town gas at a low pressure of 56in. W.G., and is of the double flow design, each side having two 36in. fans in series, the gas being drawn in from both ends and discharged in the centre.

Provision is made in this case for an additional double-sided fan, making three in series, if at any time it is desired to increase the gas pressure.

The whole of the rotor impellers are constructed of steel, and all parts are machined, ensuring perfect running balance.

Figure 16 makes the design quite clear. The single impellers consist of two steel discs, kept apart by the impellers proper, or vanes, which are of aluminium. These vanes are curved as shown by the rivet heads on the two sides of the discs, which pass right through them.

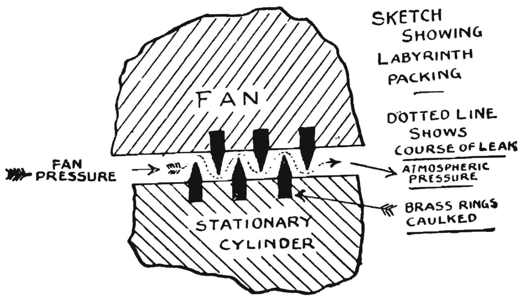

FIG. 17.

The fans of all these blowers are made out of mild steel boiler plate of 30-ton quality. The original thickness of the plate as ordered from the steel works is $\frac{3}{4}$in., and this is machined away to give a plate of a tapered section having a thickness of only an $\frac{1}{8}$in. at the periphery.

The flat or back plate of the fan is spigotted into the forged steel boss, while the front or shroud plate is flanged out at the eye, where it is provided with labyrinth packing mentioned elsewhere.

All these parts are highly polished to present the least resistance to the passage of the air.

To prevent air leaking back from the periphery to the suction or from neighbouring fans running in series, Parsons' system of labyrinth packing is adopted, a sketch of which is shown on Fig. 17.

These rings are reduced to a knife edge, and almost touch the stationary cylinder, as do the similar rings coming from the stationery cylinder to the shaft.

Any air leakage is wire drawn when passing each of these rings to such an extent, that the waste is reduced to a minimum, and the system possesses the great advantage of a gland having no actual rubbing surfaces.

When compression is appreciable in a series fan system, the width of the vanes in the final fans of the series, is reduced at the outlet.

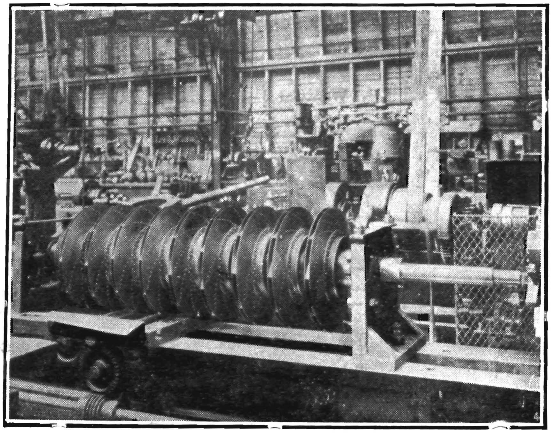

FIG. 18.

For example, Fig. 18 shows an experimental compressor spindle, containing a series of 9 fans, and this outfit was found capable of giving an output of 2,000 c. ft. of free air per minute, at a pressure of 25 pounds per square inch.

This figure shows how the width of discharge gradually decreases with increase of pressure.

The peripheral speed of these fans at the above duty was about 600 feet per second. This machine is referred to later.

Fig. 19 shows the blower cylinder cover for the spindle shown in Fig 16. The labyrinth packing is shown on the edge of the diaphragm dividing the fan chambers, and also on the edges of the dividing diaphragms.

The air, in passing from one impeller to the next in series, passes through a single series of stationary or guide vanes, placed opposite the discharge from each fan, and similar in section to those in the Parsons' Turbine, but of large section.

