Condensers

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CONDENSERS.

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LECTURE VIII.—CONDENSERS.

BY F. R. LOW.

If you build up a solid column of bricks the pressure which it exerts on its base will increase directly as the height of the column. A column ten feet in height will press twice as hard on its base as a column five feet high, and a column 100 feet high ten times as hard as a 10-foot column.

Now, the point I want to make is that the pressure per square inch of base depends altogether on the height and not on the width or diameter of the column. A column 2 feet square will, it is true, press on its base with four times the pressure of a column one foot square and of the same height, because there are four times as many bricks in it and it weighs four times as much, but there is also four times as much base to it, so that the pressure per square inch of base is entirely independent of the cross section and depends upon the height alone.

The same thing is true of water. A cubic foot of fresh water weighs 62.355 pounds at 62 degrees Fahrenheit. It is easy to remember this weight approximately, for it is the same as the degrees and 62 is a standard temperature in dealing with water. A cubic foot rests on a base of 144 square inches and is a foot high, so that the pressure per square inch on the base would be

$$62.355 \div 144 = 0.433 \text{ of a pound}$$

and for every foot in height that we build our column or fill our pipe with water we gain 0.433 of a pound pressure per square inch. If one foot or 12 inches gives us 0.433 of a pound it would take a column

$$12 \div .433 = 27.71 \text{ inches}$$

in height to exert a pressure of one pound per square inch. For every 27.71 inches in vertical height between the point at which you are measuring and the top of a column of still water there will be a pressure of a pound to the square inch, and it makes no difference whether you are measuring the pressure at the bottom of a one-eighth inch pipe, a twenty foot stand-pipe, or a lake, or the ocean itself. Every once in a while we have to explain this to the man who believes it takes more power to feed into the bottom of a tank than into the top, on account of the weight of water in the tank. The bottom of the tank holds up all the water except the column directly over the opening of the delivery pipe, so that the additional pressure on the pump is due only to the depth of water in the tank, not to the size of the body, and it is impossible to feed into the top without increasing the height of the column fully as much. It makes no difference whether the height is due to the depth of the water inside the tank or an additional length of pipe outside. The difference between the water and the column of bricks is that while the pressure of the latter can act only vertically that of the water can act in all directions so that as you lower a body into the water the pressure upon its surface in all directions increases one pound per square inch for every 27.71 inches of depth of water above it. In Fig. 1, for instance, the pressure due to the column of water *P* will act upward upon the piston *A* and sidewise upon the pistons *B* and *G* as well as downward upon the piston *D*.

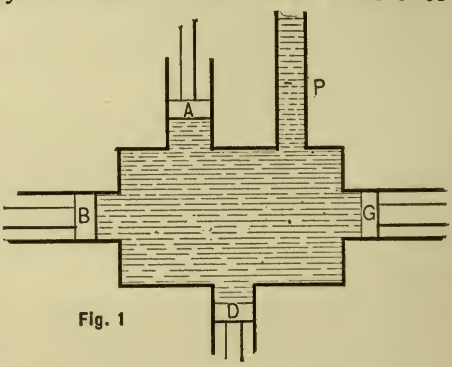


Fig. 1

We live at the bottom of an ocean of air. The winds are its currents, we can heat it, cool it, breathe and handle it, weigh it, and pump it as we would water. The depth of this atmospheric ocean cannot be determined as positively as could one of liquid, for the air is elastic and expands as the pressure decreases in the upper layers. It is variously estimated at from 30 to 212 miles. We can, however, determine very simply how much pressure it exerts per square inch.

Here is a U-tube, Fig. 2, into which a quantity of mercury has been poured. It stands at an equal height in both legs. Into one leg I pour some water on top of the mercury, and the mercury is depressed in that leg, and rises in the other. The difference in level of the mercury is a measure of the weight or downward pressure of the water.

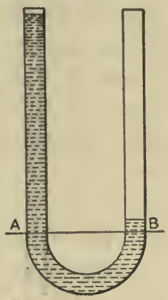
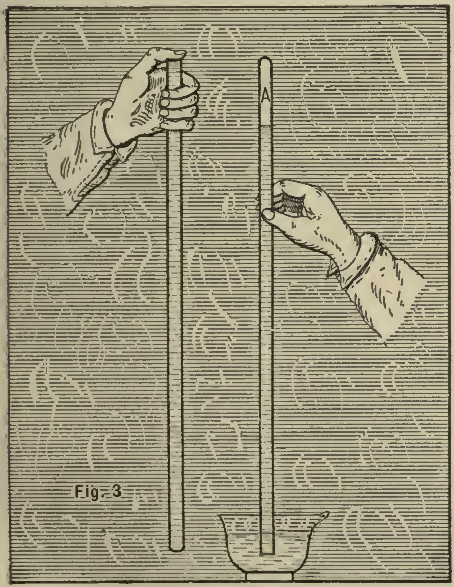


Fig. 2.

The mercury below the line *A B* balances in both legs and the mass of mercury above that line in the right leg just balances the weight or pressure of the water in the other. The pressure of the atmosphere makes no difference in this experiment, for it is exerted on both columns equally. Now we can find the pressure of the atmosphere in a similar way by making it act on one end of the mercury column as does the water here and keeping it away from the other.

Here is a glass tube about a yard in length and filled with mercury. Closing one end with my thumb to prevent a premature escape, I invert it in a bowl of mercury as in Fig. 3. This is handier than the U-tube but the principle is the same. The bowl is in effect the other leg of the tube and no matter what its size may be the atmosphere exerts a certain pressure on each square inch of its surface, except at the point covered by the tube, and here the mercury rises until it forms a column high enough to exert the same pressure per square inch, so that the height of the column is a measure



of the atmospheric pressure. This column will be approximately 30 inches, and as a cubic inch of mercury weighs about a half a pound, each two inches of height will be equal to a pound pressure, so that the pressure exerted by the atmosphere is about 15 pounds per square inch. This pressure depends first upon the nature of the atmosphere. You know that steam or aqueous vapor is lighter than air at the same pressure, so the more moisture there is in the air the lighter the column of atmosphere above us, and the less the height to which our column of mercury will rise to balance it. Also the warmer the air becomes the lighter it is. Again, if we carry our apparatus to the top of a high mountain we shall find a considerable difference in the height of the column, because we have lessened the height of the column of air above us. This arrangement, which is known as a "barometer," is therefore of use in indicating coming changes in the weather, and elevations above the sea level, at which our experiment is supposed to have been made.

We are then subjected all the time to a pressure of 15 pounds to the square inch all over our bodies, yet we suffer no inconvenience, in fact, it took mankind a long while to find it out, because the pressure is the same in all directions, it is exerted inside as well as out, and there is no unbalanced pressure. It is only when the atmospheric pressure is removed from one side and allowed to act upon another that we get any effect. In a space from which the air has been removed without allowing anything else to enter, a "vacuum" is said to exist, and the vacuum is more or less complete according to the more or less complete removal of the air. In the space *A* in Fig. 3, exists the most perfect vacuum we are able to create, for the mercury in receding has left nothing behind it, except possibly a little mercurial vapor if there have been no air bubbles and no moisture between the mercury and the glass. With this complete vacuum above it the mercury will rise about 30 inches, and we would say that we had "30 inches of vacuum." What we mean is that the pressure has been so completely removed from the space *A* that the atmospheric pressure is able to support 30 inches of mercury against the pressure that is left. Suppose we let a little air into *A*. The mercury would fall more or less according to the amount of air admitted, because this air would exert some pressure, there-would be less difference between the pressure in *A* and that of the atmosphere, and the atmosphere would be able to support a lesser column against this greater pressure. If the column now was 18 inches high we would say that we had 18 inches of vacuum, and should mean that the atmospheric pressure could support 18 inches of mercury against the pressure in our vacuum. These are the "inches" of vacuum upon the ordinary vacuum gage. When the pointer stands at 26 inches it means that there is difference enough between the pressure in the condenser and that of the atmosphere to support a column of mercury 26 inches high. If with an absolute vacuum the barometer stood at 30 inches, and if a cubic inch of mercury weighed half a pound the atmospheric pressure would be 15 pounds, and two inches would equal one pound. As a matter of fact the height of the barometer varies and mercury weighs only .49 of a pound to the cubic inch, so that the atmospheric pressure is nearer 14.7 than 15 pounds. When you put your hand over an opening into a space containing a vacuum you feel it drawn to and held down very hard upon the opening. This is due not to any attractive power of the vacuum, but to the pressure of the atmosphere upon the back of your hand unbalanced by an equal pressure on the area in contact with the opening to the vacuum.

Here is an implement which every schoolboy knows under the name of a "sucker," a circular pad of leather, thick, but pliable, with a string through its center. It has been soaking in water. I press it against the smooth wooden seat of this chair and am able, you see, to lift the chair with a string. The boys used to get themselves into disrepute with the householders in the vicinity of the school by pulling the bricks out of the sidewalk in this way. This action is not due to any attractive or adhesive property of the leather, but to the fact that there is a pressure of about 15 pounds per square inch pushing the leather against the chair, and the atmosphere, owing to the more or less complete contact of the wet leather with the surface on which it rests cannot get to the under surface to balance it. The disk is four inches in diameter, having an area of 12.5 square inches, and the atmosphere exerts a pressure on its surface of $14.7 \times 12.5 = 183.75$ pounds with which the "sucker" would resist separation from the surface to which it was attached, if the pressure was entirely removed from its under-side, and other surface, and the leather perfectly air tight.

We are accustomed to say that water is "sucked up" or "drawn up" by a pump as though there was some pulling property to the vacuum which it creates, when as a fact the water is pushed up by the atmospheric pressure acting on the surface of the water in the well. If in Fig. 3, we had a tube of water instead of mercury we should find that the water would rise in it about 34 feet instead of 30 inches. We have seen that it takes a column of water 27 71 inches high to exert a pressure of one pound, then the atmospheric pressure of 14.7 pounds could support a column of

$$\frac{27.71 \times 14.7}{12} = 33\ 94 \text{ ft.}$$

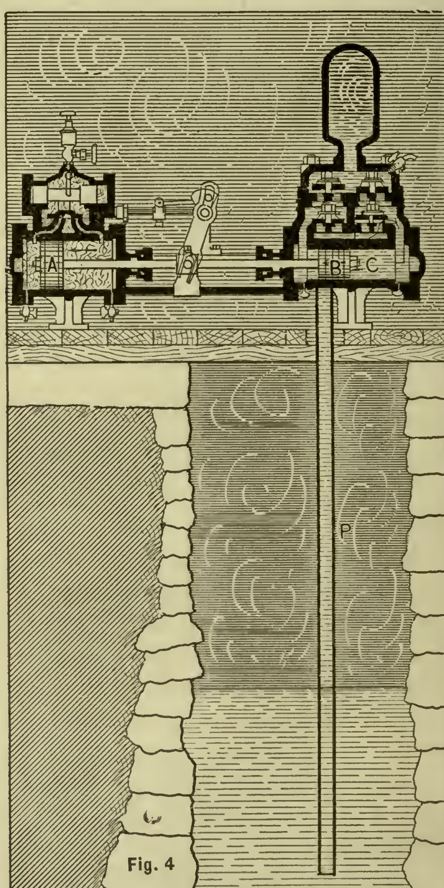


Fig. 4

In Fig. 4, we have a steam pump drawing water from a well. Steam acting on the piston *A* pushes the piston *B* toward the left, forcing the water before it through the upper valve to the discharge pipe and leaving behind it a more or less complete vacuum in the space *C*. Connected to the space *C* through the lower valves is the pipe *P*, the lower end of which is immersed in the water of the well. Here we have a reproduction (Fig. 4) of Fig. 3. The well is the bowl, the pipe *P* is the glass tube, the vacuous space *C* corresponds with the vacuous space *A*. There is this difference, however, the pipe P is not so long but that the atmospheric pressure can force the water clear through it, through the valve into the cylinder C, ready to be forced out again when the piston moves in the other direction. The difference in pressure between the cylinder and the atmosphere must be sufficient to lift the water from the level in the well to the level of the pump cylinder, to lift the valve and to induce a flow sufficiently to keep the cylinder full behind the receding piston, and these considerations limit the distance that we can place a pump above its source of supply, in other words its "lift." In the first place we cannot get a perfect vacuum in contact with water. You remember that the boiling point of water depends upon the pressure. In a boiler with a pressure of 60 pounds by the gage, the water will not boil until it is over 300°. Under the pressure of the atmosphere it boils at 212°, and as you reduce the pressure below that of the atmosphere the boiling point lowers rapidly. You can even boil water at 32° if you reduce the pressure sufficiently. In the table on page 8 are shown the relations of pressure and temperature for water at from 32° to 212°. This means that if we had an arrangement like Fig. 5, starting with a complete vacuum in chamber A the mercury in the tube would not rise above the level in the cup because there is a complete vacuum both in A and B. Now if water of 32° be introduced into A it would boil and give off vapor until the pressure in A arose to .089 of a pound, and the mercury would rise in B 181 thousandths of an inch. A complete vacuum as given by this table is 14.7 pounds, or 29.922 inches, but the introduction of the water even at 32° has reduced the vacuum to 29.922—.181=29.741 inches. The heat necessary to convert the water into vapor, which you will remember from an earlier lecture was considerable in amount and was called the latent heat, coming from the water and its surroundings, the water would be frozen, and I have seen ice made by simply spraying water into a space in which a high degree of vacuum was maintained. If the water was 60° the vacuum would be impaired .571 of an inch or .254 of a pound. This is the reason it is so difficult to pump hot

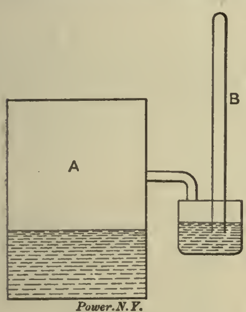


Fig. 5.

water. If the water in Fig 4 was 150° the space *C* left by the piston, instead of being a nearly complete vacuum, would be filled with steam of 3.708 pounds pressure, leaving only 14.700—3.708=

Temperature	Pressure.		Vacuum.	
	Inches of mercury.	Lbs. per square inch.	Inches of mercury.	Lbs. per square inch
Fahrenheit.	Inches.			
32°	.181	.089	29 741	14.611
35°	.204	.100	29 718	14 600
40°	.248	.122	29 674	14.578
45°	.299	.147	29.623	14.553
50°	.362	.178	29.560	14 522
55°	.426	.214	29.496	14.486
60°	.517	.254	29.405	14 446
65°	.619	.304	29 303	14.396
70°	.733	.360	29.189	14 340
75°	.869	.427	29.053	14.273
80°	1 024	.503	28.898	14 197
85°	1 205	.592	28 717	14 108
90°	1.410	.693	28.512	14 007
95°	1.647	.809	28.275	13.891
100°	1 917	.942	28.005	13.758
105°	2.229	1.095	27.693	13 605
110°	2 579	1.267	27 343	13 433
115°	2 976	1.462	26 846	13.238
120°	3.430	1 685	26 492	13.015
125°	3.933	1 932	25 989	12.768
130°	4 509	2 215	25.413	12.485
135°	5.174	2 542	24.748	12.158
140°	5.860	2 879	24.062	11.821
145°	6 662	3 273	23.262	11.427
150°	7 548	3 708	22.374	10 992
155°	8.535	4 193	21.387	10 507
160°	9.630	4.731	20.292	9.969
165°	10 843	5 327	19.079	9.373
170°	12.183	5 985	17.739	8 715
175°	13.654	6 708	16.268	7 992
180°	15.291	7 511	14.631	7 189
185°	17.044	8.375	12.878	6 325
190°	19 001	9 335	10.921	5 365
195°	21 139	10 385	8 783	4 315
200°	23.461	11 526	6.461	3 174
205°	25.994	12 770	3.928	1 930
210°	28.753	14 126	1.169	.574
212°	29.922	14 700	0.000	0.000

10.992 pounds to raise the water and force it into the pump. If the water was 212° it would give off steam equal in pressure to that of the atmosphere, and we have no available force at all.

These relations between pressure and temperature are simply those for aqueous vapor or steam. When air is present the pressure will be higher for a given temperature. For this reason the vacuum or pressure in a condenser is not that due to the temperature of its contents as given in a table of the physical properties of steam for it is not steam alone with which we are dealing but a mixture of steam and air.

You are now in a position to appreciate what is meant by "absolute" pressure. It is the pressure reckoned from a complete vacuum as are the pressures in the above table, and atmospheric pressure which is the zero of the ordinary steam gage and of what

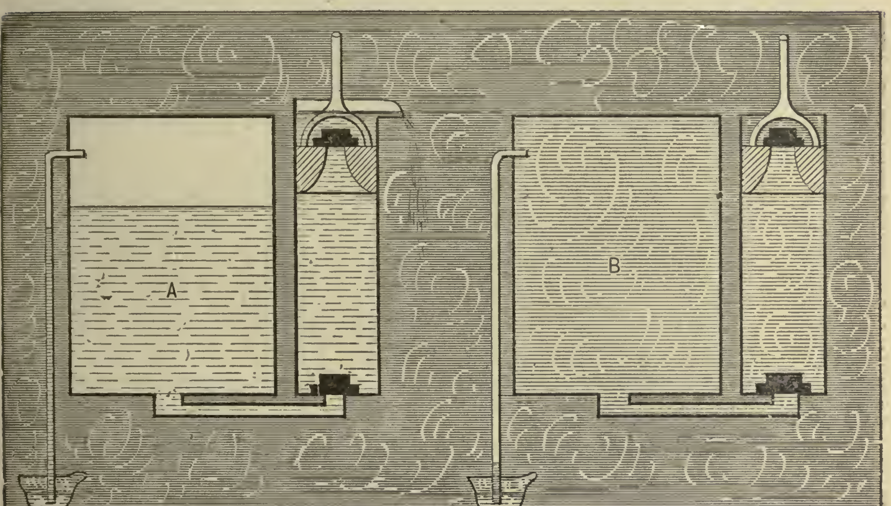


Fig. 6.

is referred to as "gage pressure" is about 14.7 pounds absolute, varying with the barometer. In order to get the "absolute" pressure then we must add the barometer's pressure, 14.7 pounds, or 15 if we do not care to be very precise, to the pressure indicated by the gage. The steam tables are given in absolute pressures, and we have to take the absolute, not gage pressure, when laying out the expansion line, or figuring problems in which expansion is involved.

There is this difference between pumping air and water, that water is either there or not there; there is no half way about it. It is neither expansible nor compressible by change of pressure, and it may be handled in mass. In Fig. 6, for instance, we have a closed vessel of water at *A* and another of air at *B*. Now when the pump connected with *A* is operated a volume of, say one-fifth, of the water is removed, the water left in the tank falls, there is nothing to take its place, and a practically complete vacuum is left behind. But in the case of the air, when one-fifth of the volume is removed by a stroke of the pump, the remainder, instead of assuming a level and leaving a vacuum at the top as the water did, expands and fills the whole space. Before the pump was operated the air was at atmospheric pressure, say 15 pounds to the square inch absolute. The operation of the pump removes one-fifth of its volume, and the remaining four-fifths expands to fill the complete volume. In this expansion, its pressure would be reduced to four-fifths of the former pressure, equal to 12 pounds, so that instead of having at once a complete vacuum in the chamber as with the water, we have only reduced the pressure three pounds below the atmospheric pressure outside, and if a column of mercury be connected with the chamber, as shown at *B*, we shall find that in the case of the air it will only stand about 6 inches in height, for the sustaining force is the difference between the inside and outside of the chamber, which is $15 - 12 = 3$ pounds, and as one inch in height exerts a pressure of one-half pound per square inch on its base, 3 pounds would balance $3 \div .5 = 6$ inches in height. In this case there is said to be 6 inches or 3 pounds of vacuum in the vessel.

By further reducing the air in the vessel, we can produce greater differences in pressure between the inside and outside and the atmosphere will press the harder toward the inside of the vessel, its pressure being measured in the inches of mercury which it will lift, or the pressure per square inch which it exerts. All questions in regard to a vacuum become plain when we consider that the atmosphere itself exerts a pressure of nearly 15 pounds, and measure everything from an absolute zero 15 pounds below the atmospheric pressure.

