under pressure is supplied beneath the fuel through the ash-pit, and above the fuel through the holes in the inner furnace door, the air issuing through these holes being under considerably higher pressure than that in the ash-pit. The object of this double admission is to secure the complete combustion of the fuel with a very moderate supply of air, and thus to increase the temperature of the products of combustion. The air supply is heated by the waste gases from 180° to 200° above its ordinary temperature, and whenever the furnace door is opened the current of air is cut off for the moment so as to prevent the sudden cooling down of the furnace.

At first sight it would appear that the forced draught system is of no special advantage where economy of fuel is a primary consideration; for, although by its use the grate area may be diminished, nevertheless, the heating surface and the total weight of boiler cannot be reduced. It must however be borne in mind that if the supply of air be properly regulated, the volume of the products of combustion will be greatly diminished, and their temperature increased; consequently, a less heating surface is required to produce the same evaporative results per pound of fuel than when natural draught is employed. Also with forced draught it is possible to make use of tubes of comparatively small diameter, and consequently a considerably increased heating surface can be obtained without increasing the dimensions of the boiler.

CHAPTER X.

CONDENSATION AND CONDENSERS.

The object and advantages of condensing steam—General description of condensers—Quantity of water required to condense steam—Objects of surface condensation for marine engines—Description of a jet condenser for a stationary engine—Description of a marine surface condenser—Air-pumps—Ejector condensers—Method of indicating the vacuum.

THE condenser may, in a certain sense, be described as having the inverse functions of the boiler; for, whereas the latter is employed to raise the medium with which the engine works to the superior limit of temperature, the purpose of the latter is to reduce the inferior limit of temperature as far as possible. The boiler fulfils its purpose by converting the feed water into steam, and the condenser by re-converting that steam after it has done its work into water.

The advantages from the thermal point of view of condensing the steam, instead of allowing it to escape into the open air at a little above the atmospheric pressure, are very easily explained by reference to the principles which enable us to calculate the maximum efficiency of heat engines (see p. 88). Suppose, for instance, that we have two precisely similar engines working with steam of 50 lbs. pressure per square inch absolute, and one provided with a condenser, while the other discharges the exhaust into the open air. Suppose, also, that the former expands down to a pressure of 3 lbs. absolute, and the latter down to a pressure of about 3 lbs. above the atmosphere, say 18 lbs. absolute.

The relative theoretical efficiencies of the two engines may be expressed as follows. The temperature of steam of 50 lbs. absolute is 280·5°, while that of steam of 18 lbs. absolute is 222°·5, and of 3 lbs. absolute is 141·5°. Then, according to the principles of the efficiency of heat engines, the maximum efficiency of the condensing engine is—

$$\frac{280{\cdot}5 - 141{\cdot}5}{461 + 280{\cdot}5} = \frac{139}{741{\cdot}5};$$

while that of the non-condensing engine is—

$$\frac{280{\cdot}5 - 222{\cdot}5}{461 + 280{\cdot}5} = \frac{58}{741{\cdot}5}.$$

Thus the condensing engine is theoretically the more efficient of the two in the ratio of 139 to 58·5, or 2·37 to 1.

From the mechanical point of view, as illustrated by the indicator diagram, the advantages of condensation are most apparent, for it enables the back pressure to be reduced from some three pounds above the atmosphere to, say, ten or eleven pounds below it. Also, as in non-condensing engines it is practically impossible to expand the steam below the atmospheric pressure, it is evident that condensation enables us to make use of much higher grades of expansion than would otherwise be possible. This latter is merely another mode of expressing the advantage explained above by reference to the principles of thermodynamics.

The condenser is an apparatus into which the steam is discharged when it has done its work, and where it comes in contact either with a jet of cold water, or else with a large area of metallic surface, one side of which is kept cool by contact with cold water. The steam on entering this chamber is instantly condensed, giving up its heat to the water; and the result would be, if a sufficient quantity of water were used, the formation of a practically perfect vacuum, were it not for the fact that the feed water usually contains a large quantity of air, which passes over with the exhaust steam into the condenser, and exerts a back pressure against the piston. In order to get rid of this air, an air-pump, driven by the engine, is fitted to the condenser, and is also made use of to pump away the water into which the steam condenses. Various types of condensers, together with their fittings, are described and illustrated on pages 429 to 437.

