MODERN CONDENSING SYSTEMS.

BY A. E. LEIGH SCANES, M.A., OF MANCHESTER.

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I. It is the object of this Paper to indicate as briefly as possible the lines along which development in condensing systems has taken place during recent years and will probably continue in the future. It will be shown that the tendency is towards smaller and cheaper plants of high vacuum efficiency (*see* Appendix I, page 216), and towards simplicity in auxiliaries as exemplified by rotary pumps. So much has already been written about the ultimate advantages of condensing systems as opposed to the initially cheaper non-condensing types of engines that nothing need be said under this head. The same is true to a lesser extent regarding the degree of vacuum it is desirable to maintain for maximum overall efficiency.

In practice it is necessary to investigate each particular case with regard to local conditions, as to the cost of coal and the water supply available for cooling. Generally speaking, it will be found that for reciprocating engines vacua of from 26 to 27 inches are most suitable, and for turbines from 27 to 29 inches. Here, and in all calculations which follow, the barometer has been assumed to be 30 inches of mercury.

For high-pressure steam-turbines an approximate saving of 5 to 6 per cent. in steam consumptions can usually be effected between 27 and 28 inches vacuum, and 7·5 to 9·5 per cent. between 28 and 29 inches and 13 and 15 per cent. respectively for low-pressure turbines.*

For low-pressure turbines in connection with winding engines and similar plants, care must be taken that air is not pumped into the condenser, if a very high vacuum is to be maintained. From the above gain there must be deducted the increased capital outlay, running cost and depreciation of the condensing plant, over one for 27-inch vacuum, which may be taken as the minimum desirable vacuum under normal conditions.

II. *Types of Condensers.*—Condensers can be divided into two main types and five divisions:—

* K. Baumann on "Recent Developments in Steam-Turbine Practice." *Inst. of Electrical Engineers*, December 1911, vol. 48, page 768.

1. *Surface.* (a) Cooling water passing through tubes, the usual form.
(b) Evaporative where steam is condensed in the tubes by a small quantity of water flowing over their exterior.
2. *Jet.* (a) High level or barometric.
(b) Low level.
(c) Ejector.

The first type is used when water is not available at a reasonable cost for boiler-feed purposes.

III. *Choice of a Plant.*—Figs. 1 and 2 (pages 182–3) show a series of curves from which it is possible to compare the relative advantages of the different types of plant. In calculating these curves in Fig. 1 allowance has been made for the original cost of condensers, cooling towers (when necessary), and foundations. In Fig. 2 depreciation at 12 per cent. per annum and running cost at $\frac{1}{2}d.$ per unit have been capitalized at 5 per cent.

If the depreciation and running cost were capitalized at 4 per cent. the difference between the curves for surface and jet condensers would not be as much, and in any case the greater freedom from troubles, smaller number of working parts, and reduced attention for cleaning, make the jet plant the most desirable when it can possibly be used. These figures do not pretend to be more than an approximate guide, as so many conditions must be taken into consideration that any hard and fast rule is impossible.

IV. (1) *Surface Condensers.*—In the earlier types of surface condensers a very low rate of heat transmission was usual, about 350 B.Th.U. per square foot of surface per degree Fahrenheit mean temperature difference (*see* Appendix III, page 217). The above rate would give condensation of 5 or 6 lb. per square foot per hour for moderate vacua and water temperature.

From the point of view of reliability much has been said in favour of large surfaces, since they were supposed to require less

The figure consists of three vertically stacked graphs, each showing the relationship between Total initial cost (£) on the y-axis and Vacuum inches Hg. (Baro. 30") on the x-axis. The y-axis ranges from 0 to 4000 in increments of 1000. The x-axis ranges from 27 to 29 in increments of 1. Each graph contains three curves: a solid line for Surface Plant, a dashed line for Low level Jet Plant, and a dash-dot line for High level Jet Plant. The graphs are for water temperatures of 60°F, 70°F, and 80°F, respectively. The cost generally increases with vacuum for all plant types, with Surface Plant being the most expensive and High level Jet Plant being the least expensive.

Vacuum (inches Hg)	Water temp. 60°F			Water temp. 70°F			Water temp. 80°F		
	Surface Plant (£)	Low level Jet Plant (£)	High level Jet Plant (£)	Surface Plant (£)	Low level Jet Plant (£)	High level Jet Plant (£)	Surface Plant (£)	Low level Jet Plant (£)	High level Jet Plant (£)
27	1100	600	500	1300	700	600	1800	1000	800
28	1300	800	700	1600	900	750	2500	1400	1100
29	1600	1100	900	2000	1200	950	3500	1900	1400

attention and cleaning to maintain a high vacuum. This may be true with moderately clean water. With some kinds of bad water, however, marine practice has shown that small surfaces are best for reducing the formation of scale, owing to the increased water velocity. Furthermore, this increase of velocity through the tubes

FIG. 3.

Heat Transmission from Steam to Water in Surface Condensers.

also increases the rate of heat transmission from the metal to the cooling fluid.

Fig. 3* shows experimental results obtained by various authorities, connecting the rate of transmission and water velocities. It will be seen that considerable difference of opinion exists, and it is necessary to observe the greatest care in details of design to obtain results corresponding to even a mean of the curves shown.

* Taken from *The Engineering Review*, 1911, vol. 24, No. 140.

A common cause of failure is throttling of the flow of steam among the tubes, a large drop of vacuum resulting between the steam inlet and the air-pump suction. Many methods have been tried to overcome this by keeping a constant steam velocity, such as spacing the tubes farther apart near the steam inlet, making the condenser-shell heart-shaped to offer a large surface at the top, gradually decreasing to the air outlet, and running passages among the tubes. The first and last have been adopted by the British Westinghouse Electric and Manufacturing Co., with great success, and in their opinion these offer the best all-round results when combined with a cylindrical shell which offers many advantages in manufacture and is mechanically strong.

TABLE 1.—BURNLEY CORPORATION.

Extract of official test on 20 per cent. overload.
 1,500-kw. Turbine, 2,100 r.p.m., 3,100 sq. ft. condenser.
 Power of auxiliaries, 28·5 b.h.p.

Time.	Throttle Press.	Throttle Temp.	Inlet Press.	Temp. before Throttle.	Vac. In. Hg. Bar. 30 in.	Inlet Water Temp.	Hot-well Temp.	Net Weight of Steam condensed.
a.m.	Lb. per sq. in.	° F.	Lb. per sq. in.	° F.		° F.	° F.	Lb. per hr.
1.30	127	456	99	470	} 28·9 mean	41	68	29,500
1.35	126	460	100	475				29,200
1.40	126	460	100	475				29,000
1.45	126	459	100	475				29,000
1.50	124	460	100	475				29,800
1.55	124	466	100	480				29,300
2. 0	124	472	101	485				29,250
2. 5	123	481	102	495				29,200
2.10	123	489	106	500				28,550
2.15	123	490	107	500				28,250

Average steam condensed, 29,105 lb. per hour. Pounds condensed per sq. ft. of surface = 9·4. Hot-well of 68° F. corresponds to a vacuum of 29·32 in. mercury. Therefore, 28·9 in. = 98·56 per cent. vacuum efficiency, that is 98·56 per cent. of 29·32 in. mercury.

FIG. 4.—*Diagram of Surface Condenser (Westinghouse).*

Fig. 4 is an illustration of this type. It will be seen that the path of the steam follows an easy curve, it being advisable to avoid baffle-plates and sudden changes of direction as much as possible. Note should be taken of the generous size of the steam-dome at the top of the condenser, obviating air-pockets and distributing the steam easily without the need of directing plates.

Small, but important, points in manufacture are :—

1. Ferrules which do not allow “creeping” of the tubes.
2. Avoidance of “expanded” tubes.
3. Ample thickness of the tube-plates.
4. Staying of the tube-plates.
5. External joints between tube-plates, condenser body and water-boxes.

This last point is very important, as any leakage in the junction between the water-box and tube-plate will be to atmosphere and not into the vacuum space, and is, therefore, immediately detected. With the condenser as shown in Fig. 4 (page 186) having a water velocity of over 5 feet per second, effective condensation can be obtained commercially for land practice at the rate of 8 to 15 lb. per square foot, depending on the degree of vacuum required. The official test at Burnley Corporation electric light station, shown in Table 1, is an example of a plant of this type. It will be seen here that with a vacuum of 28·9 inches condensation was obtained at 9·4 lb. per square foot. A few years ago this would have been considered very unwise, if not impracticable. In this case the water flows at about 7 feet per second, passing four times through the condenser.

Table 2 (page 188) gives results obtained at the official test of a condenser for the East Indian Railway Co. This condenser was designed for 45,800 lb. of steam per hour, 26 inches vacuum, 2,800 gallons per minute of circulating water at 85° F., transmission 435 B.Th.U. per square foot per degree mean temperature difference. The readings give a mean transmission of 639 B.Th.U. The vacuum efficiency in readings 3 to 6 fell off, as by an oversight the condensed steam was allowed to rise a few inches in the condenser, by over-throttling the extraction pump discharge which was controlled by a sluice-valve to give the guaranteed head. The effect of this was similar

TABLE 2.

THE EAST INDIAN RAILWAY CO.

Extract from official test on the Surface Condenser.

Read- ing.	V.	T ₁ .	T ₂ .	T ₃ .	T ₄ .	T ₅ .	δ.	S.	k.	η.
1	27.52	69.5	100.5	109	102	85.5	20.2	48,600	584	98.5
2	27.17	76	105.7	113.5	109	98	18.9	46,400	597	98.7
3	26.66	79.8	112.6	119	110.5	107	18.1	48,200	650	97.5
4	26.18	84.5	116.5	123.5	108	115.5	18.65	48,500	632	95.5
5	25.64	88.1	121.9	128.5	108	124	18.65	51,200	666	93.7
6	25.2	93	125.5	132.5	110.5	129.3	18.78	48,500	625	92
7	24.5	100.4	129.7	137	132	135.3	18.15	53,600	720	97

V = Vacuum, inches of mercury (Bar. 30 in.).

T₁ = Inlet cooling water temperature, ° F.T₂ = Outlet cooling water temperature, ° F.T₃ = Theoretical temperature of steam at inlet to condenser, ° F.T₄ = Condensed steam temperature, ° F.T₅ = Air-pump suction temperature, ° F.δ = Mean temperature difference (T₃ = T₄, see Appendix III, page 217).

k = Transmission in B.Th.U. per square foot per ° F., mean temperature difference.

η = Vacuum efficiency per cent.

to a weir in the bottom of the condenser. It will be seen in reading No. 7 the efficiency was rising again.

Fig. 5 shows an older type still largely made by British manufacturers.