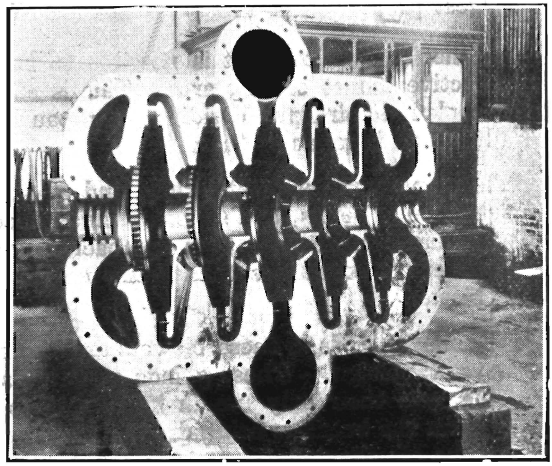

FIG. 19.

These vanes, with a minimum of eddy losses, direct the air to the next fan in series.

On the board is a sectional drawing of this blower, which the author thinks will be of interest to members, and he shall be pleased to explain any details that may not be clear to them. The glands in this machine are water packed to prevent leakage of gas out, or air leaking into the blower.

One of the chief mechanical advantages of the centrifugal blower is the possibility of very effective cooling during compression, and this is adopted for all pressures exceeding a few pounds per square inch.

In the early days of the reciprocating compressor plant, cooling water was injected during compression into the cylinder.

It was very effective, but troublesome, and in time, internal corrosion appears to have stopped its use, and now compressor manufacturers are content with water jackets and intercoolers. Water jackets, however, only have time to affect the air in close proximity with the cylinder walls—the largest proportion of the air is not directly affected.

In the Turbo blower, it is possible to have large and very effective jacket cooling area. This is effected by the water cooled diaphragms dividing neighbouring fans, and full advantage is taken of them.

The diaphragms are cast hollow, and water is circulated through them in series, and so is continually cooling the air during actual compression, thus approaching nearer to the ideal isothermal conditions than in the case of the reciprocating blower.

The shaft glands in these blowers are also of the standard labyrinth design. These glands are always on the suction side, and their only use in blowers is to prevent oil vapour from the bearings being drawn in and cementing the dust in the air channels. It is now the practice to overcome the slight suction of these glands by " packing " them with a little high pressure air from the blower outlet, which effectively overcomes the slight vacuum.

In the double flow design of blower illustrated in Fig. 36, there is no end thrust on the rotating spindle. In blowers for such pressures of 15/20 pounds per square inch pressure, the single flow design is generally adopted, which would result in an end pressure on the blower shaft.

This is entirely counteracted by fitting a rotating piston on the shaft of the proper area, which area is easily and accurately calculated. This piston is of exactly the same construction as that fitted to the parallel flow turbine for the same purpose, and on the periphery of which is the labyrinth packing.

All types of blowers are fitted with a thrust bearing, not in the ordinary acceptance of the word, but one applied merely to ensure the proper alignment of the rotating parts in respect to the cylinder.

At the Mortlake Gas Works, Sydney, there is a plant capable of delivering 17,000 c. ft. of town gas per minute, at 56in. W.G. pressure. The Gas Co. had either to put in a large new gas main for supplying Sydney, or adopt some means of delivering a very much larger quantity of gas through the existing pipes. They decided upon the latter as entailing only a fraction of the cost.

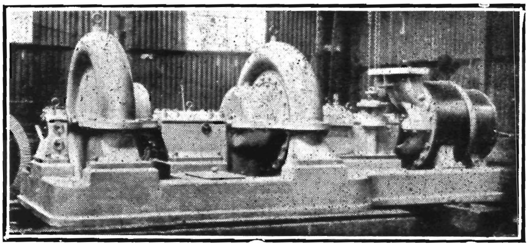

FIG. 20.

In blowing plants for purposes other than air, it should be mentioned that, other things being equal, the pressure obtained with a fan is proportional to the specific weight of the gas flowing through it.

It is now proposed to give particulars and results of tests of centrifugal blowing plants, which in the case of Messrs. Parsons, are generally in accordance with the foregoing remarks.

In the plants of Prof. Rateau's design, also centrifugal, the main features are alike, differing only in matters of detail.

Fig. 20 shows a Rateau low pressure blower with separate blower cylinders for series or parallel duty, designed to deliver 7,000 cubic feet of air at 4 lbs. pressure, or 13,000 cubic feet at 2½ lbs. pressure.

