A FEW NOTES ON THE AIR PUMP.

ABOUT the middle of last year I endeavoured to hunt up records showing the actual duty performed by the air pumps of condensing engines, and especially with reference to those fitted with surface condensers. I was surprised at what appeared to me to be a superficial treatment of the subject in those text-books to which I had access, and the difficulty I experienced in obtaining the necessary data from the records in engineering journals. I, therefore, venture to bring before you the substance of a few notes that I made on matters connected with this subject, in the hope that some of my remarks may be sufficiently unorthodox to stimulate a good discussion, and, perhaps, induce some of those who have air pumps under their charge to ascertain what duty they actually perform, and their efficiency as air displacers. The air pump has been developed along two totally different lines by two totally different classes of investigators, but in each case the object to be attained is the same, viz., to produce the lowest possible absolute pressure in the vessel being exhausted consistent with the practical limitations imposed by the conditions under which the vacuum has to be formed or maintained. The physicist very early found that valves between the pump barrel and vessel being exhausted were fatal to the production of a good vacuum unless the valves were mechanically operated. He also found that the presence in the pump of water, or other liquids, which have an appreciable vapour tension at ordinary temperatures was equally fatal to good results. The laboratory air pump has been brought to its present well known state of perfection by carefully attending to the scientific principles of its operation. Whilst the air, or other gas, is passing from the vessel being exhausted to the displacement chamber of the pump, the passage between the two is totally unobstructed, and every precaution is taken to prevent water, or other condensible vapours, from entering the pump along with the gas pumped, so that practically the pump has to deal with nothing but non-condensible gases. [By non-condensible I mean those which do not liquify under ordinary conditions.] In the laboratory, piston and jet pumps are used when low vacua only are required; special piston pumps may be used for the production of moderately high vacua, but mercury pumps are almost always employed when the highest vacua are required. Water vapour is prevented from entering the pump when high vacua are required by causing the gas being pumped to pass through a drying tube containing phosphoric anhydride before it enters the displacement chamber or the pump. Mercury pumps are usually constructed either on the principle of
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the Sprengel or Geissler pumps, modified to meet the special requirements or ideas of those using them. The Geissler pump is the only laboratory pump to which I wish to direct your attention this evening, and I do so because the Geissler cycle has been adopted in the Edwards' air pump, which has recently attracted some attention amongst marine and electrical engineers in England. The Geissler pump is essentially a plunger pump having mechanically operated trap valves, and a fluid plunger to enable it to effect perfect displacement. It usually consists of a vertical displacement chamber, having a branched opening at the bottom, and a cock, valve, or trap pipe opening at the top. A mercury reservoir is connected to the bottom opening by means of a flexible pipe, and from the branch another pipe is carried above the head of the pump, and is attached to the vessel to be exhausted. The cycle of this pump is as follows:—The reservoir containing mercury is raised above the head of the pump, and as the mercury flows into the displacement chamber, it enters and seals the branch pipe leading to the vessel being exhausted; and continuing to rise, expels the gas from the displacement chamber through the opening, or trap at the head of the pump. As the mercury reservoir is again lowered the trap, or valve, at the head of the pump closes, and cuts off communication with the atmosphere; the mercury level continues to fall, leaving a Torricellian vacuum above its surface, until that surface falls below the opening of the branch pipe, when the gas in the vessel being exhausted distributes itself by expansion, and fills the displacement chamber of the pump, though to a less pressure than previously. The mercury reservoir is then raised, and the above cycle repeated until the requisite degree of exhaustion is obtained. Should water vapour, or other condensible gas, be present, it is evident that the lowest absolute pressure which the pump can produce will be dependent on the vapour tension of the substance, corresponding with the temperature of the pump. Although the engineer has the same end in view, he cannot, for obvious reasons, employ exactly the same means as the physicist. In the first place, the engineer has to deal with comparatively enormous volumes of air, almost always associated with still large volumes of water vapour, and often with considerable volumes of water at a fairly high temperature. The partial pressure due to air in the condenser should be kept as low as possible, not only to relieve the back pressure, but also because the presence of air diminishes the efficiency of the cooling surface of the condenser, the steam having to reach that surface partly by diffusion through the air, instead of by direct flow, as it should. Figure I., curves Nos. I. and II., show for constant condenser pressures of 2lbs. and 4lbs. abs. respectively, or say for a 26in. or 22in. vacuum, the relation between the temperature of the condenser, and the ratio, which the partial pressure due to air bears to the total condenser pressure. It will be seen from these curves that the proportion of air present, increases at a slightly decreasing rate, as the temperature of the condenser falls below that at which the vapour tension of water is equal to the condenser pressure. It will also be seen that a comparatively small fall of the condenser temperature below that at which the vapour tension of the water equals the condenser pressure will account for a very considerable proportion of air present. It will be observed that under the above mentioned conditions, if the temperature of the condenser is about
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4 deg. Fah. below that corresponding with the pressure, then the partial pressure due to air will be about 10 per cent. of the absolute pressure in the condenser. Hence if we desire to determine the ratio of the air pressure to the absolute pressure in the condenser from readings of the vacuum gauge, barometer, and thermometer, indicating the condenser temperature, we must adopt methods of determining these quantities with considerable accuracy. A temperature error of \( \frac{1}{2} \) deg. Fah., either due to inaccurate graduation of the thermometer, or its location in an unsuitable part of the condenser, will produce an error of a little more than unity in a determination of the percentage of air present, and if the air present does not exceed 5 per cent. of the total volume it will be seen that an error of \( \frac{1}{2} \) deg. in the temperature means an error of over 20 per cent. in a determination of the volume, or partial pressure of the air present. The vacuum gauge and barometer, from whose indications the absolute pressure in the condenser is determined, must be sufficiently accurate to ensure that the difference in their readings which is the quantity required can be depended on to within a small part of itself. The quantity of air
which the air pump has to deal with depends on a variety of circumstances such as the volume of air and other non-condensible gases in the feed-water which enters the boilers and finally reaches the air-pump unless proper precautions are taken to remove it, the quantity of matter, which on entering the the boiler with the feed-water, is capable of being decomposed and giving off permanent gases, the extent to which leaks may occur at glands and joints in the low pressure portion of the plant; and if injectors be used to feed the boilers, the care or otherwise that is exercised to see that there is a slight overflow, and that the injector is not pumping air into the boilers. In Mr. A. E. Seaton's "Manual of Marine Engineering," 13th edition, page 245, he gives for an engine using 3 cubic feet of water per minute the theoretical volume to be pumped from the jet condenser as 114 cubic feet per minute when the absolute condenser pressure is 2lbs., the temperature 120 deg. Fah., the temperature of the injection water 60 deg. Fah., and the volume of the injection water 26 times that condensed. Again from page 247, taking 0.5 as the efficiency of the pump, its swept volume must in practice be 228 cubic feet per minute, from which we get 

\[
\text{As the total volume of water passing through the pump per minute is 81 cubic feet, the available swept volume for air, water vapour, &c., is } 228 - \frac{147}{187.5} = 1.22 \text{ cubic feet as the swept volume per lb. of steam condensed.}
\]

As the total volume of water passing through the pump per minute is 81 cubic feet, the available swept volume for air, water vapour, &c., is 228 - \[\frac{147}{187.5} \times 81 = 147 \text{ cubic feet per minute, or } \frac{147}{187.5} = 0.78 \text{ cubic feet per lb. of steam condensed.}\]