By far the most important factor that has to be considered in the rate of transmission of heat from the steam to the cooling water is unfortunately one which has often been much neglected, namely, the absolute partial pressure of air in the condenser. Many people not intimately concerned with the problem of condensation think that the only reason for lowering the partial

air-pressure in condensers is to reduce thereby the total pressure (*see* Appendix II, page 217). A most important point, however, is that the air, being admittedly a bad conductor of heat, should be

FIG. 5.—*Diagram of an older type of Surface Condenser.*

A.—Steam inlet.	D.—Cooling water inlet.
B.—Condense outlet over weir.	E.—Cooling water outlet.
C.—Air-pump suction.	

eliminated as far as possible for this reason also. If this is not done, a certain proportion of the surface near to the air-pump suction becomes "air-logged," and little condensation takes place at that point, the surface being wasted. It must, however, be

remembered that the distribution of air throughout the condenser is not equal, which affects the results of experimenters with small apparatus who have given many admirable Papers on this subject. It is obvious, in any case, that an air-pump is necessary that will deal efficiently with air at a very low absolute pressure.*

The next point of importance is Vacuum Efficiency (*see* Appendix I, page 216). It is obvious that the hotter the condensed steam is when returning as feed-water, the higher will be the overall thermal efficiency of the plant. This point is dealt with more fully under the heading "Auxiliaries." With a good system of departmental drainage a vacuum efficiency of 100 per cent. can be obtained; indeed, Professor Weighton, in his experiments at Armstrong College, Newcastle-on-Tyne, claims to have exceeded even 100 per cent., probably owing to the vacuum being measured at a point remote from the condensed steam. The author is, however, of the opinion that if serious loss of vacuum is to be avoided in the condenser between the exhaust inlet and the air-pump suction, as straight a flow of steam as possible is desirable, and that departmental drainage, which involves baffle-plates, is not to be recommended, except to a very limited extent. Referring again to the Burnley tests, it will be seen that a vacuum efficiency of 98·6 per cent. was obtained, without any special arrangement of steam-baffles. This is due to the Leblanc air-pump, which will be described later.

A condenser designed for moderately high transmission has usually three or four water-passes, to obtain the necessary velocity without undue length of tubes between plates. This means it is possible to obtain fairly short tubes which are easily cleaned, and this is a great advantage. The cost of renewals is also lower as the number of tubes to be replaced will be approximately the same,

* For one of the best Papers on "Heat Transmission," *see* Orrok, Proceedings, Am. Soc. of Mech. Engineers, Nov. 1910. *See* also "Heat Transmission," by Professor W. E. Dalby, Proceedings, I.Mech.E., 1909, page 921.

whilst their length is shorter ; moreover, there is greater freedom from troubles due to "sagging" and vibration.

It will now be seen that the performance of a surface condenser for any given set of conditions does not necessarily depend on the number of square feet of surface it contains, but that design and efficiency of air-pump play a most important part.

FIG. 6.

Diagram of a Reciprocating Air-Pump Cylinder with Automatic Discharge- and Relief-Valves.

(2) *Auxiliaries.*—(a) *Air-Pumps.*—It has already been shown that a high efficiency is impossible without a first-grade evacuator. The Hon. Sir Charles Parsons was the first to take a practical step towards solving the problem, when he introduced his vacuum augmentser. It is not intended to describe this well-known invention, but note should be taken that it absorbs from 1 to 1.5 per cent. of the main steam consumption of the turbine at full load, and, in calculating the relative overall efficiency of different

types of pumps, this fact should be taken into consideration ; * also there is the additional cost of the auxiliary condenser.

Excellent results can be obtained by use of a dry piston-pump working with an equalizing valve. The use of such valves was a great advance on previous practice ; but as the efficiency of the pump depends on very fine clearances, any wear on the pistons and valves rapidly decreases the volumetric efficiency, which some makers claim to be as high as 98 per cent. Fig. 6 is a common type of dry air-pump. It will be observed that, in addition to the slide-valve, there are at least two automatic discharge-valves and two relief-valves. Great care must be taken with the dry piston-pumps to eliminate moisture as far as possible from the cylinders ; the percentage carried over with the air should not exceed 0.15 to 0.3 per cent. if trouble is to be avoided.

Good results are also obtained with multiple cylinder wet air-pumps ; one or more cylinders, being flooded with cold water to reduce the vapour tension, are used as an air-pump, and the other removes the condensed steam at a higher temperature. This type also suffers, although to a less extent, from the disadvantages of valves which require constant attention and renewal.

(b) *Rotary Air-Pumps*.—With the object of obtaining greater simplicity, while retaining all the advantages of reciprocating pumps and augmenters, various rotary and water ejector-pumps have been developed during the last ten years, nearly all of them an evolution of the Körting ejector condenser, which was one of the earliest forms. The Leblanc pump was the first working rotary pump, and all later types follow the same general principles, except the Kolb rotary condenser, which never gave satisfaction in practice. Then came a rotary pump in which water was discharged at a high velocity through a number of rotary converging nozzles. Such a pump is illustrated diagrammatically in Fig. 7.

* Parsons and Stoney on the Steam-Turbine: Proceedings, Inst. of Civil Engineers, 1905, vol. clxiii, page 167.

Pressure is generated by centrifugal force in the central chamber A, which is rotated at a high speed. This pressure is converted into velocity through the converging nozzles BB, which are usually arranged in pairs, the action being similar to that of an old-fashioned gas-burner, the resulting water-spray taking a fan formation. These fan-shaped jets entrain air which is drawn in through the slots CC. The mixture is discharged through a series of suitably arranged guide-blades into the diffuser D, as shown in the illustration.

FIG. 7.

Diagram of Rotary Spray Air-Pump.

FIG. 8.
*Water Ejector
Type of Air-
Pump.*

1. Water inlet under pressure.
2. Air suction.
3. Combined discharge.

Fig. 9 (page 194) shows the results of a test on a pump of this type of a very large capacity, suitable for 100,000 lb. of steam per hour with a vacuum of 27.4 inches. It will be observed that the b.h.p. is practically constant for all loads. This is claimed to be an advantage, but in the author's opinion it is the reverse, the efficiency being necessarily bad at light loads. A pump of this type, however, only removes a small weight of air per b.h.p. under normal working conditions. It is obvious that this design can never be efficient owing to the loss of kinetic energy in the spraying nozzles.

A simple type of pump is shown in Fig. 8 (page 193). The water-jet or jets remove the air by ejector action. Here again the expanded volume entrained is very small, often less than twice the volume of water. Still another recent development consists of a combination of a steam augments and water-jet. The author is

FIG. 9.—Test on Rotary Air-Pump of similar design to Fig. 7.
See "The Engineer," 2 Dec. 1910.

unable to give figures of the working of this plant as there are not many in operation, and no results had been published that the author was aware of when this Paper was written, but it would be interesting to know the makers' guarantees on weight of air dealt with for a given power, including the necessary steam reckoned at 10 lb. per b.h.p.

(c) *Latest Developments in Rotary Pumps.*—Finally we come to the Westinghouse-Leblanc dry air-pump. This pump has undergone many improvements since it was originally brought out by Professor Leblanc, and its original faults have been found out by constant experiments and world-wide experience, and eliminated. The changes consist mainly in alteration to the shape of the blades and proportion of the cones, and the points of difference are so dependent on knowledge gained by experiments, and in many cases so apparently trivial, that it is outside the scope of this Paper to describe them.

The latest designs of the Westinghouse-Leblanc pumps will remove much more air per b.h.p. under similar working conditions than any other rotary pump. One peculiarity of this and of similar rotary pumps is that the expanded volume removed increases as the absolute air-pressure decreases, the contrary being the case with reciprocating pumps. A great advantage is that the working medium of the pump being at a far lower temperature than the condensed steam, the air is cooled and considerably reduced in volume before being removed. In comparing expanded volumes in various types this fact must be taken into account, which can easily be done by specifying the weight of air instead of the volume. A further advantage of the Leblanc pump is that, to save space and complication, it can in many cases be direct coupled with its extraction pump to a circulating pump running at the same speed without the use of gears, the whole set being driven by a motor or small turbine, Fig. 10 (page 196). In the latter case, as a good steam consumption cannot be attained on so small a machine, when running condensing at a suitable speed and with a simple design, it is usual to exhaust above atmospheric pressure, either into a stage of the main turbine or to a feed-water heater. A high overall efficiency can be obtained in this manner. Furthermore, the Leblanc pump in certain cases can be used as a combined air and extraction pump without reducing its efficiency.

Plate 5 shows one of two sets of pumps working at Penalta Colliery, South Wales.

Fig. 11 (page 197) is a diagrammatic representation of the method of operation of the air-pump. The particulars are as follows:—

FIG. 10.—*Space occupied by Surface Condensing Plant (Leblanc) and by Reciprocating Pumps.*

FIG. 11.—*Diagrammatic Section of Air-Pump and Tank shown in Plate 5.*

A.—Air inlet.
B.—Diffuser and water outlet.

C.—Water inlet.
D.—Valve for filling the tank.

E.—Tank.
F.—Overflow.

G.—Water guide-nozzle.

FIG. 12.—Test on Air-Pump (Westinghouse-Leblanc).
Arranged as in Fig. 11.

NOTE.—Test Points have been added to the Curves since the Discussion took place.

The pump does not take its working fluids from a seal-well below its axis as in the older types, but from the tank E with its water-level above C. The advantage of this arrangement is that should the vacuum fall for any reason the pump remains primed, and on the leakage being stopped it picks up again without attention. Supposing that the casing is full of water from the tank E, the motor or turbine is started and the wheel revolves in the direction of the arrow. The centrifugal force imparted to the water forces it out through the cone B until the case is empty. This creates a vacuum in the casing, and more water is drawn in through the guide-nozzle G. This water is cut off in a series of thin sheets, about 0.2 inch thick, which are hurled at a velocity approaching 130 feet per second through collecting cones into the diffuser B. Each sheet of water carries with it a layer of air, the combined mixture being discharged back to the tank E, where the air is separated from the water by a series of baffles between the discharge and suction of the pump. Fig. 12 illustrates a test under these conditions showing that the pump is effectively removing air even at little above atmospheric pressure. It is interesting to note that with decreased vacuum the horse-power also decreases.

It has been said against the Leblanc pump that it absorbs more b.h.p. than its reciprocating rivals. This is undoubtedly true under normal conditions for both types when the plant has to maintain a fairly low vacuum, though under equal conditions of hot-well temperature it is not the case. It will be noticed that condenser builders who use reciprocating air-pumps usually advocate a weir in the bottom of their condensers to submerge 2 to 3 per cent. of the tubes, with the object of reducing the hot-well temperature to a point at which the air-pump can operate on favourable brake horse-power terms with the rotary pump (*see* Fig. 5, page 189). A moment's consideration of overall efficiency will expose the false economy of this.

With a 28.8-inch vacuum we know that the wet air-pump has a vacuum efficiency of about 96.5 per cent., and therefore requires a hot-well temperature about 80° F., which is frequently obtained

by means of the weir, while the Leblanc pump will work with 94° F., an advantage of 14° F. over the piston-pump. Now an increase of 14° F. on the feed-water temperature means an approximate gain on coal consumption of

$$100 \times \left\{ \frac{94 - 80}{1193 - 80} \right\} = 1.25 \text{ per cent.},$$

FIG. 13.

FIG. 14.

- A Absolute pressure in condenser with "wet piston" pump.
 B " " " " "Leblanc" " "
 C Power absorbed by "Leblanc" pump.
 D " " " " "wet piston" " "
 E Net gain in B.Th.U. obtained with "Leblanc" pump.
 F Percentage of coal consumption saved with "Leblanc" pump.

with steam at 150 lb. boiler pressure. From this gain must be deducted the percentage lost in extra power for driving the pump, leaving a balance in favour of the Leblanc of 1 per cent. on the overall efficiency. In addition there is the decreased cost of a smaller condenser due to the absence of submerged tubes, less circulating water and simplicity of design and operation, there being no valves or reciprocating parts.

FIG. 15.

Tests on various types of Rotary Pumps.

Barometer during test of rotary spray pump 29.47" Hg.

