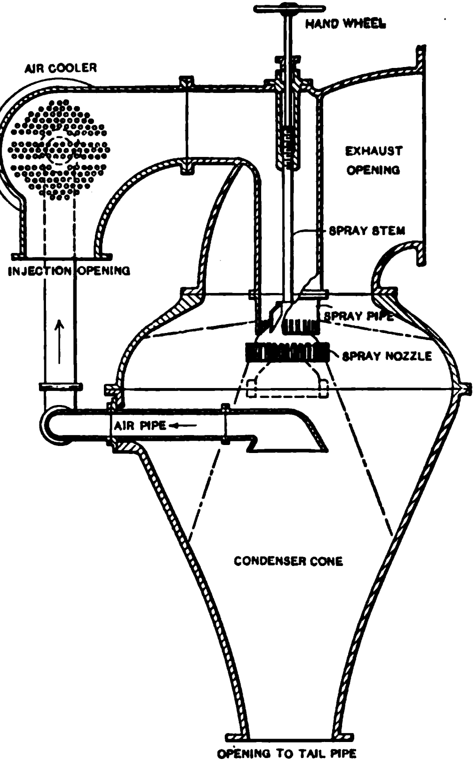

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Fig. 26. Condenser Head of the Spray Type

Fig. 27. Siphon Condenser of the Nozzle Type

Barometric or Siphon Condensers

The barometric or siphon condenser is particularly well adapted to plants in which the condensing water is suitable for boiler feeding, and also to any plant where condensation of steam only is desired, the condensing water not being used. These condensers are able to maintain a vacuum of 26 to 27 inches without the use of pumps or valves, and require practically no adjustment.

A siphon condenser of the nozzle type is shown in Fig. 27. This type consists of a steam chamber in the form of a return bend, and is fitted with a relief valve at the top which closes automatically, due to its own weight assisted by a light spring. The steam flows through the regulating nozzle *A*, while the cooling water enters at the side of the nozzle chamber, and flows in a thin sheet or film through the annular orifice formed between the nozzle and the chamber wall. The throat or combining tube *B* is just below the nozzle *A*, and is of the tapering form shown. This connects with the discharge or tail pipe and should be at an elevation of 34 feet above the surface of the hot-well. In operation, the condensing water passing through the annular orifice formed by the nozzle *A* flows downward in a cone-shaped film into the combining tube *B*, where its velocity is sufficiently increased to enable it to carry air along with it, thus producing a vacuum in the steam exhaust pipe. The steam flows downward through the regulating nozzle and into the cone-shaped film of water, where it is condensed.

A condenser head of the spray type is shown in Fig. 26. This is used in connection with water and vacuum pumps as indicated in Fig. 28. The condenser is placed about 30 feet above the hot-well, and the water falls out of it by gravity against the pressure of the atmosphere. The sectional view of the condenser cone and air cooler shows the method of distributing the condensing water within the chamber. This is done by means of a series of teeth at the lower end of the spray pipe in connection with a similar series upon the nozzle just below it. The falling water in the cone entraps a portion of the air set free by the condensation of the steam, and carries it down into the tail pipe, so that a partial vacuum is formed in the condenser and exhaust piping.

In order to obtain the highest range of vacuum without using an abnormal amount of water to carry off the air, a separate dry vacuum pump is used, by means of which the air not carried off by the water is taken from the empty space under the spray cone in the condenser. On its way to the vacuum pump, the air passes through a cooler, consisting of a large number of tubes through which the condensing water passes, and around which the air circulates on its way to the pump. The usual sizes for the exhaust pipe and connection for the condenser head for different capacities are given in Table V.

Atmospheric Condensers

A vertical section through one form of atmospheric condenser is shown diagrammatically in Fig. 29. This condenser consists of an outer shell, cylindrical in form, to which are attached upper and lower tube sheets *A* and *B*, as shown. The shell is filled with air-tubes of 4-inch wrought-iron pipe, extending about 4 inches above the upper tube-sheet. The exhaust steam enters through a special form of distributor near the bottom of the shell, and is made to circulate among the tubes by means of baffle plates not shown.

TABLE V. EXHAUST PIPE DIMENSIONS FOR SPRAY TYPE CONDENSERS

Weight of Steam to be Condensed per Hour, Pounds	Diameter of Exhaust Pipe and Condenser Connection, Inches	Weight of Steam to be Condensed per Hour, Pounds	Diameter of Exhaust Pipe and Condenser Connection, Inches	Weight of Steam to be Condensed per Hour, Pounds	Diameter of Exhaust Pipe and Condenser Connection, Inches
2,000	5	5,000	9	10,000	12
3,000	7	6,000	9	15,000	14
4,000	8	8,000	10	20,000	14

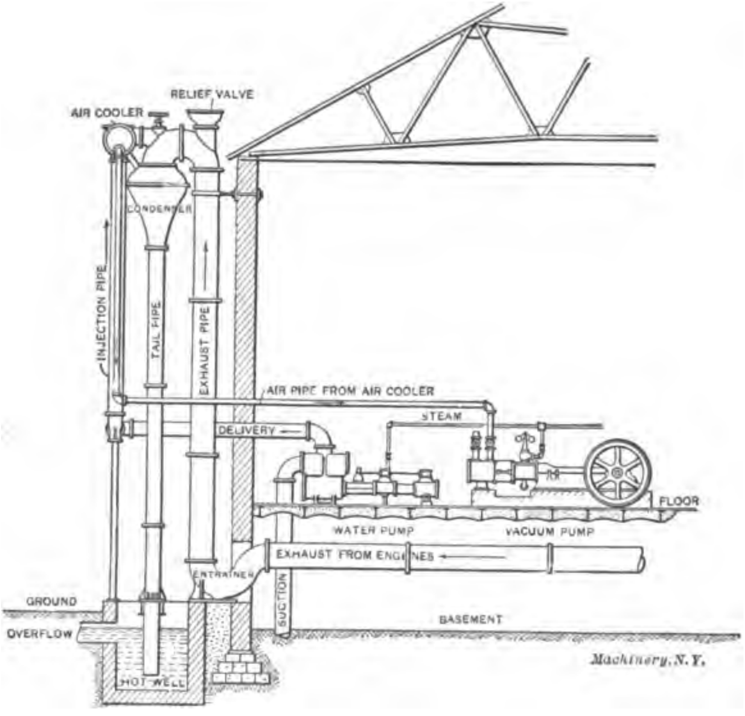

Fig. 28. Arrangement of Condenser shown in Fig. 26

The cooling effect is produced by an upward movement of air through the tubes, which is greatly increased by pumping water into the water-pan above the upper tube sheet and allowing it to trickle downward through the tubes into the cistern below. The upper ends of the tubes are notched so that the water in passing into them spreads into a thin sheet, covering the inside surface. The exhaust steam in the shell heats the tubes and the film of water, causing the latter to evaporate rapidly, thus saturating the air and causing it to pass swiftly up the tube, carrying with it large quantities of heat taken from the condensing steam. The water of condensation is taken off through a drain connection in the lower tube sheet, and the air from the upper outlet, as indicated. These pipes are connected and carried to a vacuum pump not shown in the illustration. The velocity of air-flow through the tubes is increased by carrying up a flue of light steel to a height of about 50 feet above the condenser.

