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SOME NOTES ON THE STEAM TURBINE.

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It is interesting at times, in the course of the development of an art, which is progressing in various places and by divers means, to collect as much information as possible concerning it, so that existing conditions may be compared with the past and some insight gained into the possibilities of the future. This fact, perhaps, justifies an attempt to contribute something concerning the steam turbine, even at a time when so much is being published in the current magazines and in the records of engineering societies. The matter in most of these articles is, however, largely made up of reports of single tests or series of tests on one machine and particulars of special turbines; while the object of this paper is simply to put their information in such a form that the comparisons referred to above may be easily made.

The turbine, the oldest type of steam engine, has always attracted more than an ordinary amount of attention, but the results of the epoch-making events of 1884 and 1889, when patents were awarded the Hon. Chas. Algernon Parsons and Dr. Gustaf De Laval respectively, have increased this interest to an almost unlimited degree. Trevithick, Pilbrow, Wilson, and possibly others, grasped the salient features of the modern turbine; but it needed modern workshop facilities, with the attendant accuracy of workmanship and attention to detail, to make the turbine a commercial success. And when we realize that it is not twenty years since the application was made for the first letters patent for the Parsons turbine, and that it was as late as 1891 that the first condensing turbine was produced, we cannot but wonder at the success it has achieved, and the world-wide interest it has excited. Previous to the last decade, conservative engineers in general undoubtedly looked askance at rotary engines and turbines; but it needs only a glance at the modern turbine to perceive mechanical features which must meet with approbation, while a closer examination cannot fail to call forth admiration for the ingenuity displayed and the persevering attention to detail shown in every part.

The modern parallel flow turbine is too well known to need a detailed description, but it will not be amiss to insert here the general principles of its chief types. They all depend for their action upon the conversion of the kinetic energy, caused by the expansion of the steam into work done on the rotating turbine shaft. In the De Laval turbine the expansion of the steam takes place in one or more nozzles before it reaches the turbine blades. In the Parsons this expansion takes place during the passage of the steam through the turbine, while in the Curtis turbine we have the application of both these principles. The De Laval, with its one row of blades, must, in order that the velocity of the steam leaving the blades be not excessive, have a very high peripheral velocity. In the Parsons and Curtis turbines, however, the employment of many rows of stationary and rotating vanes makes it possible to diminish the speed of the turbine shaft without reducing the efficiency.

As regards the velocity of the turbine blades, it is not difficult to find the one that is most efficient. Suppose V_1 be the absolute velocity in feet per second of the steam as it strikes the vanes, and V_2 the absolute velocity of the steam leaving the vanes, the greatest amount of energy that can be given to the turbine per pound of steam is $\frac{V_1^2 - V_2^2}{2g}$ foot pounds, and in order that this should be a maximum, V_2 must equal nothing. This is the case when the velocity of the vane is one-half the velocity of the impinging jet, and when the direction of the motion of the vane is parallel to that of the impinging and leaving jets.

This condition cannot be realized in steam turbines, though it may be noticed in passing that a close approximation to it is obtained in the case of the Pelton water wheel. But the velocities dealt with when working with steam are immensely greater than can ever be experienced with water. Thus, with a head of 200 feet, the velocity of the water entering the turbine could not exceed 113 feet per second. In the case of turbines of the De Laval type, however, where the steam expands in a diverging nozzle from initial pressure to condenser pressure, it is estimated that velocities of

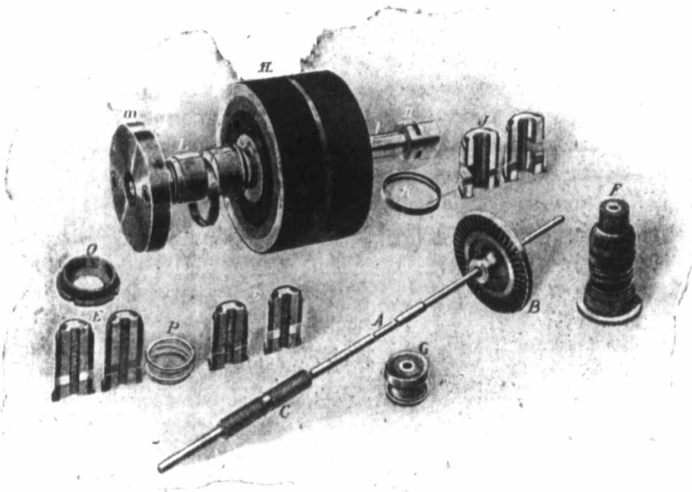

Fig. 1.