Temperature of water used 64° Fah. corresponding to a water vapour tension of 0.63" Hg. Theoretical vacuum 28.84" Hg.

Barometer during test of Westinghouse-Leblanc pump 29.78" Hg.

Temperature of water used 68° Fah. corresponding to a water vapour tension of 0.68" Hg. Theoretical vacuum 29.1" Hg.

Curve A Vacuum obtained with Westinghouse-Leblanc pump absorbing 40 B.M.P.

Curve B " " " Rotary spray " " 42 "

Curve C " " " Westinghouse-Leblanc " " 27.2 "

Curve D " " " Rotary multiple jets pump " " 40 "

Figs. 13 and 14 (page 200) show graphically the relative power absorbed by a Leblanc rotary air-pump, including rotary hot-well

FIG. 16.—*Diagram of Rotary Jet-Pump.*

Pressure is generated in the revolving chamber A, and a thin stream of water is projected across the chamber B, extraining air which is compressed in the passage D up to atmospheric pressure.

pump, delivering to 10 feet external head, compared with a reciprocating wet air-pump of good design. In Fig. 13 full load has been taken at 27 inches vacuum, and in Fig. 14 at 28 inches vacuum. The power has been given in B.Th.U. for the sake of

simplicity, an overall efficiency of 20 per cent. being assumed. The other curves give the respective total absolute pressures in the condenser, the total heat returned by the rotary pump in the hot-well in excess of that returned by the piston-pump, the net gain in B.Th.U. and the overall gain in coal consumption, assuming that live steam at 100 per cent. efficiency be used to raise the hot-well temperature before entering the economizer, in the case of the piston-pump.

In calculating the net gain in B.Th.U., allowance has been made below full load on the condenser for the decreased steam consumption for equal power outputs due to the higher vacuum maintained by the Leblanc pump; this decrease has been taken as 5 per cent. mean between 27 and 28 inches vacuum and 6 per cent. between 28 and 29 inches vacuum. In Fig. 15 (page 201) curve C shows a test on a Leblanc pump superimposed on the test of the rotary spray pump.* The pumps are of approximately the same capacity, and it will be seen that the power taken by the Leblanc is considerably less than the other, namely, 42 b.h.p. This variation in power is, of course, obtained by regulating the quantity of water passing through the pump to suit the load. If the controlling valve is left set for full load conditions, the power will, of course, remain constant at 27 b.h.p. The other curves, D and A, are for a less known (in this country) type of rotary pump, Fig. 16, which needs no detailed description, and for a Leblanc pump of approximately the same brake horse-power as the rotary spray pump.

Finally, the author would recommend that intending purchasers of air-pumps would do well to insist on makers guaranteeing the weight of air to be removed † (not expanded volume, which means nothing), the total power absorbed, including steam-jets at (say) 10 lb. per b.h.p. and the temperature of the condensed steam as

* *The Engineer*, 2 December 1910, page 594.

† Average 13 lb. of air per 22,000 lb. of steam for surface condensers in conjunction with turbines, and 36 lb. per hour for the same steam from engines. For jet condensers, the air in the water must be added, which may vary from 2.4 to 5 lb. per 10,000 gallons of water. See Fig. 25 (page 218).

delivered to the pump from the condenser, not as it leaves the hot-well pump, the real efficiency, of course, depending "on how little" heat is extracted from the condensed steam and imparted to the circulating water and lost therein.

(d) *Circulating Pumps*.—It is apparently thought by many people that there is little room for choice under this heading, that

FIG. 17.—*Test of Low-Lift Circulating Pump (Westinghouse-Rateau).*
Speed 480 revs. per min.

it is only necessary to fix the water quantity and head required and proceed to make or order a pump. This, unfortunately, is the view taken by large numbers of customers who do not avail themselves of expert advice. They are apt to look only at the initial outlay, and overlook running efficiency and reliability. A short study of the design of centrifugal pumps, however, will show that for any given set of conditions and design of pump, one set of characteristics

will give the best results. That is to say, the speed at which the pump is to work is fixed for a maximum efficiency. The first point to settle then, in the choice of a pump, is the most suitable speed having regard to all conditions of service. The next point is to consider if it is advisable to run this pump from the same motor or engine which is to drive the air-pump. With reciprocating air-pumps this is usually done with the help of gearing. This arrangement, however, is not usually liked, as even with the best design a certain amount of noise is unavoidable, which is most noticeable with a turbine installation. The best alternative is a rotary air-pump which, when possible, should be run at the same speed as the circulating pump, all gears being avoided.

When the cooling water has to be lifted against a very low head, the circulating pump unfortunately must be run relatively slowly, probably 480 r.p.m. to obtain good results. While it is possible to make a rotary air-pump to run at 480 r.p.m. even for very small capacities, it becomes expensive, and under these conditions it is better to use separate motors, the air and extraction pumps then being coupled together. Fig. 17 (page 204) shows a test on a Westinghouse-Rateau low-lift pump, designed for a low speed. When a cooling tower must be used and the head against the circulating pump is 30 feet or more, a high speed can be utilized and all the pumps direct coupled, it often being advantageous to drive them by a low-speed turbine as shown in Fig. 10 (page 196). Even with a high head the efficiency of the water pump will be relatively low, about 65 per cent. for a speed of 2,500 r.p.m., but the advantages of a self-contained unit, which does not depend on the main plant for power, usually more than compensates for this and the additional initial outlay involved.

(e) *Extraction Pumps.*—The desire for high efficiency in every detail can be carried to excess, and in the end defeat its own object. This in the past has often been the case with condensed steam extraction pumps for use with surface condensers. To save 1 to 2 horse-power, four or even six stage pumps have been built running at about 1,000 r.p.m. When it is considered, that if one of the

extraction pumps should fail, the whole plant will be seriously affected, if not shut down with some types of air-pump, it is obvious that simplicity of design and reliability are essential. The British Westinghouse Co. has developed a single-stage pump running at a high speed which gives a sufficiently good efficiency, from 40 per cent. to 60 per cent. according to size. A section is

FIG. 18.—*Diagram of Vertical Extraction Pump (Westinghouse).*

shown in Fig. 18 which is self-explanatory. These pumps can either be horizontal and direct coupled to the air-pump or, if the latter is coupled to the circulating pump and running at too low a speed, they can be driven by a small vertical motor, which gives an ideal arrangement, as the pump itself can be sunk well below the condenser floor level, ensuring a good head of water on the inlet side. Much trouble has been caused by not allowing sufficient head to overcome the pipe friction from the condenser and to give

the necessary velocity into the impeller. A sure symptom of this is intermittent discharge. Should an accident stop this pump, a Leblanc air-pump will take up its duty, and remove the condense with a very slight fall of vacuum.

(f) *Vacuum Traps*.—When it is necessary to have a condenser placed so that there is a fall in the exhaust-pipe to the engine or turbine, an efficient vacuum trap must be fitted to remove the water which collects and which would ultimately obstruct the free

FIG. 19.—*Diagram of Continuous Drain Vacuum and Steam Trap (Ogden).*

passage of the steam. A very simple and effective type is manufactured by Mr. John E. L. Ogden, and is illustrated in Fig. 19. It consists of a cylinder divided into two compartments B, B, and supported at the centre on trunnions A. The ends are connected to the vacuum space and the drainage water falls by gravity into each end alternately, causing the cylinder to tip from side to side, a simple valve C releasing the water when the slide-valve cuts off the vacuum connection.

V. *Jet-Condensers*.—(1) Jet-condensers may be used when the supply of cold water available, either from natural sources or from

a cooling system, is suitable for boiler-feed purposes. Even when this is not the case, but the water is of a quality which would rapidly "scale" or "pit" the tubes of a surface plant, jet plants can often be advantageously used in conjunction with a purifier for feed purposes. The power absorbed by the pumps is usually higher than for a surface plant, although the actual water quantity required is considerably less, the excess power being due to the water being extracted from the condenser body against the vacuum head (which is approximately 32 feet) in addition to any external pressure. The air-pump must also be larger to deal with the air liberated from the water, usually not less than 2 per cent. of its volume at atmospheric pressure.

(2) The earlier forms of jet-condensers were of the parallel flow type, in which the steam enters at the same end of the condenser as the cooling water. The air is extracted at the opposite end, and is obviously at the temperature of the discharge water. A marked improvement came with the introduction of the counter-current type, one form of which is illustrated in Fig. 20. The steam here enters near the bottom and the water at the top, whence it flows over a series of trays or through a series of holes in fine streams. The air is removed from the top of the shell where it is coldest, owing to contact with the entering cooling water through which it passes, and the size of the air-pump required is, therefore, smaller than in the parallel flow type.

(3) It is curious how improvements in any apparatus frequently revert to a modification of earlier designs. The Westinghouse-Leblanc multiple-jet condenser is an example of this. Turning to Fig. 21 (page 211), which illustrates this plant, we see that the water and steam enter at the top, but with several important improvements. The water does not flow by gravity through a number of small holes, which are liable to choke up, but is sucked in by the vacuum at high velocity through a number of nozzles of ample size, which have a spiral vane in the centre of the nozzle to give the water a rotary movement which effectively breaks it up to form as large a surface as possible.

The most important point, however, is the cone under the water-

FIG. 20.—*Diagram of Counter-Current Jet-Condenser.*

A.—Steam inlet.

B.—Water inlet.

C.—Water and condenser outlet to extraction pump or barometric leg.

D.—Water separator in air-pump suction.

E.—Air-pump suction.

R

injection nozzles, Fig. 21. The water passing through this at a high velocity gives the air a first compression into the chamber D. The disadvantage of this air being at the temperature of the discharge water is neutralized by the cooling effect of the Leblanc air-pump, which is referred to under the heading of "Auxiliaries" (page 191). The condensed steam and water are removed by the centrifugal pump C, which is on the same shaft as the air-pump, one motor or engine only being needed to work the plant. The pump C is specially designed to work against the high vacuum head of the condenser, and is so arranged that no air-lock can be formed. The external head on the extraction pump is normally from 0 to 35 feet, according to conditions, but in special cases even higher heads can be dealt with. The great advantage of this type is immediately apparent, as it can be placed directly below a turbine plant without intermediate or bend exhaust pipes, ensuring a straight flow for the steam and no loss of vacuum between the exhaust outlet and the condenser. The space occupied by this plant is much smaller than is required for any other similar type, owing to the compactness of the rotary pump.

The vacuum efficiency is not less than 98 per cent. for turbine work; the average terminal temperature difference between the theoretical temperature of the vacuum and the outlet water temperature is about 4° F., and in many cases this figure can be further reduced and sometimes eliminated, that is to say, the theoretical vacuum attained. The latter condition necessitates a very tight system and low-seal water temperature in the air-pump.

Where head room does not permit of the plant being placed immediately below the main unit, a side exhaust can usually be adopted, which arrangement was adopted by the Westinghouse Co. at the Metropolitan Railway Co.'s Power Station at Neasden. This plant will be eventually composed of five 5,000 kw. Westinghouse-Rateau impulse turbines, exhausting into five Leblanc jet-condensers. Each condenser is designed to deal with 70,000 lb. of exhaust steam per hour at a vacuum of 28.1 inches, the temperature of the injection water being 82° F., 9,000 gallons being circulated per

minute. Two of these sets are now running under full load conditions, and the guaranteed vacuum is easily maintained. The author thinks it may fairly be claimed that such results as these are the best that could be obtained under commercial working conditions.