The vapour tension corresponding with 120 deg. Fah. is 1.683lbs., which, deducted from the 2lbs. abs. condenser pressure, leaves 0.317 lbs as the partial pressure due to air, and therefore each cubic foot of the mixture of air and water vapour which leaves the condenser contains only \(\frac{1}{6}\) by volume of a cubic foot of air if measured at the condenser pressure of 2lbs. abs. The useful work which the air pump performs is the abstraction of the air and other permanent gases, and in most cases also the condensed steam from the condenser. The quantity of water vapour pumped which is in reality almost wholly liquified during the compression and discharge of the air, does not materially assist in producing or maintaining the vacuum, and its presence in the air pump diminishes its effective capacity in the ratio which the partial pressure due to air bears to the partial pressure of the water vapour. The above statement is, I am aware, diametrically opposed to the opinions of some of the leading authorities on the subject. In speaking of the duty of the air pump, Professor Jamieson says:—"And in the case of a surface condenser to free the same of the condensed water as well as the air and the vapour which have been set free." And further on he adds: "Besides this air, there would, of course, be an accumulation of vapour or low pressure steam in the condenser from the condensed water under the action of a partial vacuum, and this vapour would soon fill the condenser and also spoil the vacuum if it were not extracted by the air pump." I cannot follow Professor Jamieson in this, for he seems to imply that the steam entering the condenser is first condensed, and the vapour subsequently given off again under the action of the vacuum. Surely the vapour tension cannot at any time be less than that corresponding with the temperature of the condenser at the particular
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Point considered. If the condenser were shut off from the engine, and its temperature maintained constant by some external course of heat, the operation of the air pump could not possibly reduce the pressure below that due to the vapour tension of water at the given temperature, nor would the cessation of the air pump's action cause the pressure to rise, provided that the temperature is maintained constant. The operation of the air pump can only effect a reduction in the pressure in the condenser when the whole of the water, which may lie in the condenser, has evaporated. Again, if we regard the air pump and condenser as being at the same temperature, and assume that the non-condensible gases have been exhausted, and nothing but water and water vapour remain, the air pump will cease to pump water vapour from the condenser; for the temperatures, being the same in both pump and condenser, the vapour tension will be the same both above and below the bucket valves, or foot valves, and there will be no difference of pressure to operate them. Under the conditions of practical working, the condenser soon attains its state of steady temperature, when the absolute pressure in the condenser will be the sum of the partial pressures due to air, and water vapour corresponding with the temperature. In the above I have not considered the differences of temperature which exist in, or the differences of pressure which determine the flow of steam to different parts of the condenser. Neither have I considered the fact that the composition of the condenser contents varies from place to place, the ratio of water vapour to air being greatest at the steam inlet; and least near the condenser outlet, or in stagnant corners. These small differences, which are essential to the action of the condenser, do not affect the question as to whether an accumulation of water vapour can take place, and so spoil the vacuum, as stated by Professor Jamieson. That water vapour must be removed from the condenser with the air is a physical necessity, depending on the fact that the air and water vapour are mixed. But it must not be inferred from this that the removal of the water vapour in itself materially improves the vacuum, or that if the air only could be abstracted by the air pump that water vapour would accumulate and spoil the vacuum. This latter can only take place when the temperature of the condenser is permitted to rise, by increasing the steam supply, reducing the volume or increasing the temperature of the circulating water, or diminishing the effectiveness of the cooling surface by permitting air to accumulate in the condenser, or otherwise. The actual work to be done, in removing the air from the condenser to the atmosphere, is in most cases very small. Take the case of an ordinary air pump; when it commences to operate the condenser is at atmospheric pressure, and neglecting the weight of valves and inertia of the air, no work is expended in removing the air. Again, if the absolute pressure in the condenser is exactly that due to the vapour tension of water at the condenser temperature, there cannot be any air present, and consequently no work is spent in pumping air. Somewhere between these two limits of air pressure in the condenser, the condition of maximum work must lie. In Fig. 2 two curves are given showing the work performed in removing one cubic foot of air, from a chamber at variable pressure $P_x$ lbs. per square inch to the atmosphere or other place at a constant pressure $P_o$ lbs. per sq. inch. Curve No. 1 is drawn on the assumption that compression
takes place isothermally, and Curve No. 2 assuming adiabatic compression. In both cases the discharge is assumed to take place at 15 lbs. abs. It will be seen from these curves that with isothermal compression the point of maximum work corresponds with an absolute pressure of about 5.5 lbs. per square inch, and that when the compression is adiabatic this point corresponds with about 4.5 lbs. per square inch abs. It will also be noticed from these curves that when the vacuum is very poor the work done per cubic foot of air expelled is not materially different whether we assume isothermal or adiabatic compression. But as the vacuum improves the difference becomes greater, and when the absolute pressure is a little less than 2 lbs. per square inch, 50 per cent. more work is required, and if the pressure be 0.2 lbs. per square inch, 100 per cent. more is required when compression takes place adiabatically than when it takes place isothermally. From this it appears that air pumps, like other air compressors, and for the same reason, should not run at a high speed if they are to work economically unless some special provision be made for quickly abstracting the heat from the gases or vapours compressed. It does not follow that in practice air pumps as actually constructed will be capable of pumping air at so small an expenditure of energy per cubic foot of air as shown by these curves (even though friction and such losses be neglected). The ordinary vertical type of single-acting air pump, having foot, bucket and head valves, is almost always so constructed that the air passing from the condenser to the pump is trapped, and has to break through a water seal, in addition to having to lift the foot valves. If the condenser is well arranged, and the air pump properly located, the temperature of the water passing through the pump will be very little below the average temperature of the condenser, and therefore the vapour tension on both sides of the air pump, foot and bucket valves will be almost exactly the same as in the condenser, and practically the only pressure available to open the foot valves will be the partial pressure due to air in the condenser. It will be readily seen that under these circumstances when the bucket is at the top of its stroke the absolute pressure between the bucket and foot valves will be less than that in the condenser by the amount required to raise the foot valves and break through the water seal, if any; and that this difference of pressure must be almost wholly due to air in the condenser. As the pressure available for lifting the foot valves is approximately the difference of the partial pressures due to air in the condenser and pump barrel respectively, it is evident that the air pump really has to pump from a chamber having the average absolute pressure existing between its bucket and foot valves and not from a chamber at condenser pressure. If we assume that the partial pressure due to air in the condenser is 0.2 lbs., and that its partial pressure below the pump bucket is 0.1 lbs., which leaves very little more than 0.1 lbs. per square inch to operate the foot valves and break through the water seal, we see that for each cubic foot of air removed from the condenser at 0.2 lbs., the air pump has to deal with two cubic feet of air at 0.1 lbs. pressure. That is if 0.1 lbs. per square inch pressure is required to lift the foot valves and break the water seal, when the total air pressure is 0.2 lbs.; the discharging capacity of the pump is reduced by the action of the foot valves and seal to 50 per cent. of the value which it should have, if these defective
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arrangements did not exist. Again, more work is required to be expended to remove the two cubic feet of air at 0.1 lbs. than the one cubic foot at 0.2 lbs. pressure, even without taking into consideration the frictional losses in operating a pump having double the capacity. To make this type of pump reasonably effective the foot valves should be placed above the highest point to which water can rise below the bucket, and should be so arranged that the air can enter the pump barrel the moment the bucket starts to rise without having to bubble through the water, which should enter the pump in a stream below the air, and the foot valves should be mechanically operated, so that there is an absolutely clear passage between the condenser and pump barrel. The difference in the action of the various types of pumps can be most readily seen by considering their cycles. I shall now only refer to three types of pumps, viz.: the common vertical single acting pump, having foot, bucket and head valves; the Edwards pump, having head valves and a mechanically operated foot valve, which also serves the purpose of a bucket valve; and the jet air pump. In all cases I will assume that the pump has reached the steady state of working, and that the discharge takes place at atmospheric pressure.

COMMON VERTICAL SINGLE ACTING PUMP.—When the bucket is at the top of its stroke the pressure on its upper side will be that due to the atmosphere, and on its lower side a pressure less than that in the condenser by the amount which is required to operate the foot valves, and break through the water seal. On the down stroke, the pressure above the bucket should at once fall to that corresponding with the vapour tension of the water, but as some air is always present in the water, and therefore is not discharged during the up stroke, the pressure will fall gradually above the bucket, and rise gradually below it until the difference of pressure is sufficient to operate the bucket valves; the bucket continues to descend to the bottom of its stroke, when the whole of the air (for the steady state is reached) taken in through the foot valves during the up stroke is transferred to the top of the bucket, and during the up stroke which completes the cycle, the pressure above the bucket rises until it reaches atmospheric, when a quantity of air equal to that which passed through the bucket valves will be discharged. During this stroke the pressure below the bucket will have fallen gradually until the foot valves open, and the pressure below the bucket rises to its initial value.

EDWARDS' AIR PUMP, Fig. III.—As this pump has but comparatively recently come into prominence, it may not be out of place if I briefly describe it before proceeding to indicate its cycle. This is a very interesting pump, not only because in its operation the Geisseler cycle has been adopted, but also because of the very successful results which have been obtained by it, notwithstanding some theoretical defects in its design. The pump is a single acting vertical one, having a mechanically operated admission valve, and the usual discharge, or head valves, operated by the pressure of the fluid being pumped. The head valves are placed as usual on the cover of the pump, and for our present purpose require no further comment. The inlet valve is of the piston type, the plunger serving the double purpose of bucket and valve. The ports are arranged as follows:—A row of openings are made near the bottom end.
of the pump liner, and an annular space is also left between the end of the liner and bottom of the pump. The bottom of the pump is dished, and

An annular chamber surrounds the lower end of the pump, and forms the communication between the pump barrel and eduction pipe of the condenser.

The dished surface of the pump bottom is continued upwards and inwards by means of a lip, which enters the external annular chamber and is finally directed towards the centre of the upper row of ports. When the pump is in action the water of condensation enters the pump barrel through the annular space below the liner and collects in the dished bottom of the pump. As the plunger descends a vacuum approximately corresponding with the temperature of the pump is produced above it, till, in its descent, the upper row of ports is opened, and absolutely free unobstructed communication is established between the condenser and displacement side of the pump piston. As the piston continues to descend its lower conical surface strikes the water accumulated in the bottom of the pump and drives it along the curved surface of the lip, so that it enters the pump barrel in a thin sheet through the upper
ports without obstructing them for the passage of air.

The cycle of the Edwards' Pump is as follows:—As the piston makes the down stroke the indicator pencil would describe the line a, b, c. (In practice there may be a considerable curve at b, due to imperfect displacement). The part b, c, will be approximately parallel to the line of zero pressure, but above it, at the distance $P_w$, which represents the absolute pressure due to the tension of the water vapour. When the the ports open the pressure above the plunger will quickly rise to that in the condenser, or o, d, in the diagram, which is equal to $P_w + P$, or the rise of pressure in the pump due to the opening of the ports is $P_a$, the absolute pressure due to air only. On the up stroke the curve d, c, f, a is described, which completes the cycle and shows that compression and discharge takes place precisely as in the ordinary type of pump. The difference between the cycle of this pump and the common pump is that the absolute pressure above the plunger in the Edwards pump falls to that corresponding with the vapour tension of the water, whereas in the common pump it only falls until the difference of pressure above and below the valves is sufficient to operate them. The effect of this difference in the cycles is apparent if we assume that the two pumps are operated with their suction openings directly communicating with the atmosphere. If the common pump be worked under these conditions we shall only require to expend energy to overcome the pump friction, weight of the valves, and inertia of the air, and the work spent on pumping air is practically negligible. This is not the case, however, with the Edwards pump, for on the down stroke the difference of pressure on the two sides of the plunger is practically

the same as atmospheric pressure, or say 14.5 lbs. per square inch, but when the ports open the air rushes in, and the up stroke is completed under the same conditions as obtain with the ordinary pump, but the work spent in producing a vacuum above the plunger on the down stroke is not returned on the up stroke. Thus we see that the maximum work has to be expended to operate this pump when it is performing the minimum useful work, and the work lost per cycle will depend on the absolute pressure of the air in the vessel being exhausted, and will become less as that pressure falls towards zero. If the pump is working on a surface condenser, and the distance between them is so small that no material fall of temperature
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takes place in the water of condensation between the condenser and pump, it is apparent that the vapour tension of the water above the pump plunger and in the condenser will cancel, so that the action of the pump is the same as it would be if exhausting from a vessel containing air only at an absolute pressure equal to the partial pressure of the air in the condenser. The theoretical efficiency of the Edwards pump can be obtained as follows: on the assumption that compression takes place isothermally, which is the most favourable condition for the pump:

Let $P_b$ represent the pressure in the pump barrel at the moment that the ports close.
Let $P_w$ represent the pressure in the pump barrel during discharge.
Let $P_a$ represent the pressure due to water vapour only.
Let $P_e$ represent the pressure due to air only in the pump barrel when the ports close.