A.—Turbine shaft.

B.—Turbine wheel.

C.—Pinion.

D.—Pinion bearing, two parts.

E.—Pinion bearing, two parts.

F.—Wheel bearing with spring.

G.—Flexible bearing.

H.—Gear wheel.

I.—Gear wheel shaft.

J.—Gear wheel bearing, two parts.

K.—Oil ring.

L.—Gear wheel bearing in position.

M.—Coupling.

N.—Centrifugal governor.

O.—Gland adjusting nut.

P.—Adjusting nut for flexible bearing.

4,000 feet or more per second must be employed. This velocity can, of course, be regulated by the form of nozzle. But for economical working, as large a proportion as possible of the heat energy of the steam must be changed into the kinetic energy of the gas, the velocity of which, since it has a large specific volume, must be very high. It is stated above that for maximum efficiency the velocity of the vane should be one-half the velocity of the impinging jet. But a vane velocity of 2,000 feet per second would, of course, cause such centrifugal forces in the turbine wheel as no known material could safely bear. Turbines, with a single row of vanes using high pressure steam, must consequently run at a speed lower than the most efficient—a peripheral velocity of 1,000 feet per second being about the limit. The introduction of many rows of moving and stationary vanes at once overcomes this difficulty. The steam loses some of its velocity at each row, and so, on this principle, turbines have been made that run efficiently at speeds not much in excess of those of some high-speed reciprocating engines.

Figure 1 shows the working parts of a De Laval turbine, and Figure 2 is a sectional plan of the same. The steam enters the nozzle from the chamber D, where it is completely expanded, passes through the turbine bucket F to the exhaust chamber G. The important features are the diverging nozzle referred to above, the fact that there may be considerable clearance between the wheel, casing and nozzle, the flexible turbine shaft with its flexible bearing, the turbine wheel, made of forged nickle steel of increasing thickness from the periphery to the centre to resist centrifugal force ; above all, the high velocity of the turbine wheel, and the gear wheels required to reduce this velocity usually in the ratio of about ten to one. It is interesting to notice in passing some of the forces acting on this turbine. Suppose in a 10 H.P. turbine the speed of the turbine shaft is 24,000 revolutions per minute and the diameter of the turbine wheel 4.8 inches, the torque on the flexible spindle will be about 26 lbs. inches, and the total tooth pressure approximately 50 lbs.

Figure 3 shows a longitudinal section of the Parsons turbine, as manufactured by the Westinghouse Machine Company. In this the steam enters at A, passes through the stationary to the rotating blades through the high pressure, intermediate and low pressure cylinders, exhausting at B. As stated above, because of the many rows of stationary and rotating vanes and the reduction of speed with each pair of vanes, the speed of the Parson turbine can, by multiplying the vanes, be reduced to almost any amount. The end,

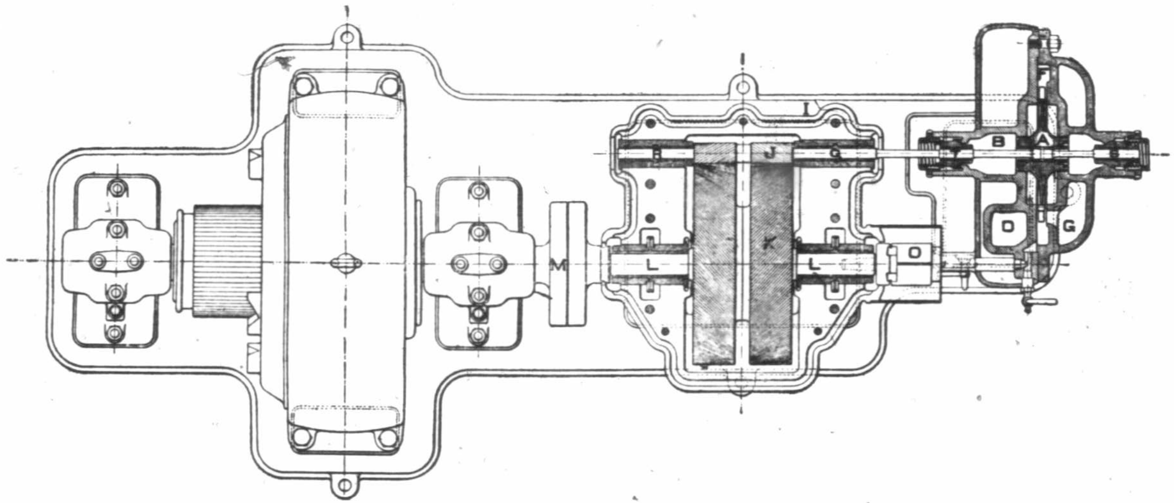

Fig. 2.