Fig. 21 shows the results of tests of a Rateau Turbine-driven blower for blast furnace work taken from a paper read by Professor Rateau.

The author regrets that he has been unable in the short time at his disposal, to get further particulars of the Rateau machine and later test figures.

It is often stated that it is difficult, without indicator cards, to ascertain the volume of air that a plant of this description is delivering, if the speed and opening are not the

TURBO-FAN OF THE BÉTHUNE MINES. (OCTOBER, 1905.)

		First Test.				Second Test.				Third Test.			
		Stages.				Stages.				Stages.			
		1st.	2nd.	3rd.	4th.	1st.	2nd.	3rd.	4th.	1st.	2nd.	3rd.	4th.
Absolute pressure at inlet ..	kg. per sq. cm.	1.033	1.59	2.70	4.62	1.033	1.46	2.54	4.43	1.033	1.61	2.88	5.18
Absolute pressure at inlet ..	lb. per sq. in.	15.18	23.37	38.69	66.44	15.18	21.31	37.83	65.12	15.18	23.66	42.33	76.14
Absolute pressure at discharge ..	kg. per sq. cm.	1.64	2.72	4.62	6.98	1.60	2.60	4.43	6.78	1.688	2.91	5.18	8.13
Absolute pressure at discharge ..	lb. per sq. in.	24.10	39.98	68.44	102.61	23.52	38.22	65.12	99.66	24.81	42.77	76.14	119.51
Speed	revs. per min.	4660	4660	4660	4660	5000	5000	4840	4840	5000	5000	4900	4900
Temperature of air at inlet ..	.. deg. C.	14	19.9	17.2	18.9	12.8	21	18	20.3	14.5	19	17.6	18.7
Temperature of air at inlet ..	.. deg. F.	57.2	67.8	63	66	55	69.8	64.4	68.5	58	66.2	63.6	65.8
Temperature of air at discharge	.. deg. C.	77.4	96.4	102.4	102	71.5	98	98	93.2	75.8	95.5	95	94.2
Temperature of air at discharge	.. deg. F.	171	206	216	215.6	160.7	208.4	208.4	199.6	168.5	203.9	203	201.5
Adiabatic rise in temperature ..	.. deg. C.	41	50	46	41	39	55	51	38	44	54	53.5	40.5
Adiabatic rise in temperature ..	.. deg. F.	106	122	114.8	105.8	102.2	131	123.8	100.4	111.2	129.2	128.3	[illegible]
Actual rise in temperature ..	.. deg. C.	63.4	76.5	85.2	83.1	58.7	77	80	72.9	61.3	76.6	77.6	75.5
Actual rise in temperature .	.. deg. F.	113.8	137.2	153	149.6	105.7	138.6	144	131	110.5	137.7	139.5	136.7
Air discharge at atmos. pressure	cb. m. per sec.	0.76				1.31				0.906			
Air discharge at atmos. pressure	cb. ft. per sec.	26				46				32			
Efficiency	.. per cent.	60.5	60.5	54	46.2	62.3	66.6	58.7	48.6	66.7	64.6	63.3	50

FIG. 21.

same as in the test conditions--which latter are generally for maximum duty.

It is, of course, of the utmost importance that a metallurgist should know exactly the volume passing through his furnace at any time. This difficulty is easily overcome by having pressure-volume curves taken at different speeds and openings.

From such curves it is an easy matter, after a direct reading of the speed and air pressure, to get at the exact volume being delivered.

The following are the results of some tests carried out with Messrs. Parsons' plants.

Fig. 22 gives the results of tests with a small blower running at the Electrolytic Works, Port Kembla, under varying conditions of speed, volume, and discharge pressures, the adiabatic efficiency being as high as 75%, and, compared to the isothermal ideal 66%, when de-

MACHINE No. 1237.

Built for The Electrolytic Smelting and Refining Co. of Australia.

This machine is to deal with 3,500 to 4,000 cubic feet of free air per minute to pressures varying between 8 and 15 lbs. The steam turbine in this case uses high-pressure steam. The compressor turbine is of the centrifugal type and is direct coupled to the steam turbine.