When an engine is run without a condenser the steam with which the cylinder is filled at the end of the stroke has to be forced out against the pressure of the atmosphere, about 15 pounds to the square inch. It is possible from the nature of steam to remove the atmospheric pressure with, in most cases, a decided gain. One pound of steam at atmospheric pressure occupies 1,642 times as much room as it does in the state of water. If therefore when the stroke has been completed and we are ready for the piston to come back we inject a little cold water into the spent steam, it will condense to about one 1600th of

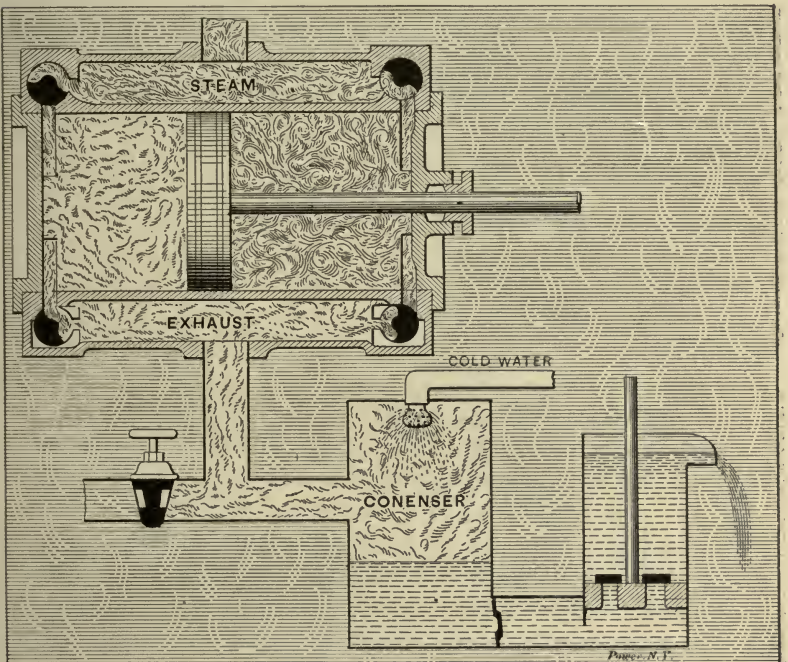


Fig. 7.

its volume, and leave a vacuum into which the piston can return without having to force back the atmosphere. This is the way the earlier engines were run, the condensation taking place in the cylinder itself, and, moreover, the vacuum was all that made the engine operative, for the steam carried was but little above atmospheric pressure. Watt's introduction of the separate condenser was his greatest contribution to the steam engine, and constituted his most important invention, for he was not as you know the inventor of the engine, but its improver. The operation of the condenser is shown in Fig. 7. The denser steam in the stuffing box end of the cylinder is pushing the piston to the left, forcing the spent steam of the previous stroke to the condenser where, instead of having to be forced out against 15 pounds pressure of the atmosphere, it is condensed by coming into contact with a spray of cold water. The condensed water, the water of injection and the air which has entered with the steam and by leakage are drawn out by an "air pump," and the comparatively small volume which it has to expel against the atmospheric pressure, leaves a large margin of power gained after that required to run the pump is deducted.

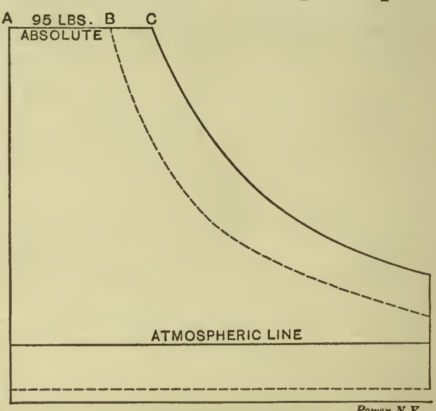


Fig. 8.

Let us first consider the nature and extent of the saving due to a condenser, and when it is and is not advisable to use it.

Suppose we have an engine with an initial pressure of 80 pounds gage,=95 pounds absolute, cutting off at one-third. The mean effective pressure, if the engine ran non-condensing and made the perfect diagram represented by the full lines in Fig. 8, would be 51.25 pounds. If we put on a condenser and reduce the back pressure from that of the atmosphere, say 15 pounds absolute, to 3 pounds absolute, the diagram, to give the same mean effective pressure representing the same load on the engine, would take the form shown by the dotted lines.

In the non-condensing diagram, the boiler has to fill the cylinder up to the point *C*, and the volume of steam at cut-off is proportional to the line *A C*. In the condensing engine the steam is cut off at *B*, and the steam is proportional to the line *A B*. Now *A C* is .33⅓ of the volume of the cylinder and *A B* is only .23256, so we have apparently saved

$$\frac{.33333 - .23256}{.33333} \times 100 =$$

about 30 per cent. (clearance neglected).

Again, suppose we have a throttle governed engine cutting off at two-thirds the stroke, with an initial pressure of 50 pounds, gage,=65 absolute, running non-condensing, it would make, theoretically, the diagram indicated by the solid lines in Fig. 9, and exert a mean effective pressure of 45.62 pounds. If we put on a condenser and reduce the back pressure to 3 pounds, in which case we should as before realize a vacuum of 12 pounds or 24 inches, the cut-off would remain at two-thirds, but the initial pressure would be lowered, as shown by the dotted lines, to 38 pounds. While the volume up to cut-off is the same in each case, the pressure is lowered, and the same volume of lower pressure steam weighs less. Suppose the size of the cylinder was such that it took a cubic foot to fill it up to cut-off. Then, when making the non-condensing diagram shown by the solid lines in Fig. 9, it would take a cubic foot of 50-pound steam (65 absolute) which would weigh .1519 of a pound. When making the condensing diagram shown by the dotted lines, it would take the same volume of 53-pound (absolute) steam, which would weigh .1255 of a pound. An apparent saving of

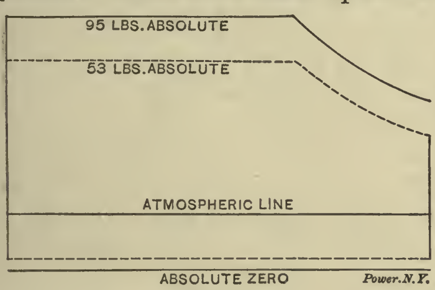


Fig. 9.

$$\frac{.1519 - .1255}{.1519} \times 100 = 17.38 \text{ per cent.}$$

This is not, however, a pure saving. The most important charge against it is the reduction of available temperature for the feed water. With an engine exhausting at atmospheric pressure the exhaust steam has a temperature of 212°, and by the use of a suitable heater it is possible to get the feed water nearly as hot. With a condenser in which the absolute pressure is reduced to three pounds, the temperature of the exhaust steam is only 141 62, and the temperature of the hot-well, or the discharge from the air-pump, would be in practice from 110° to 120°. Very careful practice might raise it to 130° but the temperature of the hot-well will always be considerably less than that due to the pressure of the steam or vapor in the condenser, on account of the impossibility of bringing every particle of steam into contact with the water when only the exact quantity of water theoretically needed to condense it is used, and the raising of the pressure in the condenser by the presence of air without a corresponding increase of the temperature. Suppose the hot-well temperature is 110° as against the 210° that we might have by running non-condensing. There would be a loss of approximately 10 per cent, for there is a gain in efficiency of one per cent for about each ten degrees we heat the feed water. Even if we kept the hot-well up to 130° there would be a fall in the available temperature of feed of 80°, or approximately eight per cent.

Again, it takes a great deal of water to condense the steam, and all this water, as well as the condensed steam and the air, which has worked in with it and by leakage, must be pumped out against the pressure of the atmosphere, so that the cost of supplying the condenser with water and of operating the air pump must be deducted from the apparent gain. There is also the interest on the extra cost of the condenser, the extra repairs, supplies, insurance and attendance if the condenser plant is large enough to require especial attention.

Here is a little extract from Peabody's Steam Tables, giving the amount of the heat contained in a pound of steam at absolute pressures of from 10 to 25 pounds, or from about 5 pounds below the atmosphere to about 10 pounds above. The column marked "Heat of the Liquid" gives the number of heat units that we would have to put into a pound of water to bring it from 32° up to the boiling point (given in the second column), at the corresponding pressure in the first column. It is unnecessary to tell those of you who have read the previous lectures that a "heat unit" or "British Thermal Unit" is the amount of heat necessary to raise a pound of water one degree. In the column marked "Heat of Vaporization" is given the "latent heat" or the number of heat units necessary to evaporate the pound of water into steam after it has been raised to the boiling point. The "Total Heat" is the sum of the two. Now suppose the terminal pressure in the cylinder, that is, the pressure at the time the exhaust valve opens, is 5 pounds above the atmosphere, or say 20 pounds absolute, then every pound of steam used will carry to the condenser 1151.5 heat units. Suppose the hot-well temperature is 120°. A pound of water at 120° contains 88.1 heat units above 32°. Suppose again that the temperature of the injection water was 60°. A pound of water at 60° contains 28.12 heat units above 32°. Then each pound of water in raising from 60 to 120° will absorb 88.1 — 28.12 = 59.88 heat units.

To condense the pound of steam and reduce it to water of 120° we must take from it 1151.5 — 88.1 = 1063.4 heat units.

Pressure, Pounds per Square Inch	Temperature, Degrees Fahrenheit.	Heat of the Liquid.	Total Heat from 32°.	Heat of Vaporization.
10	193 25	161 9	1140.9	979.0
11	197 78	166 5	1142 3	975 8
12	201 98	170 7	1143.6	972.9
13	205 89	174.6	1144.7	970 1
14	209.57	178.3	1145 8	967.5
15	213.03	181.8	1146.9	965 1
16	216 32	185 1	1147.9	962.8
17	219 44	188 3	1148 9	960.6
18	222 40	191 3	1149 8	958.5
19	225.24	194 1	1150 7	956.6
20	227.95	196 9	1151.5	954.6
21	230 55	199 5	1152.3	952 8
22	233.06	202 0	1153.0	951 0
23	235 47	204 5	1153 7	949.2
24	237.79	206 8	1154.4	947 6
25	240 04	209 1	1155.1	946 0

As one pound of water will absorb 59.88 units it will require to condense each pound of steam

1063.4 ÷ 59.88 = 17.7 pounds of injection water.

It will be noticed that the number of heat units absorbed by one pound of water is very nearly the difference in temperature between the injection water and the hot-well. This difference in the case in question would have been 120 — 60 = 60 heat units, and is near enough in any case for practical purposes. To find the amount of water required for a condenser, subtract the heat units contained in a pound of water at the hot-well temperature from the number of such units contained in a pound of steam of the terminal pressure. These values can be gotten from a table of the Physical Properties of Steam, to be found in any engineer's reference book. Divide this value by the difference between the temperature of the injection and of the hot-well, or by the rise in temperature of the circulating water in the case of the surface condenser, and you get the number of pounds of injection or circulating water required per pound of steam. Multiply this by the number of pounds of steam required per hour per horse-power, and you get the injection per hour per horse-power. Multiply this again by the horse-power developed, and you get the injection required to run a given engine with a given load.

When the only water available for injection is foul, and would make a mixture in the hot-well, that would not do to feed to the boilers, a surface condenser may be used. This is the general practice on sea-going steamers where the injection water is salt, and it is necessary to use the same boiler water over and over. Did you ever think what an immense amount of water is boiled into steam to run one of the great liners? The Paris has 30,000 horse-power. Suppose she runs on 13 pounds of steam per hour per horse-power, her boilers would evaporate over a million gallons of water a day, a good supply for a sizable town. Of course they cannot afford to foul this by mixing the salt sea water with it, so they condense it by letting it come in contact with metal surfaces kept cool by sea water flowing upon the other side, but always separated from the condensed steam. In this way it will be seen the cooling or circulating water is kept entirely separate from the condensed steam and the latter can be safely returned to the boilers, while any sort of non-corrosive liquid can be used for cooling purposes. We have heard of plants in large cities where water was taken from the sewer, passed through a surface condenser, and returned to the sewer again.

It will be noticed that the exhaust steam carries to the condenser a very large percentage of the heat which it brings from the boiler. A pound of steam at 80 pounds gage, 95 absolute, contains 1180.7 heat units. Suppose 20 pounds of this steam are required per hour per horse-power. Then 20 pounds of steam will do $33{,}000 \times 60 = 1{,}980{,}000$ foot pounds of work, one pound will do $1{,}980{,}000 \div 20 = 99{,}000$ foot pounds. As one heat unit is equal to 778 foot pounds, the number of heat units transformed to work would be $99{,}000 \div 778 = 127.4$ heat units.

$$1180.7 - 127.4 = 1053.3.$$

We have 1180.7 units of heat taken from the boiler, 127.4 of them converted into work and the balance, barring the trifling loss from radiation, going out in the exhaust. It follows that if we have any use for heat at anything under the temperature of a reasonable exhaust, it would be bad engineering to let this heat, which might be applied to the purpose, escape into the river in the overflow from a hot-well. One case then where it is inadvisable to use a condenser is where it is possible to use the exhaust steam to advantage.

Again, suppose we had 80 pounds initial pressure and instead of cutting off at one quarter we carried the 80 pounds for the full stroke, and exhausted at atmospheric pressure our mean effective pressure would be 80 pounds. Now, if we put on a condenser giving us 12 pounds of vacuum, we must reduce the initial to 68 pounds gage. The volumes used would be the same in both cases. Steam of 80 gage (95 absolute) pressure weighs .2165 of a pound; at 68 pounds gage (83 absolute), .1908, a saving of

$$\frac{.2165 - .1908}{.2165} \times 100 = 11.8 \text{ per cent.}$$

Now if we lose ten per cent by reducing the temperature of our feed water, and it takes two per cent to run the air pump, we shall be worse off with the condenser, than without it, to say nothing of the investment in it, the cost of oiling, packing, attending it, and keeping it in repair. Evidently here is another case where we would be better off without the condenser.

In a well-designed engine, the power required to operate the pumps may be less than one per cent of that developed by the main engine, and is sometimes as high as three per cent. This percentage or more of the steam supplied may be used according as the pump is operated from the engine itself, or by an independent cylinder more extravagant in the use of steam.

In order to understand one of the points that bears on the desirability of the condenser in a special case, it is necessary to understand something of the cylinder condensation. When steam contains just the number of heat units per pound given in the tables, that is, just enough to evaporate it into steam, it is said to be "saturated." This means that it is saturated with heat, not with moisture. The term is apt to be misunderstood, and I have frequently talked with engineers who could not get rid of the idea that "saturated" steam must be soaking wet. The ordinary steam that we get from boilers carries with it more or less moisture, and steam is "commercially dry" when it has no more than two per cent by weight of such moisture. If we apply heat to such steam, and dry it out or evaporate the moisture, we shall have "saturated" steam at the instant that all the moisture is gone, and if we continue the heating so as to increase the temperature above that due to the pressure, we shall have "superheated" steam.

Now, unless steam is superheated, it cannot lose a particle of heat, except by expansion, without a corresponding amount of condensation. Steam of 80 pounds gage (95 absolute) pressure has a temperature of about 324° F. As it is expanded in the cylinder after cut-off its temperature falls, and during the exhaust stroke the temperature is that due to the back pressure; 212° if the exhaust is against the atmosphere, 141.6° with a condenser reducing the absolute back pressure to 3 pounds. As a consequence, the cylinder and piston heads, the ports, and wall of the cylinder, having been in contact with this cooler steam, have had their temperature reduced and when the live steam enters at the beginning of the stroke, it finds itself in contact with surfaces comparatively chilly, and therefore has to part with enough heat to raise these surfaces to its own temperature before it can continue to exist as steam in contact with them. As a result, there is a large amount of condensation at the beginning of the stroke, and this continues up to the point of cut-off. As the steam commences to expand its temperature is reduced, the surfaces begin to give back the heat that has been expended upon them, and the water resulting from the initial condensation commences to boil under the diminished pressure, as did the water when we cooled the flask in lecture I. Meantime, however, the piston is uncovering new cylinder wall, which requires to be heated, and this action will continue to a point where the temperature which the wall has assumed equals the temperature of the expanding steam. Beyond this point all the surfaces are hotter than the steam and the re-evaporatlon is more rapid. Except on very slow running engines, however, this re-evaporation during the working stroke is not very extensive. In a good tight engine at ordinary speeds the expansion line usually agrees very well with the theoretical curve, commonly rising a little above it at the later portion, showing that the re-evaporation but little more than makes up for the condensation due to the conversion of some of the heat units into work, and to radiation. In this way the re-evaporation during the working stroke is a benefit, but the greater part of the evaporation occurs during the exhaust stroke, when the resulting steam can do no good, but is escaping to the atmosphere, or the condenser. When the pressure is reduced by the opening of the exhaust valve, the moisture in the cylinder, being above the boiling point at the reduced pressure, passes rapidly into steam, the heat for its continued evaporation being furnished by the containing surfaces, and these containing surfaces chilled by this abstraction of heat, must be heated again on the following stroke. The wall never gets as cool as the exhaust temperature, and probably never as hot as the initial steam. The longer the time it is exposed to a temperature lower than the initial and the lower the temperature of the exhaust, the greater will be its range of variation. Notice that the surfaces must give up to the outgoing steam exactly as much heat as they receive from the incoming steam. They certainly cannot give up any more, and if they did not give up as much as they got, heat would accumulate and melt them down.

This subject of cylinder condensation is one of the most interesting and important connected with steam engineering. Experiments indicate that the loss from this action is rarely less than 20 per cent in simple unjacketed cylinders of ordinary automatic engines, and it may be much more. The point I want to call your attention to in connection with our present subject is that the greater the difference between the initials and back pressures, the hotter the steam the cooler the exhaust, the greater this action and loss will be. Further, the earlier in the stroke the cut-off occurs the greater the initial condensation, because of the greater variation of temperature on the working stroke and the greater proportion of the time that the temperature of the steam in the cylinder is below that of the steam chest. The condensation will also increase with any increase in the proportion which the area of the containing surface bears to the volume of steam contained.

In an indicator diagram like Fig. 10, the space *E B* represents the volume of the cylinder including clearance up to the point of cut-off, while the shaded area is proportional to the work done. The volume that must be filled with steam at each stroke will bear the smallest proportion to the work done when as in Fig. 3 the cut-off is at such a point that expansion extends just to the line of back pressure, making the diagram end in a point; and compression extends just to initial pressure. The higher the initial pressure and the lower the back pressure, the greater will be the number of expansions used, and the greater the area of the diagram compared with the volume up to cut-off. But every engineer knows that, notwithstanding the fact that the steam accounted for by the diagram per horse-power would be the least in amount under these conditions, it would be very poor economy to run an engine with so light a load. We might continue the reduction of the diagram on these lines until the power developed is barely sufficient to run the engine itself, in which case, even if we got a very low rate of steam consumption per indicated horse-power, the little useful power we would get would be very expensive. As a matter of fact, however, we should use more steam per indicated horse-power, for the gain by expansion falls off rapidly as the number of expansions is increased, while the loss by cylinder condensation increases at a rapid rate. Conseqently, there is a point where the loss from cylinder condensation equals the gain from increased expansion, and any increase of expansion will result in a loss. The more power we can get out of the cylinder the less proportion will the radiation and frictional losses bear to the power delivered to the shafting, so that it is not found economical in practice to cut off much earlier than one-

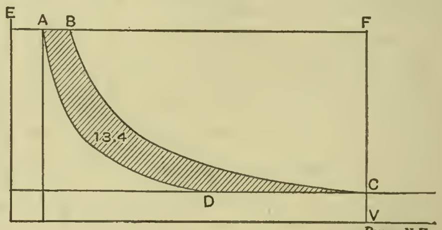


Fig. 10.

quarter stroke, in an ordinary single-cylinder non-condensing engine without jackets; nor to expand much below the atmosphere with a simple condensing engine. Obviously then, if an engine is cutting off at one-fifth stroke, or earlier, there will be little chance of increasing the economy by putting on a condenser. It is possible to extend the point of cut-off without increasing the mean effective pressure by lowering the boiler pressure, or throttling it at the engine, but here the efficiency of the high pressure steam is sacrificed, and it is still an open question how far it is safe to go in this direction. I commend it to you as a subject for profitable discussion, whether with an underloaded engine, condensing or not, it is profitable to reduce the initial pressure and if so under what circumstances and to what extent.