A detail of design of great importance in low-level jet-condensers, particularly when working with reciprocating engines, is the vacuum-breaker, F, Fig. 21. Many serious breakdowns in the past have been due to complicated mechanism failing to act efficiently in emergency and allowing the cooling water to enter the low-pressure cylinders. In the Leblanc jet an extremely simple and effective breaker is employed which eliminates this risk. For large plants two or more breakers are fitted, each capable of destroying the vacuum in a few seconds. Should the pumps stop for any reason, the air-pump also acts as a vacuum-breaker, the air having a free entrance to the condenser through the air-pump discharge-diffuser.

It has been the author's experience that troubles have been hastily attributed to back-flooding of the condenser, which have afterwards been traced to more obscure causes. In several cases when the engine has been running on light load with a very high vacuum of, say, 28.5 inches when changing over to "atmosphere," knocking has occurred in the low-pressure cylinder due to rapid condensation caused by the low temperature of the cylinder walls. Great care should be also taken not to have a rise in the exhaust-pipe between the engine and the condenser, or if unavoidable, a large and efficient vacuum and low-pressure trap should be provided (*see* Auxiliaries, page 191).

VI. *Barometric Jet-Condensers.*—These are very similar to the low-level type, with the exception that the hot water and condensed steam are removed by means of a barometric leg instead of an extraction pump. The advantage is that the risk of back-flooding through failure of the extraction pump is eliminated. The disadvantages consist of greater expense, more room occupied and loss of vacuum between the condenser and main unit due to the

TABLE 3.

The Critical Point in a Jet-Condenser.

Basis—Water 60° F. at inlet and 18 ft. mano. suction head.

Vacuum in. Hg. (Bar. 30 in.)	Vacuum Head (ft. of H ₂ O) = h .	Mano. Suction Head (ft.) = h_1 .	Head causing Flow = $h - h_1$.	$\sqrt{h - h_1}$.	Inlet Water Temp.	Outlet Water Temp.	Vacuum Temp.	Ratio water steam	Water Quantity.	Steam Quantity.	Per cent. of Full Load on Condenser when Vacuum at Full Load is	
					° F.	° F.	° F.				26 in.	28.5 in.
29	32.9	18	14.9	3.86	60	75.1	79.1	70.4	1	0.014	25.5	54.6
28.5	32.3	18	14.3	3.78	60	87.8	91.8	37.8	0.98	0.026	49.5	100
28	31.7	18	13.7	3.7	60	97.2	101.2	28.1	0.96	0.034	65	131.5
27	30.6	18	12.6	3.55	60	111.1	115.1	20.4	0.92	0.045	85.5	173
26	29.5	18	11.5	3.39	60	121.5	125.5	16.75	0.88	0.053	100	202
25	28.3	18	10.3	3.21	60	129.8	133.8	14.7	0.83	0.057	108	217
24	27.2	18	9.2	3.04	60	136.8	140.8	13.3	0.79	0.059	113	228
23	26.1	18	8.1	2.85	60	142.9	146.9	12.25	0.74	0.06	114	232
22	24.9	18	6.9	2.63	60	148.3	152.3	11.4	0.68	0.059	113	228

abnormal length of exhaust-pipe necessary. This loss may vary between 0.25 and 1.0 of mercury. In the majority of cases a low-level jet is to be preferred. The only condition where a

FIG. 22.—Diagram of
Ejector Condenser.

- A.—Water inlet.
B.—Steam inlet.
C.—Natural head of water of
about 15 feet is neces-
sary to work this plant,
or a rotary pump to give
the necessary head.

FIG. 23.

Critical Point in Jet-Condenser.

barometric plant is advisable is when the cooling water is at a level which does not necessitate an injection pump.

VII. *Simple Jet-Condensers.*—For very small steam quantities, ejector condensers are frequently employed. Fig. 22 shows one

of the best forms of the older designs. A great objection to this type is its limited air capacity, which depends on the quantity of water passing through the condenser. To overcome this objection the Westinghouse-Leblanc simple jet-condenser, Fig. 24, was designed, working on the principle of the air-pump. This plant will remove air in the ratio of from 4 to 5 times the

FIG. 24.

Diagrammatic Sectional Arrangement of Simple-jet Condensing Plant (Westinghouse-Leblanc).

water volume, giving a vacuum efficiency nearly as high as the multiple jet.

Evaporative Condensers.—The author does not propose to deal with these, as they are quite unsuitable for the high vacua now in vogue, owing to the impossibility of keeping them reasonably air-tight for any length of time.

A very important condition in the working of a condenser which draws its cooling water by means of its vacuum is frequently overlooked by users, namely, the maximum overload which can be put on the condenser without it entirely shutting down. It is obvious that, as the vacuum falls, less water will be drawn in, the effect being cumulative, until enough does not pass to condense the steam, the vacuum entirely failing. Table 3 (page 213) shows how this may be calculated, and Fig. 23 (page 214) is a curve plotted from the Table. Arrangement can always be made so that the critical point is beyond the highest steam consumption possible, if the maker is consulted with that view. Many unexplained failures could probably be traced to this cause.

In conclusion, the author wishes to thank Mr. Young for his assistance in preparing the drawings and diagrams with which this Paper is illustrated.

The Paper is illustrated by Plate 5 and 24 Figs. in the letterpress, and is accompanied by 4 Appendixes with 1 Fig.

APPENDIX I.

Vacuum Efficiency.—This is the percentage of the theoretical vacuum corresponding to the temperature of the condensed steam discharged (as obtained from steam-tables), which is maintained at the inlet to the condenser.

Example :—

Hot-well temperature, 90.0° F.

Corresponding vacuum, 28.6 inches.

Vacuum at inlet, 28 inches = 98 per cent. of 28.6 inches.

This term must not be confused with condenser efficiency, which refers to the volume of water required to condense a given quantity of steam compared with the theoretical quantity necessary.

APPENDIX II.

Partial Air-Pressure.—Dalton's law of partial pressures of gases states that if two or more gases (such as air and steam) are enclosed in a vessel, the total pressure exerted on the walls will be equal to the sum of the pressures proper to each gas at the same temperature. Therefore, in a condenser the total pressure is the pressure of steam at the temperature of the mixture plus the pressure of such air as is present at the same temperature.

APPENDIX III.

Mean Temperature Difference.—For counter-current condensers the mean temperature difference (δ) is obtained from the formula :—

$$D = T_4 - T_1$$

$$B = T_3 - T_2$$

$$\delta = \frac{D - B}{\text{Log}_e \frac{D}{B}}$$

In practice it is sufficiently accurate to take $T_3 = T_4$, which gives a lower result for δ .

T_1 = inlet water temperature.

T_2 = outlet water temperature.

T_3 = inlet steam temperature at top of condenser.

T_4 = final temperature at bottom of condenser.

APPENDIX IV.

Weight of Air passing through a Nozzle.—The weight of air passing through a properly designed nozzle may be calculated as follows (Leblanc) :—

$$W = 384 \cdot 2 \sqrt{\frac{P}{T_B + 273}}.$$

$$V = 1 \cdot 125 \sqrt{T_{B1} + 273} \times F.$$

W = weight in grammes per cm² area of neck.

P = atmospheric pressure = 1 kg. per cm².

V = volume in litres at atmospheric pressure.

T_B = atmospheric temperature in ° C.

F = area of nozzle neck in cm².

FIG. 25.—Weight of Air passing through a properly shaped Nozzle.

In English notation :—

W = lb. weight per hour.

A = area of nozzle neck in square inches.

T_n = atmospheric temperature in ° C.

$$W = \frac{20330A}{\sqrt{T_B + 273}}.$$

(See Fig. 25.)

Discussion.

The PRESIDENT (Sir H. Frederick Donaldson) said he was sure the Members would desire to accord a very hearty vote of thanks to the author for his most interesting Paper. The old proverb said that "every hen thought her own chick the finest," and the Paper seemed to adopt this attitude with reference to the particular type of condenser, the consideration of which formed the body of the author's Paper. He thought that there were members present who would not be averse to take part in the discussion, and possibly advance arguments on behalf of other types of condenser and thus lead to advantage in the usefulness of the Paper as it stood.

The resolution of thanks was carried with acclamation.

Captain H. RIALL SANKEY (Member of Council), in opening the discussion, said that the title of the Paper, "Modern Condensing Systems," might have been more accurately put as "The Westinghouse-Leblanc air-pump compared with certain other air-pumps which, in the author's opinion, are less efficient." Further, the author had omitted from his Paper other kinds of air-pumps, which, as the President had already stated, might be considered to be equally as good as the Leblanc; therefore, it was to be hoped that those pumps would be described in the discussion, so that the title of the Paper might in reality be maintained.

The author stated (page 192) that the Leblanc was the first working rotary air-pump, a statement which was interesting to him (Captain Sankey) personally, because he was one of the first engineers to examine the pump when it was made at the French Westinghouse Works at Le Havre. Even at that time it was working well, although it had the defects to which the author had referred (page 195). The Leblanc air-pump was specially suitable when a very good vacuum was required, that is, when working with steam-turbines. When working with reciprocating engines, when a vacuum of about 26 to 27 inches was desirable, he thought a reciprocating air-pump would have the advantage. The author

(Captain H. Riall Sankey.)

stated (page 193) that it was claimed as an advantage by certain makers that the horse-power required was a constant at all speeds. The author disagreed with that statement, and he (Captain Sankey) upheld the author's opinion.

Reference was made (page 195) to the original faults of the Leblanc air-pump. One of those faults was corrected prior to his visit to Le Havre, and as it illustrated the action of the air-pump he would like to refer to it. He thought he was right in saying that the idea originally held was that it worked very much like the Krting pump, namely, by the attraction of the air to the water, whereas, as explained by the author, the action was due to a series of water-pistons following each other rapidly along the pipe, and that these pistons simply sliced off, so to say, "pieces of vacuum," and buried them in the water flowing through the discharge-pipe. Obviously for that action to be satisfactory, the film forming each piston must not be broken. Now the original pump-vanes were splayed, so that the water-film was extended horizontally, and naturally broke towards the end and the pistons became leaky. The vanes were then made parallel, and he was informed that 40 to 50 per cent. better results were thus obtained.* The space between succeeding water-pistons contained steam and air at the pressure of the condenser,† and as they moved along the pipe they closed on to each other by the condensation of the steam, but the pressure remained constant and at last only air was left between them. Finally, the air became entangled in the water and was drawn out into the exhaust-pipe, which was made of the shape shown in Fig. 11 (page 197). The shape of the discharge-pipe was such that there was a good vacuum at the small or pump end and atmospheric pressure at the large or discharge end.

He did not understand what the author meant by the statement on page 195, "that the Westinghouse-Leblanc pumps will remove much more air per b.h.p., under similar working conditions, than

* Fig. 26 (page 221) illustrates the action described.

† Or rather, as pointed out by the author in his reply, the pressure in the pipe immediately above the air-pump.

any other rotary pump. One peculiarity of that and of all similar rotary pumps is that the expanded volume removed increases as the absolute air-pressure decreases," because if the above was a correct description of the action of the pump, the volume of the "slices of vacuum" depended on the number of pump-vanes and their speed. It did not matter what the pressure was there;

FIG. 26.

FIG. 27.

FIG. 28.

theoretically, there might be even an absolute vacuum and yet the same volume would be abstracted; the weight of air abstracted, however, depended on its density and therefore on the vacuum in the condenser.