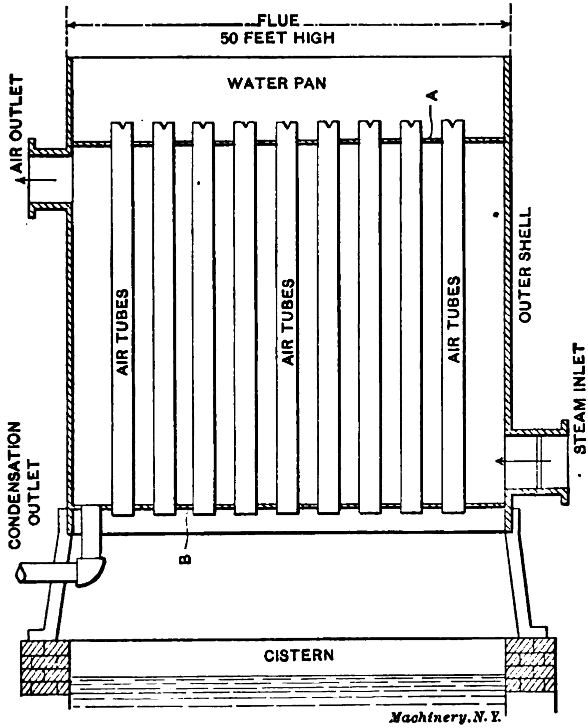

Fig. 29. Section through Atmospheric Condenser

Cooling Towers

From 20 to 30 per cent may be saved in fuel by the use of a condenser. This estimate is based on the assumption that the water used for condensing the steam can be obtained free of cost. When the plant is located in a city where the water must be obtained at regular city rates, it often happens that it is more economical to run non-condensing than to purchase cooling water. In order to do away with the water expense, so-called cooling towers are now extensively used. By means of these the condensing water may be cooled and used over and over again with a comparatively small loss by evaporation.

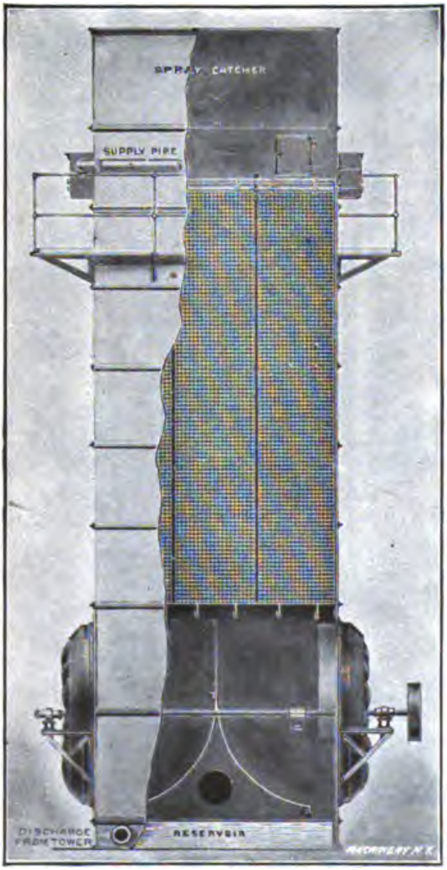

Fig. 30. Cooling Tower

There are various forms of cooling towers in use. The general principles, which are practically the same in each case, are well illustrated in Fig. 30. The tower consists of a steel shell inside of which are suspended a number of mats of a special steel wire cloth, galvanized after weaving. The mats are, in effect, a metallic sponge, capable of holding a large quantity of water in suspension, which accumulates and drips off into the reservoir at the bottom. The water to be cooled is pumped to the top of the tower and discharged through a number of distributing nozzles upon the tops of the mats. From here it drips to the bottom, exposing a large surface to the air which is forced upward by the fans placed at the bottom of the tower.

The cooling effect is due to three causes: first, radiation from the sides of the tower; second, the contact of the water with the cooler air; and third, the most important of all, the heat carried away in the process of evaporation. The proportion due to the latter cause may be easily calculated as follows: The latent heat of evaporation at 110 degrees is 1035; that is, 1035 heat units are absorbed in changing 1 pound of water from a temperature of 110 degrees into vapor at the same temperature. Suppose it is desired to cool 100 pounds of water 40 degrees. This will evidently require the removal of $40 \times 100 = 4000$ heat units. If the evaporation of 1 pound of water will absorb

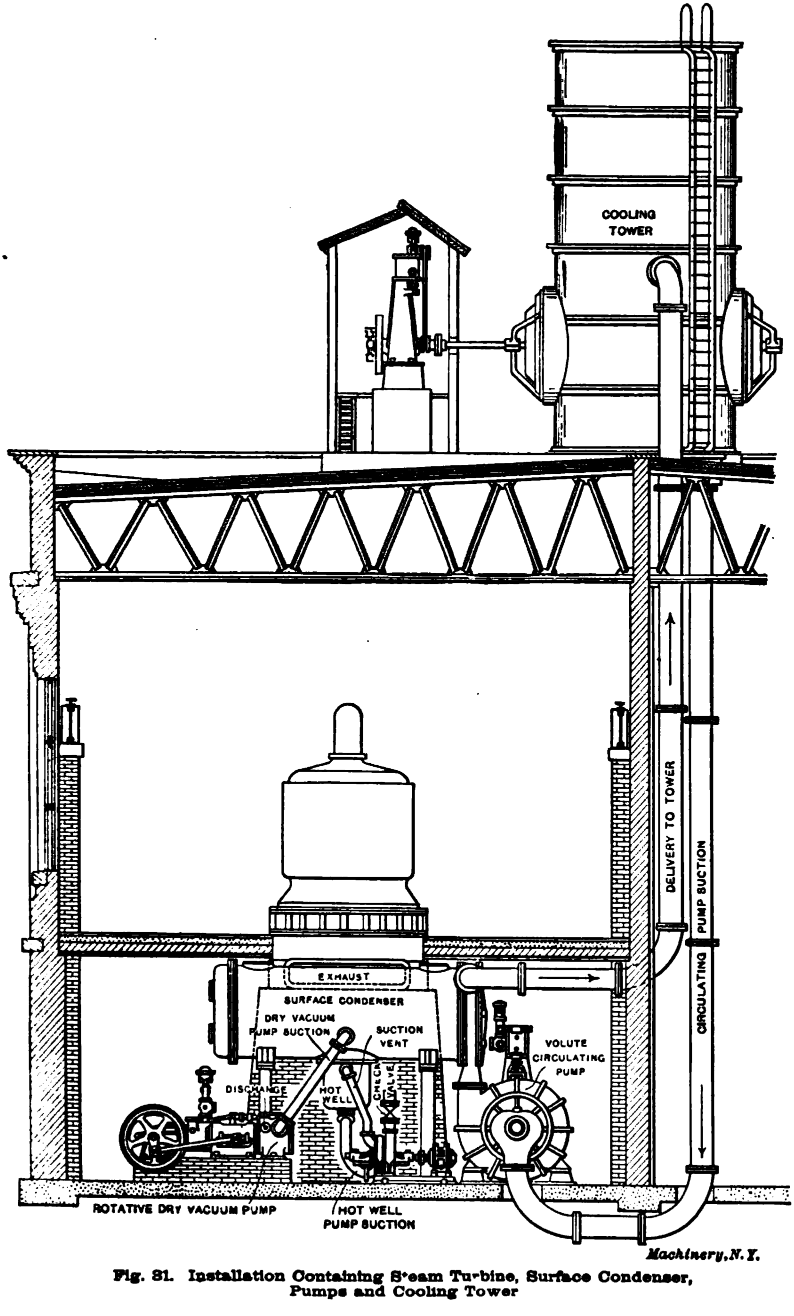

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Fig. 81. Installation Containing Steam Turbine, Surface Condenser, Pumps and Cooling Tower

1035 heat units, then 4000 ÷ 1035 = 3.86 pounds is the amount of evaporation required to remove 4000 heat units, and therefore represents the loss necessary to cool 100 pounds of water 40 degrees in this manner; that is, 3.86 pounds of water will be evaporated for every 100 pounds cooled 40 degrees in passing through the tower. With the best forms of cooling towers the temperature of the condensing water may easily be reduced from 40 to 50 degrees with a loss from evaporation not exceeding 3 to 4 per cent.