Condensers may be divided into two general classes. Those whose air pumps are driven by the main engine.

Those having their own independent motive power.

The first type includes belt and gear driven pumps as well as those directly attached to the working parts of the engine itself. The advantage claimed for them is that the power required to drive them is generated in the large economical cylinder to much better advantage that it can be in a small cylinder of a direct acting pump, such as is usually used to operate the independent condenser.

On the other hand, the advocates of the independent condenser claim that while the attached air pump is constrained to move at the same speed as the main engine or a speed proportional thereto, regardless of the amount of work it has to do, the independent air pump can be run fast or slow according to the amount of water passing, which varies with the load and the vacuum carried. They further claim that the steam from the cylinders which operate the pump can be used to heat the feed-water, thus doing away with the loss noted above, and that as practically all the steam required to run the pump is thus utilized, it does not matter if the pump is not so economical as the main engine. Many of both types of condenser are used and each has its advocates. If one is very decidedly better than the other, it will in time appear, and the fittest will survive or perhaps as in many other cases it will be found that each is particularly adapted to special circumstances.

The amount of work done by an air pump depends not upon the size of the piston, or the speed at which it runs, but upon the amount of water and air that it forces out of the condenser against the pressure of the atmosphere. In Fig. 11, when the bucket or piston rises, it leaves a vacuum behind it. Suppose that the water line *A B* just reached the diaphragm *C D* when the bucket was in its highest position, without lifting the valves. Then when the bucket descended it would leave a vacuum above it, and if no water is let in to raise the level *A B* the bucket will continue to move up and down with a vacuum above and below it, without doing any work or calling for any power except to overcome its own friction. Now if we let a little water into the chamber *E*, a corresponding amount will pass through the bucket on its downward stroke, increasing the amount above the bucket and the water line *A B* will come in contact with the diaphragm *C D* before the upward stroke is completed, lifting the valves in that diaphragm, andmaking the pump complete the stroke against the atmospheric pressure. This is where the work of the pump comes in, and this will be dependent upon the quantity of water passed, for if we put in twice the amount of water, the valves in *C D* will be open twice as long and the bucket travel twice as far against the atmospheric pressure. Of course there would be a saving, so far as friction is concerned, if the pump could be run slowly enough to completely fill at each stroke, instead of making several strokes to do an equivalent amount of work, but it is not constantly working against a vacuum as many suppose.

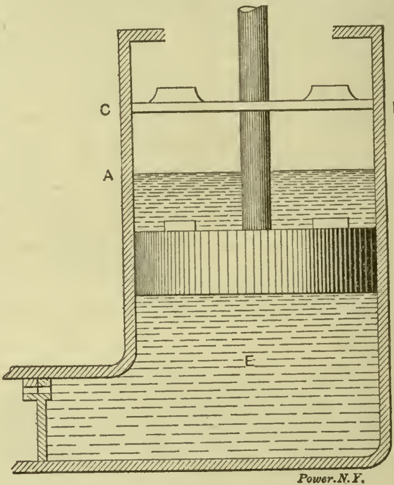


Fig. 11.

It is quite generally conceded that the vertical form of air pump, although necessarily single acting, is preferable to the double acting horizontal pump. This is due to the certainty of its action in taking water through the bucket valves, to the quick and positive closure of the valves, to the facility with which the water will collect in the bottom of the pump during the up stroke ready for the bucket when it descends. The flow is always in one direction, the water always lies on the valves so as to keep them air tight, and very little clearance is necessary between the foot and bucket valves and between the bucket and head valves. The glands around the vertical rods can be cupped and filled with water, to seal them against air leaks. It don't hurt a vacuum any to have water leak into it, but a little air will make a big difference, so that if you can keep water around a place where air is likely to get in you will have a better vacuum. I have heard of serious breaks in condensing apparatus being gotten over at sea by building a coffer dam around the fracture and keeping it full of water. This kept it sealed against the atmosphere, and some water simply went through the crack instead of through the injection valve.

Of course the air pump must be large enough to keep the condenser clear at times of maximum load or when the greatest amount of water and air is to be handled. On the other hand, it should not be too large so as to cause unnecessary loss by friction. The indications are that past practice has been too liberal in this respect and that many engines have labored along with cumbersome pumps where smaller sizes would have been ample. It would appear, too, that good design lies in the direction of short strokes and large diameters, for if we quarter the stroke of a pump and double its diameter, it will have the same capacity, the force required to overcome the friction will be exerted through only one-quarter the space, and will be no more than twice what it was before for the rubbing surface, the circumference of the bucket has only been doubled, and in a vertical pump where the bucket is always covered with water, this may be an easy fit. The larger bucket also gives greater capacity for the valves and the speed of the water through the larger passages thus afforded is slower.

If you are interested in proportioning surface condensers, I advise you to read a paper on the subject by J. M. Whitham, page 417, Vol. IX., Trans. Amer. Soc. Mech. Engrs. In it he considers all the factors bearing on variable conditions and gives formulae which meet all conditions. For the average case, he gives a very simple formula for the amount of cooling surface required.

Multiply the total number of pounds of steam condensed per hour by 17 and divide by 180.

This allows nearly one-tenth of a square foot of cooling surface per pound of steam, which would not be a bad figure to bear in mind.

A condenser may fail to work from a failure of the injection or circulating water supply, in which case the steam will not be condensed, but will accumulate in the condenser, destroying the vacuum and heating the condenser up. Relief valves which open automatically to the atmosphere when the pressure in the condenser exceeds that outside are usually provided to allow the engine to keep on running non-condensing until the trouble can be located and remedied. Secondly, a condenser may fail to work on account of the failure of the air pump to remove the water and air as fast as it comes to the condenser. Such a failure is apt to result seriously, for if there should be a vacuum in the cylinder at such a time, as there is likely to be by expansion in the low pressure cylinder of a compound or triple expansion engine, or even in a single cylinder engine when starting or stopping, or when lightly loaded, the water will draw into it and result in a break down. For this reason condensers are often, and should always be provided with a device for automatically admitting air and breaking the vacuum when the height of water in the condensing chamber exceeds a safe limit, and care must be taken that nothing occurs to slow down the air pump if indirectly connected or independent.

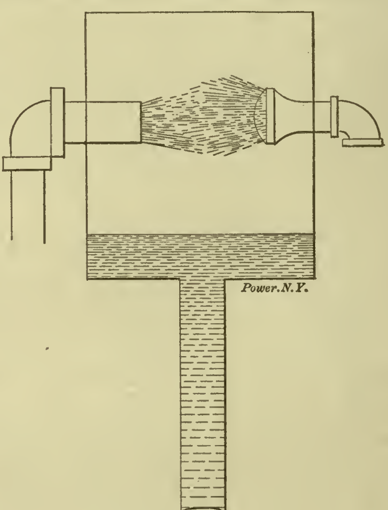


Fig. 12.

You remember the experiment we performed with the long tube of mercury. The action would be just the same with water only it would take a longer column of water to balance the pressure of the atmosphere. In Fig. 12, if the tank were originally full of water, the water would run out through the pipe until the column is just sufficient to balance the pressure of the atmosphere, which will be 34 feet more or less according to the temperature of the water and the height of the barometer. If, then, we have the pipe over 34 feet long, water will run out of the chamber by its own weight against the atmospheric pressure, leaving a vacuum in the chamber. If we let steam and cool water together into such a chamber, the steam would be condensed, the water would flow out without the necessity of a pump, and the vacuum would be maintained without the bother and expense of pumping the water out. There is one fatal objection to the operation of this ideal scheme. We have seen that the steam and the injection water bring into the condenser more or less air to say nothing of that which steals in through leakage. As the air would not fall out by gravity, it would gradually cacumulate and destroy the vacuum. This objection is very ingeniously and simply gotten over in the injector or ejector condenser, shown in Fig. 13. The exhaust steam enters through the nozzle *A*. The injection water surrounds this nozzle and issues downward through the annular space between the nozzle and the main casting. The steam meeting the water is condensed, and by virtue of its weight and of the momentum which it has acquired in flowing into the vacuum the resulting water continues downward, its velocity being further increased, and the column solidified by the contraction of the nozzle shown. The air is in this way carried along with the water and it is impossible for it to get back against the rapidly flowing steam in the contracted neck. The condenser will lift its own water twenty feet or so. When water can be had under sufficient head to thus feed itself into the system, and the hot-well can at the same time be

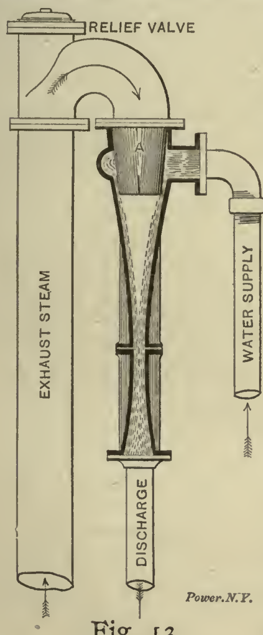


Fig. 13.

so situated as to drain itself, it makes a remarkably simple and efficient arrangement. In case the elevation is so great that a pump has to be used to force the injection, the pump has to do less work than the ordinary air pump, and its exhaust can be used to heat the feed water.

Except under exceptional circumstances, the nature of which we have tried to indicate, the gain by the use of a condenser is so great that their use is very general in plants where water can be had for condensing purposes, and it is an important point for consideration in locating a plant, whether or not a supply of suitable condensing water will be available. In large cities where water must be bought at a considerable cost, plants are run non-condensing at a great sacrifice of steam efficiency, because it would be out of the question to buy water for injection. Considerable has been done in the way of cooling water off after it has passed through the condenser, and using it over and over again. This is done by letting it trickle over a series of pans on the roof, or letting it fall in a shower through a shaft through which a current of air is circulated.

In this connection, there has recently appeared on the market, an apparatus, which appears to promise well. You know as much heat must be taken out of a pound of steam to reduce it to water of a given temperature, as would have to be put into the water to make it into steam from that temperature. Suppose the steam from an engine cylinder is discharged through a series of pipes upon the outside of which cold water is sprayed. Part of the water, as it strikes the heated surface, will be evaporated and escape into the atmosphere as vapor, but for every pound of water so evaporated a pound of steam is condensed, and can be used as boiler feed. Thus, instead of using the city water for boiler feed, we use it to spray the condenser, and use the condensed steam over and over in the boilers, and if, as it appears, and as the makers of the apparatus assure us, it takes no more water in one case than in the other, we are ahead whatever net benefit we can get out of the vacuum.

It is a mistake to strain for too high a vacuum. Of course every particle that you can save by keeping things free from air leakage is so much pure gain. What I mean is don't crowd your circulating pump or open your injection too wide just to get the last fraction of an inch of vacuum. The amount of water to be handled to get an additional half inch at the lower end of the gage is excessive, the temperature of your feed is reduced, and while it may mean less pounds of steam per hour per horse-power for the main engine, it is likely to mean more dollars per year per useful horse-power delivered. If you do not use the hot-well water for boiler feed, or if you have methods by which this may be heated, that would allow you to run a lower hot-well temperature to advantage. Suppose, for instance, you have an economizer of ample capacity, heating the water with the waste of the uptake, then it would pay you to run a higher degree of vacuum, for if your economizer is ample, it will deliver the water to the boiler at about the same temperature whether it comes to it at 100 or 130, and, so long as you do not make your pumps do as much extra work as the extra vacuum amounts to, you are ahead. When cold water is used for feed or when there is a very considerable difference between the hot-well and exhaust steam temperatures, and the hot-well water is used for feed, the water may be passed through a heater placed between the engine and the condenser. The exhaust will have a temperature of about 120° and will impart considerable heat to the feed, leaving so much less for the condenser to do.

You will understand, of course, from what has been said of the nature of a vacuum and of the nature of pumping that no pump, however powerful, can lift water out of the condenser by suction because the atmosphere cannot act upon the water to force it up to the pump. The pump must, therefore, be situated below the condenser, so that the water can fall into it by its own weight or head. Further, there must be no chance for any accumulation of air or the pump will get air bound and simply work back and forth without taking any water.

A common annoyance connected with the running of an air pump is the hammering or clattering of the discharge valves, due to the variations in pressure as the air and water are discharging. This can be avoided by connecting a small pipe with a valve into the passage leading from the water cylinder to the delivery valve, and admitting a small quantity of air, the amount to be admitted being only sufficient to overcome the hammering. This air cannot vitiate the vacuum in the condenser, as it aids the water in keeping the inlet or foot-valve closed. The pipe should extend to an elevation greater than the hot-well for otherwise the water and the air will discharge from it on the down stroke of the bucket.*

*Constructive Steam Engineering, J. M. Whitham, p. 464.

THE JET CONDENSER.

In Fig. 1 we have a vessel filled with steam at atmospheric pressure. Attached to it is a U-tube filled with mercury, open to the atmosphere at the outer end. As long as the inside and outside pressures are equal, the mercury will be at the same level in both legs of the tube, as shown. If we inject a spray of cold water into the vessel through pipe *W*, the steam will be condensed and will fall to the bottom, occupying only the small space below the dotted line *AB* in Fig. 2. The space above the dotted line will now

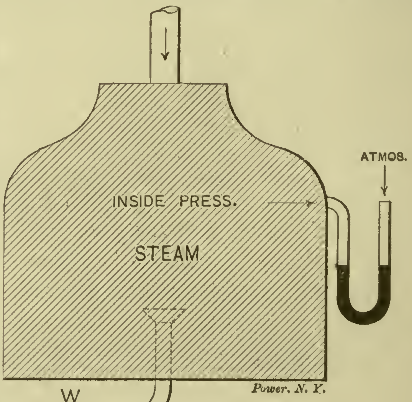


Fig. 1

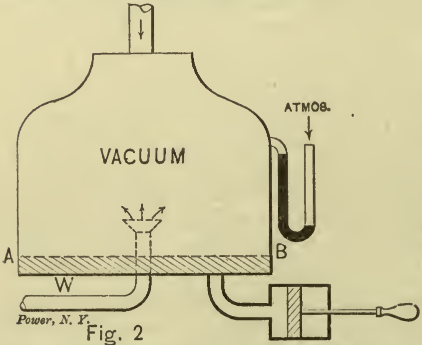


Fig. 2

be empty, or, in other words, a vacuum, and since the pressure on the inside is removed, the mercury will rise in one leg of the tube as shown. If we continue to supply steam and condensing water to the vessel and draw out the condensed steam and water as fast as it accumulates, we can maintain a constant vacuum in the vessel. This was the principle upon which the early mining pumps were operated. The piston was drawn to the top of its stroke by the descending pump plunger; steam at atmospheric pressure was admitted under the piston and condensed by a spray or jet of water, thus creating a vacuum. The pressure of the atmosphere then forced the piston down, raising the pump plunger at the other end of the beam.

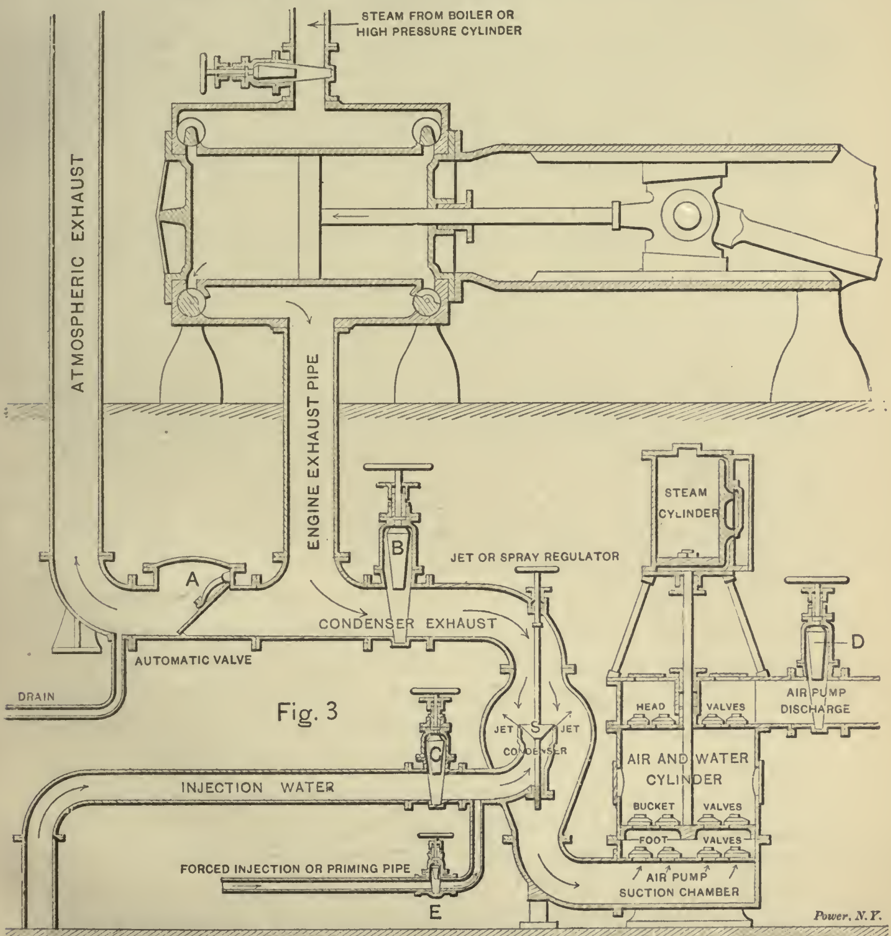


Fig. 3

The same thing is done on a larger scale and in a more scientific manner by the jet condensing apparatus of today.

An entire apparatus of this type including all pipes and valves, and connected to an engine cylinder, is shown in cross-section by

Fig. 3. The engine piston is moving to the left, and the exhaust steam is passing out through the lower left-hand port into the exhaust pipe and from there into the bottle-shaped condenser. As it enters the condenser it meets a spray of cold water issuing from the injection pipe around the edges of the cone *S*; this spray condenses the steam and the intermingled steam and water pass down into the lower part of the condenser and the suction chamber of the air pump. This leaves a vacuum in the condenser and exhaust pipe and the engine cylinder up to the piston face. When the air pump bucket starts on its upward stroke, the mingled air and water pass by gravity up through the foot valves of the air pump. When the air pump bucket descends, the water and air pass up through the bucket valves to the upper side of the bucket or plunger. The next upward stroke of the bucket forces the water out through the head valves of the pump into the discharge pipe, at the same time allowing more water and air from the condenser to pass up through the foot valves into the lower part of the air cylinder. This action is continuous and the air-pump speed must be regulated to handle the condensed steam, the water required to condense it and the air brought in by the water.

Let us consider some of the general features of the jet condenser, and particularly the apparatus shown.

First among these is the fact that the injection or condensing water and the condensed steam are mixed together. If the condensing water is pure the air pump discharge is suitable for boiler feed, but if the condensing water is impure, acidulous or salt, it is evident that the water discharged from the air pump is unsuitable for boiler use. Second, there is to be considered the type of air pump and the means by which it is driven. This pump may be of the horizontal or vertical type, single cylinder double acting, double or twin cylinder single acting or duplex; it may be independently steam driven, as in Fig. 3, or it may be driven by a belt from the main engine or shafting or by an electric motor. The independent steam driven type has the advantage of being absolutely independent of the main engine; it may be started before and stopped after the main engine, thus establishing a vacuum before the load is thrown on the engine and draining the cylinder and pipes of the water of condensation and leakage. It may be run at any speed within its limits, keeping the vacuum constant under changes of load; it may also be placed at any convenient point near the engine. On the other hand, it is more expensive to operate than the belt or electrically driven type, as the latter obtain their power at the same cost per horse-power as that of the large units. We will not discuss here the relative economy of the different types. Another point of importance is the possibility of damage or inconvenience through the failure of the condensing apparatus or the improper arrangement of the connecting pipes. In case the air pump fails to operate or the injection pipe becomes clogged, the engine must be shut down unless it is provided with another passage for the exhaust. The usual method is to provide an atmospheric exhaust outlet, which will allow the engine to exhaust into the atmosphere. As shown in Fig. 3, this outlet is provided with an automatic relief valve *A*. This is so arranged that when there is a vacuum in the exhaust pipe between the engine and condenser the atmospheric pressure on the outer side of the valve keeps it closed. If the air pump becomes inoperative, the pressure accumulates in the exhaust pipe and condenser and forces the valve open, allowing the engine to exhaust freely into the atmosphere. When the vacuum is reestablished and the inside pressure falls below that of the atmosphere, the valve closes automatically. This valve may be a special swing check or any one of a number of other special valves made for the purpose. The gate valve *B* is intended for use in case of repairs to the condenser; it may be closed tightly and the automatic valve *A* locked open, when the condenser or air pump may be repaired without interference from the hot exhaust steam. In case the condenser and air pump are connected to injection and discharge mains common to other condensers the gate valves *C* and *D* are necessary in the event of repairs to the condenser or pump; the valve *C* is, however, primarily intended to regulate the supply of injection water as will be mentioned later.