All pressures are in lbs. per square foot. The area of the plunger is taken as 1 square foot, and the stroke as 1 foot.

**on the down stroke**—
The pressure above the piston is $P_w$.

" below " $P_w + P_a$.
The work done on the pump is $P_w - P_w + P_a$.

" " $P_a$ foot lbs.

**on the up stroke**—
The pressure below the piston is $P_w + P_a$.

" above " $P_w + (\text{mean air pressure})$.
The mean air pressure is $P_a \left(1 + \log \frac{P_e}{P_a}\right)$.
The work done on the pump is

$$
(P_w + P_a + P_a \log \frac{P_c}{P_a} - P_w - P_a)
$$

foot lbs.

$$
= P_a \log \frac{P_c}{P_a} \text{ foot lbs.}
$$

Therefore, the work done per cycle is

$$
P_a \left(1 + \log \frac{P_c}{P_a}\right) \text{ foot lbs.}
$$

But the work which is theoretically required to take one cubic foot of air from a vessel where the pressure is $P_a$ lbs. per square foot, to a vessel where the pressure is $P_c$ lbs per square foot, is

$$
P_a \log \frac{P_c}{P_a} \text{ foot lbs.}
$$

The efficiency of the pump is therefore

$$
\frac{P_a \log \frac{P_c}{P_a}}{P_a \left(1 + \log \frac{P_c}{P_a}\right)}
$$

From this we see that when $\log \frac{P_c}{P_a} = 1$, the efficiency of the pump is
only 50 per cent., and that is when \( \frac{P_c}{P_a} = 2.718 \), or for a pump discharging to atmosphere \( P_a \) would lie between 5 and 6 lbs. abs. As \( P_a \) diminishes the efficiency increases and approaches 100 per cent. as \( P_a \) approaches zero.

<table>
<thead>
<tr>
<th>Values of ( \frac{P_c}{P_a} )</th>
<th>Efficiency in per cent.</th>
</tr>
</thead>
<tbody>
<tr>
<td>60</td>
<td>80 per cent.</td>
</tr>
<tr>
<td>40</td>
<td>78</td>
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<tr>
<td>30</td>
<td>77</td>
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<tr>
<td>20</td>
<td>75</td>
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<tr>
<td>15</td>
<td>78</td>
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<tr>
<td>10</td>
<td>70</td>
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<td>5</td>
<td>61</td>
</tr>
<tr>
<td>3</td>
<td>52</td>
</tr>
<tr>
<td>2</td>
<td>41</td>
</tr>
<tr>
<td>1.5</td>
<td>29</td>
</tr>
<tr>
<td>1.0</td>
<td>0</td>
</tr>
</tbody>
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Figures VI. and VII. are taken from "The Electrician," of London, 17th March, 1899, and refer to the results obtained from the tests of a three cylinder air pump, the pistons being 12" diameter and the stroke 12". The curves in Fig. VII. are for a constant speed of 150 revolutions of the crank shaft per minute. The influence of the peculiar cycle employed in this pump is rendered apparent in diagrams Nos. II. and III., where a large rise of pressure can be seen to take place when the ports open and establish communication between the pump and condenser. The small size of the diagrams, and the uncertainty as to the law of compression at such a high speed, prevent a determination of the ratio of air to water vapour present in this pump barrel. In Fig. VII. if any two equal ordinates be taken on the pump H.P. curve it will be seen that the corresponding E.H.P. ordinates are not equal, and this discrepancy amounts to about 10 per cent. in the ordinates for 6.25 H.P. The following data refer to the diagrams, Fig. VI.:

<table>
<thead>
<tr>
<th>No. of Diagram.</th>
<th>Revolutions</th>
<th>Vacuum.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Stated. From diagram.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>H.P. From &quot;Engineering.&quot;</td>
</tr>
<tr>
<td>I.</td>
<td>140</td>
<td>29.25&quot; 26.7&quot;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>15.0&quot; 14.3&quot;</td>
</tr>
<tr>
<td>II.</td>
<td>138</td>
<td>4.8&quot;     3.0&quot;</td>
</tr>
<tr>
<td>III.</td>
<td>158</td>
<td></td>
</tr>
</tbody>
</table>

The good results obtained with this type of pump (as reported in the technical journals) are, no doubt, due in part to the small clearance, and more perfect displacement rendered practicable by the absence of bucket valves; but I think the main factor is the mechanically operated foot-valve, which gives an unobstructed passage from the condenser to the pump barrel, and permits of the displacement being practically the swept volume. At high piston speeds—and especially with long eduction pipes—we might anticipate that the very short time during which the ports
are open, together with the fact that the down stroke of the plunger has just produced a retrograde movement of the air and vapour in the pipe, would prevent the pump from receiving a full charge, and that its effective displacement would be interfered with. This, however, does not appear to be the case, for although the time during which the ports are open is very short, the mass of matter to be moved is small, and the force available to move it, though small, is still capable of producing a considerable velocity in that time. To take a rough example, we may assume that the absolute pressure in the condenser is 2lb. (or, say, 26" vacuum), the partial pressure due to the air, say, 0·21lb., and the length of the pipe 10ft. Then, for each square foot of the pipes cross section, the weight of the 10ft. column of air and water vapour will be about 1·1 lbs. The difference of pressure existing between the condenser and barrel of the air pump will be about 0·21lb. per square inch as the ports open, and will diminish to zero as the pump barrel fills. If we further assume that the cross area of the pipe is 1/3 of the
pump plunger area, and that the stroke of the plunger is one foot, then, for each stroke of the plunger, if the pump barrel fills, the air and water vapour in the pipe must advance 5 feet.

We have now to determine the time which the above mentioned diminishing force will take to move the column of air and vapour in the pipe through 5ft.

Let \( L \) represent that length of the pipe, whose volume is equal to the displacement of the pump per stroke.

\( x \) be any other length.
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Let \( l \) represent the total length of pipe between the condenser and pump.

\( F \) represent the available force at the commencement of motion in poundals per lb. of matter in the \( l \) feet of pipe.

\( f \) represent the force after the gases have moved \((L-x)\) feet.

Then \( f = \frac{Fx}{L} \)

But \( f = \frac{dv}{dt} \) & \( v = \frac{dx}{dt} \)

\( \therefore \frac{f}{v} = \frac{dv}{dx} \) & \( fdx = vdv \)

\( \therefore \int fdx = \frac{v^2}{2} \)

Substituting the above value for \( f \) in this equation we get—

\( \int \frac{Fx}{L} dx = \frac{v^2}{2} \)

The value of which between the limits is—

\( \frac{F}{2} \left( L - \frac{c^2}{L} \right) = \frac{v^2}{2} \)

The velocity acquired in moving over the distance \((L-c)\) is therefore—

\( v = \sqrt{\frac{F}{L} \left( L - \frac{c^2}{L} \right)} \) ft. per second.

Again the velocity at \( c \) is \( v = \frac{dc}{dt} \)

\( \therefore dt = \frac{dc}{v} \)

\( \therefore t = \int \frac{dc}{v} \)

Substituting for \( v \) the value found above we get—

\( t = \int \frac{dc}{\sqrt{\frac{F}{L} \left( L - \frac{c^2}{L} \right)}} \)

The value of which between limits gives—

\[ t = \frac{\pi}{2} \sqrt{\frac{L}{F}} \] seconds.

As far as I can ascertain from illustrations, the ports of the Edward pump will be open for about \( \frac{1}{5} \) of a revolution of the crank shaft, and as the usual speed of moderate sized pumps appears to be about 150 revolutions per minute, the time that the ports are open will be

\[ \frac{60}{150 \times 5} = 0.08 \text{ sec.} \]
In the example taken we have

\[ F = 0.2 \times 144 \times 32 \times 17 \text{ lb. abs. per lb. of gas.} \]

\[ L = 5 \text{ ft.} \]

\[ t = \frac{3.14}{2 \sqrt{\frac{15667}{5}}} = 0.03 \text{ sec.} \]

In this we have only considered inertia, and that very imperfectly; friction and a number of other disturbing elements have been neglected. I think, however, the result indicates that inertia is not likely to interfere with the action of the pump unless the eduction pipe is very long or of small diameter. The question of the relative volumes of air and water vapour with which the air pump has to deal is a very important one, and especially so when the pump is required to maintain a good vacuum. Although the theory of the method of determining the ratio of the partial pressures due to air and water vapour respectively in the condenser is simple, in practice the nature of the quantities, and the conditions under which they may be determined, are not such as to render it easy to obtain satisfactory results. The three most obvious methods of determining this ratio are:

**First.**—To collect and measure the volume of air actually discharged by the pump, noting the absolute pressure in the condenser at the time, and the number of strokes made by the pump.

**Second.**—To take the absolute pressure and temperature at the condenser outlet. The partial pressure due to air being the difference between the total absolute pressure, and the vapour tension of water corresponding with the measured temperature.