The volume of air to be passed through a cooling tower will depend upon the temperature and relative humidity, the condenser pressure, the weight of condensing water, and its final temperature. Professor E. F. Miller gives the results of two tests which are tabulated below, and which will be found useful in estimating the air volume for average conditions. The first case (Table VI) relates to a condenser maintaining 28 inches of vacuum, and using 40 pounds of condensing water per pound of exhaust. The final temperature of the water is assumed to be 95 degrees, and the air temperature 70 degrees.

TABLE VI. VOLUME OF AIR REQUIRED FOR COOLING TOWERS

Relative Humidity of Air	Cubic Feet of Air required per Pound of Exhaust Steam	Relative Humidity of Air	Cubic Feet of Air required per Pound of Exhaust Steam
0.60	410	0.80	455
0.70	420	0.90	480

The second case (Table VII) relates to a condenser maintaining 26 inches of vacuum, and using 20.7 pounds of condensing water per pound of exhaust. A final water temperature of 119 degrees, and an air temperature of 70 degrees, is assumed.

TABLE VII. VOLUME OF AIR REQUIRED FOR COOLING TOWERS

Relative Humidity of Air	Cubic Feet of Air required per Pound of Exhaust Steam	Relative Humidity of Air	Cubic Feet of Air required per Pound of Exhaust Steam
0.60	176	0.80	184
0.70	180	0.90	188

An installation containing a steam turbine, surface condenser, pumps, and cooling tower, is shown in Fig. 31. The supply of cold water for the condenser is drawn from the bottom of the cooling tower by a centrifugal pump, and then forced through the condenser, from the top of which it again passes to the tower to be cooled. In the towers shown, the air has been forced through them by fans. Towers are also made for natural draft, in which case the flue is extended to a considerable height above the cooling surfaces.

A surface condenser requires two pumps, one called the circulating pump, for forcing the condensing or cooling water through the tubes,