In the case of a reciprocating air-pump, when the piston was in the position A, Fig. 27, there must be a certain amount of air

(Captain H. Riall Sankey.)

remaining in the clearance slightly above atmospheric pressure, and when the piston moved in the direction shown by the arrow that air had to be expanded below the condenser pressure, say to the position B, before the admission-valves opened. Hence the effective volume of the pump was from B to C. Whatever the vacuum in the condenser, the weight of air in the clearance remained the same, but as the vacuum improved, the position B would be displaced towards C until finally at some particular vacuum the air-pump would not draw at all. He therefore thought the remark made by the author, that the contrary was the case with reciprocating pumps, was correct. He agreed with what the author said about specifying the weight of air (page 195).

To anyone interested in the subject of the Leblanc air-pump he would recommend "A Note on Condensation," by M. Leblanc, which was published in *Engineering*, on 28th August 1908. The whole of the theory would there be found explained in a very clear manner. Amongst other things, the reasoning and experiments which had led up to the design of the Leblanc air-pump, were discussed. For example, M. Leblanc calculated the weight of air that would be drawn along by a stream of water. The velocity of the air close up to the stream would be that of the stream itself, and the air would be drawn along at that velocity; but at a certain distance away from the stream there would be no velocity in the air, the velocity at intermediate points would be given by a curve like that shown in Fig. 28. The velocity of the air around the stream was represented by a sort of cone surrounding the stream of water going down. In this manner M. Leblanc calculated what the diameter of the water streams should be, in order to make an air-pump of the Körting type work at about $28\frac{1}{2}$ or 29 inches of vacuum, and he found that the diameter would have to be so small as to be impractical, that in reality it would be a fog. It was that conclusion that led him to devise the Leblanc air-pump.

The author gave a formula in Appendix IV (page 218) for determining the amount of air passing through a nozzle, and he (Captain Sankey) could fully recommend this formula. He had

experimented with this method at Le Havre where they had a series of very carefully shaped converging nozzles, so that the air would issue in parallel streams. Nozzles were prepared of 1, 2, up to 6 millimetres in diameter, and the diameter was measured with extreme care. The pressure at the inlet would be that of the atmosphere, say 14.7 lb. per square inch abs., the pressure at the outlet was that of the air-pump, say 1 or 2 lb. abs. As was well known, the velocity obtained in a nozzle of that kind was the critical velocity, or the velocity of sound, when the outlet pressure was less than 53 per cent. of the inlet pressure, namely, 14.7 in this case. So long, therefore, as the pressure in the air-pump was less than 14.7×0.53 , say 8 lb. per square inch, the critical velocity obtained in the nozzle, which in round figures was 1,000 feet per second, or as exactly given in Appendix IV, 313.5 metres per second. Therefore, knowing the diameter of the nozzle, the weight of air could be accurately calculated, so long as the outlet pressure was less than about 8 lb. per square inch. The formula in the Paper was based on these considerations. It would be of interest to point out that, so long as there was a pressure in the air-pump less than 8 lb., it was impossible to hear the working of it, simply because the velocity was equal that of sound, and the noise could not get back through the nozzle; but the very instant a pressure of 8 lb. was obtained, by admitting more air the noise of working was heard, and that was a definite proof that the critical velocity was the velocity of sound.

Mr. MARK ROBINSON (Member of Council) said that he had the pleasure of being with Captain Sankey during the investigation of the Leblanc condenser at Le Havre, and he had felt much interested in the design, though he was not fully a believer in the doctrine of slices of water cut off and serving as pistons to carry down what Captain Sankey had aptly described as "pieces of vacuum" between them. The rotating wheel flung the water down the converging channel or nozzle with sufficient force to ensure its making its way out against atmospheric pressure, and

(Mr. Mark Robinson.)

the Paper showed that everything depended on the velocity given to these fragments of water, and of course to the air entrained amongst them. The air had no choice but to go forward with the water, and go out with it at the other end.

In the barometric condenser the expulsion of the water against atmospheric pressure was due to gravity only, that is, to the weight of a lofty column of water, which overbalanced the difference of pressure between the atmosphere below and the vacuum at top. The downward velocity of the water played no part in the expulsion of the water, but a certain velocity was required for another purpose altogether, namely, to ensure that when the air became entrained in or mixed with the water, it should be carried downwards with it, and prevented from returning to the condenser by merely rising up again through the water. He remembered some interesting experiments carried out by Captain Sankey at Rugby some years ago, which proved that a downward velocity of more than 1 foot per second was unnecessary for this purpose. In practice the figure adopted was 2 feet, but 1 foot per second was in fact sufficient to carry down any bubbles of air which had once been fairly "entangled" in the water. Without having seen the experiment, no one would believe how helpless these bubbles were in the water. One foot per second sufficed to conquer the most powerful bubbles. The very little bubbles hardly attempted to rise at all, even in still water; bigger ones rose slowly, while even very big bubbles made but a poor fight of it, though the curious way in which, in rising, they turned themselves inside out, and took the shapes of parachutes, much as jelly fish would, was extremely interesting to watch. The experiment was carried out by forcing air, in varied quantities, into the bottom of a column of water contained in a long glass tube of large diameter, in which the rate and nature of the ascent of the bubbles (the water being still) could be clearly seen. With a barometric condenser, therefore, the rate of downward flow might be very moderate, for when once the air was in the water it had to go. The trouble was to get it into the water. In this connection he might refer to one of their experiences at Rugby, with which Captain Sankey was also

acquainted. The firm had installed a big Willans engine at a cotton mill, to which was applied a barometric condenser. A centrifugal pump forced water up to the requisite height for discharge into the condenser, and the discharge was so arranged as to give a diverging cone of water, spread out with considerable surface; this condensed the steam effectively. A reciprocating dry air-pump, driven by a belt, was fitted, drawing air by a pipe which was connected originally with the upper part of the condenser. This connection had to be shifted to a lower point in the condenser body, below and, in fact, inside the conical spray of water, and then the apparatus proved very efficient, and gave complete satisfaction. On one occasion the belt slipped off, and the pump stopped, but the attendant was greatly surprised to find that the vacuum suffered no change whatever. As a fact, the air had always been satisfactorily entrained by the sheet of water in the condenser, and carried with it into the stand-pipe; once there it could not get back, and it was duly carried away, without requiring any help from an air-pump. An analogous state of things existed in the Leblanc pump, only instead of the outside pressure being overcome by gravity, it was overcome by velocity. He greatly doubted, however, whether the water going down the cone-pipe really remained as a series of neat (and very thin) pistons, such as had been described, with the water-vapour and air shut in between them. Such pistons would be subject to a good deal of "ring friction," and they would probably break up before getting far down. However, they entangled the air somehow amongst them, and that was all that was wanted.

Perhaps the best mechanical analogue of the Leblanc pump, different as it appeared, was a *chain-pump*. Imagine a condenser, with a jet of water falling through it, placed on the top of the descending branch of a chain-pump. A certain quantity of water-vapour and air, as well as water, would be below each chain-bucket (or piston) as it entered the head of the stand-pipe, and would be carried down by it to the bottom, and so out to the atmosphere. The Leblanc water-pistons, which certainly existed in some form, even if not in the very neat shapes offered in the drawings, acted

(Mr. Mark Robinson.)

just as the chain-pistons would act. Air was shut in below the water-pistons and could not get back, and the pistons had velocity enough to shoot themselves out of the delivery-pipe at the other end, atmospheric pressure notwithstanding. No doubt the chain-pistons, though effective enough when they once started the down journey, would make a sad hole in the vacuum where they entered the condenser again, and they would of course be a mechanical impossibility. Such troubles were escaped by Leblanc by not bringing his pistons back for use over again, but by making new ones every time out of the water required for condensing the steam.

Many forms of ejector-condenser had been tried, but they all failed to deal with more than very small quantities of air. The area of the exposed surface of the water-jet had to be very small owing to the necessity for keeping the "rod" of water straight, and in a non-scattering form, by passing it through a succession of petticoat pipes and nozzles, with only very limited openings between them by which the steam could give access to the water. Hence it was not possible to obtain such intimate mixture of the air and the water as would enable any great quantity of air to be carried off. The essence of the Leblanc invention was the *intermittent discharge* of the water, with spaces between the succeeding portions of water—or "pistons," as the makers called them. It was a brilliant attempt, apparently successful, to combine the good points of the piston and the ejector types.

Captain H. RIALL SANKEY agreed with Mr. Robinson's remarks so long as there was only a moderate vacuum, but he disagreed if the vacuum was good. The point of M. Leblanc's theory was that the water streams were unable to draw a sufficient weight of air beyond a certain vacuum. Barometric condensers never gave a better vacuum than 26 or 27 inches at the most, unless fitted with air-pumps. For better vacua the water-pistons became necessary. Strictly, only two water-pistons were required; the air enclosed would then become entangled in the water in the discharge-pipe, and could not rise again against the stream of water,

as explained by Mr. Robinson. Mere attraction by water streams could not effect the entanglement of the air if the vacuum were good.

Mr. G. P. MAIR desired to refer to Fig. 17 (page 204), where the author showed the test figures of a centrifugal pump for circulating water through condensers. It would be noticed that there was a curve showing the relationship between the head and quantity, when running at constant speed. Taking the head of 20 feet, there were two separate quantities represented, namely, 200 gallons a minute and something like 2,800 gallons a minute. That meant, that the pump could work and give two separate discharges, or in other words, the pump was designed to work over the unstable portion of the characteristic curve. In practice, pump-makers designed pumps to work on the stable portion, as the unstable portion was obviously not suitable. The next point he wished to mention in connection with the curve was the variation in the power. He did not consider that the pump shown was designed in accordance with modern practice; most pump-makers—and there were now a great many of them—designed pumps which ran at constant power, under a varying head, which was a necessity in connection with condensing work. For instance, taking a circulating pump delivering through a surface condenser, drawing from and delivering to the same level and where the siphonic action could be taken full advantage of, the head was then simply due to the friction in the pipes and condenser. If a smaller amount of water was required, then the head fell; and, in accordance with the properties of this particular pump, if the head fell, the power as shown by the curve would rise considerably and might overload the motor.

He also desired to ask the author a question in connection with the extraction pump. The author had there shown a very simple form of centrifugal pump for drawing water out of condensers. He would like to know what head was necessary to force the water into the disk. He believed there were some pumps made to work on as low a head as about 1 foot.

Mr. JAMES GOLDER said he desired to ask the author a question on a practical point referred to on page 216, where the statement was made: "It is obvious that, as the vacuum falls, less water will be drawn in, the effect being cumulative, until enough does not pass to condense the steam, the vacuum entirely failing." The author considered that that was due to the failure of the injection-water. Some of the members had probably had experience with such plant on mixed-pressure turbines, where the condenser was not overloaded so much with the steam as with the sudden inrush of air. In the early days of mixed-pressure turbine manufacture, when the change-over gear did not operate when it should have done, the pressure in the supply-pipe would fall below atmosphere, thereby drawing in air. Sometimes the full blast of the winding engine, started after a long stop, would send a long pipe-line of air into the condenser, which would be very effectively dealt with by the reciprocating type of air-pump, the vacuum falling only momentarily from, say, 27 to 18 inches, and recovering quite readily. But when that occurred in the case of the Leblanc condensers, if the vacuum fell only 5 inches, the result usually meant that the plant would go over to atmosphere. This sometimes resulted in the complete shutting down. He would like to know whether the author had overcome such a difficulty, so that his pump was capable of dealing with sudden inrushes of large quantities of air, due to imperfections in the apparatus governing the mixed-pressure turbine.