Quantity of water required to effect condensation.—Suppose the steam is expanded in the cylinder down to a pressure of say 4·5 lbs. per square inch, the temperature corresponding to which is 158°; and suppose, further, that the temperature of the final mixture of condensed steam, and of condensing or injection water is to be 110°, then for every pound of steam which enters the condenser the injection water will have to absorb the total heat of the steam of 158° above the water of 110°.

The total heat of steam of 158° is 1,130 thermal units; subtracting from this the heat of water at 110°, which is approximately 110 thermal units, we have 1,020 units, which have to be absorbed by the injection water. The quantity of the latter required obviously depends upon its initial temperature. Suppose the latter to be 50°, each pound of it can absorb 110 − 50 = 60 thermal units by rising in temperature to 110°. Therefore the total quantity of water required is $\frac{1020}{60} = 17$ lbs. This, therefore, is the *minimum* quantity of water required under the given circumstances.

It is obvious from the foregoing that the quantity of injection water required in any given case depends upon the final pressure of the steam, the initial temperature of the injection water, and the temperature at which it is required that the mixture of injection water and condensed steam should be maintained.

Let T_1 = the temperature of the steam when the exhaust opens.

Let $L =$ the latent heat of the steam at this temperature.
Then $T_1 + L =$ the total heat in thermal units of 1 lb. of the steam.
Let $T_2 =$ the temperature of the injection water.
Let $T_3 =$ „ „ final mixture.
Let $W =$ the weight in pounds of the injection water per pound weight of steam.

The injection water in rising from T_2 to T_3 absorbs—

$$(T_3 - T_2)\ W \text{ thermal units.}$$

The pound of steam in falling from the condition of steam at T_1 to that of water at T_3 gives out—

$$T_1 + L - T_3 \text{ thermal units.}$$

Now the heat lost by the steam must equal that gained by the injection water. Hence we have—

$$(T_3 - T_2)W = T_1 + L - T_3.$$

$$\therefore \quad W = \frac{T_1 + L - T_3}{T_3 - T_2}.$$

The numerator of this fraction is the expression for the total heat of steam of the temperature T_1 over and above the heat contained in water at the temperature T_3, which (see p. 100)

$$= 885{,}200 + 235{\cdot}46\ (T_1 - 212^\circ) - 772\ (T_3 - 32) \text{ foot-lbs.}$$

Reducing and dividing by 772, so as to obtain thermal units, and substituting the result in the above formula for W, we get—

$$W = \frac{1114 + {\cdot}3T_1 - T_3}{T_3 - T_2}.$$

Example.

Find the amount of injection water required when the exhaust steam has a pressure of 19·5 lbs. absolute, the injection water a temperature of 60°, and the required temperature of the mixture 110°. The temperature T_1 of steam of the above pressure is 227°. Hence—

$$W = \frac{1114 + {\cdot}3 \times 227 - 110}{110 - 60} = 12{\cdot}4 \text{ lbs.}$$

Surface condensation.—In former times, when the pressure of steam rarely exceeded 35 lbs. per square inch, jet condensers were universally used for marine engines. The boilers were fed from the hot mixture of condensed steam and injection water, which contained nearly as large a per-centage of salt and other solid matters as the sea-water itself. The necessity of making marine engines more economical of fuel led to the abandonment of jet condensers at sea, and the substitution for them of surface condensers, in which the steam is condensed by contact with a cold metallic surface, all mixture of the condensed steam and the injection water being avoided.

There were two principal reasons for this change. The first was that when jet condensers were used the boilers could only be fed with salt water, which during the process of evaporation became constantly more and more saturated with salt, and which would, unless special measures were taken, eventually deposit its solid contents in large masses on the heating surface, and thus destroy the boiler, or render it useless. The only way to avoid this was from time to time to blow off large quantities of the brine in the boilers, and to supply its place with corresponding quantities of feed water. The hot water blown away from the boiler involved, of course, a large loss of heat, the amount of which depended on the state of saturation which the water was allowed to reach before blowing off. If the latter be allowed to reach three times the density of sea-water, the loss of heat in blowing out would be 7·4 per cent. of the total heat supplied to the boiler; and if less densities were permitted, the loss was considerably greater. The maximum density permissible was three times that of sea-water. The average loss of heat due to blowing off may fairly be set down as equal to from 12 to 15 per cent. of the total fuel supply. In order to save this loss it was necessary to feed the boilers with fresh water, and to use the condensed steam over and over again as feed water. This of course could only be effected by keeping the steam separate from the condensing water, and hence the introduction of surface condensers.