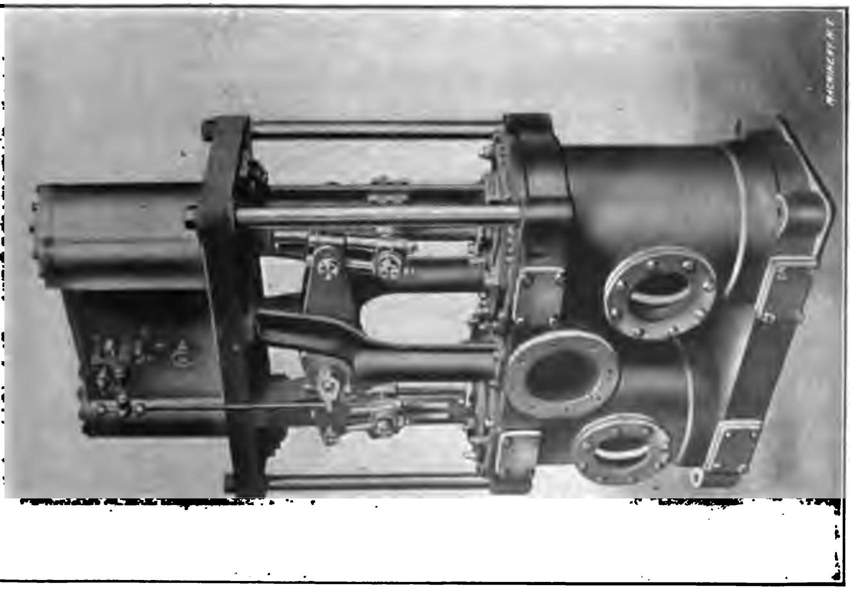

Fig. 32. Direct-acting Twin Vacuum Pump

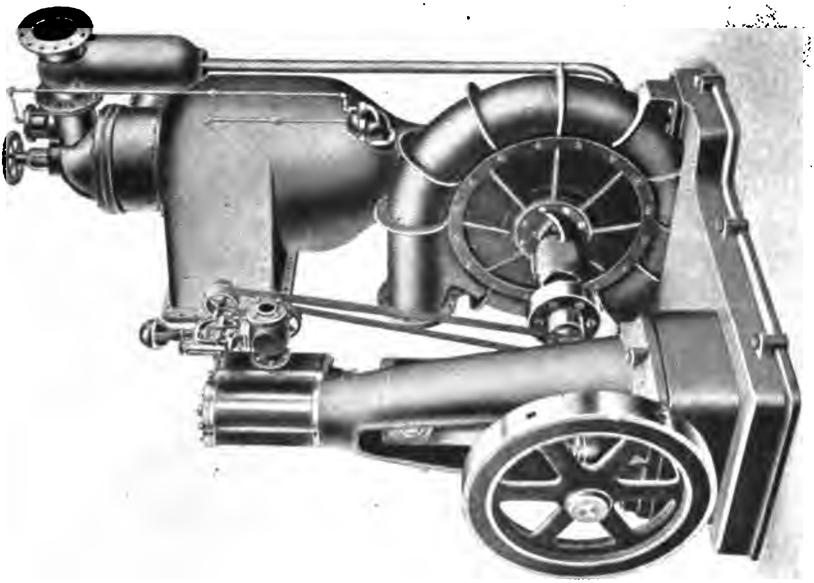

Fig. 33. Condenser with Centrifugal Pump

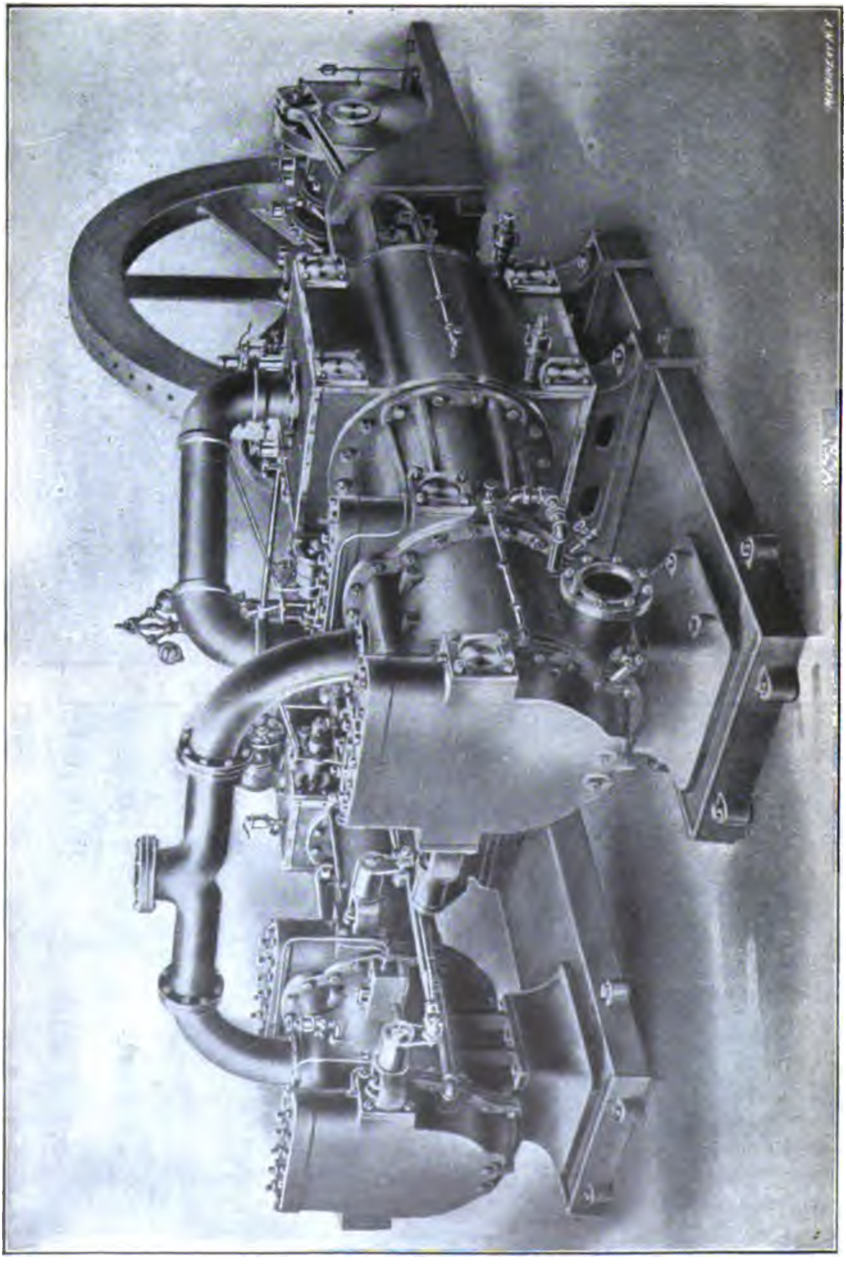

Fig. 34. Large Duplex Pump for Power Plant Service

and one called a vacuum or air pump for withdrawing the condensed steam and the air which it may contain. Separate pumps may be used for this purpose or the two may be combined with a single steam cylinder as shown in Fig. 21. Both direct-acting steam pumps and those of the centrifugal type are used for circulating pumps, while the direct-acting pump only, either of the vertical or horizontal type, is commonly employed for removing the air and condensation.

A direct-acting vertical twin vacuum pump, of the single-acting type, is shown in Fig. 32. This type of pump is used in connection with both surface and jet condensers, and has the advantage of occupying a comparatively small floor space.

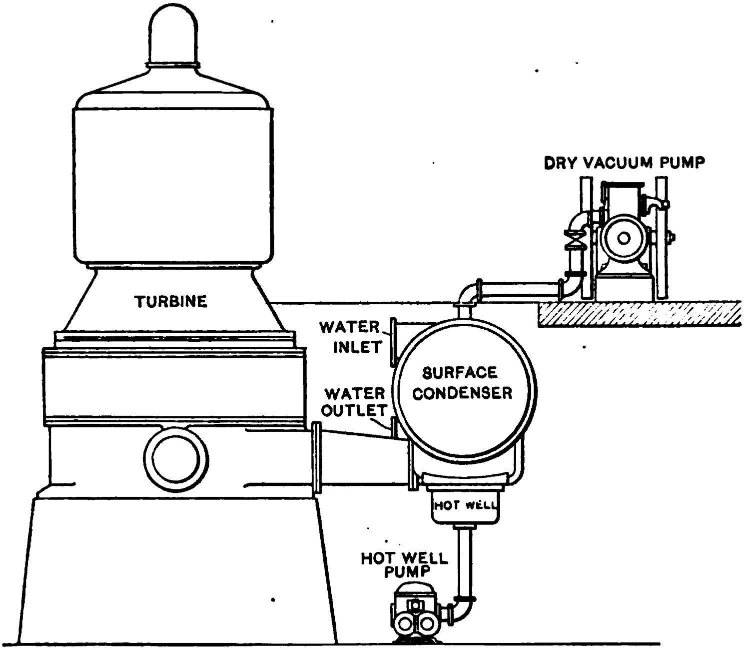

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Fig. 35. Diagrammatical View of Steam Turbine with Surface Condenser

Jet condensers, as commonly used, require simply a vacuum pump, the cooling water being drawn into the condenser head by suction. Both direct-acting steam pumps and centrifugal pumps are used for this purpose. Fig. 24 shows an outfit equipped with a direct-acting horizontal pump, while in Fig. 33 the condenser is provided with a centrifugal pump driven by a direct-connected steam engine. In some cases a pump is also used for forcing the water into a jet condenser instead of drawing it in by suction.

The high vacuum necessary to obtain the best efficiency from a steam turbine has brought into use the dry vacuum pump. With this arrangement the air and condensed steam are removed from the condenser by means of separate pumps. Dry vacuum pumps are usually of the rotative type, and require a higher grade of workmanship than is necessary with pumps used for a moderate vacuum. A large duplex pump of this type, designed especially for power plant service, is shown in Fig. 34. The air cylinders are at the left, and the steam cylinders between them and the main shaft.

A steam turbine arranged for high vacuum, and employing a surface condenser, is shown in Fig. 35. The dry vacuum pump is shown at the right, and is so connected that it removes the air from the top of the condenser. The condensed steam falls into the hot-well beneath the condenser, and is removed by the hot-well pump. A similar arrangement for a jet condenser is shown in Fig. 36.

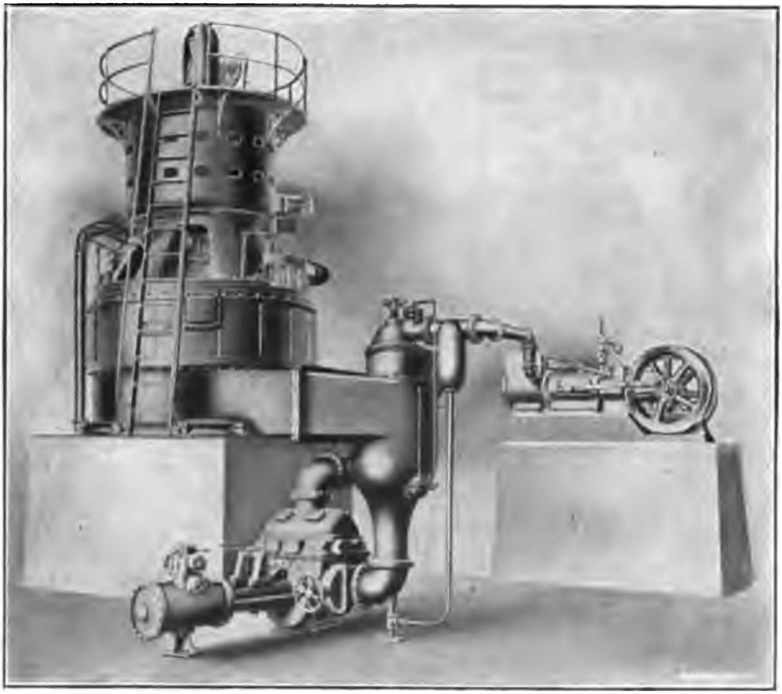


Fig. 36. Steam Turbine Installation with Jet Condenser

The circulating pump should be proportioned to handle the required volume of cooling water the same as for tank service, and in the manner already described. A common method of determining the size of the air cylinder is by use of the following equation:

$$V = W \times C \qquad (12)$$

V = piston displacement, in gallons per minute,

W = weight of steam condensed, in pounds per hour,

C = 0.045 for double-acting horizontal pumps; 0.022 for single-acting vertical pumps.

CHAPTER III

STEAM AND WATER PIPING

Some of the more important points to be kept in mind when designing a system of piping for a power plant are as follows:

1. The route between the boilers and engines should be made as direct as practicable under existing conditions.
2. Provision should be made for expansion and contraction, so that excessive strains will not be thrown upon the pipe and fittings.
3. The piping should be so supported and anchored that vibration will be eliminated, so far as possible.
4. All supply piping should be of such size as to avoid undue losses in pressure and excessive velocities. Low velocities should be the rule in exhaust lines, in order to minimize the back-pressure upon the engines.
5. Proper joints and packing should be used to make the system steam- and water-tight, and all pipes and fittings should be thoroughly insulated to prevent excessive loss of heat from radiation.
6. Special attention should be given to the matter of drainage; the condensation from separators, main headers, and all other low points in the system should be thoroughly removed and returned to the boilers.
7. Separate headers should be provided for the engines and auxiliaries, and the whole system should be divided into sections, or else duplicated so far as practicable.