The following table gives the figures obtained on the official trial at the various pressures :—

RESULT OF TRIALS, 3RD JUNE, 1910.

Air Outlet Pressure, lbs. per sq. in.	14.75	12.7	9.9	9.8	7.75
Air Inlet Temp., °F.	72.5	75	75	75	76
Air Outlet Temp., °F	194	186	181	166	161
Jacket Water Inlet Temp., °F.	63.5	64.6	65.1	66.4	67.5
Jacket Water Outlet Temp., °F.	74	74	73.8	73.8	74.5
Speed, Revs. per min.	5325	5250	5250	4750	4750
Barometer	29.73	29.73	29.73	29.73	29.73
Vol. of Air at Blower Inlet, cu. ft. per min	3540	3940	4290	3470	3820
Temperature due to Adiabatic Compression	197.9	188.5	168.6	166	154
H.P. in Air, calculated Adiabatically	175	176	156.5	124.5	116
Total B.H.P.	242	234.8	234.3	174.5	172.5
Efficiency of Comp. $\frac{\text{A.H.P.}}{\text{B.H.P.}}$	72.5%	75%	67%	71.5%	67.5%
Gauge Pressure, lbs. sq. in.	153	150	150	156	150
Superheat, °F.	100	104	104	99	98
Vac at Cyl. Bar. 30"	25.98"	25.98"	25.98"	25.98"	25.98"
Steam Consumption, lbs. per hour	4208	4165	4165	3400	3390
Lbs. of Steam per Air H.P. (Adiabatic)	24	23.6	26.6	27.5	29.2
Lbs. of Steam per B.H.P.	17.4	17.75	17.75	19.5	19.7
H.P. required to Compress Air Isothermally	157	155	140	114	102
Efficiency of Comp. $\frac{\text{Iso Air H.P.}}{\text{B.H.P.}}$	65%	66%	60%	65%	59%

FIG. 22.

livering 3,940 cubic feet of free air at 12.7 pounds per sq. inch pressure.

The temperature of the jacket water has increased to 74° Fah., which is equivalent to 59.8 H.P. This amount added to the thermal power in the compressed air, is equivalent to 234.8 H.P., and this figure divided into the adiabatic horse power in the air, gives the adiabatic efficiency of the blower as 75% as per column 2.

As far as the steam consumption is concerned, this Turbine exhausts into a condensing plant, common for all the reciprocating steam plant in the station, and the vacuum, it will be noted, as a result, is low. A vacuum of say 28in., which could easily be attained under the conditions at this port, would reduce the steam consumption per adiabatic air

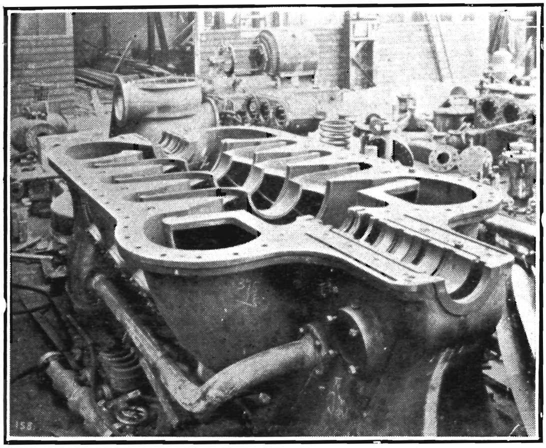

FIG. 23.

horse power to 21.3 lbs. and per B.H.P. hour to 16 lbs.—a good result for such a small turbine.

This blower supplies converter vessels 10ft. 6in. x 7ft. 6in., each vessel having eleven one inch tuyeres, holding on an average 9 tons of 50% copper matte, and producing in $2\frac{1}{2}$ hours when blowing 3,500 cubic feet per minute at 12 lbs. pressure, $4\frac{1}{2}$ tons of blister copper.