thrust is counterbalanced by three rotating pistons placed on the turbine shaft. Many of the details of the Parsons turbine are worthy of special study. The method of preventing leak past the balancing piston and the manner of getting the turbine shaft through the case are examples.

The Curtis turbine, unlike the two types described, has a vertical shaft in sizes above 500 kilowatts. It is perhaps best described in the maker's own words—

"Each stage or element of the Curtis turbine essentially consists of a group of expanding nozzle sections, which delivers steam to the first of a group of wheels or rings of buckets. Between the successive rings of buckets rows of stationary buckets, called 'intermediates,' are placed in the region opposite to the group of nozzles, the function of these intermediates being to reverse the motion of the steam received from one set of moving buckets, and to deliver it against the following set of moving buckets in an effective direction. The steam from one group of nozzles may thus be passed by the action of successive intermediates through several rows of moving buckets, the number of such rows associated with a single group of nozzles being governed by various mechanical and theoretical conditions. The group of nozzles imparts motion to a column of steam, most of the energy of the steam expansion being transformed into this motion. This motion is then fractionally abstracted by the passage of the steam through the successive rows of moving buckets.

"The above described circle of operations takes place in what is known as one stage of the Curtis Steam Turbine, and it is generally desirable to use two or more of such stages, in order that the expansive force of steam may be effectually utilized. Where a plurality of stages is used, the turbine conditions are so arranged that all the stages, under normal conditions, will perform approximately equal amounts of work. All the losses and efficiencies of one stage take the form of heat in the steam, and are therefore more or less available as motive force in the succeeding stages.

"In our first commercial machines we adopted two such stages, three or four rows of moving buckets being used in each stage. In some of our later machines we have adopted four stages, with two rows of moving buckets in each stage. Under certain other conditions, other numbers of stages and arrangements of buckets will doubtless be adopted."

Parsons turbines have been running in England for over twelve years, a sufficient length of time to permit of some idea being formed as to their durability.

At Newcastle, a Parsons machine ran for 36,000 hours without interrupted service, and at the end of the run there was no per-

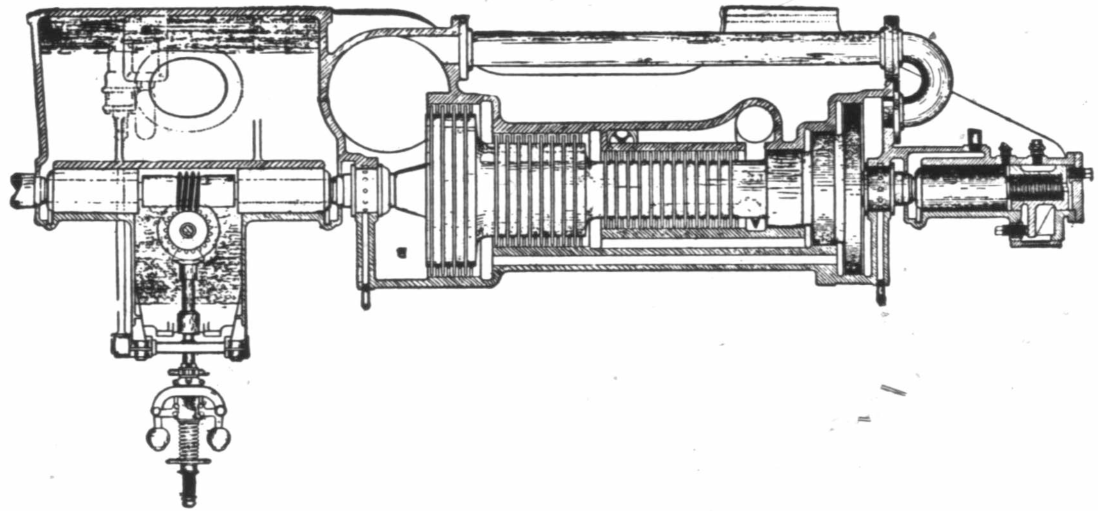

Fig. 3.

ceptible wear on the blades. The oldest Westinghouse-Parsons machine has been running for four years only. The repairs, however, during this time are said to have been light and of a minor character, with no perceptible wear on the blades.