Piping Materials

Under the heading of piping materials are included the different kinds of pipe used in power plant construction, fittings, valves, etc. Although designated as wrought-iron pipe, the pipe commonly used in power plant work is made of wrought steel. When of good quality as to malleability and ductility, with joints properly welded, there is no advantage in using the more expensive wrought-iron pipe.

Pipe is classed according to weight or thickness of shell, being known as standard, extra strong, and double extra strong. Taking the weight or thickness of standard pipe as 1, the thickness of extra strong pipe is 1.4, and of the double extra strong, 2.8. All of these weights of pipe have the same outside diameter, the additional thickness of metal being added to the inside of the pipe. The standard pipe of commerce, commonly known as "Merchant" pipe, is lighter than the standard or "full weight" pipe. Manufacturers usually specify that the latter may vary 5 per cent from the standard, but as a matter of fact, it almost invariably falls below.

Pipe 3 inches in diameter and smaller is commonly butt welded, and larger sizes lap welded. While it is possible to make larger pipe by the former process, the cost is greater, and it has, therefore, become the practice of manufacturers to limit the diameter of butt welded pipe to 3 inches.

Standard weight pipe is used for exhaust lines and for pressures up to 125 pounds, although some engineers consider it safe for pressures of 200 pounds or more. Heavier weights, however, are generally used for higher pressures, and also for pipes which, due to their location, are especially subject to corrosion. Table VIII gives data and dimensions relating to standard weight pipe.

TABLE VIII. STANDARD PIPE DIMENSIONS

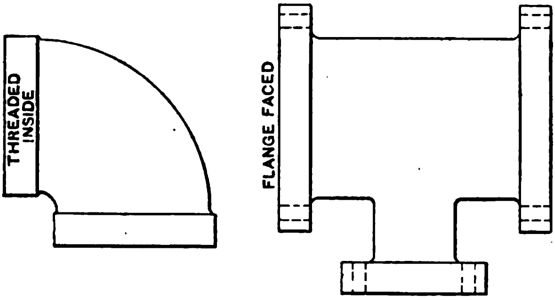
Diameter			Thickness of Wall, Inches	Internal Area, Square Inches	Exposed Surface per Foot, Square Feet	Weight per Foot, Pounds	Number of Threads per Inch
Nominal, Inches	External, Inches	Actual Internal, Inches					
1	1.31	1.05	.13	.86	.345	1.67	11½
1¼	1.66	1.38	.14	1.50	.434	2.24	11½
1½	1.90	1.61	.14	2.04	.497	2.68	11½
2	2.37	2 07	.15	3.36	.621	3.61	11½
2½	2.87	2.47	.20	4.78	.752	5.74	8
3	3.50	3.07	.22	7.39	.916	7.54	8
3½	4 00	3.55	.23	9.89	1.047	9.00	8
4	4.50	4.03	.24	12.73	1.178	10.26	8
5	5.56	5.04	.26	19.99	1.456	14.50	8
6	6.62	6.06	.28	28.89	1.734	18.76	8
7	7.63	7.02	.30	38.74	1.988	23.27	8
8	8.63	7.98	.32	50.04	2.258	28.18	8
9	9.62	8.94	.34	62.73	2.520	33.70	8
10	10.75	10.02	.37	78.84	2.814	40.00	8
11	11 75	11.00	.37	95.03	3.075	45.00	8
12	12.75	12.00	.37	113.10	3.337	49.00	8

Spiral riveted pipe is a pipe frequently used for low-pressure piping of large size, as in heating work, and for exhaust lines. Its advantages over wrought-iron pipe are its lightness and lower cost. Pipes which are to carry hot water should be of brass, owing to the corrosion of wrought-iron or steel when used for this purpose. Pipes of this class include the connections between boilers and feed pumps, the attachment of boiler accessories, and all connections around hot-water boilers for lavatory purposes. Seamless drawn brass tubing is made in sizes to correspond with standard wrought-iron pipe.

The fittings used for making up wrought-iron pipe are usually of cast iron, although wrought-iron flanges are used to some extent in the best class of high-pressure work. For the smaller sizes of piping up to 2 or 2½ inches screwed fittings are commonly used, an elbow of this pattern being shown in Fig. 37. In the case of larger pipes, the flange fitting is almost universally employed, it being much easier to connect where the space is limited, and also easier to take down in case of repairs. A flanged tee is shown in Fig. 38. Cast-iron fittings are made in a great variety of forms to meet almost any requirements which may occur in the design of a system of piping. They are made regularly in two weights: standard, for pressures up to 125 pounds per square inch, and extra heavy, for higher pressures up to 250 pounds.

Gate valves are generally used in power plant work except where an angle valve will answer the purpose and take the place of a fitting. They offer but little resistance to the passage of steam through them, and if placed with the spindle in a horizontal position, or vertical with the hand-wheel at the top, they cannot form pockets in the piping for the accumulation of condensation. For all important lines of piping, the rising spindle valve, shown in Fig. 39 should always be used, so that the engineer can tell at a glance whether it is open or closed.

Valves 6 inches and larger in size are usually provided with a by-pass as shown in Fig. 40. This allows the steam to be admitted slowly when first turning it on, and also equalizes the pressure on both sides of the main valve so that it can be opened easily, without scoring the faces of the gate and seat. Globe valves may be used in small vertical pipes, drip lines, etc.; when placed in horizontal pipes, care should be taken to have the stem horizontal. Check valves for vertical feed pipes should be of the "spring" type. Swing checks are commonly used in horizontal pipes, but sometimes give trouble by chattering or beating at each stroke of the feed pump.



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Figs. 37 and 38. Pipe Fittings

The matter of pipe joints is an important one in the design and construction of a power plant. One of the simplest joints is that shown in Fig. 41. In this case the pipes are threaded with a full taper and screwed into the flanges by power, so that the ends project slightly, and are then faced off in a lathe, a thin cut being taken from the whole face of the flange. The joints are commonly packed with a corrugated copper gasket placed inside the bolt circle. This form of gasket is very durable, often lasting as long as the pipe. Piping made up with flanges of this kind is easily disconnected for repairs, as a section may be removed without springing the joint apart, which is a matter of much importance in the case of long mains of large size.

Another type of joint often used is shown in Fig. 42. Here the pipes are threaded and screwed into the flanges as before, but instead of being plane surfaces, the flanges are tongued and grooved as indicated, with a copper gasket placed at the bottom of the groove. This makes a very satisfactory and durable joint, the only objection being the necessity of springing it apart when removing a section for repairs.

Fig. 43 shows a joint in which the flanges are shrunk on the pipe, the ends of which are turned up as shown. No gasket is required in this case, a ground joint being used instead. In the very best class of high-pressure work, where the cost is no objection, it is customary to use wrought-iron flanges welded to the pipe. This makes the tightest and most durable joint possible. One method of making a joint in a line of spiral riveted pipe is shown in section in Fig. 44. In other cases flanges are riveted to the pipe and bolted together with a gasket between them in the usual manner for low-pressure work. In the case of exhaust lines, the plain flange is used, with a vulcanized rubber gasket or one of similar material. Table IX gives data regarding flanges for low- and high-pressure work.