**Third.**—From indicator diagrams taken from the air pump.

Each of these methods is open to objection. If the first is employed it is necessary to attach a separator of considerable volume to the air pump discharge, and, even then, an appreciable proportion of the air pumped may pass through the separator, either dissolved in the water, or carried through mechanically in form of minute bubbles. Again, if there are foot valves, or traps of any kind between the air pump and condenser, we should require to determine the absolute pressure below the pump bucket instead of in the condenser, and this would be much more difficult owing to fluctuations produced by the motion of the bucket. If the second method is employed, both the absolute pressure and the temperature must, as I have previously pointed out, be taken with considerable accuracy. The determination of the temperature is especially difficult, as small differences exist at different parts of the condenser, and even considerable differences may exist between the water and water vapour passing into the pump. If an excessive quantity of circulating water is employed, the water of condensation may be cooled considerably below the average temperature of the condenser, and errors in the determination of the temperature would probably mask the desired quantities altogether. In the third,
where the air pump works slowly it is probably the simplest and most reliable of the three. If the pump works sufficiently slowly to justify the assumption of isothermal compression, the partial pressure due to air can be determined as follows, whilst the partial pressure due to water vapour can be determined with a fair degree of accuracy from the temperature of the water discharged by the pump, or from the barometer reading and diagram. From the area of that part of the diagram produced by air determine the mean pressure in the usual manner. Then since the mean pressure is $P_a \left( \log \frac{P_e}{P_a} + 1 \right)$ we can, by taking different suitable values of $P_a$ and calculating the corresponding mean pressures, obtain three or four points of the curve, giving the mean pressure as a function of $P$.

From this curve we can obtain the initial pressure corresponding with mean pressure determined from the diagram. Before taking the area of the diagram, a vertical line should be drawn through it, at that point where the discharge of water commences, and the area contained between this vertical and the admission line, must be rejected. If the volume of water passing through the pump is not known, or was not observed when the diagrams were taken, it can be approximately determined thus:—

Through the lowest point of the admission curve, draw a line parallel to the atmospheric line, and from this draw and measure any three ordinates preferably not far removed from that part of the compression curve which makes an angle of 45 deg. with the atmospheric line, and measure the length of the discharge stroke yet to be completed in each case. Calling the ordinates $P_1$, $P_2$, $P_3$, and the corresponding abcissae $V_1$, $V_2$, $V_3$ respectively, the initial air pressure $A$, and the length of the stroke during
which water is discharged $W$ we then have

$$(P_1 + A) (V_1 - W) = (P_2 + A) (V_2 - W) = (P_3 + A) (V_3 - W)$$

from which we find

$$W = \frac{(P_2 - P_1) (V_2 - V_1) + (V_2 - V_1) (P_2 - P_1)}{P_1}$$

Diagrams Nos. I, II, III, Fig. V., are enlargements of diagrams taken from a dry air pump of a sugar refining vacuum pan. (The originals were kindly given to me by Mr. W. Fyvie.) The pump is 18in. diameter, 24in. stroke, double acting, and was running at about 45 revolutions per minute when the diagrams were taken. Table No. 1 gives the more important data referring to these diagrams.
NOTES ON THE AIR PUMP.

<table>
<thead>
<tr>
<th>No. of diagram Fig. V.</th>
<th>Mean pressure of air diagram below absolute</th>
<th>Lowest absolute pressure shown by</th>
<th>Partial Pressures Due to air, lbs. abs.</th>
<th>Due to water vapour, lbs. abs.</th>
<th>Temperature of vacuum pan, Fah.</th>
<th>Per cent. of air present</th>
<th>Condenser temperature from vapour tension, Fah.</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>0.30 lbs. abs. 1.59 lbs. abs.</td>
<td>0.062</td>
<td>131</td>
<td>140 deg.</td>
<td>5.9</td>
<td>101 deg.</td>
<td>124</td>
</tr>
<tr>
<td>II</td>
<td>1.65 &quot; &quot; 2.14 &quot;</td>
<td>0.22</td>
<td>1.92</td>
<td>160 &quot;</td>
<td>10.3</td>
<td>143 &quot;</td>
<td>119</td>
</tr>
<tr>
<td>III</td>
<td>0.43 &quot; 2.49 &quot;</td>
<td>0.284</td>
<td>2.11</td>
<td>170 &quot;</td>
<td>11.0</td>
<td>143 &quot;</td>
<td>102</td>
</tr>
</tbody>
</table>

If we assume that this pump had been working on an ordinary surface condenser of a steam engine, instead of on a vacuum pan, when the diagrams shown in Fig. V. were taken, the corresponding condenser temperatures can be determined from the partial pressures due to water vapour. These temperatures are given in the last column of Table No. 1. If now we assume that the temperature of the air pump were reduced to 75 deg. Fahr. and maintained at that temperature, whilst the absolute pressure and temperature in the condenser remained as before, we should have in the first case (1.39 lbs. - 0.427 lbs. due to water at 75 deg. Fahr.) = 0.96 lbs. due to air. That is the partial pressure due to air in the pump would then be 11.7 times as great as previously, and the capacity of the air pump could be reduced to \( \frac{1}{11.7} \) part of its original value without affecting its useful capacity as an air pump. In the third case we should have (3.49 - 0.427) = 3.06 lbs. due to air; therefore, the practical pressure due to air would be 3.06 = 8 times as great as when operated at the higher temperature, and the volume of the pump could be reduced to \( \frac{1}{8} \)th of its former value, or the stroke and diameter could both be reduced to half their original values without impairing the vacuum. From this it appears that the ordinary air pump is many times the necessary size to enable it at the same time to pump the water of condensation from the condenser to the hot well. As the gross ineffectiveness of the ordinary air pump is due to the fact that it is generally required to pump hot water from the condenser to the hot well, which latter is kept at as high a temperature as is consistent with a reasonable vacuum, it appears that if we desire to make the air pump effective as such, we must either reduce the temperature of the water leaving the condenser to a very much lower value than is usual, or else we must employ a separate pump to remove the hot water, and leave the air pump to perform its legitimate function at the lowest attainable temperature. The lowest temperature practically obtainable is that of the circulating water. It should therefore be our endeavour to maintain the temperature of the air pump and gases entering it as nearly as possible the same as that of the circulating water. This object may to some extent be attained by causing the circulating water to first flow round the air pump through a water jacket, and then through a small subsidiary condenser attached to the air pump inlet, thus reducing the temperature of the entering gases and vapour to the lowest practicable
limit. Of course, in this case, a separate pump would have to be employed to remove the hot water from the condenser. The subsidiary condenser has, I believe, been employed in a few cases with beneficial results, but I have been unable to obtain any definite data concerning them. It may, I think, be urged against this system with good reason that the subsidiary condenser and additional pump (even though it would not require to be much larger than the feed pump) is a more objectionable complication than the larger air pump. A much simpler arrangement by which a low air pump temperature could be maintained would be to let the circulating pump perform the additional service of air pump, arranging it much on the lines of the old jet condenser, and using the independent pump to abstract the hot water from the condenser and return it to the hot well. With such a system no further complications would be introduced, and the temperatures of the respective pumps would be maintained as nearly as possible in conformity with theoretical requirements, instead of as now, diametrically opposed thereto. The same waste of power in discharging the water against atmospheric pressure would take place, as occurs in ordinary jet condensers. The work done on the water entering the condenser or pump is not returnable when piston pumps are used. But if the ejector type of jet condenser is used the work done on the entering water by atmospheric pressure is at all events in part returned, and abnormal waste of power in discharging water cannot be urged against it. To this arrangement I will refer later on. Again looking at the diagrams, it will be seen that a loop is formed in No. 1. The compression curve falls below the admission line for a considerable distance, clearly showing that the pump in this case acts to some extent as a condenser. If the cold water which passes through this pump for the purpose of lubrication had been brought more intimately into contact with the air and vapour entering the pump, this condensation would have taken place before the completion of the admission stroke, and a larger volume of air would have entered the pump, thus increasing its effective capacity.