Another source of trouble in jet condensing engines is the possibility of getting water into the engine cylinder and so wrecking it. Suppose the air pump to be running but slowly, or to stop entirely, so that it will not draw out the injection water as fast as it runs into the condenser. Eventually the water will flood the condenser and pipes, enter the cylinder and wreck it. To render this impossible, two methods are adopted: one is the application of a vacuum-breaking device to the condenser; the other is to so arrange the spray cone and condenser neck that an accumulation of water will reduce the surface of the spray and break the vacuum.

Fig. 4 shows a patented vacuum-breaker furnished on all Geo. F. Blake & Knowles' condensers. Its action will be understood from the cut. When the water rises in the condenser to the level *AB*, it lifts the float *F*, which in turn lifts the air valve *V* from its seat, admitting air to the exhaust pipe and engine cylinder through the pipe *P*, thus breaking the vacuum. This, of course, equalizes the inside and outside pressures, and prevents any more water from flowing into the condenser. The engine exhaust will then accumulate until it acquires sufficient pressure to lift the valve *A*, Fig. 3, and the engine will exhaust into the atmosphere.

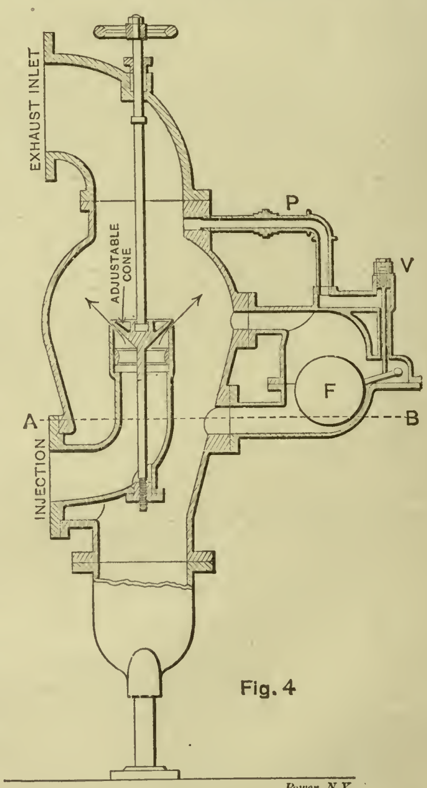


Fig. 4

Fig. 5 shows the arrangement of condenser neck and spray cone used upon the Worthington condensers to accomplish the same result. In this case the water is sprayed downward, and as the condenser neck is quite small, the rapid condensation is due only to the large surface exposed by the spraying water. Owing to the small size of the condenser, any accumulation of water rapidly diminishes the condensing surface until the spray itself is submerged, leaving only the small annular ring of water at *A B* to act on the large volume of steam from the engine. The surface of this ring is far too small to condense the steam and the pressure immediately accumulates and either the valve *A*, Fig. 3 opens, allowing the engine to run non-condensing or the exhaust steam blows out through the injection pipe and pump valves.

Again, the engine itself may draw water up into the low pressure cylinder. Suppose a compound engine having a low pressure cylinder of 4 times the area of the high pressure. In starting up or shutting down the engine the throttle is barely cracked, as usual, admitting throttled steam of, say, 20 pounds absolute pressure for the full stroke. At the end of the stroke this seam will be admitted to the low pressure cylinder, where it expands to 4 times its volume, or to about 5 pounds absolute pressure. This is equivalent to a vacuum of about 20 inches. Now suppose the air pump to be almost or entirely stopped and the injection valve to be open as usual. Then when the low pressure piston starts on its return stroke, the exhaust valve opens, connecting the cylinder under 20 inches of vacuum with the exhaust pipe and condenser, also under a vacuum; the atmospheric pressure will continue to force water up into the condenser, and, if the air pump cannot remove it, up into the engine cylinder. This would be prevented by the vacuum-breaking device shown in Fig. 4. This brings us to the proper method of starting and stopping an engine with an independent condensing apparatus. To start the apparatus, proceed as follows: Open slightly the injection valve *C* and start up the air pump to its normal speed. This produces a vacuum in the pipes and condenser, drains them of all water, and causes the injection water to flow into the condenser. When

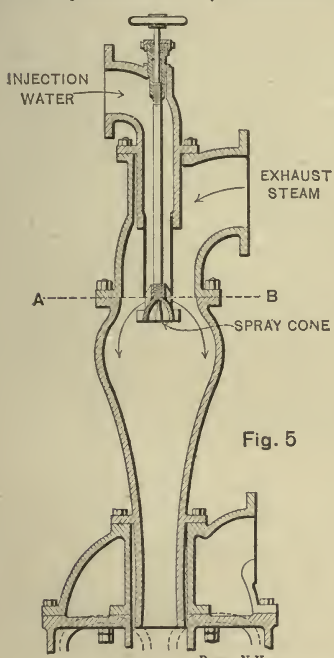


Fig. 5

the vacuum is established, as shown by the gage, open the throttle and turn the engine over slowly, warming it up. Then bring the engine up to speed, throw on the load and regulate the amount of injection water by the valve *C*. The wheel on the top of the condenser is used only for regulating the thickness of the spray and has nothing to do with the supply of injection water.

The speed of the air pump and the amount of injection water must be regulated according to the load on the engine and the amount of vacuum desired.

When several condensers are connected to a common injection main, it sometimes happens that starting up the air pump of an idle condenser will fail to bring water in through the injection branch. This is partly owing to the fact that the greater vacuum already established in the other condenser draws the water away from the condenser in question, but in a greater measure it is due to the fact that a flow of water at a high velocity is already established to the other condensers. This stream of water requires some force to break its flow and to divert a portion of it into a branch pipe, just as the stream of water from a hose nozzle will remain a smooth rod of water for some distance from the end of the nozzle, or just as the jet of water into an injector tube passes the spills or overflow holes without losing a drop of water through them.

In such event, recourse must be had to the forced injection or priming pipe shown in Fig. 3. This forced injection takes its supply from a source under a very slight head or pressure, such as a surge tank slightly elevated or the city water supply. If the water will not come to the condenser, allow the air pump to run, close main injection valve *C*, open fully priming valve *E*, and admit water until a vacuum is formed in the condenser; then open gradually injection valve *C* and close priming valve *E* gradually, when it will be found that the flow of water to the condenser is established. When this forced injection does not overcome the trouble entirely, it will usually be found that the injection pipes are too small, making the velocity of flow too great. In such cases, the velocity of flow should be decreased by increasing the size of the injection main and branches.

When shutting down an engine with an independent condensing apparatus, close the engine throttle first, and when the engine has stopped, and not until then, close the injection valve *C*, and lastly shut down the air pump. By shutting off the water supply before the air pump is stopped, the water already in the condenser and pipes is pumped entirely out and there is no danger of it getting into the engine cylinder.

THE SURFACE CONDENSER.

Suppose that we have a cylindrical vessel arranged as in Fig. 6, with a pipe through the center, leaving an annular space outside of the pipe. If we fill the annular space with steam and run a stream of cold water through the pipe, the steam will condense upon the cold pipe surface and fall to the bottom of the vessel, leaving a vacuum above it. If we draw off this condensed steam and the air it brought in with it, we can refill the vessel with steam and, by running more cold water through the pipe, condense this new steam; this operation can be continued indefinitely and a constant vacuum maintained in the vessel. In this case, we see that the exhauster draws out only the condensed steam and entrained air, the cooling water being kept entirely separate.

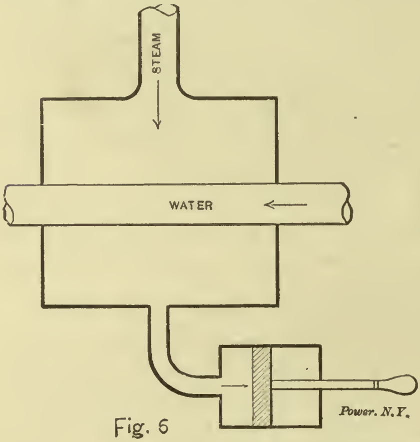


Fig. 6

In the jet condensing arrangement shown in Figs. 1 and 2, it is plain that the condensing water flows into the vessel on account of the vacuum therein; while in this case the part of the vessel which is under a vacuum is, as we said above, entirely separate from the condensing water, making it necessary to force the water through the pipe by some other means

This is exactly the manner in which the surface condenser operates.

Fig. 7 shows a sectional view of a complete surface condenser and pumps.

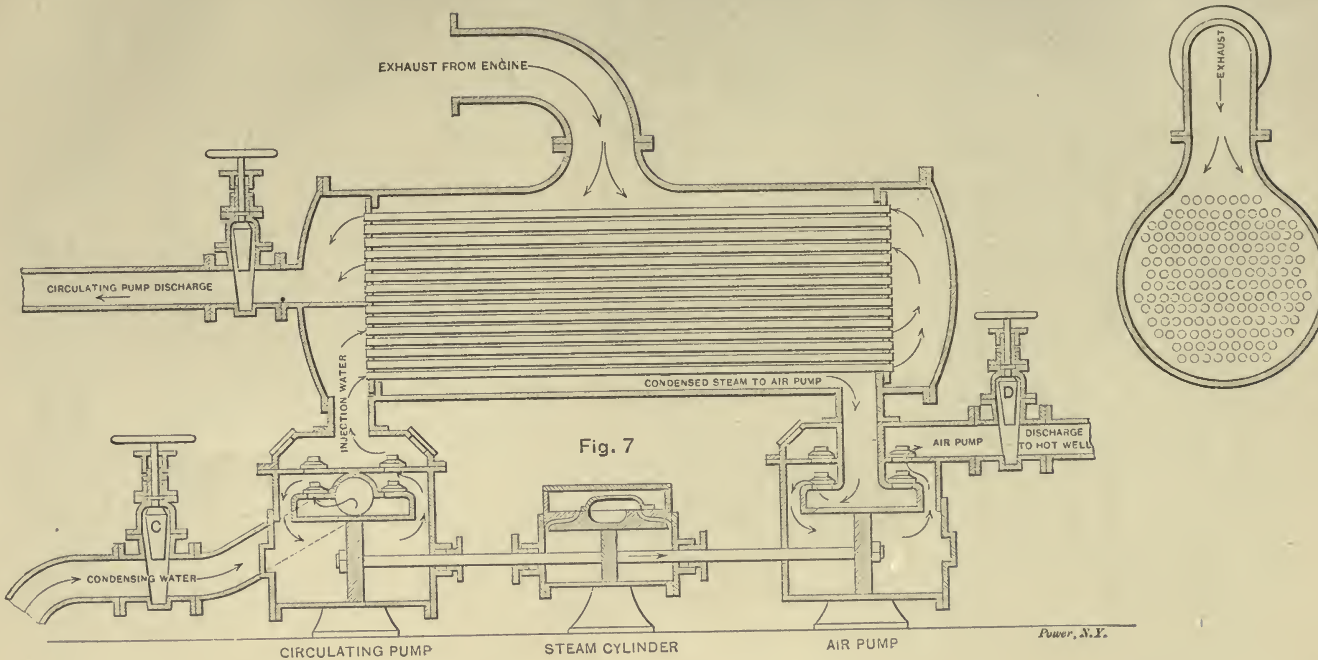

EXHAUST FROM ENGINE
CIRCULATING PUMP DISCHARGE
INJECTION WATER
CONDENSED STEAM TO AIR PUMP
C
D
CONDENSING WATER
AIR PUMP
DISCHARGE TO HOT WELL
EXHAUST
CIRCULATING PUMP
STEAM CYLINDER
AIR PUMP
Power, N.Y.

Fig. 7

The exhaust steam from the engine enters the condenser through the elbow on top; it then expands and fills the space outside of and between the condenser tubes. The circulating pump shown at the left draws its water by "suction" from any convenient source and circulates it through the tubes, keeping them cold. The exhaust steam is condensed by contact with these cold surfaces and falls to the bottom of the condenser. It is then drawn off by the air pump shown at the right, and is usually discharged into a hot well. The drawing shows clearly that the condensed steam, being outside the tubes, is kept entirely separate from the condensing water, which is forced through the tubes. Evidently then, the condensed steam discharged by the air pump may be used over again in the boilers, even if the cooling water is unfit for use, the only objection to this being the oil brought down by it from the engine cylinder and steam chest. There are several more or less satisfactory methods of extracting this oil or grease from the water; a discussion of their merits is beyond the province of this article. The condensing water is handled by a separate pump and does not flow into the condenser, as in the case of a jet-condensing apparatus; it may be salt or impure, and unless warm water is required for some outside purpose it is discharged to waste.

It is thus seen that a surface condensing apparatus requires two pumps of comparatively small size as against one large pump for the jet condenser. It is considerably more expensive than the latter, and is seldom used except where it is desirable to return the condensed steam to the boilers.

The possibilities of trouble from it are less than in the jet condenser. There is no way in which the condensing water can get into the engine cylinder; while the condensed steam might, under certain conditions of air pump operation, accumulate until it reached the top of the condenser, it could not get into the cylinder, for the condensing surface would be entirely submerged and the accumulated pressure would force open the automatic atmospheric relief valve and allow the engine to exhaust into the atmosphere.

There is the same need of an atmospheric exhaust outlet as in the case of the jet condenser and for precisely the same reason.

The piping between the engine, the elbow on the condenser, and the atmospheric exhaust should be the same as in Fig. 3. In the type of apparatus shown, the condenser is directly attached to the pump or pumps. This is not at all necessary; the air or circulating pump or both may be placed at any convenient point and connected with the condenser by pipes. The air pump, however, should be placed below the condenser, so that the condensed steam may go to it by gravity. Nor is it necessary to use the type of pump shown by the drawings. The pumps may be horizontal or vertical; of the single or double-acting single-cylinder type; the single-acting twin-cylinder type; or the duplex type. Fig. 8 shows a horizontal double-acting cylinder air pump, independently

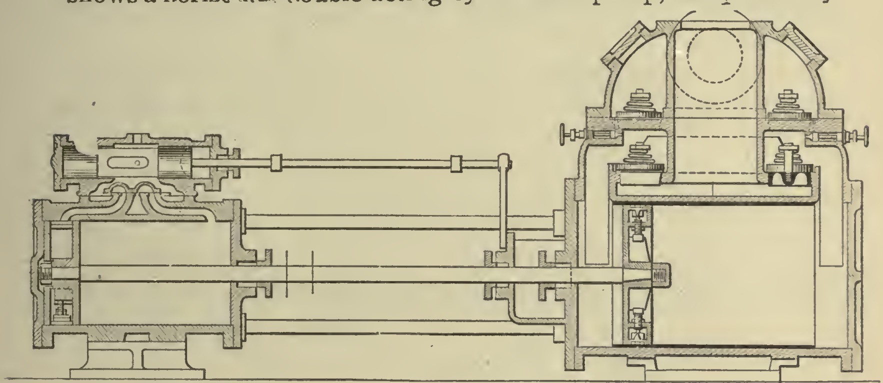


Fig. 8.

steam-driven, which is frequently used with both jet and surface condensers. The condenser outlet is connected with the suction inlet of the pump, and its operation is the same as any double-acting pump.

Centrifugal pumps are frequently used as circulating pumps, and various combinations of pumps and condensers are made.

Fig. 9 shows a surface condenser equipped with 3-cylinder vertical reciprocating air pumps and centrifugal circulating pumps, both pumps being driven by electric motors. The details of the condenser itself vary: for instance, a jet condenser is frequently box-shaped instead of bottle or cone-shaped as in Fig. 3; a surface condenser may be rectangular in cross section instead of cylindrical, as in Fig. 7; the steam may be inside and the water outside the tubes of a surface condenser, instead of as shown in Fig. 7; etc., etc. The principle is the same, however, in all arrangements; in a jet-condensing apparatus the steam is condensed by contact with a jet or spray of cold water and the air pump handles the condensed steam, the condensing water and the air entrained in both; in a surface-condensing apparatus the steam is condensed by contact with a cold surface and the air pump handles

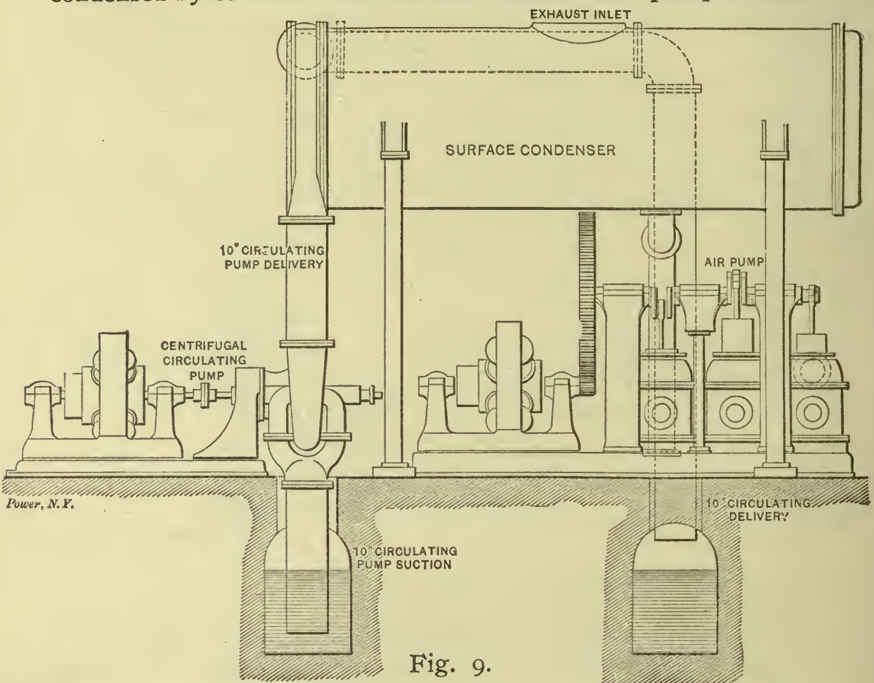


Fig. 9.

the condensed steam and the air, while the circulating pump handles the condensing or circulating water alone.

THE INJECTOR OR SIPHON CONDENSER.

Fig. 10 is a small scale reproduction of the jet condensing apparatus described in Fig. 3. It will be seen that with the arrangments shown, an air pump is required to pump out the condensed steam, the condensing water and the air brought in by both.

If the hot well were lowered to a point about 34 feet below the condenser, as shown by the dotted lines, it will be seen that an air pump is not required to remove the condensed steam and the water. This will be plain if it is remembered that a perfect vacuum of 30 inches in the engine exhaust pipe will not support a column of water in the discharge pipe more than 34 feet high; so that if any water is supplied to the condenser in excess of this 34-foot column it will pass through the condenser and out of the discharge pipe without the aid of a pump. Now, if the neck of the condenser be contracted, as in Fig. 11, the velocity of this falling water will be greatly increased, and the water will carry out with it not only the condensed steam but the air, leaving a vacuum in the exhaust pipe.