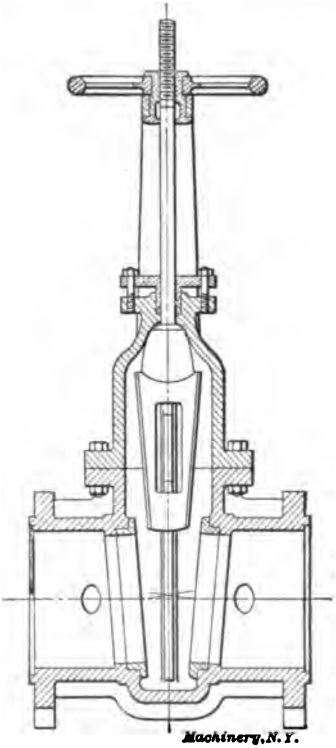

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Fig. 39. **Rising Spindle Valve**

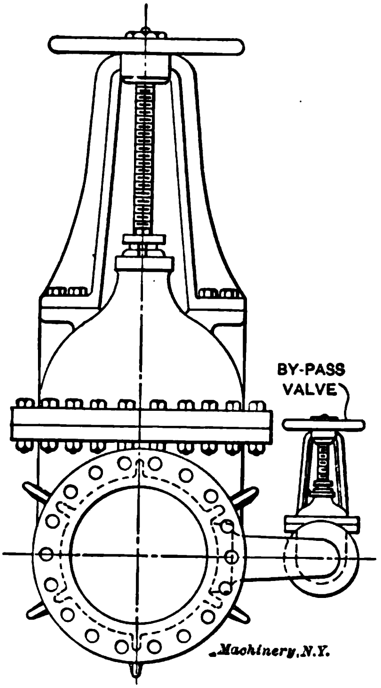

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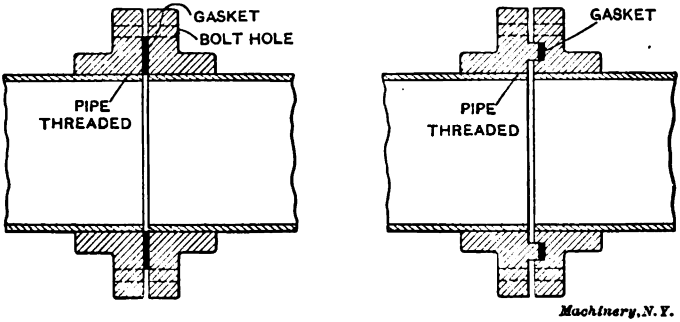
Fig. 40. **Large Valve with By-pass Valve**

In the design of a system of piping, either for power or heating, especial care must be taken to allow for the strains due to expansion and contraction. Although this is of importance in heating work, it becomes doubly so in the case of high pressures, both because of the greater expansion due to the higher temperature, and the more serious results in case of fracture under high pressure. Table X gives the expansion for each 10 feet in length for different steam pressures, in an atmospheric temperature of 50 degrees F.



Figs. 41 and 42. Types of Pipe Joints

There are three methods commonly used for taking up the expansion in pipes:

1. By using sweep bends in place of cast-iron elbows, and arranging the piping so as to provide the maximum amount of flexibility or spring.
2. By the use of swivel joints.
3. By using expansion or slip joints.

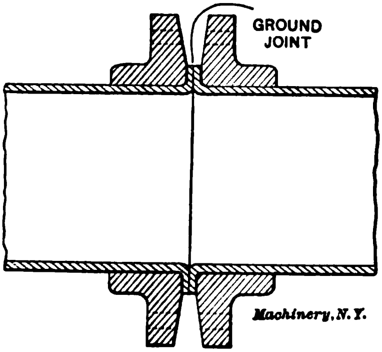


Fig. 43. Pipe Joint with Flanges Shrunk on the Pipe

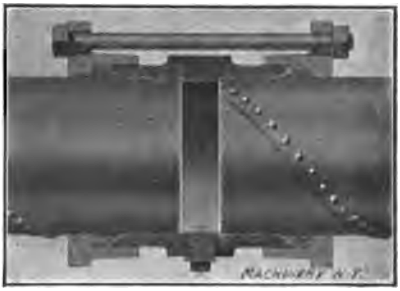


Fig. 44. Pipe Joint in Spiral Riveted Pipe Line

The first method should always be employed, so far as possible, and should be supplemented by the other two where more flexibility is required. A swivel joint is shown in Fig. 46. The main and its continuation, together with the offset, are clearly indicated in the illustration, and any lengthening of the former is taken up by a turning movement at the two swivels which are packed to prevent leakage. A slight turning movement of this kind is kept tight more easily than a sliding movement.

A balanced slip joint is shown in section in Fig. 45; the path of the steam is indicated by the arrows. The joint is made tight by means of two glands with packing, and is balanced by exposing the end *A* of the inlet pipe to atmospheric pressure. A type of expansion joint sometimes used in large outbound exhaust pipes, where there is practically no internal pressure, is shown in Fig. 47. This form is particularly adapted to spiral riveted pipe, to which the flanges *A* are riveted. Flexibility is secured by means of a copper ring attached to the flanges by the cast-iron rings *B*, which are secured by bolts as shown.

TABLE IX. PIPE FLANGES

Diameter of Pipe	Diameter of Flange		Thickness of Flange		Number of Bolts		Diameter of Bolt Circle		Diameter of Bolts	
	Pressure		Pressure		Pressure		Pressure		Pressure	
	125 Pounds	250 Pounds	125 Pounds	250 Pounds	125 Pounds	250 Pounds	125 Pounds	250 Pounds	125 Pounds	250 Pounds
2	6	6½	¾	⅞	4	4	4¾	5	⅝	⅝
2½	7	7½	13/16	1	4	4	5½	5⅞	⅝	¾
3	7½	8¼	13/16	1⅛	4	8	6	6⅝	⅝	⅝
3½	8½	9	⅞	1 3/16	4	8	7	7¼	⅝	⅝
4	9	10	15/16	1¼	4	8	7½	7⅞	¾	¾
5	10	11	15/16	1⅜	8	8	8½	9¼	¾	¾
6	11	12½	1	1 7/16	8	12	9½	10⅝	¾	¾
7	12½	14	1 1/16	1½	8	12	10¾	11⅞	¾	⅞
8	13½	15	1⅛	1⅝	8	12	11¾	13	¾	⅞
9	15	16	1⅛	1¾	12	12	13¼	14	¾	⅞
10	16	17½	1 3/16	1⅞	12	16	14¼	15¼	⅞	⅞
12	19	20	1¼	2	12	16	17	17¾	⅞	⅞

TABLE X. EXPANSION OF PIPE AT DIFFERENT TEMPERATURES

Pressure, Pounds Gage	Temperature, Degrees F.	Expansion in 10 Feet, Inches	Pressure, Pounds Gage	Temperature, Degrees F.	Expansion in 10 Feet, Inches
10	240	0.148	90	331	0.219
20	259	0.163	100	337	0.224
30	274	0.175	110	344	0.229
40	286	0.184	120	350	0.234
50	297	0.193	130	355	0.238
60	307	0.199	140	360	0.242
70	316	0.207	150	365	0.246
80	323	0.213	...	...	.....

Steam and exhaust pipes should not only be strongly supported, but the supports should also be arranged to allow for the movement due to expansion and contraction. Overhead pipes, if not too large, are commonly hung from the ceiling construction by adjustable hangers. Fig. 48 illustrates a method of attaching a hanger to a brick wall by means of an iron bracket and anchor bolts. A similar type of hanger, clamped to the lower flange of an I-beam, is shown in Fig. 49. In both of these cases any movement of the pipe is cared for by the swinging of the hook upon the supporting rod. In Fig. 50 the hook is replaced by a bolt and nut, and flexibility is secured by a spherical washer beneath the nut, shown in detail in Fig. 51.