The Jet Pump.—Any one who has devoted a few minutes to the study of the “ejector condenser” cannot fail to have been struck with the beautiful simplicity of this piece of apparatus, which serves the dual purpose of condenser and air pump combined. It is smaller, lighter, less costly, and less liable to get out of order than any other type of condensing plant of equal capacity. It is, however, only in its secondary capacity, as an air pump, that I wish to direct your attention to it this evening. We have already noticed the application of the water jet in the laboratory air pump, and as is well known this type of pump is capable of producing a vacuum corresponding closely with the vapour tension of the water passing through it, provided that the water is not too strongly charged with permanent gases. The water pressure required to operate these pumps is not great: a pressure at the jet of less than 4 lbs. per square inch is capable of producing the best attainable vacuum at the existing temperature. The jet air pump possesses several important features which appear to render it specially suitable for use with the condenser. For, if the whole of the circulating water is caused to pass through the jet pump on its way to the surface condenser, we should fulfill the condition of maintaining the pump at the lowest possible temperature. The greater part of the water vapour
entering the jet pump would be immediately condensed on the ever fresh surface of cold water presented to it, and as an air pump it will only have to deal with the air, and the water vapour associated with it corresponding with the slightly raised temperature of the circulating water. A 100 I.H.P. engine using, say 18lbs. of steam per I.H.P. hour requires 1800lbs. of steam per hour. If we allow per lb. of steam condensed, say 25lbs. of circulating water, and one cubic foot displacement of the air pump; then if we take the efficiency of the air pump at 50 per cent., the effective displacement will be 900 cubic feet per hour. With a vacuum of 24ins., or say an absolute pressure of 3lbs., the 900 cubic feet would weigh 7·7lbs., (assuming no air present) and the number of heat units in this weight of low pressure steam would be roughly 8,400. But as we have $1800 \times 25 = 45,000$ lbs. of circulating water passing through the pump in the same time it will be seen that no appreciable rise of temperature will take place in it, to detract from its effectiveness in the surface condenser. Again it will be seen that the loss of 7·7lbs. of steam per hour per 100 I.H.P. is not a serious quantity, being less than $\frac{1}{3}$ per cent. This would of course have to be made good by the addition of fresh water to the hot well. In the above I have assumed that the hot water is drawn from the condenser by means of a separate piston-pump, and that the circulating pump is designed to force the water against 10lbs. extra back pressure to enable it to operate the jet pump with certainty. Again, with the jet pump, there are no valves, water traps, or other obstructions, between the jet and condenser; and as the pump operates continuously the air and water vapour flows through all pipes and passages at a constant velocity; a condition which minimises losses due to friction of pipes, etc. The very intimate mixture of air and water in the jet pump enables it to effect compression isothermally, and, therefore, with the least expenditure of power under this head. Further, the work done on the water flowing through the jet, by the atmospheric pressure, is not lost (as in the case of the piston pump previously mentioned), but increases the kinetic energy of the strain by an amount sufficient to enable it to discharge itself into the atmosphere again. On the other hand, the jet pump has the disadvantage of requiring a separate pump to be used to abstract the hot water from the condenser; and in common with other jet apparatus it is inefficient, owing to the loss of energy, in the formation of eddies, and due to internal friction. Whether the second disadvantage will condemn the jet pump or not depends almost wholly on the relative efficiencies of the jet and piston pumps, not forgetting the power required to operate the pump for abstracting the hot water from the condenser. If we take the case previously mentioned of a 100 I.H.P. engine and assume that the absolute pressure in the condenser is 2lbs., and that the partial pressure due to air is 0·2lbs. abs., then the air pump would only have to remove 900 cubic feet of air at 0·2lbs. per square inch from the condenser to the atmosphere at, say 15lbs. per square inch. If we assume compression to take place isothermally, which is the condition most favourable to the piston air pump, the actual work required to be done in the removal of air, exclusive of all friction of the pump is 111,456 foot lbs. per hour, i.e. about 0·056I.H.P., or 0·056 per cent. of the indicated power of the engine. The power required to discharge the condensed steam would be 0·027 I.H.P.
The total theoretical power required to abstract the air and condensed steam from the condenser is therefore 0.083 H.P., or 0.083 per cent. of the indicated power of the engine. As an ordinary air pump for such an engine would absorb at least 1 H.P., the efficiency, or ratio of useful work done to actual work expended, would in this case be 8.3 per cent., an efficiency which surely will not preclude the jet pump from competing, notwithstanding its defects and attendant water pump. If a jet pump be employed, operated by the circulating water on its way from the circulating pump to the condenser, the circulating pump will not have to be altered in size, but must be designed to deliver the water against an extra back pressure of, say, 10 lbs. per square inch. We may fairly assume that the motive power employed to drive either the piston air pump in the previous case, or the circulating pump in this case, costs the same per horse power hour. We therefore have to charge against the jet pump the extra power supplied to the circulating pump to enable it to discharge against the increased back pressure, due to the insertion of the jet pump in the circulating system. The additional power required to operate the circulating pump will therefore be

\[ \frac{1800 \times 25 \times 10 \times 144}{62.5 \times 60 \times 32,000} = 0.58 \text{ H.P.} \]

If we add 10 per cent to this for increased friction of pump glands and pistons, under the higher pressure, we get 0.588 H.P. The theoretical work required to pump the hot water from the condenser has already been found to be 0.027 H.P. As, however, small piston pumps cannot be expected to deliver much more than 50 per cent. of the power applied at piston rod as water horse power, we can take the actual power required to operate the hot water pump at 0.054 H.P., and the total power taken by the jet pump and hot water pump combined will be 0.637 H.P. instead of at least 1 H.P., which would be required by a piston air pump. We have just seen that the actual work required to exhaust the air is 0.056 H.P., and the work required to operate the jet, including extra losses in the circulating pump, is 0.53 H.P. The efficiency of the jet pump need not, therefore, exceed 11 per cent. From the fact that the ‘ejector condenser’ when supplied with water under a head of about 15 ft. is found to perform the double duty of condenser and air pump, it does not appear to be unreasonable to anticipate that it will perform either of these functions independently if supplied with water under the same head. If it was attached as an air pump to a surface condenser, and thus relieved of the duty of condensing the steam from the engine, the temperature of the jet would be as low as practicable, and certainly considerably lower than when used as an ejector condenser, and would thus be better able to deal with the air.

With a view to ascertaining whether the jet pump has an efficiency which would justify its use in conjunction with a surface condenser I tested a small one in which the water jet was about \( \frac{1}{3} \) of an inch in diameter. The method of testing was to measure, by means of a gas meter, the actual quantity of air that was drawn through the pump, a cock being inserted between the meter and pump so that the vacuum in the pump chamber could be controlled, whilst permitting the meter to measure the air at atmospheric pressure, and thus ensure that any errors due to leakage...
would reduce the apparent efficiency and tell against the pump. The vacuum in the pump chamber was measured by means of a mercury gauge supplied from the same reservoir as a barometer, so that the difference of the readings taken on the same scale gave the absolute pressure. The water pressure at the jet was taken by a good gauge which had been tested against a mercury column, and when low pressures were employed the mercury column itself was used. The quantity of water passing through the pump was determined by allowing the pump to discharge into a tank of known capacity and noting the time taken to fill it. The efficiency of this pump was found to vary from 8 per cent. to 12 per cent., depending on the water pressure and vacuum. With a water pressure of 40 lbs. per square inch at the jet, and a vacuum of 25 in., the efficiency was 8 per cent. This pump was then attached to a small surface condenser taking the exhaust steam from an engine using 400 lbs. of steam per hour. Although the condenser was much too small for this quantity of steam, the vacuum was maintained at from 24 in. to 25 in. of mercury. The jet pump was using a little less than one cubic foot of water per minute at a pressure of 22 lbs. per square inch, or, say, roughly, \( \frac{1}{3} \) H.P. at the jet. During this test the air discharged by the pump was measured and found to be about 4'5 cubic feet per hour, measured at atmospheric pressure. As the feed-water was not quite 7 cubic feet per hour, it will be seen that the volume of air discharged was over 60 per cent. of the volume of the feed-water, and, therefore, when we remember the small size of the plant, the existence of considerable air leaks will be apparent. The above result is not very bad for so small a plant. Subsequently a larger jet pump was tested with 6440 lbs. entering the condenser per hour. The vacuum was maintained at from 26'4 in. to 27 in. of mercury, and the power supplied to the water jet was 1'4 H.P. For a plant of these dimensions we may safely assume that 75 per cent. of the power applied to the pump pistons will be available as water H.P. The circulating pump will, therefore, take 1'76 H.P. extra to enable it to operate the jet; and the condensed steam pump will take about 0'16 H.P., giving a total of 1'92, or, say, 2 H.P., to maintain the vacuum for a 300 I.H.P. engine. The efficiency of this jet pump was tested in the same way as the smaller one, and it was found to vary from 10'5 per cent. to nearly 17 per cent., depending as before on the water pressure and vacuum. The dependence on this latter factor is evident from Fig. II. When the water pressure was 12 lbs. per square inch at the jet, the vacuum 25'4 in. and the barometer 29'6 in., the efficiency was 14'5 per cent., and the volume of air discharged was 0'41 of the volume of water passing through the pump, the volume of air being measured at the pressure of the vacuum chamber, or 4'35 in. of mercury. In Fig. IV. curve No. II shows the variation of efficiency of this pump with a constant water pressure at the jet of 12 lbs. per square inch for different values of the vacuum in the chamber being exhausted; and curve No. 1 shows the variation of efficiency of the same pump with variation of water pressure at the jet, the vacuum remaining constant at 25'4 in. of mercury. It will be seen from the curves in Fig. II. that the work actually required to pump air, assuming the displacement of the pump per minute to remain constant, diminishes after the pressure in the chamber, being exhausted, falls below
about 5lbs. per square inch; and, therefore, though the vacuum may be higher than that corresponding with 5lbs. abs., the energy supplied to the jet must not be reduced. For if the energy supplied to the pump just sufficed to operate it when the abs. pressure was only, say, 3lbs., and an air leak slightly increased the air pressure, the pump would be unable to cope with it, and the vacuum would rapidly fall to that value shown by the curve where the work ordinate is equal, but on the other side of the maximum. Indeed, whenever the vacuum is higher than about 20in. of mercury the power supplied at the jet must be greater than that absolutely necessary or a condition of instability exists, and the vacuum is liable to fall to the other point on the curve, where the ordinate is the same height and a condition of stability is restored. The jet pump, in conjunction with the surface condenser, appears to be specially suitable for use in pumping plants where the water required to operate the jet could be drawn from the main pump discharge, and thus dispense with the separate circulating pump. The results obtained with the larger jet pump mentioned above are fairly good, and when it is remembered that the efficiency was not more than 15 per cent., at the time the test was taken, it will be seen that there is a large margin to work on, and probably with a better designed pump a much higher efficiency could be obtainable. How this higher efficiency is to be obtained, or on which detail of the construction of the pump it primarily depends, I am not prepared to say. My object in introducing this application of the jet pump to your notice this evening is to obtain the benefit of the criticisms and suggestions of those who may address themselves to the subject during the discussion which, I hope, will follow, and give some interest to this otherwise dry subject. With a view to furthering this object I will give as briefly as possible an outline of the method of dealing with the subject which I adopted before having the actual pump made. The subject can be divided with advantage into two sections; the first, dealing with the method of introducing and mixing the air with the water as it issues from the jet; the second, dealing with the best form to give the discharge tubes of the pump to enable it to deliver the air and water into the atmosphere with the least expenditure of energy. The first can, I think, only be dealt with experimentally, and was therefore left for this method of treatment. In dealing with the second I thought some assistance should be derived from a consideration of what appeared to me to be reasonable assumptions concerning the behaviour of the mixture of air and water in passing through the discharge pipe of the pump. As the water jet has to carry the air along with it through the discharge tube by friction, and in the state of small bubbles distributed through its mass, and as a difference of pressure exists between the ends of the discharge tube, it is clear that there is a tendency for the air to blow back against the direction of motion of the water, and thus a circulation of a part of the air entering the pump may take place and cause a loss of energy. Again, as the air is carried down the discharge pipe, it must be compressed from the pressure which exists in the vacuum chamber, to the pressure at the discharge end of the pump; therefore the ratio of the volume of air to water diminishes. The risk of air finding its way back against the water jet must increase as the ratio of the volume of air to water increases, and also as the difference of pressure per unit length of
discharge pipe increases. We can thus see that where the ratio of the volume of air to water is great, the difference of pressure per unit length of tube must be small, and that this difference of pressure per unit length of tube may be increased as the ratio of these volumes diminish, or towards the discharge end of the tube. I have therefore taken as one condition to be fulfilled, that the rate of increase of pressure at any point along the tube shall vary as