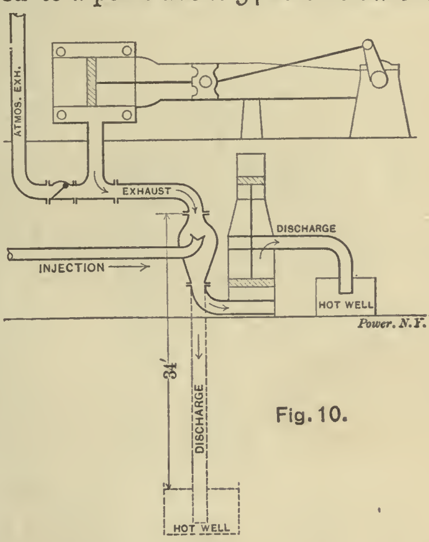


Fig. 10.

This is the principle of the injector or siphon condenser, one type of which, the Bulkley, is shown in cross-section in Fig. 12. As will be seen from the figure, it is not necessary to place the condenser below the engine, as in Figs. 10 and 11. All that is required is to have the column of water between the condenser and the hot well as great or greater than the vacuum will support, so that the constant supply of condensing water will produce a continuous downward flow. In the arrangement shown the hot well is a little below and the condenser is above, the engine level, an ordinary tank pump being used to elevate the condensing water.

The exhaust steam enters the top of the condenser and passes through the inner cone or nozzle *C*. The condensing water enters the condenser at the side and passes downward around the exhaust nozzle in a thin conical film. The exhaust steam is condensed within this hollow cone of falling water, and the condensed steam and the condensing water then fall vertically through the condenser, and discharge pipe. In passing through the neck of the condenser the water acquires sufficient velocity to draw out with it the entrained air, leaving a vacuum in the exhaust pipe and engine cylinder. The lower end of the discharge pipe is sealed by the water in the hot well. It is necessary to provide a large condensing surface, as well as a high velocity for the injection water; in most condensers of this type this is provided for by bringing the exhaust steam in through the cone or nozzle *C*, and the injection water in through the annular space outside the cone. This forces the condensing water to take the shape of a hollow cone into which the exhaust steam is discharged.

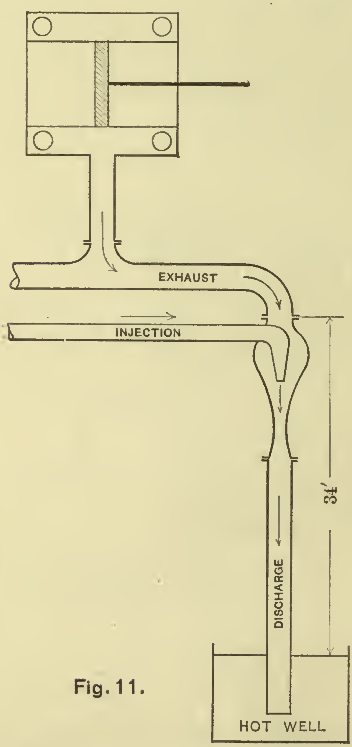


Fig. 11.

In the Knowles Spirojector condenser, which is otherwise similar to the Bulkley, the cone *C* has cast on its face vanes which compel the injection water to assume a spiral or whirling motion as it passes through the condenser to the discharge.

In Fig. 12 a pump is shown for lifting the condensing water; if the level of the injection water supply is not more than, say, 20

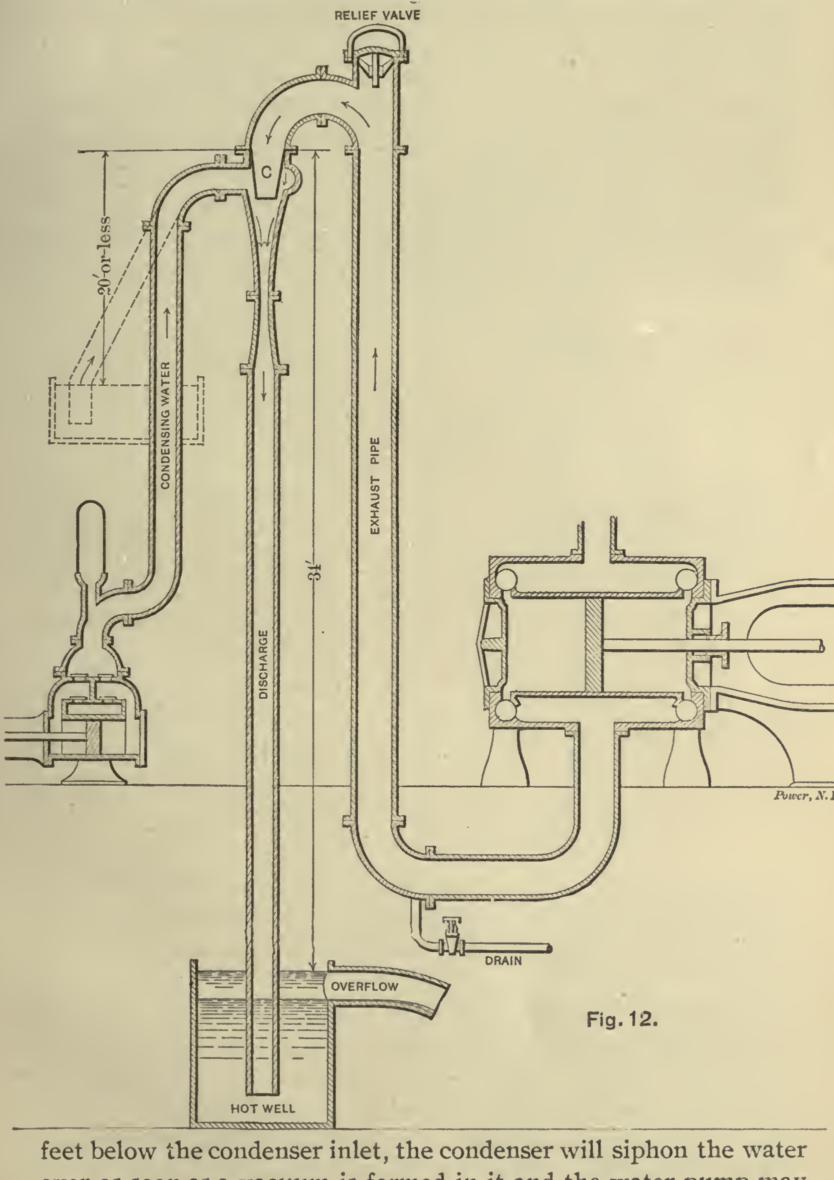


Fig. 12.

feet below the condenser inlet, the condenser will siphon the water over as soon as a vacuum is formed in it and the water pump may be dispensed with. As 20 feet is about the limit to which water may be continuously lifted by the siphoning action, it follows that when the water supply is more than 20 feet below the condenser a pump must be used. The arrangement with a pump shown in Fig. 12 is sometimes modified by the insertion of a tank (shown in dotted lines) at about the lower limit of the siphon. This is convenient when a single-acting or single-cylinder tank pump is used to lift the water; such a pump gives a more or less intermittent flow, whereas a practically constant flow is required by the condenser. In the tank arrangement, the pump discharges intermittently into the tank and the condenser siphons continuously from the tank. Fig. 13 shows the arrangement of condenser and pipes for a siphoning apparatus, when no pump is used. In this figure there is shown a cross connection at the supply level between the injection and discharge pipes. As we said before, the vacuum must be formed in the condenser before it will siphon water; by opening the starting valve *S* water is admitted to the discharge pipe, and in falling through the pipe it draws the air out with it, forming enough vacuum in the upper pipes and condenser to draft the injection water up to the condenser. The starting valve should then be closed and the water supply should be regulated by valve *J*. When the injection supply is at the extreme lower limit of the siphon, say 20 feet below the condenser, this arrangement of starting valve is not always satisfactory; in such cases the cross connection may be omitted and a small priming pipe *P*, shown in dotted lines in Fig. 13, may be run from the boiler feed pump discharge to the condenser inlet. As soon as the injection water appears the valve *p* may be closed and the feed

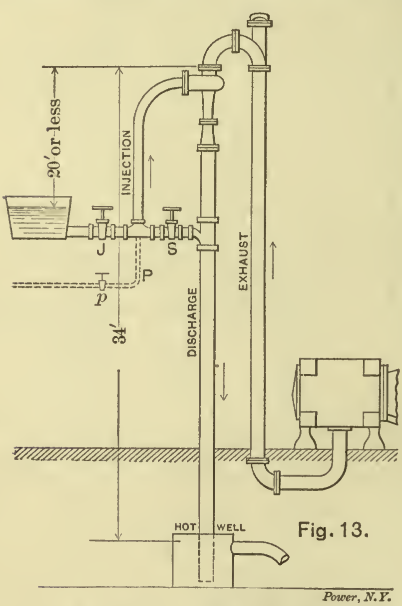


Fig. 13.

pump may resume its usual duty. When a pump is used to elevate the water, the starting valve or priming pipe and the injection valve *J* are of course omitted, as the vacuum is formed by forcing water directly into the condenser, and the water supply is regulated by the pump speed.

This type of condenser is suitable for many locations, and if properly made and connected will maintain a good vacuum. It is economical in operation and has no moving parts to wear or to get out of order. There is no way in which water can get into the engine cylinder unless it is allowed to accumulate in the pocket formed by the exhaust pipe, and not even then unless atmospheric pressure is admitted to the exhaust pipes through the uncovering of the water supply or discharge pipes. A drain pipe placed as shown in Fig. 12 will serve to drain out the exhaust pipe before the engine is started and removes all danger from this source. Pumping action of the low pressure cylinder cannot draw water up from the hot well on account of the height, and water drawn

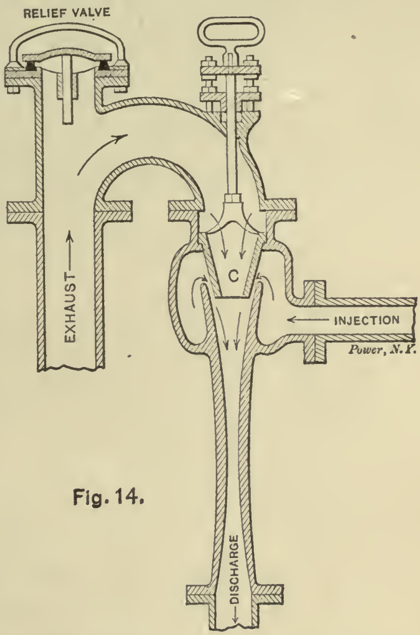


Fig. 14.

up from the injection supply would fall into the discharge pipe and not into the exhaust pipe, by reason of the construction. Another condenser of this type, but differing slightly from the above in detail, is the Baragwanath water jacket condenser shown in Fig. 14. In this condenser, as in the others, the exhaust enters at the top and the injection at the side, and the exhaust nozzle is surrounded by cold water. The water chamber is larger than in the others, and the shell of the condenser is prolonged inside the water chamber, forming an inverted cone, into the end of which the condensing nozzle *C* projects. This nozzle is adjustable by means of the spindle shown and can be set to admit precisely the right amount of water.

Each of the three condensers mentioned herein, *i. e.*, the Bulkley injector, the Knowles Spirojector and the Baragwanath water jacket, is supplied with an automatic atmospheric relief valve similar to that shown in Fig. 13; this valve discharges directly into the atmosphere, so that when the top of the condenser is inside the building it is necessary to use a relief valve in the pipe line and to carry the atmospheric exhaust pipe out of doors. This valve may be of the swinging check type shown in Fig. 3, or any one of several well-known types. The injection water may be supplied under a head or a pump of either the reciprocating, the rotary or the centrifugal type may be used. The head against which these pumps work is evidently quite small, since the vacuum in the condenser will take care of the upper 18 or 20 feet of the lift.

THE EXHAUST STEAM INDUCTION CONDENSER.

The operation of this condenser is based upon the same principle as that of the steam injector. As most of our readers know, the operation of the injector is as follows: A jet of steam enters through the steam tube at a high velocity and induces the air in the suction pipe and injector body to pass out with it; this leaves a vacuum and allows the atmospheric pressure to force in water. The steam is condensed by this water and the velocity of the steam is imparted to the water; then the energy in the moving column of water is sufficient to overcome the pipe friction, lift the check valve and force the water into the boiler against the pressure.

An inspection of Fig. 15 shows that this is exactly the operation of the induction condenser. The exhaust steam enters through the valve *E* and passes through the inclined perforations into the central tube *T*, as shown by the arrows. Owing to the velocity of its movement the air in the condenser and the injection pipe is drawn out with it, and the atmospheric pressure on the injection supply forces the condensing water up through the pipe and into the tube *T* as shown. The exhaust steam is condensed by this water and a vacuum is left in the condenser and exhaust pipe. The original velocity with which the water entered the condenser and the added velocity due to the exhaust steam enable the mingled steam and water to overcome the atmospheric pressure on the discharge end and pass out into the hot well, just as the water from the injector overcomes the resistance due to friction and pressure and passes into the boiler. From this we see that the velocity of the discharge is sufficient to draw out the air and to get rid of the condensing water and condensed steam; so that no air pump is required as in the case of a jet or surface condenser, nor a 34-foot "tail" column, as in the injector or siphon condenser.

The condenser shown in Fig. 15, however, has its limits of operation. We have just seen that the operation depends upon the velocity of the discharge; it is plain that when the condenser lifts its injection water, as shown in the figure, this velocity must be almost wholly imparted by the exhaust steam. Then if the load on the engine is variable or if the condenser is too large for the engine, there will be times when the small amount of exhaust steam

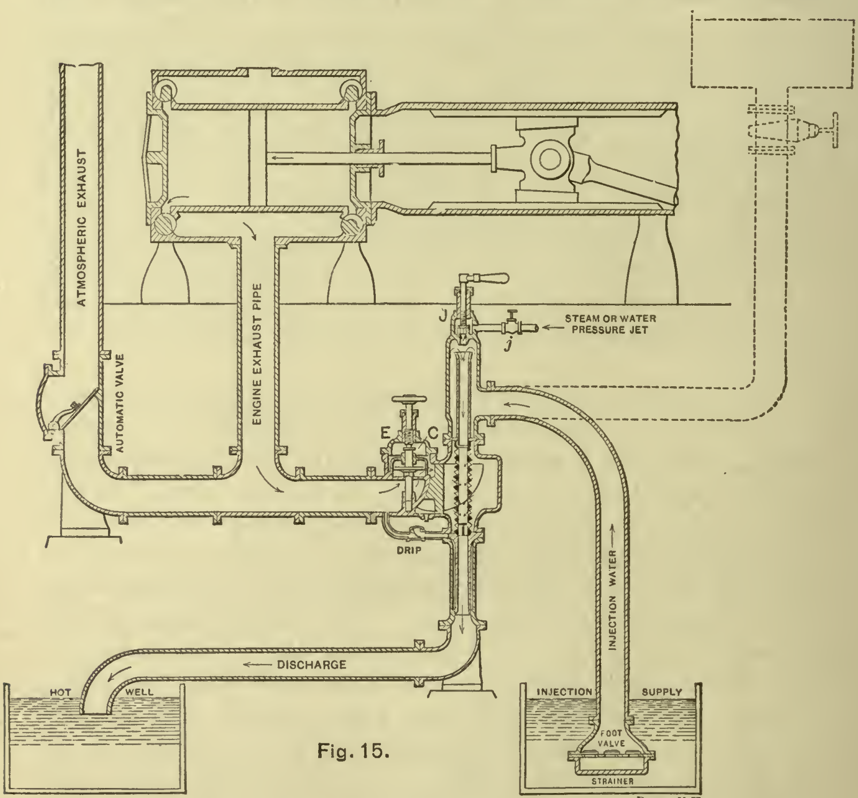


Fig. 15.

furnished by the engine will not be enough to impart the required velocity to the large volume of water and the condenser will not operate satisfactorily. In other words, the volume of exhaust steam must be, within limits, in proportion to the volume of water which it keeps in motion, too little steam being unable to induce the flow of water and too much steam affecting the vacuum. The minimum amount is that which will increase the temperature of the water at least 30° F. and the maximum amount is that which will not cause a rise of more than 50° F. in the water temperature.

In cases where the condenser takes its water under a head, as shown by dotted lines in Fig. 15, this objection does not apply, for then the velocity of the water is that due to the head and is independent of the exhaust steam.

In order to guard against the trouble due to a varying amcunt of exhaust steam, the condenser shown in Fig. 16 has been devised. It is called the adjustable capacity condenser in order to distinguish it from the fixed capacity condenser shown in Fig. 15. Both the condensers shown were designed by Korting and are identified with L. Schutte in America.

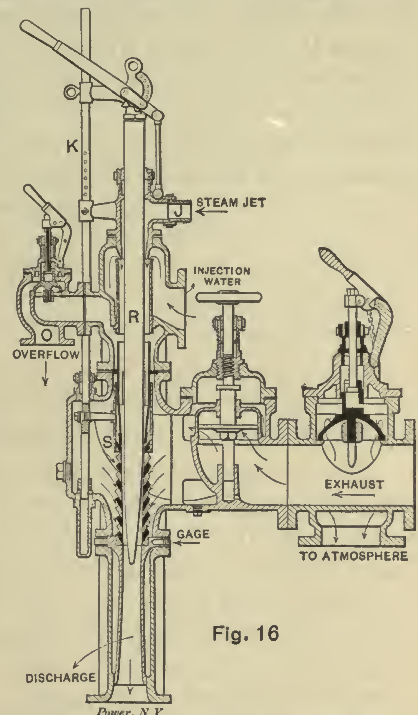


Fig. 16

The adjustable condenser shown in Fig. 16 is provided with a movable ram *R* inside the central water tube and a sleeve *S* outside the tube. The ram is tapering and controls the volume of water admitted by increasing or diminishing the annular space between its surface and the inside of the tube; while the sleeve *S*, by covering more or fewer openings in the tube, governs the area of the exhaust inlet and consequently the velocity of the exhaust steam. The relative positions of the ram and the sleeve can be regulated by the extension rod *K*, so that the machine can be adjusted for almost any condition of load. In fact, the machine may be adjusted to work satisfactorily at any point from ¼ or 1-5 of its capacity to full capacity; this is particularly desirable, as we have said, when the load is variable and the injection water must be lifted. On the other hand, the range of the fixed capacity condenser is only from about one-half capacity to full capacity. For high suction lifts, the live steam jet or water pressure jet *J* is used to bring the water to the condenser, and an escape is provided through the overflow valve *O* just as in the case of the steam injector. This starting jet and overflow valve are necessary only when starting, and may both be omitted when the suction lift is very small or when water is supplied under a head. It will be seen that in this condenser, as in the jet and siphon condensers previously described, the condensed steam and the condensing water are mixed together, so that the water from the hot well can not be used for boiler feed unless the condensing water is pure. An inspection of Fig. 15 shows that an atmospheric outlet must be provided for the exhaust, just as in the cases of the jet, surface and siphon condensers. The figure shows a special swing check valve for this purpose, while on the adjustable capacity condenser in Fig. 16 is shown the automatic valve usually furnished with this type of condenser.

In this condenser there is no 34-foot tail column, and in case the low pressure cylinder of the engine acts as a pump, water may readily be drawn up into the engine.

This is prevented by the arrangement of the stop valve *E*, through which the exhaust steam enters the condenser. The valve itself is not fixed to the spindle, but is free to move vertically within the limits set by the seat at the bottom and the collar *C* on the spindle at the top. It thus allows steam to pass out under it from the engine into the condenser, but acts as a check valve against the passage of water in the opposite direction or from the condenser to the engine. It may also be used as a stop valve by lowering the spindle until the collar *C* locks the valve to its seat. Since the seat of this valve is practically at the top of the exhaust pipe, it is advisable to drip the pipe on the engine side of the valve, as shown in Fig. 15, to prevent any accumulation of condensation. This drip may be piped into the condenser as shown, with a check valve arranged to prevent the return of water from the condenser to the exhaust pipe.

The directions for starting or stopping an engine equipped with either type of this condenser are very simple. It is always desirable to start the condenser and form the vacuum before starting the engine, as we have mentioned in connection with the other condensers.

To start the engine, proceed as follows:

When the condensing water is under a head, turn on the condensing water and when a vacuum is formed start up the engine. When the condensing water must be lifted, open the steam or pressure jet valve *j*, and as soon as this has lifted the water start the engine. The operation of the condenser will begin as soon as the engine exhaust reaches the condenser and when the vacuum is formed the suction or lifting jet may be turned off. In shutting down, stop the engine first, when the operation of the condenser will cease if the condensing water is under a suction lift; if the water supply is under a head, stop the engine first and then shut the valve in the water supply pipe.