Mr. SCANES, in reply, said that he wished first to thank the President for his remarks. On Captain Sankey's interesting account of his experiences at Le Havre (page 219) he had no criticism to make, since they not only ran parallel with his Paper but amplified it. He (Mr. Scanes) had purposely avoided giving more theoretical details in the Paper than were to be found there, because other writers had previously analysed the mathematics of the Leblanc system very carefully, notably Professor Leblanc, as mentioned by Captain Sankey. The Paper was chiefly intended to be a help to engineers requiring a practical knowledge of modern rotary

condensing systems. He agreed with Captain Sankey that the earlier pumps had splayed vanes, and in this connection he was reminded of an experience of his own recently. He had to go to Wales to dismantle a pump which was one of the first made. The plant had been in operation for five or six years. Of course, great improvements have been made during this time, and arrangements had been made with the owners of the condenser to replace the pump parts, with the modern design taking less power. When the old pumps (which had splayed vanes) were stripped for inspection, it was seen that the air-pump was polished throughout inside, and there were no signs of wear. It had never been touched, and he doubted whether any other type of pump would wear so well in ordinary colliery practice.

With regard to Captain Sankey's remarks that the steam and air between the sheets of water were at the pressure of the condenser, he would say that it was nearly, but not quite, the case; obviously they had to be at a lower absolute pressure. If the steam were entering at the top of the condenser at A, Fig. 4 (page 186), while air was being drawn into the pump through C, unless the vacuum at C was higher than at A, there would be no flow at all. Practically, it was a difference of only 2 or 3 millimetres, and, therefore, almost negligible.

Referring to Captain Sankey's remarks about expanded volume, it would be seen that in Fig. 12 (page 198), which was an actual test of a Westinghouse-Leblanc pump, the pounds of air removed were given on the bottom scale, and the cubic feet of expanded volume removed were given on the left hand vertical scale, the curve being marked "volume." Taking 20 lb. of air per hour, the expanded volume was something over 1,000 cubic feet per minute, but at 200 lb. of air per hour, the volume was reduced to 100 cubic feet: the volume was therefore ten times greater at the lighter absolute pressure. Neither weight nor volume was constant. He would refer to the comparison with reciprocating pumps again in his written communication (page 246).

Mr. Mark Robinson had made some very interesting remarks on barometric condensers, with which he largely agreed. The

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sister Westinghouse Company in America had recently installed three barometric jet-condensers of the Leblanc type on the edge of a cliff with a fairly high barometric leg. Each had an air-pump. When the plant was started, it was found that the air-pumps were not needed when working on a vacuum up to 27 inches. The customers thought they had been the victims of sharp practice, as air-pumps had been sold to them which were not required. He could not agree with Mr. Robinson's views that there were not actually any slices or sheets of water in the cones. He (Mr. Scanes) had carried out at Manchester some hundreds of tests on different cone formations, which had proved to his satisfaction that there were such sheets. If, for instance, the sheets did not touch equally on both sides of the cone on entering, there was a perceptible leakage back of air, which could be explained by the sudden insertion of a plug into a cylinder. Put in at an angle there would be a certain back leakage, but put in dead square, there would be no leakage. That explained one reason why some cone formations gave better results than others. He thought that interesting results might be obtained if one of the important Institutions could be induced to carry out experiments with a cinematograph with a pump using coloured water in glass cones. The speed could be reduced and the action watched.

Mr. Mair had referred (page 227) to the tests of the low-lift circulating pump shown in Fig. 17 (page 204). In reply, he desired to say that that pump had an efficiency of 80 per cent., a figure seldom exceeded in practice. He believed the plant had been working for two years, and, as it had given no trouble, it was evident that it had been designed on the right lines. It was impossible to have a more efficient pump, and his firm had installed many more, working on exactly the same principle. Mr. Mair was practically correct in regard to the extraction pump; this required about 1 foot head of water on the inlet side. With the ordinary type of pump, the water was drawn in owing to the rotor of the pump creating a vacuum in the pump casing, the flow being directly due to the difference from atmospheric pressure. When there was a vacuum in the condenser at A, Fig. 29, it was obvious that

the pump could not produce a higher vacuum at B; the only way to produce a flow in that pump was by having a static head, and this, with a properly designed pump, could be reduced to about 1 foot. There, were, however, many ways of getting over the difficulty. It was only necessary to put a small steam-ejector into the pump, as indicated; quite a small amount of steam would suffice to give the necessary pressure for the water to rise into the pump, Fig. 30. Such an injector was used both in marine and land work.

He regarded as imaginary Mr. Golder's difficulty with rotary pumps when used with mixed-pressure turbines (page 228). Some

FIG. 29.

FIG. 30.

half of the pumps his firm had installed were for use with mixed-pressure turbines, and the trouble referred to did not occur. Of course it was necessary to put in a pump of sufficient capacity to deal with the amount of air likely to be extracted. In Fig. 11 (page 197) a self-charging air-pump was shown, in which it was not possible for the vacuum to fail; if, for any reason, it fell, it would pick up again when the source of trouble was removed. Mr. Golder had referred to Fig. 23 (page 214), showing the critical point in a jet-condenser, but he thought that Mr. Golder had rather missed the point. If in a condenser the vacuum was reduced through air leakage, the effect would be the same as if it were

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reduced by an overload of steam—perhaps not quite so rapid, but very similar. The points of failure in both cases would be similar, and his firm had guarded against that by putting in an amply large pump. Should there still be any doubt, the arrangement shown in Fig. 11 (page 197) would entirely overcome it, as the plant would pick up from atmospheric pressure if necessary without alteration. [See page 246.]

Communications.

Mr. E. R. BRIGGS desired to draw attention to the curves showing experimental results by various authorities on the heat transmission from steam to water in surface condensers, shown in Fig. 3 (page 184). It was interesting to note that, with the exception of curves 5, 8 and 10—namely, Hagemann's, Allen's, and Orrok's—all the curves indicated that the rate of heat transmission varied as the cube root of the velocity of the water in the tubes. Hagemann's, Allen's, and Orrok's results showed that the heat transmission varied as the square root of the velocity. Hausbrand, in "Evaporating, Condensing and Cooling Apparatus," page 39, quoted Mollier's formula, calculated from Ser's experiment, as—

$$K = 3,300 \sqrt[3]{v} \text{ in metric units, which gave}$$

$$K = 453 \sqrt[3]{v} \text{ in English units.}$$

Joule's equation was given as:—

$$K = 1,750 \sqrt[3]{v} \text{ in metric units, or}$$

$$K = 240 \sqrt[3]{v} \text{ in English units.}$$

Both these figures were lower than the curves given by the author. Mollier also deduced from Hagemann's experiments the following expression for K .*

* t_a = steam temperature, t_a and t_e inlet and outlet water temperature.

$$K = 50 + \left[1,000 + 10 \left(t_d + \frac{t_a + t_e}{2} \right) \right] \sqrt{v} \text{ in metric units, or}$$

$$K = 5.63 + \left[72.6 + 0.626 \left(t_d + \frac{t_a + t_e}{2} \right) \right] \sqrt{v} \text{ in English units.}$$

A point in connection with heat transmission in surface condensers which did not appear to have received due attention, although it was recognized by many modern designers, was the speed of the steam across the tube. The advantage of this was expressed in the shape of a formula by Hausbrand (page 263), where he gave the value of the coefficient

$$K = 750 \sqrt[3]{V} \sqrt[3]{0.007 + v}.$$

Neglecting the value of the 0.007, as it was small compared with v , this might be expressed in English units as—

$$K = 57 \sqrt[3]{V} \sqrt[3]{v}.$$

Hausbrand stated that this formula was founded on observations made in actual practice, on large and small condensers of most varied forms. V denoted the velocity of the steam when it entered the condenser (initial velocity), and v the mean velocity of the cooling water. It was well to note that Hausbrand agreed with Joule and Ser in that he gave the coefficient of heat transmission as varying with the cube root of V , and not as the square root of V , as given by Hagemann. A glance at Fig. 3 (page 184) showed how widely the coefficient varied, for example, at 3 feet per second, it varied from 380 to 750. It was to be hoped that this matter would be further gone into in connection with the investigations on heat transmission by the Institution.

The author appeared to base his chief claim on the advantages of the Leblanc air-pump by disparaging other types. This was a pity, as the rotary pump had its advantages, which one would expect to be more clearly defined by a representative of the British Westinghouse Co. Exception must, however, be taken to some of the statements made by the author in connection with the augmenter plant. Special attention was drawn (page 191) to the fact that the augmenter absorbed from 1 to 1½ per cent. of the main steam consumption of the turbine at full load, and it was

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emphasized that this should be taken into account. As a matter of fact, the writer's firm (Messrs. Willans and Robinson, Ltd.), found that the consumption was lower than this, in many cases being as low as 0.6 per cent. Actual tests were made on the consumption of an augmenter for a plant dealing with 22,700 lb. of steam per hour, and it was found that the consumption of the augmenter varied from 140 to 150 lb. per hour. Taking the higher of these two figures, the consumption was 0.66 per cent. As a set-off against the consumption, the fact must not be overlooked that with an augmenter, the air-pump was much smaller than with a non-augmenter plant, consequently, the total power, including the augmenter steam, was less than the power required to drive an air-pump of a non-augmenter plant, speaking of reciprocating pumps in both cases. The cost was also greatly exaggerated; the reduction in size of the air-pump as a rule more than paid for the cost of the augmenter jet and the small surface condenser necessary.

With regard to the horse-power of the rotary air-pump, this was at least twice that required for a corresponding Edwards air-pump, and curves, showing very careful experiments on both pumps for equal duties, were given by Mr. Dexter of The Mirrlees Watson Co. in the Paper read before the Glasgow Technical College Scientific Society on 13th November 1909. If the increased power were taken into account, then the capitalized running and depreciation costs given by the author in Fig. 2 (page 183) would show an increase of quite 50 per cent. for the Leblanc pump as against the Edwards pump. This way of looking at the problem could not be over-emphasized, and very little consideration would show that the capitalized running costs of the Leblanc system amounted to a sum greatly in excess of the slightly extra initial cost for an augmenter plant and Edwards air-pump, as compared with a rotary pump.

There was another point in connection with this question, that is, the size of the surface condenser. Some two years ago, Messrs. Willans and Robinson, Ltd., quoted for a turbo plant complete with surface condenser, augmenter and Edwards air-pump. Before the order was definitely placed, it was decided to fit a Leblanc

FIG. 31.—For 5,750 square feet Willans Surface Condenser.

FIG. 32.—For 14,050 square feet Surface Condenser for Sydney Trams.

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air-pump in place of the Edwards pump and augmentor. The question was gone into very thoroughly with the Société Anonyme Westinghouse, who refused to guarantee the results with their air-pump, unless the surface of the condenser was increased by at

FIG. 33.—End View of one of the 3 sets of 7,000/750 Turbo-Alternators and Condensing Plant for Sydney Trams (White Bay).

least 10 per cent. Here was an actual concrete case, showing that the vacuum augmentor was able to sweep the tubes of air in such an efficient manner that the coefficient of heat transmission could be raised 10 per cent.