The second reason which led to the abandonment of jet condensers was that, in order to effect any considerable economy in the engine as distinguished from the boiler, it was necessary to resort to higher pressures of steam. Now if the temperature of the steam be raised above 280°, which is that due to 35 lbs. per square inch above the atmosphere, the sulphate of lime contained in the sea-water is deposited in hard and insoluble layers all over the boiler surface, and destroys the efficiency of the heating surface. Hence for this reason also the use of fresh water in the boilers, and consequently of surface condensation, became a necessity.

The amount of water required to condense the steam when surface condensers are used depends upon the efficiency of the cooling surface in abstracting the heat, and this again depends upon the thickness and conductivity of the metallic surfaces, their condition as to cleanliness, and the difference in temperature between the two sides.

In spite of the fact that the difference between the temperatures is much less than in the case of boilers, the efficiency of the cooling surface of the condenser in abstracting heat is far greater than that of the heating surface of the boiler in transmitting it. This is due in part to the fact that the condenser surfaces are much thinner than are the tubes and plates of a boiler, and they are also as a rule much cleaner. For these reasons condensers have usually less than half the surface found necessary for the boilers. Peclet found experimentally that sheet copper backed by water of the temperature of from 68° to 77° was capable of condensing 21 lbs. of steam per square foot per hour; while Joule, adopting special means, condensed as much as 100 lbs. per square foot in the same time; but in practice it is usual to allow one square foot for every 13 lbs. of steam of the terminal pressure usual in compound marine engines. And even with this large allowance of surface the amount of cooling water required is about forty per cent. greater than with jet condensers. The usual allowance is about 30 lbs. of water per pound of steam for vessels which run in the temperate zone, and about 35 lbs. for the tropics.

In order to realise the advantage due to maintaining the greatest possible difference of temperature between the two sides of the metallic surfaces, the cooling water must be kept in a constant state of circulation through the condenser by means of a special pump, which removes the water which has been warmed by the condensing steam from contact with the plates. As the air-pump of a surface condenser is only required to pump the condensed steam and air, it may be considerably smaller than the pump of a jet condenser; but in spite of this advantage surface condensers are much larger, heavier, and more costly than those in which the steam comes into direct contact with the cooling water.

Examples of condensers.—Fig. 187 gives a transverse and longitudinal section of a jet condenser, applied to the quick-running Allen stationary engines. The plunger of the pump A is worked direct by the engine piston-rod prolonged backward. The exhaust steam enters the condensing chamber B showed in transverse section, by means of the pipe S. It there meets with the jet of water D, which enters by the pipe E, and is condensed. Every time the plunger is with-

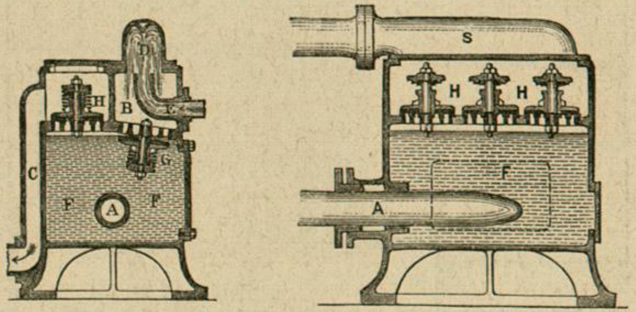

Fig. 187.

drawn, the pressure in the condensing chamber predominates over that in the pump chamber F, the valves, of which one is shown at G, open, and the mixed air, vapour, and water enter F. When the plunger returns, it displaces its own volume of water, the level of the water in F rises, and forces