At the present time there are in England from 600 to 800 turbine plants either actually installed or sold. These aggregate 200,000 H.P. The largest unit installed is 3,500 H.P. A special feature might be noted here in this connection, that many of the plants in which the first installations were made have added further turbine horse power. On the continent, Messrs. Brown, Boveri & Co., of Baden, Switzerland, manufacture the Parsons turbine. At the end of 1902 they had sold twenty plants, aggregating 29,000 H.P., the largest unit being 3,000 H.P. On this side of the Atlantic, the Westinghouse Machine Company of Pittsburg have made and have in service turbines to the amount of 6,500 kilowatts, while upwards of 5,000 kilowatts more have been shipped. The total turbine power already installed and in process of erection amounts to 110,000 kilowatts. Fifty-seven units will be in operation before the end of the next nine months.

There are unfortunately no figures at hand giving the total horse power of the De Laval turbines installed later than the year 1896, when it was said to be 23,000. Since that time some 13,000 H.P. has been installed in the United States alone.

Recent large contracts for and installation of Parsons and Westinghouse-Parsons turbines include among others the following:—

For the Philadelphia Rapid Transit Co., Philadelphia, Pa.	3 units	5,000 K.W. each
De Beers Consolidated Mines, Kimberley, South Africa	2 "	1,000 " "
Metropolitan District Ry., London (England)	8 "	5,000 " "
Metropolitan Railway Co., London (England)	3 "	3,500 " "
Cleveland, Elyria & Western Ry., Cleveland, Ohio	2 "	1,000 " "
West Penn. Ry. & Ltg. Co., Pittsburg, Pa.	3 "	1,000 " "
Rapid Transit Subway Construction Co., New York, N.Y.	3 "	1,250 " "
Penn. R.R. Long Island Power House	3 "	3,500 " "

For the Metropolitan Railway Co.'s plant, the turbines are constructed by the Parsons Steam Turbine Co., and are guaranteed to have a combined efficiency of 17 lbs. of steam per kilowatt hour, delivered at full load, and 20¼ lbs. of steam for each kilowatt hour, delivered at half load, the boiler pressure being 160 lbs. per square