Fig. 23 shows a centrifugal blower cylinder of a plant of about the same capacity as that at Messrs. Hoskins, Lithgow, and for the same purpose. This machine delivers 26,000 c. ft. of free air per minute against a normal pressure of $8\frac{1}{2}$ lbs. per square inch, and a maximum emergency pressure of 10 lbs. per square inch. The speed of revolution ranges from 2,700 to 2,900 per minute.

Fig. 23 shows the air cylinder bottom of this machine, and will give some idea of the intricacy of the casting.

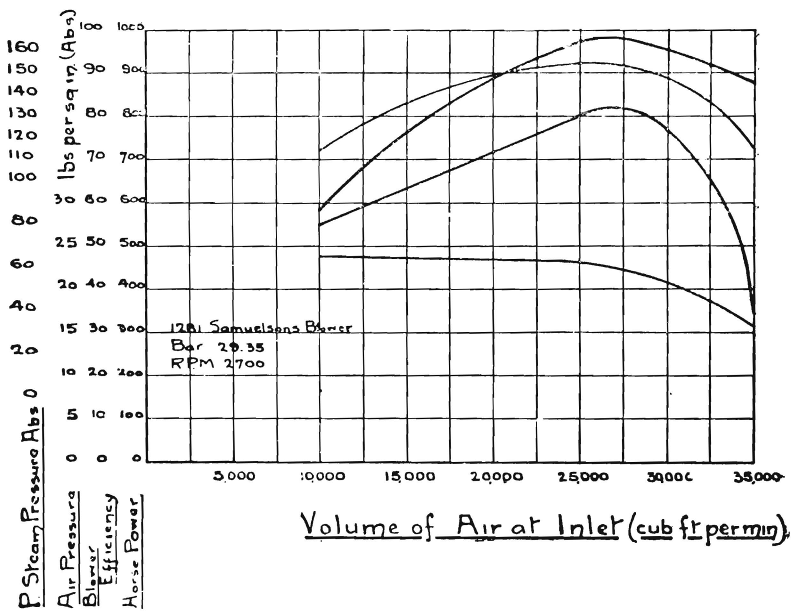

FIG. 24.

The three blank flanges on the side are cleaning doors for the water jacket spaces, and at the left hand end of this cylinder will be seen the blower shaft with its dummy piston in position, and a pair of the flanged shroud plates forming part of the fans.

Fig. 24 shows the characteristics of this machine with absolute air pressures. It will be noted that these centrifugal blowers can be made with a very flat characteristic curve, especially in machines of large capacity. The curves show the air pressure, the blower efficiency, the brake horse power, and the steam pressure in pounds absolute at the first row of turbine blades, all plotted to air volume.

It will be noticed that the efficiency reaches a maximum of $81\frac{1}{2}\%$. This efficiency was taken by the usual heat loss method, that is, a comparison between the increase of heat units as actually measured between the inlet and outlet of the blower with the number of heat units which it is calculated to require when being compressed adiabatically over the same range.

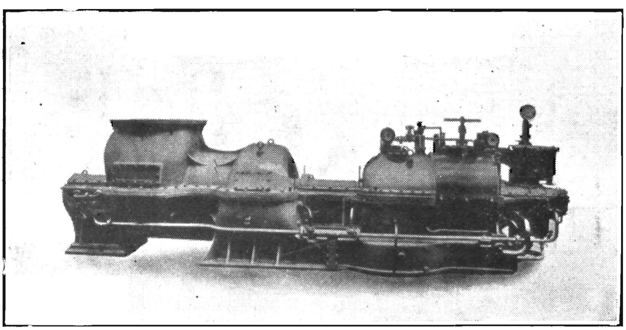

FIG. 25.

The heat units consumed by the water jackets are measured, but there are small errors introduced by radiation, etc., which reduce this apparent efficiency of $81\frac{1}{2}\%$ to an actual efficiency of 78%. As far as the author knows this is the highest figure ever attained for such plant, and as good, if not a great deal better, than that for the best make of reciprocating blowers, or centrifugal pumps.

On the board is a sectional drawing of this machine, showing four fans in series, each of 45 inches diameter.