The comparison made in Fig. 10 (page 196) between the Leblanc plant and an augmenter plant was not a fair one, inasmuch as the author had shown a small water-seal tank below the basement floor level, which he stated was not now in use, as the arrangement shown in Fig. 11 (page 197) was substituted for it. A glance at this would show the large size of the tank in proportion to the air-pump. It would be noticed in Fig. 11 that a large overflow-pipe was shown connected to this tank. Owing to the importance of keeping the temperature of the water as low as possible, it was

FIG. 34.—Condenser Tube-plate Joint and Details of Ferrules.

necessary to change it continuously, the make-up being a very serious item. In some cases it was possible to couple the inlet and overflow from the tank to the circulating water-main, the overflow being led to the suction side of the circulating pump, and the inlet from some pressure part, the inlet on the tank being controlled by a ball-float or other equivalent. Unfortunately, it was not always possible to arrange the pipes in this way, and considerable difficulty arose in making suitable provision for the large quantity of cooling water required by the air-pump tank. In one example, separate

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coolers had to be provided, and in another case, an extra pump had to be added, in order to lift the water up to the pond of the cooling tower; the tank being in the engine-room basement, it was not possible to make a connection for the discharge-water at a point having a pressure low enough to cause the water to flow away from the tank.

The design of the condenser shown on Fig. 4 (page 186) was in accordance with the practice of most modern makers. Fig. 31 (page 235) showed the arrangement of a Willans surface condenser. This particular case was a 3-flow one, but a 4-flow one was more often used, in order to keep a higher velocity of water in the tubes. Fig. 32 showed another surface condenser having two inlet branches. This lent itself to a convenient arrangement with large turbines, in which the exhaust was taken from the turbine by means of two legs. This condenser was one of three recently supplied by Messrs. Willans and Robinson, to condense the steam from three 7,000-kw. Willans turbines supplied to the New South Wales Government. Fig. 33 (page 236) showed the arrangement of these sets, from which it would be seen that, with the double exhaust legs and the condensers placed at right angles to the turbines, an excellent distribution of steam was obtained without baffling. A small detail, but one of importance, was in well rounding the inlet to the ferrules, as shown on Fig. 34. This nozzling effect on the ferrules helped to reduce the friction of the water entering the tubes.

Mr. CHARLES ERITH wrote that the advantages of small steam-turbines for direct driving of auxiliaries were very great. Small steam-turbines were necessarily designed, primarily for reliability, and secondarily for low cost; and there was no valid reason for designing them so as to approach the main turbine in steam consumption per horse-power. The practice of making small steam-turbines to exhaust to the low-pressure stages of the main turbine could not be justified. In such cases, the latent heat of the small turbines was lost in the circulating water of the condenser, and the overall efficiency of the small turbine was only

about two-thirds the efficiency of the main turbine. The only sound practice was to take the exhaust of all such auxiliaries to a suitable open-type feed-water heater, condensing the exhaust at atmospheric temperature by the feed-water, and so recovering its latent and sensible heat. In this way, the thermal efficiency of pumps and other auxiliaries was necessarily many times greater than that of the main turbine with its condenser. Experience showed that even if saturated steam were used in simple, inexpensive, "fool-proof," small steam-turbines driving the various pumps direct, the feed-water heater was capable of returning to the boilers all the heat of the available exhaust. Under such conditions, the use of superheated steam for small turbines was rarely justified, and there was no occasion whatever for complication and added cost in these auxiliaries. The case was totally different from the main turbine, where high superheat, high vacuum, refinement of design, and consequent low steam consumption, were well worth paying for.

The advantage of taking power for pumps, etc., direct from boilers was undeniable. The substitution of high-speed rotary motion for low-speed reciprocating motion, and the absence of oil in the exhaust, would alone make the small turbines exhausting to the feed-heater desirable. The small turbine had all the simplicity of an electric motor for driving pumps, etc., and it was free from the risk of electrical leaks; it would stand the hardest usage, and in combination with a suitable feed-water heater and purifier, its efficiency was far in advance of the main turbine.

Mr. OWEN A. PRICE wrote that he would like to make some remarks from the point of view of the engineer responsible for the design and satisfactory operation of centrifugal extraction pumps. In referring to extraction pumps, the author said (page 206), "Much trouble has been caused by not allowing sufficient head to overcome the pipe friction from the condenser and to give the necessary velocity into the impeller." From the immediate context, it was evident that this remark was applied to pumps extracting from surface condensers, but anyone with much experience of centrifugal

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pump or condensing work knew it was equally applicable to pumps extracting water from low-level jet-condensers. The writer had heard of these "cavitation" or fluctuating discharge troubles in extraction pumps of nearly every well-known maker in the country. The pump was usually blamed, yet when properly understood, a centrifugal pump was so simple in operation that there was nothing that could be wrong with it. Anyhow, the pump engineer was usually called in to correct mistakes, and these remarks were offered as suggestions to engineers who might have to investigate troublesome cases.

As a general result of the writer's experience, he had been forced to the opinion that in many cases insufficient head on the pump suction, as stated by the author, had not been the real cause of the cavitation or irregular flow from the pump. The cause had been either in too small a connection from the pump to the condenser, thus bringing about too high a water velocity, or the shape and volume of the condenser bottom itself had been at fault. With regard to the amount of head usually allowed (that is, the distance from normal water-level in the condenser to the pump centre) some pump-makers were quite content with 3 or 4 feet, while others insisted on 10 feet, an amount which generally required the pump to be placed in a deep pit. In the writer's opinion, a high head above the pump was not essential, provided sufficient attention were paid to the following four points: (1) Water should rain down on a portion only of the surface of the bottom pond, leaving the other part of the surface free for escaping bubbles to the air-pump suction. (2) The surface of the water should be in communication with the part of the condenser subject to the highest vacuum, thus assisting the escape of bubbles. (3) Downward flow in the bottom pond should be very slow indeed. (4) A large volume of dead water should be in immediate communication with the pump suction.

Granted that these were the correct principles of design, the arrangement shown in Fig. 21 (page 211) would appear to be ideal for the satisfactory operation of the water extraction pumps. The writer would like to ask if the author had ever had cavitation

troubles in pumps so arranged, also what was the approximate working level of water in such a condenser, and how high this water must rise before the vacuum breaker was opened.

On the other hand, Fig. 20 (page 209), judged by the principles just laid down, illustrated an arrangement which, though very common, was very unsuitable for the most satisfactory working of the extraction pumps. Every case of a troublesome pump that the writer had come across had been either connected with such a condenser or with a surface condenser having some similar defects. In Fig. 20 the water rained down on to the whole surface of a small volume of water, which itself was either a much agitated vortex or was in considerable commotion; each drop of water trapped a globule of air or water vapour and carried it down beneath the surface, where it was drawn into more quickly moving water and was carried to the pump. (Possibly large gulps of air were carried down the central core of the vortex.) Also, the surface of the water was in communication with the part of the condenser subject to the lowest vacuum, and the pump drew from a quickly running stream. In fact, in Fig. 20 the conditions for the extraction pump were in direct contrast to those of Fig. 21 (page 211), and yet many engineers would expect the pumps themselves to behave similarly in both cases.

With the assistance of a gauge-glass on the delivery side of an extraction pump, the writer was once able to examine the state of the water in a troublesome plant provided with a condenser designed on the lines of Fig. 20. The result was enough to convince any engineer that the water would be very troublesome to pump. It was no exaggeration to say that the water simply contained myriads of tiny bubbles. These bubbles on the pressure side of the impeller ranged from $\frac{1}{8}$ to about $\frac{1}{16}$ inch in diameter, and the writer estimated there were about 500 bubbles to the cubic inch. There was a pressure of about 15 lb. at the pump delivery, and, as the pressure in the condenser was about minus 14 lb., these bubbles must have been about thirty times the volume when they entered the pump suction, and the remarkable thing really was that the pump dealt with the water at all. At the pump delivery, the

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(Mr. Owen A. Price.)

bubbles represented from $1\frac{1}{2}$ to 3 per cent. of the total volume delivered. Regarding the exact nature of these bubbles and the apparent presence of so much air, the writer would like to ask the author if he had any evidence that an appreciable amount of dissolved air separated from the injection water under the high vacuum present in these jet-condensers.

It was quite well known that bubbles considerably reduced the output of a centrifugal pump, hence a pump to deal with such

FIG. 35.—*Modified Sketch of Condenser illustrated in Fig. 20 (page 209).*

water must be designed for a larger output than was actually required for it. This, however, would mean an inefficient pump, and the real remedy was to correct the condenser bottom and the connection to the pump, on the lines of the principles laid down. The condenser of Fig. 20 was easily modified, as shown by the amended sketch, Fig. 35. In conclusion, he would like to point out, that the fact that a troublesome plant had been improved by increasing the head over the pump suction did not prove that the same plant would not have worked satisfactorily with the smaller

head under better conditions; the effect of increasing this head, was of course to compress the bubbles on the suction side and this must necessarily bring about an improvement. The writer did not notice any mention of condensing systems for dealing with fluctuating load; the correct design of a satisfactory extraction pump to run at constant speed and to adjust itself automatically to fluctuating load was a most interesting but difficult problem.

Mr. ROBERT ROYDS wrote that, in studying the diagrams showing the results of tests in Figs. 9, 12, 13, 14, 15, 17 and 23, he was struck by the fact that the author had not thought it worth while to show a single experimental point. In the writer's opinion, this was a particularly objectional method of representing the results of experiments, and was likely to give the impression that the curves had probably been obtained by a free exercise of the draughtsman's imagination. On reading through the text one might find additional grounds for suspicion in this direction as, for instance, in the statement on page 195, that "The latest designs of the Westinghouse-Leblanc pumps will remove much more air per b.h.p. under similar working conditions than any other rotary pump." This was too much like the statements to be found in abundance in makers' catalogues and would probably be valued as such, since no proof was given. The only result of such a statement would be that most of the important makers of air-pumps would feel it their bounden duty to contradict the statement, and possibly adduce the results of tests to support their own view. The same remarks would also apply to the statement on page 210 that "The space occupied by this plant is much smaller than is required for any other similar type, owing to the compactness of the rotary pump." Again, of what possible value to anyone was the statement on page 217 that "The weight of air passing through a properly designed nozzle may be calculated as follows"? Then followed a formula, but no statement as to how it was obtained, or by what methods, or for what conditions.

The curves of heat transmission shown in Fig. 3 (page 184) were somewhat misleading in that whilst each experimenter no doubt

(Mr. Robert Royds.)

worked as accurately as possible, differences of conditions arose, no account of which was taken in the diagrams in Fig. 3. Only three of the curves were obtained from condensers of reasonable dimensions, the others related to experiments made with comparatively small apparatus. The average rate of heat transmission in a surface condenser not only varied with the velocity of the circulating water, but also with the amount of steam allowed to enter the condenser. It was easy to realize that, if only a comparatively small quantity of steam were allowed to flow continuously to a condenser, the steam would be condensed on the first few tubes and the remainder of the tubes would be flooded with air. Although the rate of heat transmission would no doubt increase with the velocity of the water under such conditions, the absolute values would necessarily be small compared with those obtained with a larger steam supply and a corresponding smaller number of tubes flooded with air. The curves shown in Fig. 36 illustrated this point. These showed the results of experiments made by the writer on a small high-speed condenser consisting of two brass tubes, one inside the other, and enclosed in a cast-iron shell. The steam passed through the inner tube and then to the outside of the outer tube, whilst the water passed through the annular space $\frac{1}{16}$ -inch wide between the two brass tubes. The total tube-condensing surface was 2.5 square feet. The quantity of steam flowing into the condenser was adjusted so as to preserve a definite outlet temperature of the condensed steam and the air for each series of tests at the values marked on the curves, the steam condensing under atmospheric pressure conditions in every case. The inlet temperature of the water varied between different tests from about 49° F. to 59° F. These results clearly illustrated the effect of attempting to cool air and water of condensation, and, of course, this feature was fairly well known to condenser experts. There was hardly any doubt but that the discrepancies exhibited between the curves in Fig. 3 (page 184) could be explained in the light of the results shown in Fig. 36, but only a few of the experimenters represented on page 184 thought it worth while to publish the values of the temperatures of the condensed steam and air at the condenser outlet.