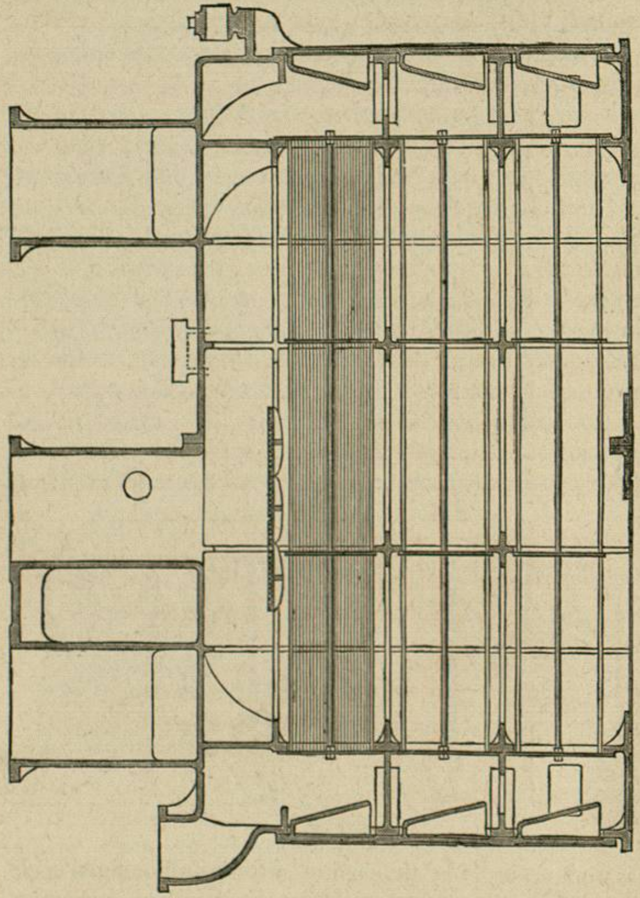

Fig. 188.

out the air and vapour, and a certain amount of water, through the valves HH, the hot water flowing away through the pipe C to a receptacle called the hot well. The valves G and H, which are made of india-rubber discs, are closed by spiral springs which exert a pressure equivalent to a quarter of a pound per square inch. With very quick-running engines these springs are found preferable to so arranging the valves that they may close by their own weight. It will be noticed that the valves G G are seated on a slope.

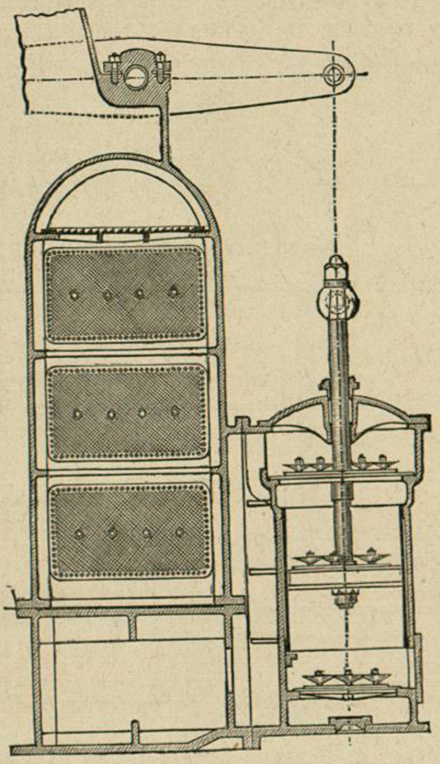

Fig. 189.

This is to allow the air and vapour as they come through to escape easily to the surface of the water in F, and so avoid the possibility of the plunger working in a mixture of air and water, which would injure its efficiency. The surface of the water in this type of condenser is the real air-pump, for it is by its rising and falling that the air is expelled and admitted. The velocity with which the surface rises and falls is, on the average, about 30 feet a second, while the velocity of the plunger is 800 feet.

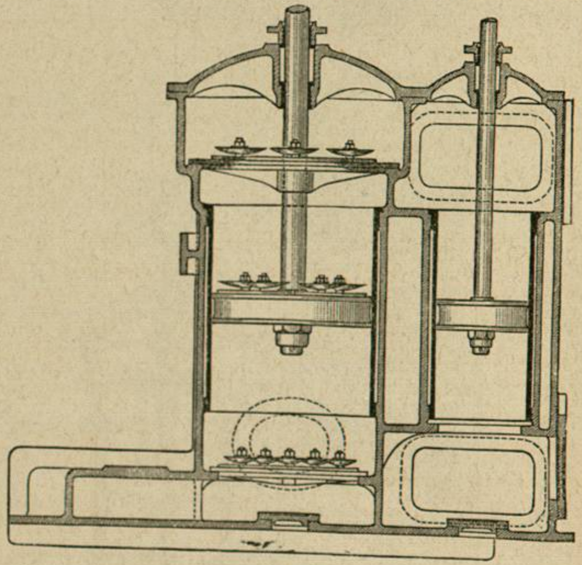

Fig. 190.