Fig. 25 shows a very interesting combination. This blower is driven by exhaust steam and deals with 4,000 cubic feet of free air per minute against a pressure of 20 lbs. gauge. It is used as a "primary" compressor on a pair of reciprocating compressors which were originally designed for compressing from atmospheric pressure up to 80 lbs. per sq. inch. The reciprocating compressors are engine-driven, and they are run non-condensing, the exhaust steam from them being taken into the exhaust turbine of the Turbo plant, in order to compress the air to 20 lbs. pressure before it enters the low pressure air cylinders of the reciprocating plants.

In this way the combined plant gives double the output of free air that it did originally, and at the same time the reciprocating engines are run slower. On the board is a sectional drawing of this machine, which consists of seven fans in series which run at 5,000 r.p.m.

The principle of the "Exhaust Steam Turbine" is sufficiently well known.

Steam, in expanding from atmospheric pressure to that of a 28in. vacuum, is capable of doing practically the same amount of work as steam expanding from the ordinary boiler pressures of say 150 lbs. per sq. inch, to that of the atmosphere. The reciprocating engine as a rule is unable to appreciate vacua higher than about 25in, whilst the turbine can swallow the whole barometer, so that in many cases, where compressing plant has to be extended, and where condensing water is available for a good vacuum, the present example should be taken into consideration as a very cheap and effective means of duplicating existing compressor plants.

Fig. 26 gives the test figures of this exhaust steam plant. The exhaust steam from the reciprocating compressors at an absolute pressure of 15.9 lbs. per sq. inch, amounts to 9,040 pounds per hour, which, when exhausting to a high vacuum of 29.2in., barometer 30.0in., exerted a brake H.P. of 323, delivering 3,880 c. ft. of free air at a pressure of nearly 21 pounds per square inch, the adiabatic efficiency of the blower being 76.1%.

The exhaust steam was passed through a superheater to ensure dryness, and it will be noted that it is dry steam, having 2° Fah. superheat, a superheat figure which could not possibly affect the steam consumption results, the low pressure steam consumption per adiabatic A.H.P. amounting to 36.75 pounds and 28 pounds per B.H.P. hour.

MACHINE No. 1235.

Built for Messrs. The Burradon and Coxlodge Coal Co., Ltd.

This plant consists of a Turbo Centrifugal Blower direct coupled to an exhaust steam turbine. It is designed to deal with a normal duty of 4000 cubic feet of free air per minute, and to compress it to 20 lbs. per square inch. The exhaust steam is obtained from two reciprocating air compressors, and is slightly above atmospheric pressure. The turbo compressor is to be used for feeding the existing reciprocating air compressing plant with air at 25 lbs. pressure, which will have the effect of doubling their capacity, and, as the turbine uses exhaust steam, without any increase of steam consumption or boiler power

The following is a summary of the official trial :—

Air Inlet Temperature, °F.	72.1
Air Outlet Temperature °F	210
Air Outlet Pressure, lbs. per sq. in.	20.95
Jacket Water Inlet Temperature, °F	58.6
Jacket Water Outlet Temperature, °F.	71.6
Speed, Revolutions per minute	5025
Barometer	29.29"
Volume of Air at Blower Inlet, cubic feet per minute	3880
Temperature due to Adiabatic Compression	232
Horse power in Air, calculated Adiabatically	246.0
Total B.H.P.	323.0
Efficiency of Compressor, % $\frac{\text{A.H.P.}}{\text{B.H.P.}}$	76.1%
Exhaust Steam, Gauge Pressure, lbs. per sq.	1.2
Superheat, °F	2.0
Vacuum at Cylinder, Barometer 30"	28.51"
Steam Consumption, lbs. per hour	9040
Lbs. of Steam per Air H.P. (Adiabatic)	36.75
Lbs. of Steam per B.H.P.	28.0
Horse-power required to Compress Air Isothermally	220
Efficiency of Compressor, $\frac{\text{Iso Air H.P.}}{\text{B.H.P.}}$	68%

FIG. 26.

Comparing the results of this machine with Rankine's ideal, his perfect engine would, under the same steam conditions, consume 207 B.T.U. per H.P. The actual B.T.U. supplied per B.H.P. is 515, so that the overall thermal efficiency of this low pressure plant between temperatures amounts to 30.5%.