CONDENSER CAPACITIES.

In column 4 of Table I are given the number of heat units required to raise a pound of water from zero Fahrenheit to the corresponding temperatures T in column 3.

In column 5 are given the numbers of so-called "latent" heat units L required to convert a pound of water at the corresponding temperatures T into dry saturated steam of the same temperature, the corresponding pressures being given in various units in columns 1 and 2.

Column 6 gives the total number of heat units H required to raise a pound of water from zero F. to the corresponding temperature and to convert it into steam of that temperature. It is the sum of the corresponding values in columns 4 and 5.

For example, in a vacuum of 25 inches (column 1), which is the same thing as an absolute pressure of 2.417 pounds to the square inch (column 2), water will boil at 133 degrees F. (column 3). It will take 133.21 heat units to raise a pound of water from zero F. to this temperature (column 4), and 1,021.295 more heat units (column 5) to evaporate the pound at that pressure and temperature, making a total of 1,154.505 (column 6).

To make a pound of steam at an absolute pressure of 20 pounds (column 2) from a pound of water at 110° would require

$$1,183.454 - 110.110 = 1,073.344$$

heat units, because the pound of water has already 110.110 heat units in it (column 4) above what it would have at zero F., and there are in a pound of steam of 20 pounds pressure absolute 1,183.454 heat units above the number in a pound of water at zero (column 6).

Condensation is the reverse of evaporation. If we wish to

TABLE I.—PHYSICAL PROPERTIES OF STEAM.

1	2	3	4	5	6
Pressure		Temperature.	Heat Required to Convert a Pound of Water at Zero F. into Steam.		
Vacuum by Gage. Inches Mercury.	Absolute. Lbs. per sq in.	T Degrees F.	To Raise the Temperature to the Boiling Point. h Heat Units.	To Convert the Water at the Boiling Point into Steam. L Heat Units.	Total. H Heat Units.
29.74	.089	32	32	1091.700	1123.700
29.72	.100	35	35	1089.615	1124.615
29.67	.122	40	40.001	1086.139	1126.140
29.62	.147	45	45.002	1082.661	1127.665
29.56	.176	50	50.003	1079.187	1129.190
29.49	.212	55	55.006	1075.709	1130.715
29.40	.254	60	60.009	1072.231	1132.240
29.30	302	65	65.014	1068.751	1133.765
29.19	.359	70	70.020	1065.270	1135.290
29	.425	75	75.027	1061.788	1136.815
28.90	.502	80	80.036	1058.304	1138.340
28.72	.590	85	85.045	1054.8 0	1139.865
28.51	.692	90	90.055	1051.355	1141.390
28.27	.809	95	95.065	1047.850	1142.915
28	.943	100	100.080	1044.360	1144.440
27.89	**1.**	102	102.086	1042.964	1145.050
27.69	1.094	105	105.095	1040.870	1145.965
27.34	1.265	110	110.110	1037.380	1147.490
27	1.435	114.34	114.470	1034.344	1148.814
26.95	1.462	115	115.129	1033.886	1149.015
26.50	1.682	120	120.149	1030.391	1150.540
26	1.931	125	125.169	1026.896	1152.065
25.85	**2.**	126.266	126.440	1026.010	1152.450
25.42	2.213	130	130.192	1023.389	1153.590
25	2.417	133	133.21	1021.295	1154.505
24.79	2.526	135	135.217	1019.898	1155.115
24.06	2.876	140	140.245	1016.395	1156.640
24	2.9	141.293	141.543	1015.491	1157.034
23.81	**3.**	141.622	141.877	1015.254	1157.131
23.26	3.270	145	145.275	1012.890	1158.165
23	3.399	146.528	146.808	1011.823	1158.631
22.37	3.707	150	150.305	1009.385	1159.690
22	3.9	152.05	152.37	1007.945	1160.315
21.78	**4.**	153.070	153.396	1007.229	1160.625
21.39	4.191	155	155.339	1005.876	1161.215
21	4.373	156.74	157.09	1004.452	1161.542
20.29	4.729	160	160.374	1002.366	1162.740
20	4.863	161. 63	161.543	1001.552	1163.095
19.74	**5.**	162.33	162.722	1000.727	1163.449
19.08	5.324	165	165.413	998.852	1164.265
19	5.36	165.317	165.730	998.632	1164.362
18	5.855	169	169.45	996.035	1165.485
17.74	5.981	170	170.453	995.337	1165.790
17.71	**6.**	170.123	170.577	995.249	1165.826
17	6.346	172.6	173.07	993.513	1166.583
16.27	6.704	175	175.497	991.818	1167.315
16	6.837	175.9	176.4	991.180	1167.580
15.67	**7.**	176.910	177.425	990.471	1167.896
15	7.329	178.98	179.51	989.019	1168.529
14.65	7.500	180	180.542	988.298	1168.840
14	7.82	181.877	182.427	986.980	1169.407
13.63	**8.**	182.910	183.481	986.245	1169.726
13	8.311	184.668	185.25	985.014	1170.264
12.87	8.375	185	185.591	984.774	1170.365
12	8.802	187.272	187.885	983.360	1171.245
11.60	**9.**	188.316	188.941	982.434	1171.375
11	9.293	189.82	190.46	981.375	1171.835
10.92	9.33	190	190.643	981.247	1171.890
10	9.784	192.21	192.89	979.674	1172.564
9.56	**10.**	193.240	193.919	978.958	1172.877
9	10.275	194.53	195.22	978.052	1173.272
8.79	10.38	195	195.697	977.718	1173.415
8	10.767	196.742	197.46	976.480	1173.946
7.53	**11.**	197.768	198.496	975.762	1174.258
7	11.258	198 9	199.64	974.964	1174.604
6.46	11.52	200	200.753	974.187	1174 940

6	11.7	200.747	201.514	973.654	1175.168
5.49	**12.**	201.960	202.737	972.800	1175.537
5	12.24	202.924	203.712	972.120	1175.832
4	12.73	204.772	205.58	970.815	1176.395
3.93	12.766	205	205.813	970.652	1176.465
3.45	**13.**	205.885	206.709	970.025	1176.734
3	13.222	206.722	207.557	969.433	1176.99
2	13.714	208.522	209.377	968.162	1177.539
1.42	**14.**	209.560	210.428	967.427	1177.855
1.17	14.122	210	210.874	967.116	1177.990
1	14.205	210.3	211.17	966.9115	1178.0815
	14.696	212	212.900	965.700	1178.600
Gage Pressure lbs. per sq. inch.					
.304	15	213.025	213.939	964.973	1178.912
1.304	16	216.296	217.252	962.657	1179 909
2.304	17	219.410	220.409	960.450	1180.859
3.304	18	222.378	223.419	958.345	1181.764
4.304	19	225.203	226.285	956.343	1182.628
5.304	20	227.917	229.0 9	954.4 5	1183.454
6.304	21	230.515	231.676	952.570	1184.246
7.304	22	233.017	234.218	950.791	1185.009
8.304	23	235.432	236.672	949.072	1185.744
9.304	24	237.752	239.029	947.424	1186.453
10.304	25	240.000	241.314	945.825	1187.139
11.304	26	242.175	243.526	944.277	1187.803
12.304	27	244.281	245.671	942.775	1188.446
13.304	28	216 326	247.748	941.321	1189.069
14.304	29	248.310	249.769	939.905	1189.674
15.304	30	250.245	251.738	938.925	1190.263

reduce a pound of steam at an absolute pressure of 20 pounds to water at 110° we shall have to take out of it

$$1,183.554 - 110.110 = 1,073.344$$

heat units, because the pound of steam contains 1,183.454 units, of which 110 110 will remain in the water, always reckoning from zero Fahrenheit.

It will be seen by comparing columns 3 and 4 that the heat in the water is very nearly the same as the temperature of the water, the increase on account of the greater specific heat at higher temperatures being less than one heat unit between 32° and 212°. It will be sufficiently accurate for our purpose to consider the heat in the water the same as the temperature, *i. e.*, to consider column 4 equal to column 3, letting a heat unit represent the amount of heat necessary to raise a pound of water one degree irrespective of the temperature.

If we represent by t the temperature at which the injection or circulating water comes to the condenser and by T the temperature at which it leaves, the number of heat units absorbed by each pound will be approximately

$$T - t.$$

The number of heat units to be taken out of a pound of steam to condense it is, as we have seen above,

$$H - h,$$

i. e., the total heat in the steam less the heat in the resulting water.

By dividing the number of heat units to be abstracted by the number absorbed by each pound of condensing water we find the quantity, Q, of water required to condense a pound of steam

$$Q = \frac{H - h}{T - t} \qquad (1)$$

Where $H =$ the total heat in a pound of steam of the given pressure,

$h =$ the heat in a pound of water at the temperature of the condensed steam,

$t =$ the temperature at which the condensing water enters the condenser,

$T =$ the temperature at which the condensing water leaves the condenser.

TO FIND THE AMOUNT OF WATER REQUIRED TO CONDENSE ONE POUND OF STEAM.

RULE:—*From the total heat in one pound of steam of the given pressure subtract the heat in one pound of water at the condenser temperature. Divide the remainder by the rise in temperature of the injection or circulating water in passing through the condenser. The quotient will be the number of pounds of water required to condense one pound of steam.*

In considering the condensation of steam in an engine the value of H must be taken at the terminal pressure, not at the counter-pressure or vacuum line. The engine delivers the steam to the condenser at the pressure existing at the point of release, and if the steam were dry saturated each pound would carry to the condenser the number of heat units given in column 6 of Table I. There is, however, little difference in this value for the entire range of terminal pressures met with in good practice, and the quality of the steam may vary widely. There is little use straining after extreme accuracy in this particular, and we shall be entirely safe for the average case and not far from right in any case if we assign to H the maximum probable value of 1,190, corresponding closely to a terminal pressure of 30 pounds absolute.

Call the temperature of the air pump discharge τ, which, allowing one heat unit per degree of temperature, would equal h.

Substituting these values for H and h in formula 1 we have

$$Q = \frac{1190 - \tau}{T - t} \qquad (2)$$

APPROXIMATE RULE FOR THE QUANTITY OF CONDENSING WATER REQUIRED PER POUND OF STEAM CONDENSED.

RULE:—*Subtract the temperature of the air pump discharge from* 1,190 *and divide the remainder by the rise in temperature of the condensing water.*

Table II has been computed by this formula and gives the values of Q, *i. e.*, the pounds of water required to condense a pound of steam for condenser temperatures of from 90 to 130 and with from 5 to 90 degrees of difference in the condensing water.

In a jet condenser where the condensing water is mingled with

TABLE II.—POUNDS OF WATER REQUIRED TO CONDENSE ONE POUND OF STEAM.

$$Q = \frac{1190 - \tau}{T - t}$$

T—t	Temperature of air pump discharge. τ														
	90	95	100	102	104	106	108	110	112	114	116	118	120	125	130
5	220	219	218	217.6	217.2	216.8	216.4	216	215.6	215.2	214.8	214.4	214	213	212
10	110	109.5	109	108.8	108.6	108.4	108.2	108	107.8	107.6	107.4	107.2	107	106.5	106
15	73.3	73	72.7	72.5	72.4	72.3	72.1	72	71.9	71.7	71.6	71.5	71.3	71	70.7
20	55	54.7	54.5	54.4	54.3	54.2	54.1	54	53.9	53.8	53.7	53.6	53.5	53.2	53
25	44	43.8	43.[illegible]	43.5	43.4	43.4	43.3	43.2	43.1	43	42.9	42.9	42.8	42.6	42.4
30	36.7	36.5	36.3	36.3	36.2	36.2	36.1	36	35.9	35.9	35.8	35.7	35.7	35.5	35.5
35	31.4	31.3	31.1	31.1	31.0	31	30.9	30.8	30.8	30.7	.0.7	30.6	30.5	30.4	30.3
40	27.5	27.4	27.2	27.2	27.1	27.1	27	27	26.9	26.9	26.8	26.8	26.7	26.6	26.5
45	24.4	24.3	24.2	24.2	24.1	24.1	24	24	23.9	23.9	23.9	23.8	23.8	23.7	23.5
50	22	21.9	21.8	21.8	21.7	21.7	21.6	21.6	21.6	21.5	21.5	21.4	21.4	21.3	21.2
55	20	19.9	19.8	19.8	19.7	19.7	19.7	19.6	19.6	19.6	19.5	19.5	19.4	19.4	19.5
60	18.3	18.2	18.2	18.1	18.1	18.1	18	18	18	17.9	17.9	17.9	17.8	17.7	17.7
65	16.9	16.8	16.8	16.7	16.7	16.7	16.6	16.6	16.6	16.5	16.5	16.5	16.5	16.4	16.3
70	15.7	15.6	15.6	15.5	15.5	15.5	15.4	15.4	15.4	15.4	15.3	15.3	15.3	15.2	15.1
75	14.7	14.6	14.5	14.5	14.5	14.4	14.4	14.4	14.4	14.3	14.3	14.3	14.3	14.2	14.1
80	13.7	13.6	13.6	13.6	13.6	13.5	13.5	13.5	13.5	13.4	13.4	13.4	13.4	13.3	13.2
85	12.9	12.8	12.8	12.8	12.8	12.7	12.7	12.7	12.7	12.6	12.6	12.6	12.6	12.5	12.5
90	12.2	12.2	12.1	12.1	12.1	13	12	12	12	11.9	11.9	11.9	11.9	11.8	11.8

the steam T and τ become identical and formula becomes

$$Q = \frac{1190 - T}{T - t}$$

Table III has been computed by this formula and gives the value of Q, *i. e.*, the number of pounds of injection water required per pound of steam with the injection water from 35 to 100 degrees and the air pump discharge from 90° to 140°.

SIZE OF AIR PUMP FOR JET CONDENSER.

If we designate by W the weight of steam to be condensed per hour, the number of pounds of water to be handled per hour by the pump will be

$$W(Q + 1).$$

For example, if we had a 100 horse power engine using 20 pounds of steam per hour per horse power the weight W of steam to be condensed per hour would be $20 \times 100 = 2{,}000$ pounds. If it takes 20 pounds of water to condense a pound of steam, then for each pound of steam condensed we shall have

$$20 + 1 \text{ pounds}$$

of water to pump, 20 pounds of injection and one pound of condensed steam. For 2,000 pounds we shall have

$$W\,(Q + 1) = 2{,}000\,(20 + 1).$$

TABLE III.—POUNDS OF INJECTION WATER REQUIRED PER POUND OF STEAM CONDENSED.

Temperature of Discharge from Air Pump	Entering Temperature of Injection Water t.													
	35	40	45	50	55	60	65	70	75	80	85	90	95	100
T	Pounds of Condensing Water Required per Pound of Steam. $Q = \frac{1190 - T}{T - t}$													
90	20 0	22 0	24.4	27.5	31.4	36.7	44.0	55.0	73.3	110.0	220.0	....	....	....
92	19.2	21.1	23.4	26.1	29.7	34.3	40.7	49.9	64.6	91.5	156.8	549.0	....	....
94	18 6	20 3	22.4	24 9	28.1	32.2	37.8	45.7	57.7	78 1	121.8	274.0	....	....
96	17.9	19.5	21.4	23.6	26.7	30.4	35.3	42.1	52.1	68.4	99.4	182.3	..	....
98	17.3	18.8	20.6	22.7	25 4	28 7	30.1	39.0	47.5	60.7	84.0	136.5	364.0	....
100	16.8	18.2	19.8	21.8	24.2	27.2	31.1	36.3	43.6	54.5	72.7	109.0	218.0	....
102	16.2	17.5	19.1	20.9	23.1	25.9	29.4	34.0	40.3	49.5	64.0	90.7	155.4	544.0
104	15.7	17.0	18.4	20.1	22.2	24.7	27.8	31.9	37.4	45.2	57.2	77.6	120.7	271.5
106	15.3	16.4	17.8	19.4	21.3	23.6	26.4	30.1	35.0	41.7	51.6	67.7	98.5	180.7
108	14.8	15.9	17.2	18.7	20.4	22.5	25.2	28.5	32.8	38.6	47.0	60.1	83.2	135.2
110	14.4	15.4	16.6	18.0	19.6	21.6	24.0	27.0	30.9	36.0	43.2	51.0	72.0	108.0
112	14.0	15.0	16.1	17.4	18.9	20.7	22.9	25.7	29.1	33.6	39.9	49.0	63.4	89.8
114	13.6	14.5	15.6	16.8	18.2	19.9	22.0	24.5	27.6	31.6	37.1	44.8	56.6	76.9
116	13.3	14.1	15.1	16.3	17.6	19.2	21.1	23.3	26.2	29.8	34.6	41.3	51.1	67.1
118	12.9	13.7	14.7	15.8	17.0	18.5	20.2	22.3	24.9	28.2	32.5	38 3	46.6	59.6
120	12.6	13.4	14.3	15.3	16.5	17.8	19.5	21.4	23.8	26.7	30.6	35.7	42.8	53.5
122	12.3	13.0	13.9	14.8	15.9	17.2	18.7	20.5	22.7	25.4	28.9	33.4	39.6	48.5
124	12.0	12.7	13.5	14.4	15.4	16.7	18.1	19.7	21.8	24.2	27.3	31.4	36.8	44.4
126	11.7	12.4	13.1	14.0	15.0	16.1	17.4	19.0	20.9	23.1	26.0	29.6	34.3	40.9
128	11.4	12.1	12.8	13.6	14.5	15.6	16 9	18.3	20.0	22.1	24.7	27.9	32.2	37.9
130	11.2	11.8	12.5	13.2	14.1	15.1	16.3	17.7	19.3	21.2	23.6	26.5	30.3	35.3
132	10.9	11.5	12.2	12.9	13.7	14.7	15.7	17.1	18.6	20.3	22 5	25.2	28.6	33.1
134	10.7	11.2	11.9	12.6	13.4	14.3	15.3	16.5	17.9	19.6	21.6	24.0	27.1	31.0
136	10.4	11.0	11.6	12.3	13.0	13.9	14.8	16.0	17.3	18.8	20.7	22.9	25.7	29.2
138	10.2	10.7	11.3	12.0	12.7	13.5	14.4	15.5	16.7	18.1	19.8	21.9	24.5	27.7
140	10.0	10.5	11.1	11.7	12 4	13.1	14.0	15.0	16.2	17.5	19.1	21.0	23.3	26.2

It takes in round numbers 28 cubic inches to make a pound of water at ordinary condenser temperatures, so that if we multiply this by 28 we shall get the number of cubic inches of water to be pumped per hour. By dividing this by 60 we get the number of cubic inches to be pumped per minute, and our formula so far becomes

Water pumped per minute=

$$W\,(Q + 1)\,\frac{28}{60} \text{ cubic inches.}$$

But in addition to the water we have a considerable quantity of air to handle, and we cannot count upon an efficiency of 100 per cent, so we must provide a pump of a displacement considerably more than the volume of the water. Let us call the ratio of the pump displacement to the volume of the water R. For instance, if the displacement were twice the volume of the water or the pump were allowed to half fill with water, R would be 2. Then the pump displacement D in cubic inches per minute would be

$$D = W(Q+1)\frac{28}{60}R.$$

If there is any standard value for R, if we can determine what proportion of their displacement it is safe or advisable to allow pumps of the different types to fill, we can combine this standard value of R with the fraction $\frac{28}{60}$ into a coefficient K and the formula becomes

$$D = K\,W(Q+1). \qquad (3)$$

For instance suppose it was the usual practice to use a pump having a displacement of twice the volume of the water to be pumped, *i. e.*, to let the pump fill half full of water, then R would equal 2, and

$$K = \frac{28}{60}R = \frac{28 \times 2}{60} = .93$$

and $D = .93\,W(Q+1)$.

Table IV gives the values of K for various values of R. If the pump fills with water to 60 per cent. of its displacement (column 1), *i. e.*, if the displacement is 1.667 times the volume of the water (column 2), the value of K would be .78 (column 3).