Figs. 188, 189 illustrate longitudinal and transverse sections of a modern marine surface condenser for a compound engine capable of working up to 3430 horse-power. It contains 3402 brass tubes of $\frac{3}{4}$ inch diameter external and 15 feet long. The total cooling surface is 10,018 square feet, being at the rate of a square foot of surface to ·342 horse-power. The tubes are arranged in three horizontal nests, and in them

the cooling water circulates, entering the bottom and being discharged after passing through the top nest. The steam enters at the top, passes round the outside of the tubes, and is distributed evenly by means of the perforated plates, of which some are shown in section above the top tubes in fig. 188.

The air-pump which is shown in section in fig. 189 is worked by a lever from the cross head of the main engine. It is of the single-acting vertical type, and is similar in principle to the ordinary lift-pump. When the piston or bucket ascends, it draws the condensed steam, air, and vapour through the lower or foot valves, and at the same time lifts whatever has passed through the piston-valves on the down stroke through the head valves shown at the top of the pump, after passing which the water flows away to the hot well. There are two of these air-pumps to the condenser in question, each having a stroke of 3 feet, a diameter of 34 inches, and a combined discharging capacity of about $\frac{1}{21}$ of the volume of the low-pressure cylinder. The barrels are of gun-metal, and the valves are indiarubber discs of the type shown in fig. 191, where the full lines represent the disc, above which is a curved metal guard plate which prevents the valve rising too high, and by its shape ensures a quick return of the valve to its seat, when the pressure which causes it to open is removed.

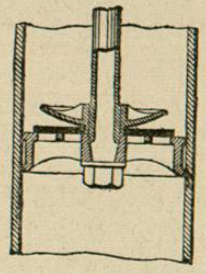

Fig. 191.

Fig. 190 shows another section of the air-pump, together with the circulating pump which forces the cold water through the tubes. The two are cast together in one piece. The circulating pump is a double action force-pump of 3 feet stroke and 20 inches diameter. When the engines are making 55 revolutions, it is capable of discharging nearly 9 lbs. of water per square foot of cooling surface per minute.

The circulating, like the air pump, is worked by a lever from the cross head of the main engine. Very often centrifugal pumps are used for circulating the water in surface condensers, and not infrequently they are driven by a separate engine.

Condenser tubes are almost invariably made of brass, which is sometimes tinned on both surfaces. Copper, though a better conductor, is never used, as the fatty acids formed in the condenser from the lubricating materials carried over by the steam from the cylinders attack the metal and form salts of copper, which, becoming dissolved in the condensed steam, are carried back into the boiler where they act most injuriously on the iron plates.

The packing of the ends of the tubes, so as to make a steam-tight joint, is a troublesome and expensive operation. The method in most general use is shown in fig. 192. In the thickness of the tube plates small stuffing boxes are formed, the tape packings at the bottom of which are tightened up by means of screwed ferrules. When the tubes are set vertically, the bottom ferrule is flanged so as to overlap the end of the tube to prevent the latter from dropping out, should the packing become loose.

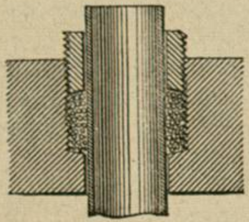

Fig. 192

Air pumps.—Nothing is more important in a condenser than the design of the air-pump. If the condenser is of the old-fashioned type the pump has to discharge not only the condensed steam, and the condensing water, but also the large amount of air which is always present in sea water, and which of course expands in volume when raised to the temperature of the condenser. It may be stated that, on an average, when the cooling water has a temperature of 60° and the condenser 120°, the discharging capacity of the air-pump should be from thirty-six to forty times the volume of the water into which the steam condenses. Hence its theoretical capacity may be calculated when we know this latter quantity, and the number of revolutions made by the engine per minute, and also the nature of the pump, whether double or single acting. The actual dimensions are determined when we know the efficiency of the pump—that is to say, the ratio which its actual bears to its theoretical discharging capacity. Vertical single acting air-pumps are by far the most efficient. They are almost always used with vertical engines, but when the latter are of the horizontal type, the use of a double-acting horizontal pump is often unavoidable. The actual efficiency of a single-acting vertical pump is, in the most favourable circumstances, about 60 per cent. of the theoretical, but should not in general be taken as more than 50 per cent. The average efficiency of horizontal double-acting pumps is about 35 per cent., or in other words the actual size of such a pump should be about three times greater than is theoretically necessary.