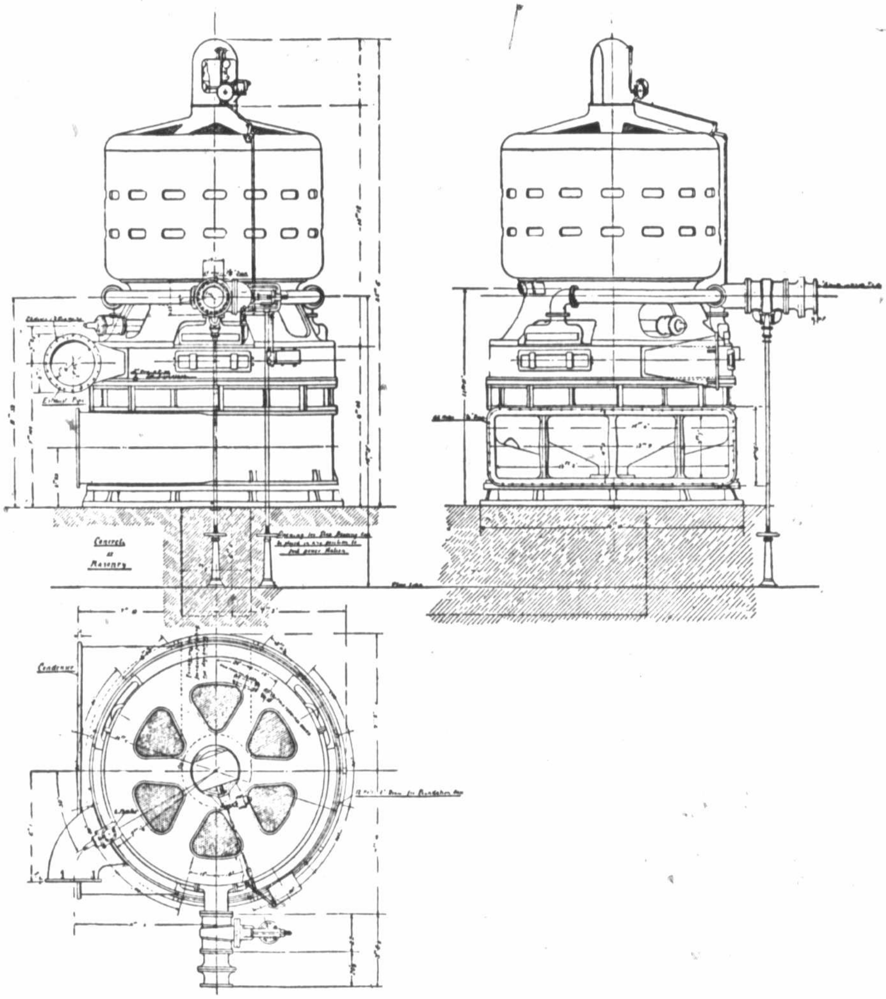

Fig. 4.

inch, with the steam superheated 180°F., 90% vacuum in the condenser.

Recent large contracts for and installations of Curtis turbines include—

Commonwealth Station, Chicago..	1 unit	5,000 K.W.
Lane Cotton Mills, New Orleans..	3 units	500 " each
Fulton Bag and Cotton Mills, Atlanta, Ga.	2 "	500 " "

In all, 200,000 H.P. of Curtis turbines are said to be under contract.

These figures show that the turbine must now be seriously considered a rival of the reciprocating engine. For, while it is true that in America it is used almost entirely for driving electric machinery, yet in England it has already been employed as a blowing machine (the air compressor being a counterpart of the turbine) for driving centrifugal pumps with high lifts, for ventilating purposes, and for marine work. In these various positions its steady growth is the best indication of its performance; and it need not be restricted to these alone, for it is excellently adapted to other services, where its high speed is not a positive disadvantage.

Comparing the steam turbine with the reciprocating engine, it is seen that the former has the following points of advantage:—The turbine has no valve gear, no vibration, is very light, and requires only sufficient foundation to bear its weight. It is the more simple of the two. The torque on the shaft is uniform, and there are no moving parts to be brought to rest and accelerated twice in every revolution. Condensation should be small, and full advantage is taken of low exhaust pressures. With three cylinder reciprocating engines, on the other hand, about the same results are obtained, with a 70%, 80% and 90% vacuum. The turbine is compact. It is impossible to give figures of general application, but it has been calculated that it requires about 80% of the floor space of the vertical engine of the same power and one-half the engine room capacity; about 40% of the floor space required by a horizontal engine and a correspondingly smaller amount of engine room capacity. It is a comparatively simple matter to erect and test at the maker's plant. It is admirably suited for the use of superheated steam. The steam consumption is about the same as that of the reciprocating engine when new, but since there are no rubbing parts, the wearing of which causes leakage, this consumption should be approximately constant throughout the life of the turbine. Its consumption varies less than that of the reciprocating engine over wide ranges of loading. No cylinder lubrication is required by the turbine, in consequence of which the exhaust is pure, a matter of considerable importance where water is dear, while difficulties that are unavoidable in extracting the oil are not encountered. Incidentally because of this, less work is required in the boiler room.

The turbine is, because of the uniformity of its driving force, specially suitable as a prime mover for such a system as alternators running in parallel. With it there is no tendency to produce those periodic fluctuations of speed which occur during every revolution of a reciprocating engine. The problem of speed regulation is consequently much simplified. To effect this, it is only necessary to supply a governor, which will keep down fluctuations of speed due to a sudden change of load, prevent surging, and give the drop in speed from no load to full load that is necessary for parallel operation.

It is worthy of notice that all Westinghouse-Parsons turbines installed are running alternators in parallel, and their operation in this connection is guaranteed to be satisfactory.

Looking at the disadvantages, it must be noticed that, with some types of turbines, it is difficult to get the shaft through the case. All, for reasons referred to above, must have excessive speeds that do not permit of belt drives. Where many rows of vanes are used, the clearances must be small, causing expense because of the accurate workmanship required. In the De Laval type, however, the clearances may be very considerable—are, in fact, from two to five millimeters.