\[
\frac{\text{vol. of water}}{\text{vol. of air}}
\]

at that point. Again, as the flow is turbulent, I have considered that the rate at which energy is being wasted at any point along the tubes, on friction eddies, etc., may be assumed to be proportional to the square of the velocity of the mixture at that point. As my object was to determine under assumed conditions the best form to give the discharge tube, with a view to minimising the time spent on experiments I made the further assumption that the efficiency was a known quantity. Having made these assumptions, I endeavoured to solve the problem in the following manner, but always kept in mind the fact that any deductions based on such an incomplete theory could only be regarded as indicating the direction in which alterations of form should be made, and not as giving the correct form. I first obtain an expression giving the radius of the tube in terms of the known quantities for any assumed value of this pressure. Then with successive values of the pressure determine the corresponding values of the radius and draw a curve giving the radius as a function of the pressure. Then from the law of variation of the pressure along the tube which I have considered likely to give good results, the pressures corresponding with various lengths are determined, and a curve is drawn showing the relation between the pressure and distance along the discharge tube. The length of the discharge tube is then divided into any number of parts, and from the last curve drawn the pressure corresponding with each length is taken, and from the first curve the radius corresponding with the respective pressures can be obtained. These radii are set up as ordinates on the points along the axis of the discharge pipe from which they have been found, and the curve drawn through the tops of these ordinates was taken to be the correct form for the tube, or as indicating the direction in which an alteration should be effected if the efficiency of the finished pump differed materially from its assumed value. The work, which must be done in moving 1 lb. of the mixture of air and water (afterwards called mixture) from one where the pressure is $P_b$ lbs. per square inch absolute to another place where the pressure is $P_x$ lbs. per square inch is as follows: $- (P_x \text{ being greater than } P_b)$, let 1 lb. of mixture contain $(w)$ cubic feet of water, and $Ab$ cubic feet of air at the
pressures $P_b$.  

The work required to displace the $A_b$ cubic feet of air is

$$144 P_b A_b \log \frac{P_x}{P_b} \text{ ft. lbs. } \ldots \ldots \ldots \ldots \ldots (1)$$

The work required to displace the $(w)$ cubic feet of water is

$$144 (P_x - P_b) W \text{ ft. lbs. } \ldots \ldots \ldots \ldots \ldots (2)$$

The work done in displacing the 1 lb. of mixture is

$$144 \left\{ \frac{P_b A_b \log \frac{P_x}{P_b}}{P_b} + (P_a - P_b) w \right\} \text{ ft. lbs. } \ldots \ldots \ldots \ldots \ldots (3)$$

As I have assumed that energy is being wasted at any point along the tube at a rate proportional to the square of the velocity of the mixture at that point, each pound can be regarded as losing energy at the rate $K V_x^2$ ft. lbs. per second; where $K$ is a constant to be determined. If $T$ is the whole time in seconds occupied by 1 lb. of mixture in moving from one end of the discharge tube to the other, then the loss in the time $dt$ will be $K V_x^2 dt$, ft. lbs., and the whole loss of energy per lb. in the time $T$ will be

$$\int_0^T K V_x^2 \, dt, \text{ ft. lbs. } \ldots \ldots \ldots \ldots \ldots (4)$$

As I have further assumed that the efficiency of the pump is known, the losses can be taken at $\theta$ per cent. of the energy supplied. The initial velocity $(V)$ of the water jet is due in part to the pressure applied externally and in part to the vacuum. This latter part just suffices to carry the water out against the back pressure of the atmosphere. Let $V_o$ represent the velocity which the water would acquire in flowing into a vacuum equal to that produced by the pump, then the energy which must be supplied from an external source per lb. of mixture is

$$\frac{V_b^2 - V_o^2}{2g} \text{ foot lbs. } \ldots \ldots \ldots \ldots \ldots (5)$$

By assumption $\theta$ per cent. of this is wasted in passing through the tube; we must therefore have

$$\frac{V_b^2 - V_o^2}{2g} \times \frac{\theta}{100} = \int_0^T K V_x^2 \, dt, \ldots \ldots \ldots \ldots \ldots (6)$$

But

$$V_x = V_b \times \frac{\text{vol. at } x}{\text{vol. at } b} \times \frac{\text{area of tube at } b}{\text{area of tube at } x} \text{ ft. per sec. } \ldots \ldots \ldots \ldots \ldots (7)$$

$$\therefore \quad V_x = V_b \times \frac{v_x}{v_b} \times \frac{\pi \gamma_b^2}{\pi \gamma_x^2} \ldots \ldots \ldots \ldots \ldots (8)$$

As the air compression takes place isothermally we have

$$P_b A_b = P_x A_x$$
NOTES ON THE AIR PUMP.

Hence the volume \( v_x \) per lb. of mixture at \( x \) is

\[ v_x = A \, \frac{P_b}{P_x} + w \]  

...(10)

Substituting this value for \( v_x \) in eqn. (8) we get

\[ V_x = V_b \left( \frac{A_b \, P_b + P_x \, w}{\gamma_b} \right)^2 \]  

Again, as one of the desired conditions of action is that the increase of pressure at any point along the tube shall be proportional to the ratio of the vol. of water to the vol. of air at the same point we must have

\[ \frac{d\,P_x}{dx} \text{ varies as } A_b \, \frac{w}{P_b} \]  

...(12)

\[ \therefore \quad \frac{d\,P_x}{dx} = \frac{\beta \, w}{A_b \, P_b} \frac{P_x}{P_x} \]  

...(13)

where \( \beta \) is a constant,

\[ \therefore \quad dx = \frac{A_b \, P_b}{\beta \, w} \times \frac{d\,P_x}{P_x} \]  

...(14)

But

\[ \frac{dx}{V_x} = dt \]

\[ \therefore \quad dt = \frac{1}{V_x} \left( \frac{A_b \, P_b}{\beta \, w} \frac{d\,P_x}{P_x} \right) \]  

...(15)

If in eqn. (6) we substitute this value for \( dt \), and for \( V_x \) its value as found in eqn. (11) we get

\[ \frac{(V_b^2 - V_o^2)}{200 \, g} = K \int P_c \, V_b \left( \frac{A_b \, P_b + P_x \, w}{v_b \, P_x \, \gamma_x^2} \right)^2 \times \frac{A_b \, P_b}{\beta \, w \, P_x} \, d\,P_x \]  

...(16)

Integrating this we obtain

\[ \frac{(V_b^2 - V_o^2)}{200 \, g} = K \left\{ N \, \text{Log.} \frac{P_c}{P_b} + M \left( \frac{1}{P_b} - \frac{1}{P_c} \right) \right\} \]  

...(17)

when

\[ N = \frac{V_b \, \gamma_b^2 \, A_b \, P_b}{v_b \, \gamma_x \, \beta} \]  

and

\[ M = \frac{V_b \, \gamma_b \, A_b^2 \, P_b^2}{v_b \, \gamma_x^2 \, \beta \, w} \]  

...(18)

If in the right hand member of eqn. (17) we change \( P \) into \( P_x \) we obtain an expression for the energy wasted per lb. of mixture in moving
to the point at distance \( x \) from the inlet. If this be added to the energy usefully spent over the same distance the sum so obtained should equal the loss of kinetic energy represented by the change of rectilinear velocity of the mixture, therefore

\[
\frac{V_b^2 - V_x^2}{2g} = 144 \left\{ P_b A_b \log \frac{P_x}{P_b} + (P_x - P_b) W \right\} + K \left\{ \frac{N}{g} \log \frac{P_x}{P_b} + M \left( \frac{1}{P_b} - \frac{1}{P_x} \right) \right\} \tag{19}.
\]

From which we get

\[
V_x = V_b - 2g \left[ 144 \left\{ P_b A_b \log \frac{P_x}{P_b} + (P_x - P_b) W \right\} + K \left\{ \frac{N}{g} \log \frac{P_x}{P_b} + M \left( \frac{1}{P_b} - \frac{1}{P_x} \right) \right\} \right] \tag{20}
\]

But \( V_x \) has been found in terms of the dimensions of the pipes, &c., in eqn. (11). If this value of \( V_x \) be squared, then it can be equated to the value of \( V_x \) found in eqn. (20). Equating these two expressions for \( V_x \) and reducing we arrive at the following quadratic equation, from which the radius \( (\gamma) \) of the tube can be determined in terms of the constants and the variable pressure \( P_x \), which latter can have successive values assigned to it, and a curve can be drawn showing \( \gamma \) as a function of \( P_x \).

\[
\gamma = \sqrt{\frac{D + \sqrt{D^2 + 4CE}}{2C}} \tag{31}
\]

Where

\[
C = V_b^2 - 288g \left\{ P_b A_b \log \frac{P_x}{P_b} + (P_x - P_b) W \right\}
\]

\[
D = 2 \frac{K V_b^2 A_b P_b}{v_b^3} \beta^2 \left\{ \log \frac{P_x}{P_b} + A_b \frac{P_b}{W} \left( \frac{1}{P_b} - \frac{1}{P_x} \right) \right\}
\]

\[
E = \frac{V_b^2 \gamma_b (P_b A_b + P_x W)}{v_b^2 P_x^2}
\]

The only constants whose values have not yet been determined are \( \beta \) and \( K \). \( \beta \) must be determined from eqn (13) thus

\[
\frac{d}{dx} \frac{P_x}{P} = \frac{\beta w}{A_b P_b} \tag{22}
\]

Integrating we get

\[
\log \frac{P_x}{P} = \frac{\beta w}{A_b P_b} dx + \text{(constant} = H) \tag{23}
\]

When \( x = 0 \), \( P_x = P \)
NOTES ON THE AIR PUMP.