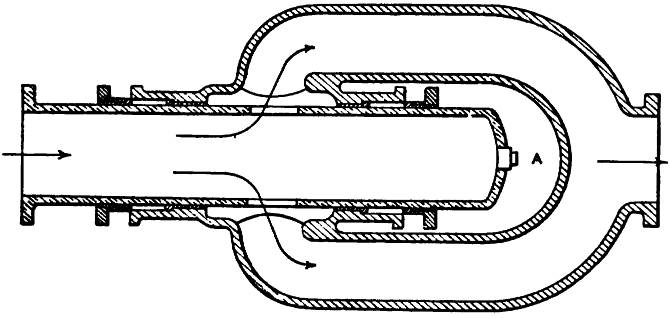


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Fig. 45. Balanced Slip Joint

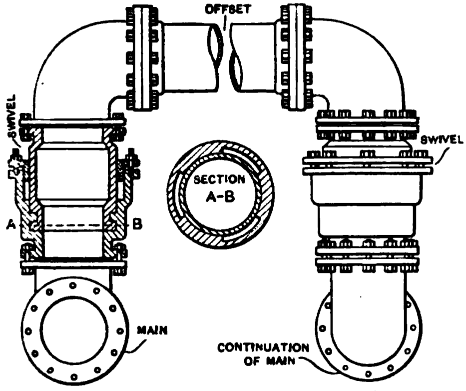

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Fig. 46. Swivel Joint

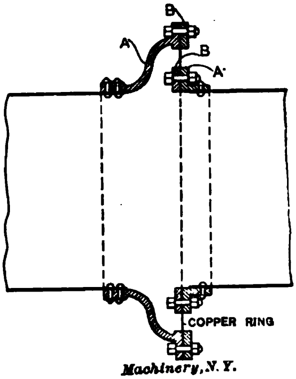

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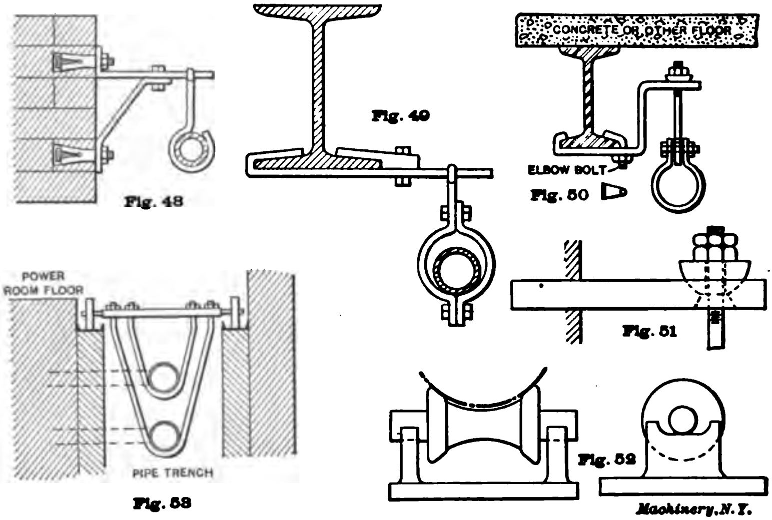
Fig. 47. Expansion Joint for Large Exhaust Pipes

Pipes which run near the floor or over boiler tops are commonly carried on rolls of the general form shown in Fig. 52. These are made of cast iron, and are usually supported on brick piers.

An arrangement for carrying pipe lines in a trench is illustrated in Fig. 53. In this case the pipes are supported in loops of strap iron hung from an overhead cross-bar with wheels at each end, running on channel iron tracks as shown. Any movement of the pipe due to expansion is taken care of by these overhead trucks.

Pipe Conduits

There are various ways of constructing underground conduits for carrying pipe lines from one building to another, or through basement rooms where it is desired to keep them below the floor. In some cases concrete trenches are used covered with slate or slabs of concrete, while in others the mains are run inside of tile piping with cemented joints.



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Figs. 48 to 53. Pipe Supports and Hangers

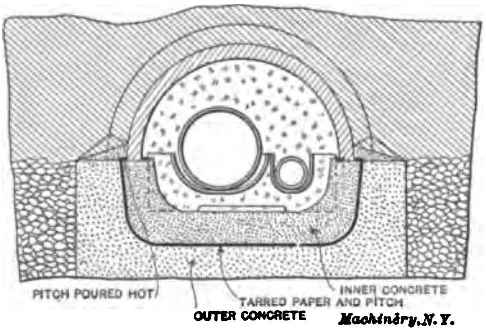

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Fig. 54. Waterproof Pipe Conduit

A waterproof conduit of good design is shown in Fig. 54. The upper part of this conduit is formed by one-half of a tile pipe of suitable size, split lengthwise, while the lower part is made up of concrete. The joints are made water-tight by means of pitch and tarred paper, as shown. For heavy mains, electric cables, etc., it is necessary to construct tunnels of brick or concrete of sufficient size for a man to pass through when inspecting.

Insulation

All steam and exhaust piping should be protected with some good form of sectional covering to prevent loss of heat. In the case of high-pressure piping, there are two reasons for this: first, to reduce the condensation, and thus prevent useless waste of heat; and second, to keep the boiler and engine rooms as cool as possible. With the exhaust lines the latter reason is the more important, unless the steam is utilized for a heating system. All apparatus such as feed-water heaters, surface condensers, receiving tanks, etc., should also be covered in a similar manner.

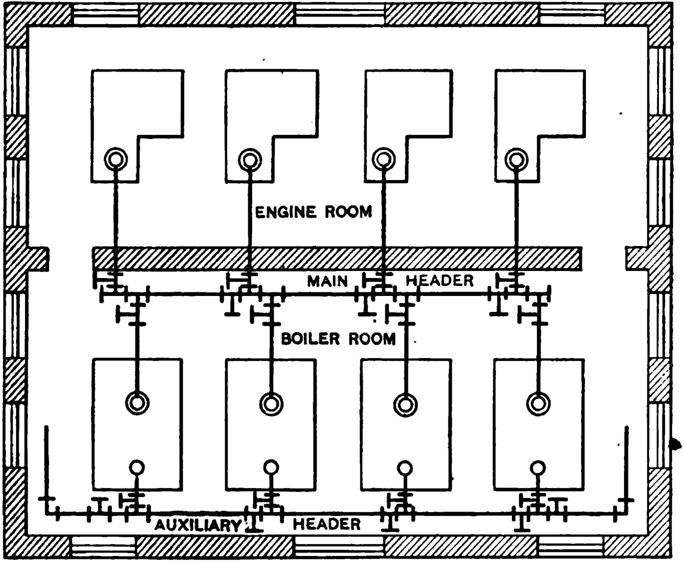

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Fig. 55. Piping Arrangement of a Power Plant

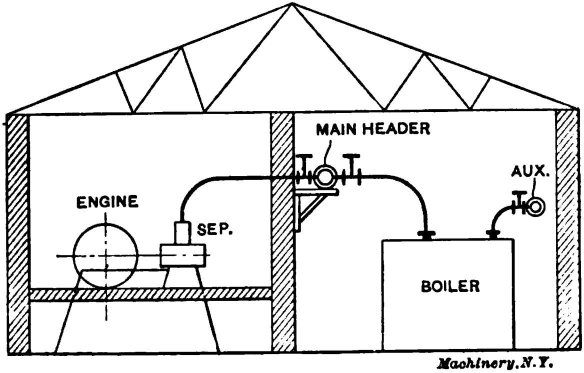

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Fig. 56. Elevation of Power Plant in Fig. 55

Pipes are usually insulated with sectional covering provided with a canvas jacket and held in place by thin metal bands. Valves and fittings are commonly covered with a plastic material and finished with canvas, the same as the piping. In the highest class of work moulded covering is often used in place of the plastic material, so that the covering may be removed for purposes of inspection or repairs, and then be replaced without injury.