In order to establish the values of K used in current practice with various kinds of pumps we have obtained from such manufacturers as were willing to furnish them the data contained in Tables V to XIII. From the data in columns 2 to 6 the displacement D in cubic inches per minute has been computed, corrected for the rod, when its diameter (column 3) was known. Columns 9 and 10 give the value of W, *i. e.*, the weight of steam condensed per hour which the pump is adapted to take care of by the builder's rating. The values in column 10 are all reduced to 20 pounds of injection water to 1 of steam. When the builder's rating is based upon a different value of Q it is to be found in column 8. In columns 13 and 14 are the values of K corresponding with the ratings for jet and surface condensers respectively.

Take, for example, the Conover vertical single acting air pump, Table V. For a jet condenser the value of K for everything but the smallest size is .75 (column 13). In column 11 is given the capacity computed by the formula as given at the head of the column, which is simply a transposition of formula 3, taking $K = .75$ and the values there given will be seen to run very close to the builder's rating.

Knowing the value of K the process becomes very simple. Suppose we have a 500 horse power engine using 20 pounds of steam per hour per horse power; that the temperature of the injection water will be for considerable periods as high as 65°, and we want to keep the hot well or condenser temperature down to 110°. What size Conover pump should we require? We see from Table III that it will take 24 pounds of injection water per pound of steam; then

$Q = 24$;

$W = 500 \times 20 = 10{,}000$;

$K = .75$;

and

$D = K\,W\,(Q + 1) = .75 \times 10{,}000 \times 25 = 187{,}500$ cu. in. per min.

The sizes nearest to this capacity are the numbers 11 and 12

TABLE IV.—VALUES OF K FOR DIFFERENT RATIOS OF DISPLACEMENT TO VOLUME OF WATER HANDLED.

1 Per cent of Air Pump Displacement Filled with Water. $100\frac{V}{D}$	2 Ratio of Air Pump Displacement to Volume of Water. $R = \frac{D}{V}$	3 Value of K. $K = \frac{28}{60}R$
		Surface.
5	20	9.33
5.26	19	8.87
5.56	18	8.40
5.88	17	8.00
6.25	16	7.47
6.66	15	7
7.14	14	6.53
7.69	13	6.07
8.33	12	5.60
9.09	11	5.13
10.00	10	4.67
		Jet.
33.33	3	1.4
35	2.941	1.33
36	2.778	1.30
38	2.632	1.23
40	2.5	1.17
42	2.381	1.11
44	2.273	1.07
46	2.174	1.02
47	2.143	1
48	2.083	.97
50	2	.93
52	1.923	.90
54	1.852	.86
56	1.786	.83
58	1.724	.80
60	1.667	.78
62	1.613	.75
64	1.562	.73
66	1.515	.71
66.67	1.5	.70
68	1.471	.69
70	1.429	.67
72	1.389	.65
74	1.351	.63
75	1.333	.62
76	1.316	.61
78	1.282	.60
80	1.250	.58

(which, by the way, have the same size of air cylinder but different steam cylinders), having a capacity of 158,340, and the numbers 13 and 14, with a capacity of 204,781 cubic inches per minute. The purchaser can determine if it is safe in his case to take the next smaller or whether the next larger is necessary.

For the double-acting horizontal pumps the value of K runs very close to unity in several of the tables, although some of the makers rate them so high as to bring K down to a figure which gives them a considerably greater efficiency than the vertical single-acting pumps. What incongruities there are in the columns of K are evidently due to erratic ratings, for there should be no reason why one pump of the same kind and make should have a greater displacement per pound of water handled than the size next to it, saving always that very small sizes may be less efficient than the larger. Where the diameter of the rod is not given there would be a greater proportional difference between the net and gross displacement in the smaller sizes, which would call for a larger value of K for a very small pump where the displacement is uncorrected for the rod. As for the difference in the values of K as given by the ratings of the different makers we must leave our readers to judge whether there should be so much difference in the efficiency of the respective pumps that one could be allowed to fill over 78 per cent., while the other fills less than 50 per cent. of the stroke.

From the data presented the conclusion would appear warranted that a safe value for K would be .75 for vertical single-acting pumps and unity for horizontal double-acting pumps, and that pumps selected by the following formulas would be very close to what the makers would recommend for the capacity required:

For horizontal double-acting pumps:—

$$D = W(Q + 1) \qquad (4)$$

For vertical single-acting pumps:—

$$D = .75\, W(Q + 1) \qquad (5)$$

TO DETERMINE THE SIZE OF AIR PUMP REQUIRED FOR A JET CONDENSER.

Rule:—*Multiply the number of pounds of steam to be condensed per hour by one plus the number of pounds of injection water required per pound of steam. The product will be the required pump* *displacement in cubic inches per minute for a horizontal double-acting pump. For a single-acting vertical pump multiply the above product by* .75.

In column 11 of the tables are given the capacities calculated by formula 4 or 5, according to the type of pump. In all excepting a few instances of abnormally high rating the capacities will be seen to agree substantially with the rated capacities of the builders in column 9.

We are informed by one of the makers that they find the long rating perfectly safe, because they always use the size next above the required computed capacity. For instance, Dean Bros. rate their $7 \times 12 \times 12$ at 5,535 pounds of steam per hour, with 26 pounds of injection to one of steam, which gives a value for K of .632. This requires the pump to fill with water to almost 74 per cent. of its volume (Table IV), a condition which could not be counted upon to preserve a good vacuum with a double-acting horizontal pump, but this pump is used for everything between 2,300 and 5,500. With a capacity of about 2,950 its coefficient becomes unity and its capacity ample. Much of the time, too, the temperature of the injection would be such that less than 26 pounds of injection would be used, and it is improbable that the maximum conditions of load and injection temperature will come together. On a pinch, too, the speed of the pump could be increased above that given in columns 5 and 6. We think, however, that the pump should be rated at what it will do continuously and comfortably under the given conditions, and the engineer be allowed to decide how much he wishes to exceed that capacity in the maximum demand which he is likely to put upon it. The reduction of the ratings to a common unit of $K = 1$ for horizontal double-acting pumps and $K = .75$ for single-acting vertical pumps (column 11) makes this possible and easy. It should be noticed that this column is computed for 20 pounds of injection to one of steam.

In selecting a pump, it should be remembered that the work done depends upon the volume of air and water delivered against the atmospheric pressure, and not upon the size of the pump. In Fig. 4, which is a section of the Conover single-acting pump, suppose the volume of air and water in the chamber B to be so small that the air would not be compressed enough to lift the head valves. On the upward stroke the pump performs the work of lifting the water resting on the piston and of compressing the air.

On the downward stroke the water, in descending, gives back the work required to lift it, and the air in expanding the work required to compress it, so that no work is done except that required to overcome friction. When the volume of air and water in *B* becomes so great that the head valves are lifted the remainder of the stroke is completed against the pressure of the atmosphere, and the work done is proportional to the volume of air and water

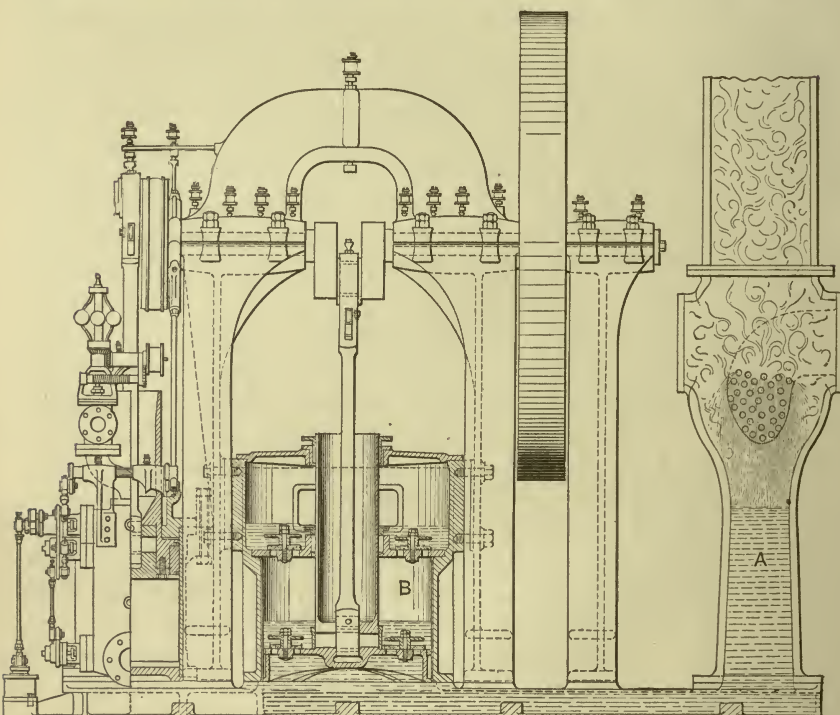
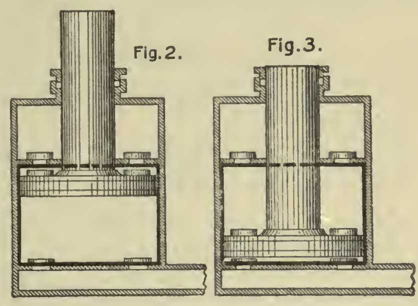


Fig. 4.

delivered at atmospheric pressure. Aside, then, from the additional investment in and friction of the larger pump, it is at no disadvantage over the smaller.

In computing the displacement of a single-acting pump, it should be noted that when a foot-valve is used, the pump is to an extent double acting. This will be seen by reference to Figs.

2 and 3. When the piston is in its highest position, as in Fig. 2, we have the volume between the head and the foot valves containing only the piston. When the piston is in its lowest position (Fig. 3) this volume has been further reduced by the volume of the trunk, and a volume equal to the cross sectional area of the trunk, multiplied by the length of the stroke, must have been discharged through the head valves on the downward stroke. When the area of the trunk becomes equal to that of the annular space around it, the pump discharges as much on the downward as upon the upward stroke. Where no foot-valve is used, as in Fig. 1, the water is held against the piston by the head at *A*, avoiding the disagreeable chug which occurs when the piston is allowed to come against a confined body of water.



The factor of uncertainty in air pump work is the air entering by leakage. This air entering at the atmospheric pressure of, say, 15 pounds, expands in a vacuum of 26 inches to over 7 times its original volume, and this increased volume must be provided for in the displacement of the air cylinder. This factor is therefore an exceedingly important one, and is very variable. Some plants are tight, others are not. Generally plants using a small amount of steam per horse power have a large leakage factor, because while the amount of steam used per horse power is decreased, the leakage is liable to go the other way, the pressure in the lower pressure cylinder being much of the time below that of the atmosphere.

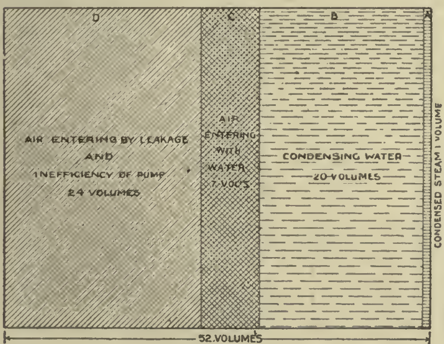


Fig. 1.

SIZE OF AIR PUMP FOR SURFACE CONDENSER.

In a jet condenser, the air pump has to handle a comparatively large amount of water. In the surface condenser it handles a small volume of water and a large volume of air. Assume 20 pounds of injection to one of steam condensed, air carried by the water one-third the volume of the water when expanded, and an air pump with a displacement about two and a half times the volume of the water to be moved. Under these conditions we should have in Fig. 1:

At *A* the condensed steam = 1 vol.

At *B* the condensing water = 20 vols.

At *C* the air entering with the water = 7 vols.

And at *D* the space which it is found necessary to provide for the air entering by leakage and to cover the deficiencies of the pump, consisting of 24 volumes.

In a surface condenser we have to handle only the water represented by *A*. The space *B*, therefore, can be cut out of the diagram, as can also the space *C*, as the only water which can bring in air is the comparatively small quantity used for make-up. We are liable to have as much air enter by leakage into a surface as into a jet system, so that the portion of the space *D* which is required for air entering by leakage, and this is the greater portion of it, will remain unchanged. The portion required to cover the lack of perfection in the pump will diminish with the volume pumped. With a pump of half the size a less number of cubic inches of volume will be lost by failure to fill, slippage, failure of valves to seat, etc.

If *D* were left as it is, we should have a displacement 25 times the volume of the water pumped. The practice of the pump companies generally is to give the air pump a displacement equal to 20 times the volume of the condensed steam, if it is a horizontal double-acting, and 12 times if it is vertical single-acting. The Conover company give their pumps a displacement of only 10 times the volume of the condensed steam.

The values of *K* for these ratios are:

R		*K*
10	=	4.67
12	=	5.6
20	=	9.33

For general use these values may safely be taken at 9 for a horizontal double-acting, and at 5 for a vertical single-acting pump. Since there is no injection the volume of water to be handled is simply W, and the formulas become:

FOR A HORIZONTAL DOUBLE-ACTING PUMP

$$D = 9W \tag{6}$$

FOR A VERTICAL SINGLE-ACTING PUMP

$$D = 5W \tag{7}$$

or generally

$$D = KW \tag{8}$$

TO DETERMINE THE SIZE OF AIR PUMP REQUIRED FOR A SURFACE CONDENSER.

Multiply the number of pounds of steam to be condensed per hour by 9 *for a horizontal double-acting or by* 5 *for a vertical single-acting pump. The product will be the air pump displacement required in cubic inches per minute.*

In column 12 of the tables are given the capacities calculated by formula 6 or 7, according to the type of pump. A comparison between columns 10 and 12 will show how nearly the formula with the constants chosen will come to the builder's rating. Of course the general formula 8 may be employed, the user choosing his own value of K.

CONDENSING SURFACE REQUIRED.

In the early days of the surface condenser it was thought necessary to provide a cooling surface in the condenser equal to the heating surface in the boilers, the idea being that it would take as much surface to transfer the heat from a pound of steam to the cooling water and condense the steam as it would to transfer the heat from the hot gases to the water in the boiler and convert it to steam. The difference in temperature, too, between the hot gases and the water in the boiler is considerably greater than that between the steam in the condenser and the cooling water. Steam, however, gives up its heat to a relatively cool surface much more readily than do the hot furnace gases, and the positively circulated cooling water takes up that heat and keeps the temperature of the surfaces down, while in a boiler the absorption depends in a great measure upon the ability of the water by natural circulation to get into contact with the surface and take up the heat by evaporization. It has been found, therefore, that a much smaller surface will suffice in a condenser than in the boilers which it serves.

The Wheeler Condenser and Engineering Company, who make a specialty of surface condensers, say that one square foot of cooling surface is usually allowed to each 10 pounds of steam to be condensed per hour, with the condensing water at a normal temperature not exceeding 75°. This figure seems to be generally used for average conditions. Special cases require special treatment. For service in the tropics the heating surface should be at least ten per cent. greater than this estimate: Where there is an abundance of circulating water the surface may be much less, as with a keel condenser, where 50 pounds of steam is sometimes condensed per hour per square foot of surface; or a water works engine, where all the water pumped is discharged through the condenser and not appreciably raised in temperature, probably condensing 20 to 40 pounds per hour per square foot of surface.

Mr. J. M. Whitham, in a paper upon "Surface Condensers," presented to the American Society of Mechanical Engineers,* gives the following formula for calculating the surface required:

$$S = \frac{WL}{180\,(T - t)}$$

Where

S = the surface in square feet,

W = the weight of steam condensed per hour,

L = the latent heat of steam at the condenser temperature,

T = the temperature of the condenser, or the air pump discharge,

and t = the average temperature of the circulating water, *i. e.*, the sum of its initial and final temperatures, divided by 2.

For ordinary conditions this reduces to

$$S = \frac{17\ W}{180}$$

or one square foot of heating surface to about 10.6 pounds of steam condensed per hour.

This refers to the ordinary arrangement of horizontal brass tubes of small diameter to the surface condenser as ordinarily

*See Trans. A S. M. E., Vol. IX, p. 417.

used. With other arrangements of surface, etc., it might not apply.

CONDENSING WATER PER HORSE POWER IN GALLONS.

The value of Q can be reduced to gallons by dividing by 8.25, for since one pound equals 28 cubic inches one pound equals $\frac{28}{231} = \frac{1}{8.25}$ of a gallon.

If we let $S =$ the steam required per horse power per hour, the weight of steam to be condensed per hour for a given engine will be

$$W = HP \times S$$

The condensing water required in pounds per hour will be

$$WQ = HP \times Q \times S$$

in gallons per hour,

$$\frac{HP \times Q \times S}{8.25}$$

or, in gallons per minute,

$$\frac{HP \times Q \times S}{60 \times 8.25} = \frac{HP \times Q \times S}{495}$$

This is $\frac{QS}{495}$ gallons per minute per horse power.

Taking 20 pounds of steam per horse power per hour and 25 pounds of condensing water per pound of steam the condensing water per horse power =

$$\frac{25 \times 20}{495} = \frac{500}{495} \text{ gals. per min.,}$$

or just about a gallon per minute per horse power, which is a a much used value.

SIZE OF INJECTION MAIN.

The injection main should be so proportioned that the velocity of flow does not exceed 300 feet per minute, a velocity which will be closely approximated by making the diameter of the pipe the square root of the quotient of the pounds of condensing water required per hour divided by 6,000.

For example, to condense 2,000 pounds of steam per hour with 24 pounds of injection per pound of steam would require

$24 \times 2,000 = 48,000$ pounds of injection water per hour.

Divide this by 6,000 and extract the square root and you have the pipe diameter,

$$48,000 \div 6,000 = 8$$

The pipe should evidently be 3 inches, the square of 2.5, the next lower size being 6.25.

If there were no air present in the condenser the temperature for a given vacuum or absolute pressure would be that given in Table 1. There is in the condenser, however, the pressure not only of the steam arising from the water, but that of the enclosed air, so that for a given condenser temperature the absolute pressure will be higher, *i. e.*, the vacuum will be less than indicated by the table. With a temperature of air pump discharge of 120°, for instance, we should have, if there were nothing present but the water and the steam arising from it, an absolute pressure of 1.682 pounds per square inch, or a vacuum of 26.5 inches. We cannot have a lower absolute pressure, *i. e.*, a better vacuum, with this temperature, because the water at this temperature will continue to give off steam of this pressure and keep the condenser full of it, no matter how much capacity we have to our air pump. Now, if we admit a little air we shall have an additional pressure with no increase of temperature, and of course there will be some air present. For this reason it is common to see condensers running with a discharge temperature of 100°, which by the table should give a vacuum of over 28 inches, with the gage showing 26 or less.

TABLE V.—CONOVER VERTICAL SINGLE-ACTING.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Builder's Number.	Air Cylinder.			Speed (maximum)		Displacement per Minute D Cubic inches.	Capacity Builder's Rating. Pounds of Steam Condensed per Hour W			Capacity Computed.		Value of K Computed from Builder's Rating.	
	Diam. of Bore, inches.	Diam. of Trunk, inches.	Length of Stroke, inches.	Working Strokes per min.	Feet Travel per min.		Jet $Q=$	Jet $Q=20$	Surface $K=10$	Jet $Q=20$ $K=75$ $W=\frac{D}{K(Q+1)}$	Surface $K=5$ $W=\frac{D}{K}$	Jet $K=\frac{D}{W(Q+1)}$	Surface $K=\frac{D}{W}$
1& 2	9	$4\frac{1}{2}$	4	90	30	17795	......	1000	3800	1130	3559	.85	4.68
3& 4	11	$4\frac{1}{2}$	5	80	$33\frac{1}{3}$	31652	......	2000	6800	2009	6330	.75	4.65
5& 6	$13\frac{1}{2}$	$5\frac{3}{8}$	6	80	40	57816	......	3600	12400	3671	11563	.75	4.66
7& 8	15	$6\frac{5}{8}$	7	80	$46\frac{2}{3}$	79654	......	5000	17100	5057	15931	.75	4.65
9&10	17	$7\frac{1}{4}$	7	80	46_{3}	103992	......	6600	22300	6603	20798	.75	4.66
11&12	20	8	8	75	50	158340	......	10000	33900	10051	31668	.75	4 67
13&14	22	$8\frac{3}{8}$	9	70	$52\frac{1}{2}$	204781	......	13000	43800	13008	4095	.75	4 67
15&16	25	$9\frac{3}{4}$	10	70	$58\frac{1}{3}$	291347	......	18500	62400	18500	58269	.75	4.67
17&18	28	$10\frac{1}{2}$	11	70	$64\frac{1}{6}$	407453	......	26000	87200	25870	81491	.75	4.67
19&20	32	11	12	70	70	595745	......	37500	127600	37825	119149	.75	4.67
21&22	34	12	14	70	$81\frac{2}{3}$	778933	......	49500	166800	49456	155786	.75	4.67
22&23	36	13	16	65	86_{3}	920556	......	59000	197100	59083	184111	.75	4.67
24&25	40	14	16	60	80	1058602	......	67500	226700	67213	211720	.75	4.67

TABLE VI.—DEAN BROS.' SINGLE CYLINDER HORIZONTAL DOUBLE-ACTING.