When \( x = c \), \( \log P_c = \frac{\beta w c}{A_b P_b} + H \)

Substituting for \( H \), we get

\[
\log P_c = \frac{\beta}{A_b} \frac{w c}{P_b} + \log P_b
\]

\[
\therefore \beta = \frac{A_b P_b}{w c} \log \frac{P_o}{P_b} \quad \ldots \quad (24)
\]

If this value of \( \beta \) be substituted in eqn. (17) the value of \( K \) can be determined. From eqn. (23) we can determine the relation between \( x \) and \( P_x \) which is

\[
x = \frac{A_b P_b \log P_x}{\beta w} \quad \ldots \quad (25)
\]

Substituting the value of \( \beta \) found in eqn. (24) and reducing we get

\[
x = \frac{C \log P_x}{\log P_o} \quad \ldots \quad (26)
\]

From eqn. (21) we get \( y \) in terms of \( P_x \). From eqn. (25) we get \( P_x \) in terms of \( P_x \). Therefore, if \( y \) and \( X \) be worked out for the same successive values of \( P_x \), we obtain the data from which the curve of the discharge pipe can at once be drawn.
A Few Notes on the Air Pump.

Discussion April 11th, 1900.

Mr. G. Higgins said he considered the paper of great importance, as it has the merit of breaking new ground, which is always a good thing in engineering subjects. The text books do not go nearly far enough into such matters as this, and we can well thank Mr. Stone for his valuable paper. Although he had carefully read the paper and weighed minutely the assumptions that the author made, he had not had sufficient time to check every extension of the equations which he works out. As far as he could see, Mr. Stone's assumptions are, on the whole, justifiable; and his methods of making his calculations appear to be correct. With one exception, these are not open to question, and that exception is where Mr. Stone crosses swords with Professor Jamieson, on page 4. The difference between Professor Jamieson and Mr. Stone is this:—The former seems to allow an interval of time to elapse after the piston has commenced to move for evaporation to follow up the piston and fill the void left by the piston with water vapour. If evaporation were instantaneous, Mr. Stone would be quite right. If it is not instantaneous, and does not cause the vapour pressure to become at once equal to what it would be after a little time, then Professor Jamieson would be right. He could not remember ever meeting with any accounts of experiments showing that evaporation is not instantaneous. It cannot be quite so, but the rate at which it takes place under given conditions of temperature and pressure is a subject about which he was not aware that there was any information. The effect of Mr. Stone's assumption is apparent in one of his first calculations.

Page 10, re the Edwards pump.—It is true that the pressure above the piston on the down stroke is the pressure of water vapour; but the speaker did not think that $P_w$ has the same value above as below the piston. He thought, assuming that there is water vapour in the pump chamber on the down stroke, that while the piston is moving down an appreciable time has to elapse before evaporation has gone on to such an extent as to restore the vapour pressure in that time to what it should be, having regard to the conditions of pressure and temperature. It may be that evaporation takes place so very rapidly that the vapour pressure remains practically constant.

On page 12, the indicator diagrams throws some light on the subject. The R.H. line corresponds to the down stroke of the piston. If the vapour pressure were the same from the beginning to end of the stroke, the lower line would be much more nearly horizontal than it is. This assumption is the only one questionable, but it affects the whole of the arguments.

On the first page Mr. Stone shows that no valves should exist be-
A FEW NOTES ON THE AIR PUMP.

between the condenser and the air pump, because if these valves have any weight, work must be done in lifting them, and certain pressure must remain in the condenser, exceeding that in the pump chamber by an amount sufficient to lift them. In that respect the Edwards valve is a great improvement on the old-fashioned single-acting air pump. A few years ago, the speaker had a case where the vacuum had been 22 in. and fell to 18 in. without any apparent reason. The engineer examined all the pipes and joints, and could find nothing in the shape of a leak; the valves also being found satisfactory. Some time afterwards they were working away with the lower vacuum, and the engineer told him he had effected a great improvement, i.e., he had put in brass valves in the air pump instead of the old rubber ones. After these heavy valves were taken out they got back to the original vacuum. On figure 2, page 3, one curve represents the foot lbs. of work in expelling air assuming adiabatic conditions, and the lower curve assumes isothermal. In the ordinary air pump, he did not think any practical difference in speed would cause any such difference in the result as we get by assuming the pump to be compressed isothermally or adiabatically, and would be glad to be enlightened on this point. If an appreciable time is taken for evaporation then Mr. Stone’s argument for slow speed in the case of air pumps would not hold good. On the down stroke of an Edwards pump, if the piston can outrun the evaporation, it follows that the quicker we run the better, and keep up the lower pressure in the condenser. However, he thought, it is advisable in many cases to use an independent air pump instead of having one worked by levers from the main engine, so that its speed can be regulated to the best advantage, having due regard to various other circumstances.

Re Page 6, line 24—

If the clearance be diminished, so that all the water in the pump chamber is forced through the bucket valves, there would then be very little water for the air to force its way through. In the case of the Edwards pump, one of the conditions is that the clearance is very small; but that is in connection with another matter.

Re page 7, lines 8 and 9—

Re water entering in a stream below the air. It is the pressure in the condenser that forces the water through the foot valves. He could hardly see how we can have water flowing in under air. Unless the air pressure is made use of we cannot expect to force the two to flow together. He was thinking of the ordinary type of air pump.

Page 11, re the advantages of the Edwards pump—

Another advantage is that the piston in moving actually mechanically forces the water out of the chambers into the pump chamber above the plunger. Mr. Stone’s investigation of the jet pump is very admirable, but as to the best shape and the best dimensions of the discharge pipe, he should like to see a drawing of one of these pipes, to ascertain how far a carefully-designed discharge pipe departs from the ordinary cylindrical form of pipe, and as to what saving in fuel results from the careful design.
Mr. W. Fyvie

I think this very important and exhaustive paper, contributed by the President, to be a very valuable addition to the literature of the Institute, and reflects great credit for the very careful and able manner in which he has handled this important and much neglected subject, "The Air Pump as Applied to the Steam Engine."

I do not know of any other part connected with the steam engine that has been so much neglected by modern engineers; I think the majority of air pumps are not so well designed as those made by Watt nearly 100 years ago.

I regret to say that I have not had the time to study the paper sufficient, to attempt its discussion as I would like, but must content myself with a few remarks.

**Vertical Single Acting Air Pumps.**

Having had some little experience with this type, and I am aware of its many shortcomings, which are as nothing compared with those to be met with in some vertical and horizontal double acting pumps, with all their mysterious air-traps and pockets.

In operating the common single-acting vertical pump, I have generally got the best results by removing the foot valve or valves, and at the same time removed what is usually a troublesome member. In a paper read before the Institute of Mechanical Engineers, on November 5th, 1896, by Mr. Michael Longridge, on "Breakdowns of Stationary Engines," he gave a list of 1000 breakdowns previous to 1894; no less than 210, or 21 per cent., were due to air pumps and their gearing; 88, or about 42 per cent., of which were due to buckets and valves; 32, or about 15 1/4 per cent., were due to the foot valves alone. This gives 327 per cent. of the 1000 breakdowns as being due to the troublesome member I referred to above—the foot valve. Such records, added to one's own experience, tends to produce a feeling of anxiety.

On page 7, line 32, referring to the up stroke of the pump, should this not be plus the pressure due to head of water over discharge valves, which is often a considerable item?

Another type of pump that has given very good results, and is interesting to follow its cycle of operations. This is the double-acting air pump, fitted with a "pressure equalising apparatus." This apparatus opens communication and equalises the pressure between the two ends of the pump when the plunger comes to rest at each end of its stroke, just when the valves are all closed. This is suitable for wet or dry pump, but most effective with the dry. It is also applicable, and sometimes used, on air compressors.

"Edwards' Air Pump."

I agree with the author that this type of pump is an improvement over those of the older type, and is likely to come largely into use. It is something after the style of the vertical single-acting pump, with the foot valve removed, including other important features in its formation.

I am very much interested with the "Jet Pump," as applied and
described by the author, and sincerely hope he may succeed in something near the results given in his paper, when applied to actual every-day practice.

A good many years ago I used ordinary vertical single-acting air pumps, side by side with that very simple and attractive apparatus, the "Ejector Condenser," but I never succeeded in getting the same results, although they were fixed to the places and personally approved by the inventor, Mr. Morton; but they may be much improved since then. Still, for all those years they have been before the public, they are not largely used, that I am aware of. There are many practical difficulties to be overcome.