Referring to the cross-sectional view at the right-hand side of Fig. 4 (page 186), it would be noticed that the draughtsman had indicated what he imagined would be the direction of the flow of steam through the condenser. The resistance of the path round the left-hand side of the condenser shown in the cross-sectional view

FIG. 36.—*Steam Condensed at 212° F.*

would be much greater than at the centre, and it was doubtful whether there was any appreciable flow in the former path. At any rate there was a likelihood that the contents of the condenser near the left-hand side were stagnant and flooded with air. This condition would appear to be aggravated by the designer having left out some of the tubes near the centre portion at the top of the condenser.

Mr. SCANES wrote that, in further reference to Captain Sankey's remarks, he noticed that Captain Sankey agreed with his statement that the characteristic of reciprocating and rotary pumps were divergent, and he wished to amplify Fig. 27 (page 221) by tests on a Westinghouse-Leblanc and a modern direct-acting steam-driven wet and dry cylinder air-pump, arranged with a cooler for the seal water used in the "dry" cylinder. This latter pump was naturally the most efficient type of reciprocating pump to be obtained from a thermal point of view, as the wet air-pump worked approximately at the temperature due to the vacuum, but the dry cylinder worked at a much lower temperature and dealt with a maximum air quantity. The Leblanc pump was one supplied to the New Goch Mine of the General Mining and Finance Co., South Africa. It would be noticed, Fig. 37, that the tests on both pumps were reduced to a working hot-well temperature of 96° F., corresponding to a theoretical vacuum of 28.3 inches. It would also be noted that the steam consumption of the two pumps was about the same, varying of course with the steam conditions, which in the case of the Leblanc had been taken as 180 lb. per square inch boiler pressure, dry saturated, 5 lb. gauge back-pressure to feed-water heater.

For a vacuum of 28 inches (turbine) the Leblanc pump had a capacity of 20 lb. of air per hour, equivalent to 34,000 lb. of steam (*see foot-note on page 203*). The reciprocating pump could not reach 28 inches vacuum with the temperature given. With a vacuum of 27.75 inches, the reciprocating pump would remove 15 lb. of air, corresponding to 25,400 lb. of steam, against 34 lb. of air or 57,500 lb. of steam for the Leblanc pump. For engine work at 26.5 inches vacuum, the pumps were of equal capacity, but the balance was in favour of the Leblanc owing to its greater simplicity. Moreover, it was not usual to put in reciprocating pumps of such an expensive type for low vacua, with the result that the performance of the reciprocating pump would be worse, down to at least 25 inches vacuum. A vacuum below this point was seldom used in modern practice. He (Mr. Scanes) thought that these figures effectively disproved Captain Sankey's remark that the Leblanc pump was only useful for turbine work.

FIG. 37.—Comparison of Performance of Westinghouse-Leblanc Rotary Air-Pump with Wet and Dry Cylinder Reciprocating-Pump, with Intercooler.

(Mr. Scanes.)

He thanked Captain Sankey for pointing out that he had omitted to state that the air-nozzle formula only held good for a pressure in the air-pump of 0.53 of the atmospheric pressure. Of course, in practice, such a low vacuum was not required.

He had had time to consider Mr. Mair's remarks on the instability of the pump characteristic shown in Fig. 17 (page 204),

FIG. 38.

and he wished to explain that this pump had to work against a total head of 20 feet and deliver 2,500 gallons per minute. This head was made up of about 12 feet friction and 8 feet static head.

Referring to Fig. 38, it could be seen that the total head would vary with the water quantity, and that the actual head at any moment was always well below the characteristic head, which was

of course obtained by throttling by means of a valve. With 200 gallons of water a minute, a head of 20 feet could only be obtained by means of a sluice-valve and throttling.

Mr. Briggs's remarks on transmission (page 233) only confirmed his opinion expressed on page 190 as to the variation in recorded results. He quite agreed that steam velocity entered largely into the design of a condenser, and it was also necessary to keep this velocity as nearly as possible constant through the condenser from the inlet to the air suction. He noticed that Mr. Briggs thought that 1 to $1\frac{1}{2}$ per cent. of the main steam consumption was too high an allowance for a Parsons augmenter. The figure was quoted from a Paper by the inventor, and he understood that in most cases it was a fair one. But taking Mr. Briggs's figure of 0.66 per cent., he thought that this compared very unfavourably with the rotary pump, for example. Turning to Fig. 12 (page 198), under average conditions the absolute air-pressure was about 0.5 inch of mercury in the air-pump; it would be seen that the pump under this condition removed 40 lb. of air per hour, corresponding to 68,000 lb. of steam per hour (*see* foot-note, page 203). This represented about 5,000 kw. The air-pump took 34 b.h.p. (Fig. 12) and if 3 b.h.p. were allowed for the extraction pump, there was a total of 30.6 kw. (motor 90 per cent. efficiency) or 0.61 per cent. of the main output, which was less than the steam alone required with an augmenter. Accepting for the moment Mr. Briggs's statement that the rotary pump took twice the power of the reciprocating pump (which the author did not admit), the Edwards pump would require 18.5 b.h.p., representing 0.305 per cent. of the main output, and adding this to the augmenter (0.66 per cent.) we obtained 0.965 per cent. or 58 per cent. more power for the augmenter combination.

With regard to the relative horse-power of the two types, the author pointed out that he did not make any claim for the Mirrlees Watson Leblanc pump; his figures and remarks applied only to pumps of the British Westinghouse design. Regarding the amount of surface required with the Westinghouse-Leblanc pumps, he could assure Mr. Briggs that his firm had built a condenser to deal with 22 lb. of steam per square foot of surface with a

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vacuum of 28.5 inches and a water temperature of 60° F. He had found, however, that customers were nervous of installing such a small surface for land plants, and his firm had recently increased the size of the plants in making guarantees, to be "in the fashion," difficulty having been experienced in convincing customers of the absolute safety of small surfaces when properly designed.

He was pleased that Mr. Briggs pointed out that Fig. 10 (page 196) did not represent the latest practice in arrangements, owing to the seal-tank being placed below the floor level; however, this tank could usually be placed in the foundations of the turbine or partially under the swell of the condenser, when space was of importance. A very small tank only was necessary, say a capacity of 35 cubic feet. He thought the question of make-up water was greatly exaggerated; he did not know of any case where the water could not be removed without the aid of an additional pump, and he could not remember that his firm had ever supplied one. When necessary, the Leblanc pump could be made to discharge against a sufficient head to carry the water out of a basement. He agreed with Mr. Briggs that rounded inlets to condenser ferrules were of importance; the British Westinghouse Co. had always followed this principle.

He agreed generally with Mr. Erith's remarks.

Mr. Owen Price had sent a very interesting communication, with which the author agreed to some extent. Before the Leblanc type of low-level jet plant was introduced, makers had found great difficulty in obtaining satisfactory results due to fluctuations in the water discharge, and he (Mr. Scanes) had heard Mr. Dexter, of Messrs. Mirrlees Watson, say something similar in a Paper he had read recently at Leeds. At the same time, provided the pump was properly designed, a large head of water on the inlet side was unnecessary, 1 foot only being required above the pump. The author could not agree that air-bubbles should be helped to escape; in fact, one of the great advantages of the Leblanc jet was the capacity for removing air by the water extraction impeller, which greatly assisted the air-pump, and reduced its size. This type of plant did not suffer from cavitation, and it gave such a steady water-level in the condenser body that it was safe to allow the free surface to be

within 6 inches of the float operating the vacuum breaker. He agreed that Mr. Price's design was a great improvement on Fig. 20 (page 209); the latter being very unsuitable in the author's opinion, principally owing to the small bore of the discharge-pipe and consequent high water-velocity into the extraction-pump, which would certainly be consequently adversely affected by air-bubbles.

The Leblanc type was nevertheless better than either Fig. 20 or Fig. 35 (page 242), for the reasons already given. Moreover, this design was admirably adapted for fluctuating loads, as it would satisfactorily remove the water even with a vacuum approaching the theoretical, for the water temperature.

The author thought it a pity that Mr. Royds should have made such sweeping remarks (page 243), without first having carefully read the Paper. Mr. Royds objected to the curves in Figs. 9, 12, 13, 14, 17 and 23. At the top of Fig. 9 (page 194) it was stated that this was copied from *The Engineer*, and the author could scarcely be held responsible for the absence of test-points in the curves of other makers; he, however, admitted it would be better practice if all makers followed Mr. Royds's suggestions, but he, however, knew of no recent Paper on Air-Pumps in which it had been done. Fig. 12 (page 198) had now the test-points inserted, and it would be seen that the draughtsman had not much scope for his artistic abilities. The author could not see that any useful purpose would be served by introducing the *calculation* points in Figs. 13 and 14, as they necessarily fell on the curves. The same remarks applied to Fig. 15 as to Fig. 9. The writer did not propose to reprint Fig. 17, but he assured Mr. Royds that the curve was no more inaccurate than Fig. 12. If Mr. Royds would refer to page 216, he would see that Fig. 23 was plotted from Table 3, and the plotting could be checked. Regarding the air-nozzle formula, the author pointed out that the method of calculation was the same as for steam passing through a nozzle, and he referred Mr. Royds to Professor Leblanc's article in *Engineering*, 28th August 1903, and to any modern standard text-book on Steam-Engines.

Mr. Royds complained that most of the experiments shown in Fig. 3 (page 184) were carried out on small plants. The author would

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point out that he made the same remark on page 190, and he could not understand why Mr. Royds then proceeded to illustrate his remarks with experiments carried out on a surface of 2.5 square feet. Further, Mr. Royds's results were misleading for illustrating heat transmission, as the apparatus appeared to have combined the functions of a condenser and water-cooler, the latter effect being the last thing condenser-makers would desire, their principal aim being to keep the condensate at as high a temperature as possible. If Mr. Royds turned to Table 2 (page 188), he would see, moreover, that with a condensed steam temperature of 132° F. a transmission rate of 720 B.Th.U. was obtained, as against a figure between 200 and 400 B.Th.U. indicated on his curves for a water-velocity of approximately 5 feet per second, which was used in the East Indian condenser. It would, therefore, appear that his tests were another example of the misleading results of laboratory experiments. Mr. Royds's final remarks would appear to be to a certain extent justified, but actual practice had proved that this design of baffle gave the best possible results in operation.

In conclusion, the author regretted that makers of various types of rotary air-pumps had not attempted to refute his claim that the Westinghouse-Leblanc air-pump was the best; it would appear in consequence that, referring to the President's opening remarks that "every hen thought her own chick the finest," he might have added, "and in this instance with some justification."

MODERN CONDENSING SYSTEMS.

Plate 5.

Turbine Driven Circulating and Extraction Pumps, Pennalla Colliery.

(Westinghouse.)

Duty 82,000 lb. steam per hour, 27.5" vacuum with water at 72° F.