1	2	3	3½	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.	Air Cylinder.				Speed (maximum).		Displacement per Minute, D Cubic Inches.	Capacity. Builders' Rating, Pounds of Steam Condensed per Hour, W.			Capacity Computed.		Value of K Computed from Builders' Rating.	
Inches.	Diam. of Bore, Inches.	Diam. of Rod, Inches.	Diam. Tail Rod, Inches.	Length of Stroke, Inches.	Working Strokes per min.	Feet Travel per min.		Jet $Q=26$	Jet $Q=20$	Surface $R=20$	Jet $Q=20$ $K=1$ $W=\frac{D}{K(Q+1)}$	Surface $K=9$ $W=\frac{D}{K}$	Jet $K=\frac{D}{W(Q+1)}$	Surface $K=\frac{D}{W}$
3	4	$\frac{13}{16}$	...	3½	103	30	4440	183	242	337	211	349	.874	9.33
3½	5	$\frac{13}{16}$	...	4	90	30	6979	294	378	515	332	534	.879	9.33
4	4½	$\frac{13}{16}$	...	5	84	35	6575	277	356	520	313	539	.879	9.33
4	6	$\frac{13}{16}$	...	5	84	35	11770	493	634	871	560	903	.884	9.33
4	5½	$\frac{13}{16}$	...	7	84	50	13823	592	761	1040	658	1078	.864	9.33
4	8	$1\frac{1}{16}$	...	7	84	50	29290	1253	1611	2243	1395	2325	.866	9.33
5½	9	$1\frac{5}{16}$	...	10	72	60	45317	2287	2940	4317	2158	4476	.733	9.33
7	12	$1\frac{5}{16}$	...	12	70	70	94436	5535	7116	8531	4497	8844	.632	9.33
7	12	$1\frac{5}{16}$	...	15	60	75	101181	5928	7622	10302	4818	10680	.632	9.33
7	12	$1\frac{5}{16}$	...	18	53	80	107252	6775	8711	12796	5107	13266	.586	9.33
7	14	$1\frac{11}{16}$	$1\frac{11}{16}$	15	60	75	136533	8064	10368	14146	6502	14664	.627	9.33
8	14	$1\frac{11}{16}$	$1\frac{11}{16}$	18	53	80	144725	9220	11854	16975	6892	17597	.581	9.33
8	16	$1\frac{11}{16}$	$1\frac{11}{16}$	18	53	80	189678	12037	15476	22244	9032	23060	.583	9.33
10	18	$1\frac{15}{16}$	$1\frac{11}{16}$	18	53	80	240291	15242	19597	27212	11442	28210	.583	9.33
10	18	$1\frac{15}{16}$	$1\frac{11}{16}$	20	51	85	256915	16194	20821	29156	12234	30225	.587	9.33
12	20	$2\frac{3}{16}$	$1\frac{15}{16}$	24	50	100	388508	23529	30252	42374	18500	43927	.611	9.33
12	22	3	$2\frac{3}{16}$	24	50	100	468396	28479	36616	50605	22305	52461	.609	9.33
12	24	3	$2\frac{3}{16}$	24	50	100	558721	33887	43569	59213	26606	61385	.611	9.33
14	30	3	$2\frac{5}{8}$	24	50	100	875775	52950	68078	93716	41704	97152	.612	9.33
14	36	3	$2\frac{5}{8}$	24	50	100	1264575	76252	98038	127512	60218	132187	.614	9.33

TABLE VII.—KNOWLES SINGLE CYLINDER HORIZONTAL DOUBLE-ACTING.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.	Air Cylinder.			Speed (maximum).		Displacement per Minute D Cubic inches.	Capacity, Builder's Rating. Pounds of Steam Condensed per Hour W			Capacity Computed.		Value of K Computed from Builder's Rating.	
Inches.	Diam. of Bore, Inches.	Diam. of Rod, Inches.	Length of Stroke, Inches.	Working Strokes per min.	Feet Travel per min.		Jet $Q=30$	Jet $Q=20$	Surface $R=$	Jet $Q=20$ $K=1$ $W=\frac{D}{K(Q+1)}$	Surface $K=9$ $W=\frac{D}{K}$	Jet $K=\frac{D}{W(Q+1)}$	Surface $K=\frac{D}{W}$
5½	8	......	7	150	87½	52778	1560	2302	5657	2513	5864	1.09	9.33
6	9	......	10	140	116⅔	89064	2240	3306	9543	4240	9893	1.28	9.33
6	10	......	12	120	120	113097	3340	4930	12122	5385	12566	1.10	9.33
8	12	......	12	120	120	162864	4800	7085	17456	7755	18096	1.09	9.33
8	14	......	12	120	120	221673	6400	9446	23759	10556	24630	1.12	9.33
10	14	......	16	90	120	221673	6540	9653	23759	10556	24630	1.09	9.33
10	16	......	16	90	120	289440	8540	12605	31022	13783	32160	1.09	9.33
12	18	......	18	80	120	366437	10800	15940	39275	17450	40715	1.09	9.33
12	18	......	24	60	120	366437	11000	16236	39275	17450	40715	1.07	9.33
12&14	20	......	24	60	120	452392	13340	19699	48483	21542	50266	1.09	9.33
14	22	......	24	60	120	547387	16140	23822	58669	26066	60821	1.09	9.33
14	24	......	24	60	120	651441	19200	28339	69822	31021	72382	1.09	9.33
16	26	......	24	60	120	764539	22540	33269	81944	36406	84949	1.09	9.33
16	28	......	24	60	120	886680	26140	38583	95035	42223	98520	1.09	9.33

TABLE VIII.—LAIDLAW-DUNN-GORDON SINGLE CYLINDER HORIZONTAL DOUBLE-ACTING.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.	Air Cylinder.			Speed (maximum).		Displacement per Minute, D Cubic inches.	Capacity, Builder's Rating. Pounds of Steam Condensed per Hour W			Capacity Computed.		Value of K Computed from Builder's Rating.	
Inches.	Diam. of Bore, Inches.	Diam. of Rod, Inches.	Length of Stroke, Inches.	Working Strokes per min.	Feet Travel per min.		Jet Q = 25	Jet Q = 20	Surface R =	Jet Q = 20 K = 1 $W = \frac{D}{K(Q+1)}$	Surface K = 9 $W = \frac{D}{K}$	Jet $K = \frac{D}{W(Q+1)}$	Surface $K = \frac{D}{W}$
5	6	......	8	75	50	16964	752	936	2300	807	1884	.863	7.38
7	8	......	12	75	75	45238	2019	2500	6200	2154	5026	.862	7.30
8	10¼	......	12	75	75	74264	3336	4131	8700	3536	8251	.856	8.54
8	12	......	18	55	82.5	111969	5021	6216	11300	5331	12441	.857	9.91
10	14	......	18	55	82.5	152401	6839	8468	17000	7257	16933	.857	8.96
12	16	......	18	55	82.5	199049	8907	11028	21500	9478	22116	.859	8.79
12	18	......	24	45	90	274828	12328	15263	29000	13087	30536	.857	9.48
14	20	......	24	45	90	339293	15228	18861	35000	16156	37699	.856	9.70
14	22	......	24	45	90	410540	18435	22825	42000	19549	45615	.856	9.78
16	24	......	24	45	90	490581	21964	27194	......	23361	....	.859	....
16	26	......	24	45	90	573404	25695	31813	......	27305	....	.858	....
18	28	......	24	45	90	665010	29908	37029	......	31667	....	.855	Av. 8.87
18	30	......	24	45	90	763409	34198	42341	......	36352	....	.858	....
20	32	......	24	45	90	869590	38769	48000	......	41409	....	.863	....

TABLE IX.—SNOW SINGLE CYLINDER HORIZONTAL DOUBLE-ACTING.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.	Air Cylinder.			Speed (maximum).		Displacement per Minute, D Cubic inches.	Capacity, Builder's Rating, Pounds of Steam Condensed per Hour W.			Capacity Computed.		Value of K Computed from Builder's Rating.	
							Jet			Jet $Q=20$ $K=1$	Surface $K=9$	Jet	Surface
Inches.	Diam. of Bore, Inches.	Diam. of Rod, Inches.	Length of Stroke, Inches.	Working Strokes per min.	Feet Travel per min.		$Q=$	$Q=20$	Surface $R=$	$W=\frac{D}{K(Q+1)}$	$W=\frac{D}{K}$	$K=\frac{D}{W(Q+1)}$	$K=\frac{D}{W}$
3	3½	...	6	85	42½	4329	....	300	....	206	481	.69	....
4	3¾	...	8	75	50	6627	....	....	1000	316	736	...	6.63
4	4¾	...	8	75	50	10637	....	600	....	507	1182	.84	....
5¼	5¾	...	8	75	50	15580	....	....	2000	742	1731	...	7.79
5¼	6¾	...	8	75	50	21471	....	1200	....	1022	2386	.85	....
6	7	...	10	75	62½	28864	....	....	4000	1374	3207	...	7.22
6	8½	...	10	75	62½	42559	....	2350	....	2027	4729	.86	....
7	8½	...	12	75	75	51070	....	....	6000	2432	5674	...	8.51
6&7	10	...	12	75	75	70686	....	3900	10000	3366	7854	.88	7.07
7&8	12	...	12	75	75	101790	....	5600	....	4847	11310	.87	....
8	12	...	18	55	82½	111969	....	6300	14000	5332	12441	.85	8.
10	14	...	18	55	82½	152401	....	8600	18000	7257	16933	.84	8.47
10&12	16	...	18	55	82½	199049	....	11200	24000	9479	22117	.85	8.29
10&12	18	...	18	55	82½	251925	....	14000	32000	11996	27992	.86	7.87
12&14	20	...	18	55	82½	311018	....	17500	36000	14810	34558	.85	8.64
14&16	22	...	18	55	82½	376329	....	21000	40000	17920	41814	.85	9.41
14&16	24	...	18	55	82½	447866	....	25000	....	21327	49763	.85	....
14&16	22	...	24	45	90	410540	....	23000	45000	19550	45616	.85	9.12
16&18	24	...	24	45	90	488581	....	27000	50000	23266	54287	.86	9.77
18&20	26	...	24	45	90	573404	....	32000	....	27305	63712	.85	....
20&22	30	...	24	45	90	763509	....	43000	....	36358	84834	.84	....

TABLE X.—WORTHINGTON DUPLEX HORIZONTAL DOUBLE-ACTING.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.	Air Cylinder.			Speed (Maximum).		Displacement per Minute, D Cubic Inches.	Capacity Builder's Rating, Pounds of Steam Condensed per Hour, W.			Capacity Computed.		Value of K Computed from Builder's Rating.	
	Diam. of Bore, Inches.	Diam. of Rod, Inches.	Length of Stroke Inches.	Working Strokes per min.	Feet Travel per min.		Jet		Surface	Jet $Q = 20$ $K = 1$	Surface $K = 9$	Jet	Surface
Inches.							$Q =$	$Q = 20$	$K = 20$	$W = \frac{D}{K(Q+1)}$	$W = \frac{D}{K}$	$K = \frac{D}{W(Q+1)}$	$K = \frac{D}{W}$
$5\frac{1}{4}$	$4\frac{3}{4}$	....	5	108	45	19146	....	600	2052	911	2127	1.52	9.33
6	$5\frac{3}{4}$	....	6	90	45	28044	....	1300	3006	1335	3116	1.03	9.33
$7\frac{1}{2}$	$7\frac{1}{2}$	....	6	120	60	63618	....	2200	6819	3029	7069	1.38	9.33
$7\frac{1}{2}$	7	....	10	91	76	70197	....	3100	7524	3342	7800	1.08	9.33
$7\frac{1}{2}$	$8\frac{1}{2}$	....	10	102	85	115760	....	5500	12407	5512	12862	1.00	9.33
$7\frac{1}{2}$	$10\frac{1}{4}$	....	10	90	75	148529	....	7200	15919	7073	16503	.98	9.33
9	12	....	10	93	78	211723	....	10000	22693	10082	23525	1.00	9.33
12	14	....	10	102	85	314038	....	13300	33659	14954	34893	1.12	9.33
12	15	....	10	102	85	360488	....	15600	38678	17166	40054	1.10	9.33
12	15	....	15	68	85	360488	....	17100	38678	17166	40054	1.00	9.33
14	17	....	15	68	85	463039	....	22000	49629	22049	51449	1.00	9.33
14&17	19	....	15	68	85	578401	....	27000	61994	27543	64267	1.02	9.33
14&17	22	....	15	..	..	....	....	....	....	....	....	....	....
18	22	....	18	..	..	....	....	....	....	....	....	....	....
18	24	....	18	..	..	....	....	....	....	....	....	....	....
18	26	....	18	..	..	....	....	....	....	....	....	....	....
18	29	....	18	..	..	....	....	....	....	....	....	....	....

TABLE XI.—DEANE STEAM PUMP CO. SINGLE CYLINDER HORIZONTAL DOUBLE-ACTING.

1		2	3	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.		Air Cylinder.			Speed (Maximum).		Displacement per Minute, D Cubic Inches.	Capacity Builder's Rating, Pounds of Steam Condensed per Hour, W.			Capacity Computed.		Value of K Computed from Builder's Rating.	
Inches.		Diam. of Bore, Inches.	Diam. of Rod, Inches.	Length of Stroke, Inches.	Working Strokes per min.	Feet Travel per min.		Jet $Q=29$	Jet $Q=20$	Surface $R=25$	Jet $Q=20$ $K=1$ $W=\frac{D}{K(Q+1)}$	Surface $K=9$ $W=\frac{D}{K}$	Jet $K=\frac{D}{W(Q+1)}$	Surface $K=\frac{D}{W}$
3½	..	5	....	5	98	40	9425	410	586	820	449	1037	0.766	11.494
4½	..	6	....	5	98	40	12572	580	829	1160	646	1508	0.78	11.7
5½	..	8	....	7	85.7	50	30159	1300	1857	2600	1436	3351	.773	11.6
6	..	8	....	12	65	65	39207	1700	2429	3400	1867	4356	.77	11.53
6	..	9	....	12	65	65	49621	2140	3057	4280	2363	5513	.773	11.59
6	8	10	....	12	65	65	61261	2650	3785	5300	2917	6807	.77	11.56
8	10	12	....	12	65	65	88218	3800	5429	7600	4200	9802	.773	11.6
8	10	13	....	12	65	65	103529	4500	6429	9000	4930	11503	.766	11.5
8	10	14	....	12	65	65	120073	5200	7429	10400	5718	13341	.769	11.54
8	10	14	....	18	50	75	138544	6000	8571	12000	6597	15394	.77	11.545
10	12	16	....	18	50	75	180954	7880	11257	15760	8617	20106	.765	11.48
10	12	18	....	18	50	75	229022	9900	14143	19800	10906	25447	.771	11.56
12	14	18	....	24	50	100	305363	13220	18886	26440	14541	33929	.77	11.55
14	16	20	....	24	50	100	376990	16320	23314	32640	17952	41888	.77	11.55
14	..	22	....	24	50	100	456156	19740	28200	39480	21721	50684	.77	11.55
16	..	24	....	24	50	100	542867	23500	33571	47000	25850	60318	.77	11.55
16	..	26	....	24	50	100	637115	27600	39420	55200	30339	70790	.7694	11.54
18	..	28	....	24	50	100	738903	32000	45714	64000	35186	82100	.7694	11.54
18	..	30	....	24	50	100	848230	36720	52457	73440	40392	94248	.77	11.55

TABLE XII —DEAN BROS TWIN CYLINDER VERTICAL SINGLE ACTING PUMP WITH JET CONDENSERS

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.	Air Cylinder.			Speed (Maximum).		Displacement per Minute, D Cubic Inches.	Capacity, Builder's Rating. Pounds of Steam Condensed per Hour, W			Capacity Computed.		Value of K Computed from Builder's Rating.	
Inches.	Diam. of Bore, Inches.	Diam. of Rod, Inches.	Length of Stroke, Inches.	Working Strokes per min.	Feet Travel per min.		Jet $Q=26$	Jet $Q=20$	Surface $R=$	Jet $Q=20$ $K=.75$ $W=\frac{D}{K(Q+1)}$	Surface $K=$ $W=\frac{D}{K}$	Jet $K=\frac{D}{W(Q+1)}$	Surface $K=\frac{D}{W}$
7	12	$1\frac{5}{16}$	10	72	60	80945	4066	5228	....	5130	....	.74	....
8	14	$1\frac{11}{16}$	10	72	60	110032	5534	7115	....	6986	....	.74	....
9	16	$1\frac{11}{16}$	12	70	70	167951	9834	12644	....	10664	....	.63	....
10	18	$1\frac{15}{16}$	12	70	70	212517	12453	16011	....	13493	....	.63	....
10	20	$1\frac{15}{16}$	12	70	70	263771	15373	19765	....	16747	....	.63	....
12	22	$2\frac{3}{16}$	15	60	75	340426	19927	25620	....	21614	....	.63	....
12	25	3	18	53	80	464918	29401	37801	....	29519	....	.59	....
14	30	3	18	53	80	670973	42338	54435	....	42600	....	.59	....

TABLE XIII.—BARR HORIZONTAL DOUBLE-ACTING.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Diameter of Steam Cylinder.	Air Cylinder.			Speed (Maximum).		Displacement per Minute, D Cubic Inches.	Capacity, Builder's Rating. Pounds of Steam Condensed per Hour, W			Capacity Computed.		Value of K Computed from Builder's Rating.	
							Jet		Surface	Jet $Q=20$ $K=1$	Surface $K=9$	Jet	Surface
Inches.	Diam. of Bore, Inches.	Diam. of Rod, Inches.	Length of Stroke, Inches.	Working Strokes per min.	Feet Travel per min.		$Q=27$	$Q=20$	$R=20$	$W=\frac{D}{K(Q+1)}$	$W=\frac{D}{K}$	$K=\frac{D}{W(Q+1)}$	$K=\frac{D}{W}$
3½	5	....	5	120	50	11778	420	560	1262	560	1309	1.00	9.33
4½	6	....	5	120	50	16962	606	808	1817	808	1885	1.00	9.33
5½	8	....	7	$85\frac{7}{10}$	50	30156	1077	1436	3231	1436	3351	1.00	9.33
6	8	....	12	75	75	45234	1617	2156	4846	2156	5026	1.00	9.33
8	10	....	12	75	75	70686	2523	3366	7573	3366	7854	1.00	9.33
10	12	....	12	75	75	101700	3630	4810	10896	4840	11300	1.00	9.33
10	14	...	12	75	75	138510	4825	6596	14840	6466	15390	1.02	9.33
10	14	....	18	60	90	166212	5936	7915	17808	7915	18468	1.00	9.33
12	16	....	18	60	90	217080	7753	10337	23259	10337	24120	1.06	9.33
14	18	....	24	45	90	274752	9812	13083	29438	13083	30528	1.00	9.33
16	20	....	24	45	90	339228	12115	16154	36346	16153	37692	1.00	9.33
16	22	....	24	45	90	410508	14661	19548	43983	19548	45612	1.00	9.33
18	24	....	24	45	90	488484	17446	23261	52338	23261	54276	1.00	9.33
20	28	....	24	45	90	664956	23748	31664	71245	31664	73884	1.00	9.33

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