And I wish the author every success, and am watching with keen interest for the good practical results I hope he may obtain, as the Jet Pump is simplicity itself, and little about it that is likely to break down, thus removing a source of anxiety that is ever present in the engine-room.

The author, on page 6, states that the absolute pressure between bucket and foot valves will be less than that in condenser by the amount necessary to raise the foot valves and break through the water seal. I admit this is a consideration of serious import in some air pumps; but if proper materials are used for the valves, and a fairly well constructed pump, this should not be of much moment. The construction so that the weight of the valve and the water, acting by gravitation, will assist in opening same.

A horizontal double-acting air pump of this type I will try to roughly illustrate on the blackboard.

I may state I have got better results with this pump than any other I have used. The plunger is a hollow cylinder with hemispherical ends, and is constructed that when nearly wholly immersed in water, will just float, water being used to fill the necessary clearance at the ends, suction and delivery valves being on the top.

Mr. J. T. Noble Anderson said he considered the Institute was under a debt of obligation to Mr. Stone for having brought this matter before them in such a complete manner. Judging from the paper, and the discussion thereon, the general impression left on their minds was that horizontal and double-acting types of air pumps are altogether bad.

Mr. Fyvie—Not all; a good many of them.

Mr. Stone—I did not mention them.

Mr. Anderson—There is no necessity for the heavy valves referred to by Mr. Higgins. A valve of two or three discs of thin, flat sheet brass is quite sufficient to act as a foot valve for a horizontal air pump, and such a valve adds no appreciative weight to the work. In the case of unsatisfactory horizontal pumps he had seen, the reason is apparent, i.e., they have been designed too small; in other cases they are placed with the valves higher than the discharge from the condenser, and he has seen them with an absolute trap for the water between the condenser and the valves of the air pump. There is considerable loss due to this. With a fairly well designed horizontal
A FEW NOTES ON THE AIR PUMP.

pump, laid down properly near the condenser, with light valves, he did not think there would be much cause for complaint. The Ballarat pump has been working for more than twelve months, and there have been no complaints, and it gives over 26in. vacuum, which, for 1400ft. above sea level, he thought, is not at all bad. He also knows of three near Maryborough which work well and give good vacua, and he scarcely thought they will be thrown out as rubbish just yet! Of course, he did not think they are as good as the Edwards Pump. He would like to have a further opportunity to discuss this paper after having looked thoroughly into it.

Mr. Fyvie said he did not condemn all horizontal pumps. The best he had to do with had been horizontal pumps; but, he said, the majority of them are bad.

Professor Kernot could only call to mind one result concerning an engine with horizontal air pump. That engine was heavily loaded, passing a large quantity of steam through the cylinder, and it gave absolutely the finest vacuum (by the gauge on the condenser and the indicator on the cylinder) that he ever saw. Re the ejector condenser, he might say that they have had ejector condensers at the Electric Light plant for a good many years, but had some trouble at first, as they were too small, and had to be enlarged. But since they have been fitted with no-return valves they have had no further trouble. However, they will have to go, because the Yarra water is getting very bad, and the engines will have to be fitted with surface condensers. The whole question is of very great importance, and he congratulated the Institute in having such a good discussion upon it. If Mr. Stone has an opportunity of carrying out his system on an actual engine, he should be glad to hear the results.

Mr. Stone said he would like to thank those gentlemen who have taken part in the discussion, especially Mr. Higgins, who has been most critical, and, therefore, most friendly. Re the proposed jet pump, Prof. Kernot and a number of other gentlemen have been to the Railway Electric Light Station, where there is one working away quietly, and he expects to get good results from it; the vacuum varies from 26 to 27 1/2 inches. His object in bringing this matter before the Institute was not to teach, but rather to learn, and to call attention to this apparently simple subject, so as to gain the assistance of others who have had opportunities of obtaining more mathematical knowledge, and endeavour to discuss the matter more perfectly.

Discussion May 2nd, 1900.

Mr. W. Stone, in reply to Mr. G. Higgins, said: The statement made by Professor Jamieson—viz. : “Besides this air there would, of course, be an accumulation of vapour or low pressure steam in the condenser from the condensed water under the action of a partial vacuum, and this vapour would soon fill the condenser and also spoil the vacuum of it where not extracted by the air pump,” appears to me as difficult to explain, even though we admit the tentative hypothesis suggested by Mr. Higgins. If for the time being we go one step further than
Mr. Higgins and assume that our air pump abstracts its full swept volume of low pressure steam from the condenser, it simply means that the condenser has so much less steam to condense. The quantity abstracted is condensed in the air pump, and in that way we may regard the air pump as an additional capacity to the condenser. If we assume the air pump to have one cubic foot displacement per lb. of steam condensed, and that the steam pressure in the condenser is 2 lb. avs., then since 1 lb. of steam at this pressure occupies 172 cubic feet, and our air pump has only abstracted 1 cubic foot of steam, its useful capacity as an addition to the condenser is only 0.6 per cent. under these conditions. The improvement in the vacuum in the condenser due to its being relieved of 0.6 per cent. of the steam entering it surely is not what Professor Jamieson refers to when he says “it would soon fill the condenser and spoil the vacuum if it were not abstracted by the air pump.” I cannot see that the question of instantaneous, or slow evaporation, can affect the correctness or otherwise of Professor Jamieson’s statement, excepting in the very subordinate sense to which I have just referred. The vacuum in the condenser depends on its temperature and the absolute partial pressure due to air, and so far as I can see no other factor comes into the question. The question raised by Mr. Higgins as to the correctness of my assumption of practical instantaneous evaporation undoubtedly is important, for as he points out, many of my equations cease to be true if the pressure on the pump piston is materially less than that corresponding with the temperature of the water. When we remember that one cubic foot of steam at 2 lb. avs. weighs only 0.0058 lb., or about 41 grains, that its latent heat of evaporation is only about 6 B.T.U., that evaporation can take place from the whole interior wetted surface of the pump, and, above all, that ebullition can take place throughout the whole mass of water in the pump the moment that the pressure falls below that corresponding with the temperature, I do not think my assumption is materially at fault. Of course, as Mr. Higgins properly says, evaporation undoubtedly is important, for, as he points out, many pressure to account for the motion of the steam in following up the piston. However, much larger quantities than these have been neglected in my treatment of the whole subject. I am glad that Mr. Higgins has called attention to this important point, for undoubtedly clear ideas and accurate conclusions concerning it are essential when we wish to design air pumps intelligently. With reference to Mr. Higgins’ remarks concerning the R.H. line in the diagrams on page 12, it is perfectly true that the slope could be accounted for by the assumption of comparatively slow evaporation. I think, however, that this slope is generally considered to be due to air re-expanding which has been mixed with the water in the pump, or even dissolved in it during compression. An examination of the three diagrams referred to by Mr. Higgins appears to me to support this view, for it will be seen that in the top diagram where the vacuum is the best, and consequently the least quantity of air will be present, the slope of the R.H. line is much less than in the others, where the vacuum
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is poor and more air is present. On page 14 I have only made a rough attempt to ascertain the effect of the weight of vapour moved on the useful displacement of the Edwards pump, and have disregarded friction, etc. I would ask that the discussion be adjourned so as to enable Mr. J. T. Noble Anderson to give us the benefit of his able criticism, after he has looked thoroughly into the question in accordance with his intention as expressed at our last meeting.

Discussion, June 6th, 1900.

Mr. Turner (in the chair) mentioned that the discussion had been adjourned pro tem. to enable Mr. J. T. N. Anderson to speak, but as he was not very well, and not likely to be out of an evening for some little time, he thought that, if no other member desired to offer any remarks, Mr. Stone should now reply.

Prof. Kernot said he was inclined to agree with Mr. Stone. He did not see how Prof. Jamieson's arguments were reconcilable with the first principles of evaporation, vapour pressure, and contact with liquids. As the matter is to be referred to Prof. Jamieson, it would be further interesting when they had his reply. It was true that the subject was ignored. He had just received Prof. Ewing's new book on the steam engine, and he seemed to ignore the condenser altogether, as if it were something beneath his notice! He (Prof. Kernot) thought that Mr. Stone had shown plainly that the condenser and air pump of the steam engine had received very scant courtesy at the hands of the various authorities on the subject, and he had done good service in calling attention to what is a region of vague and unsatisfactory information. As a rule, there had not been a very great deal of difficulty with condensers; they had done their work fairly well, and people seemed satisfied, but they should be worked up to the highest state of efficiency, as much as the other parts of the engine.

Mr. Turner mentioned that two cases had come under his notice recently where manufacturers had been carrying on with vacua of 22in. to 24in., whereas they might have had 26in if they had known better. In Melbourne there was great room for improvement in the matter of condensers.

Mr. Stone, in reply, said he did not think there was anything further for him to reply to. At the last meeting he had endeavoured to reply to the questions raised. He agreed with Professor Kernot as to the importance of the condenser. The steam engine was a heat engine, and its efficiency depended on the highest and lowest temperatures, between which they could work—the highest temperature was determined by the boiler construction, and the lowest by the condenser and air pump. This question of vacuum seemed to be of special importance in connection with the steam turbines. From data which he had, the steam consumption in these machines dropped very rapidly with an improvement in vacuum when they got down to an absolute pressure of 2lbs. As another 2in. of mercury meant double the volume of steam, it was easy to see that a great difference in the work attainable would accrue. Some of the "Parsons" and "De Laval" turbines
were designed to work with a final pressure of 1 lb. absolute. If that were brought up to 2 or 3 lb. absolute, the steam consumption was increased by, perhaps, 15 or 20 per cent.; but that was a matter outside the present subject. He desired to thank the members for the kind way in which they had dealt with his paper; but he regretted that they had not found more to criticise.
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