In surface condensers the pumps have only to discharge the condensed steam, and any small quantity of air which comes over by leakage ; but as surface condensers are generally arranged to act with a jet in case of necessity, it is usual to make the pumps much larger than is ordinarily necessary, though not so large as if jet injection were the rule.

Ejector condensers.—Fig. 193 illustrates a type of condenser which has no air-pump nor other moving parts. It is similar in principle to the injector described on page 409. Water having a pressure due to a few feet of head enters by the pipe A and flows through the nozzle B. Exhaust steam from one cylinder enters by the pipe C at a velocity varying with its pressure. If the pressure is 5 lbs. absolute, the velocity of the steam entering a vacuum of 25 inches of mercury is about 1200 feet per second. The entering steam surrounds the nozzle B, and is condensed on coming in contact with the cold jet issuing from B, and increases ten energy of the stream, just as the boiler steam imparts energy to the feed water of an ordinary injector. When the condenser is used with a two-cylinder engine, the exhaust steam from the second cylinder is arranged to enter by the pipe M, and is condensed by the combined jet issuing from the nozzle F. The steam from the second cylinder further increases the energy of the jet which passes through the nozzle N and into the trumpet-shaped pipe P, where its velocity is gradually reduced. With a steam pressure equal to 14·7 lbs. per square inch absolute at the commencement of exhaust, and a double-cylinder engine, the energy imparted to the jet is sufficient to raise the discharged water to a height of 6 or 8 feet above the condenser; or, if there is no head of condensing water to start with, the apparatus can raise the water from a level of 6 to 8 feet below the condenser. In the latter cases, however, the apparatus must be started by means of a jet of boiler steam, introduced through the pipe D and down the hollow spindle E. The rate at which the condensing water flows to the nozzle B is regulated by shifting the spindle E up or down by means of a handwheel and screw.

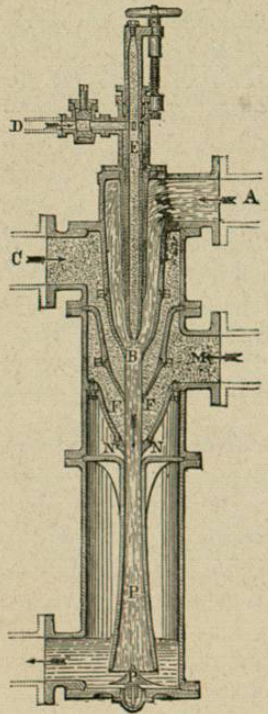

Fig. 193.

By means of these ejector condensers a perfectly steady vacuum can be maintained of 24 to 25 inches of mercury. The initial temperature of the cooling water should not, in general, exceed 60°, and the quantity of water supplied should be such that its temperature, on issuing from the condenser, would not be raised more than 20° to 25°. Experiments have however been made in which the rise of temperature was 64° and the final temperature 120°; but the vacuum at this temperature fell about two pounds per square inch. It must be borne in mind that, with condensers of this type, the whole of the power usually employed in driving the air-pump is saved, the energy of the exhaust steam being alone sufficient for the purpose of discharging the condensing water and the condensed steam.

Vacuum indications.—The amount of vacuum is indicated by a Bourdon pressure gauge, graduated in inches of mercury. It does not show the actual pressure in the condenser, but the difference between that pressure and the external atmosphere. For instance, if the barometer stood at 30 inches, and the indication of the vacuum gauge were 26 inches, the actual pressure would be $30-26=4$ inches of mercury. If the pressure in the condenser remained constant, and the outside barometer varied, the indications of the vacuum gauge would vary correspondingly.

In order to know if the vacuum is good or bad, notice must always be taken of the state of the barometer. The maximum attainable vacuum depends on the temperature at which the condenser is maintained. Thus if the latter be 120°, the pressure of vapour formed at this temperature is 1·68 lb. per square inch; then if the barometer stand at 30 inches of mercury, or 14·7 lbs. per square inch, the maximum vacuum attainable will be $14{\cdot}7-1{\cdot}68=12{\cdot}02$ lbs. per square inch.