High-pressure Piping

Nearly all modern power plants are designed along two general lines, one of which is shown in plan and elevation in Figs. 55 and 56. In this arrangement the engines and boilers are placed back to back, with a fire wall between. This arrangement is very compact and reduces the distance from the boilers to the engines to a minimum. The main header is preferably located in the boiler room, as it may be the means of avoiding serious injury to the engines and electrical apparatus in case of accident, thereby allowing the plant to continue in operation after cutting out the damaged sections by means of the valves provided for that purpose. It is always best to use an independent steam header for the auxiliaries, such as pumps, heaters, etc., as shown in Fig. 55.

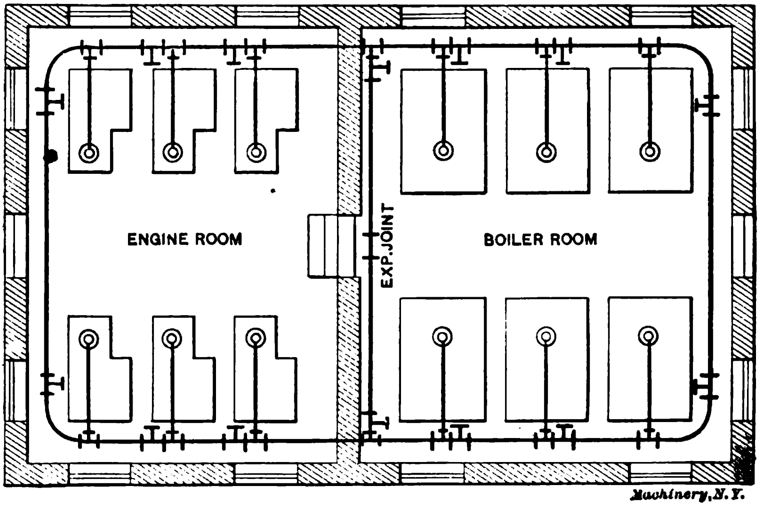

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Fig. 57. Alternative Piping Arrangement for a Power Plant

It will be noted in Fig. 56 that the level of the engine room is somewhat above that of the boiler room. This is a very desirable arrangement as it allows a space for condensers and exhaust piping below the engines. This makes it possible to keep the engine room cooler, prevents damage from steam and water in case of leaks, removes the heavy and unsightly exhaust piping from direct view, and facilitates the matter of drainage.

Flexibility is secured by using sweep bends in the boiler and engine connections with the main header, as shown in Fig. 56. Valves are so provided that any engine or boiler can be cut out of service independently of the others, and in addition to this, the main header is divided into sections.

In the second arrangement of power plants, the boiler and engine rooms are placed end to end as in Fig. 57, and the ring system of piping employed. Valves are so placed in the main as to divide it into sections which may be cut out in case of accident or repairs. In some plants the more important lines of piping are duplicated, making it practically impossible to put the apparatus out of service by an accident to the piping.

Pipe Sizes

There are various ways of computing the high-pressure pipe sizes in power plant work. In general, the velocity of flow should not exceed 6000 feet per minute. A formula often used for determining the size of supply mains, when the length is not great, is given below:

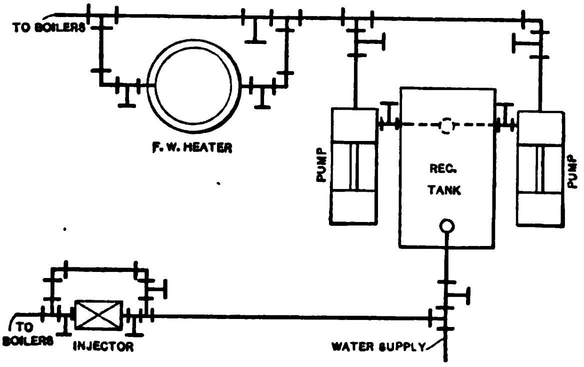

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Fig. 58. Feed Piping for Non-condensing Plant

$$d = D\sqrt{\frac{S \times \text{R. P. M.}}{36{,}000}} \qquad (13)$$

in which

d = diameter of steam pipe, in inches,
D = diameter of engine cylinder, in inches,
S = length of stroke, in inches,
R. P. M. = revolutions per minute of engine.

Feed Piping

Three typical layouts for feed piping are shown in Figs. 58, 59 and 60. The first of these applies to a non-condensing plant with a feed-water heater. The supply from the city main enters as shown, one branch leading directly to the boilers through an injector provided with a by-pass. The other branch leads to a receiving tank from which the water is pumped to the boilers as indicated. The feed-water heater is placed in a by-pass and can be used or not as desired.

In Fig. 59 the piping is arranged for a surface condenser. The branch leading to the injector is the same as before. The other branch goes directly to the pumps with an off-take to the hot-well provided with a float valve. The connections between the pumps and boilers, including the feed-water heater, are the same as before. The discharge from the air pump goes to the hot-well, from which it may be fed into the boilers by the pumps if desired.

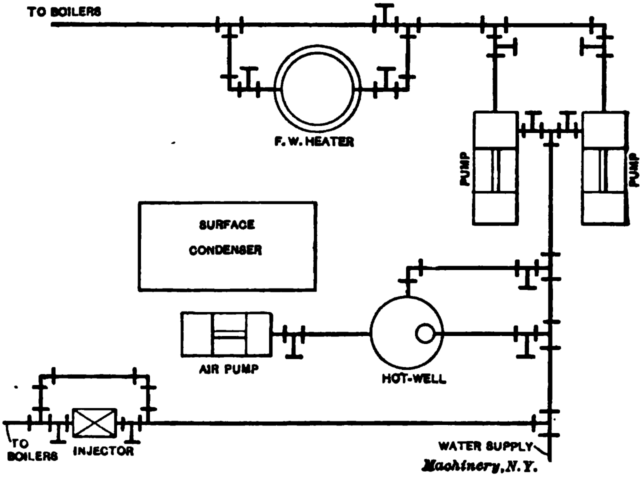

Fig. 59. Feed Piping for Plant with Surface Condenser

Fig. 60 shows a typical arrangement for a condensing system using a jet condenser, and having both primary and secondary feed-water heaters. The receiving tank and injector arrangements are the same as in Fig. 58, the only difference being that there are two heaters instead of one. The primary heater is placed next to the pumps and is supplied with exhaust steam at condenser pressure from the engines. The secondary heater is next to the boilers and is furnished with exhaust from the feed pumps and other auxiliaries. The discharge from the condenser pump is turned into the sewer.

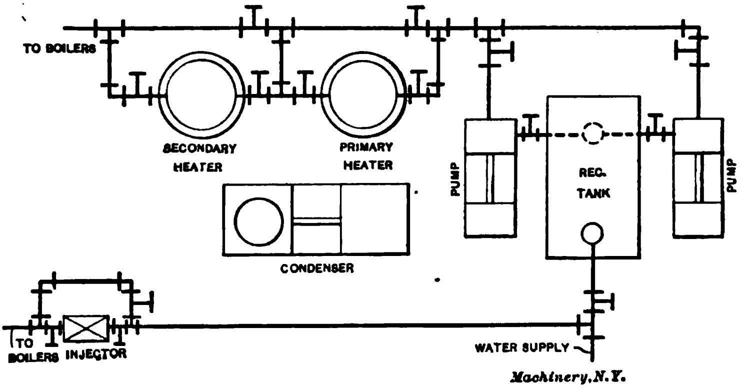

Fig. 60. Feed Piping for Plant with Jet Condenser

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