It would be surprising if, for a time, the initial cost of turbines were much below that of reciprocating engines. The experimental work of years undertaken by the producers has undoubtedly involved great expenditure, and it is only right that they should receive remuneration in proportion to the incurred expense and to the risk involved. Speaking generally, the first cost of a turbine and its alternator will not differ much from that of a cross compound Corliss engine with its alternator of good manufacture. When, however, the cost of foundations, engine room capacity, and floor space is taken into account, any advantage in price is probably with the turbine. The cost of attendance, repairs, oil, etc., should be less in the case of a turbine than of a reciprocating engine.

As regards economy, it will be seen from the following tables of results of trials that the consumption is not much different from that of the best reciprocating engines when running at most efficient loads. At light loads the turbine ought, from its construction, to have an advantage over the reciprocating engine.

The turbine is admirably adapted to the use of superheated steam, the smaller fluid friction, due to the use of a rarer gas and the elimination of water, having a marked influence on the economy. Just how great a reduction in the consumption superheating will ultimately effect is not known, but the trials already made to determine this show very satisfactory results.

The best economy recorded in the annexed results of Parsons turbines occurs in the trials of a 1,000 kilowatt machine, built by Messrs. C. A. Parsons & Co. for the Newcastle and District Electric Lighting Company. The trials were conducted by Mr. Hunter, engineer for the Company. The vacuum was 26.5", the initial steam pressure 145 lbs. per square inch (gauge), and the superheat, 237°F. The lowest consumption recorded, 17.7 lbs. of steam per kilowatt hour, is equivalent to 13.2 lbs. per E.H.P. per hour, or expressed in B.T.U. is 268 B.T.U. per E.H.P. per minute. Taking the combined efficiency of turbine and dynamos (there were two placed tandemwise), as 83%, the calculated consumption of steam per I.H.P. per hour is 11.0 lbs. This corresponds to a thermal consumption of 223 B.T.U. per I.H.P. per minute. This same turbine using steam at 138 lbs. per square inch initial pressure (gauge), superheated 71°F., with 26 inches vacuum in the condenser, took 21.5 lbs. steam per kilowatt hour. This corresponds to 16 lbs. per E.H.P. per hour, or 300 B.T.U. per E.H.P. per minute. The advantages of superheating and the higher steam pressure are obvious.

The best results at hand of trials on a Westinghouse-Parsons machine show a consumption of 12.4 lbs. of steam per E.H.P. per hour. Taking the efficiency of the combined plant as above, the calculated steam per I.H.P. per hour is approximately 10.3 lbs. This corresponds to 246 B.T.U. per E.H.P. per hour. These trials were made on a 1,500 kilowatt machine, with an initial steam pressure of 150 lbs. (gauge), 140°F. superheat, and a vacuum of 28 inches.

The trials giving this very low consumption were made by the Westinghouse Machine Company, who vouch for their accuracy, and the results are substantiated by three distinct tests.

The trials for the 1,000 kilowatt turbo-alternator, built by Messrs. C. A. Parsons & Co. for the city of Elberfeld, were made by Mr. W. H. Lindley and Professors Schröter and Weber. A complete account of these trials, which were very exhaustive, may be found in the *Revue de Mécanique* for November, 1900. The best consumption recorded —19.43 lbs. per kilowatt per hour—is equivalent to 14.43 lbs. per E. H.P. per hour, or, assuming an efficiency of 83% for turbine and alternator, the calculated steam per I.H.P. per hous is 11.8 lbs. The steam pressure was 129 lbs. per (square) inch (gauge), with 18.4°F superheat and the vacuum 28.2 inches. The consumption expressed in B.T.U. is 270 B.T.U. per E.H.P. per minute and 264 B.T.U. per I.H.P. per minute, a résult agreeing very closely with the previous one.

In the trials made by Professor Ewing on the 500 kilowatt Parsons turbo-alternator, at the Cambridge Electric Supply Co.'s plant, with a steam pressure of 145 lbs. per square inch (gauge), vacuum 25.4 inches, the consumption was 24.4 lbs. per kilowatt per hour, corresponding to 18.2 lbs. per E.H.P. per hour, or 274 B.T.U. per E.H.P. per minute. With the same assumption as above, the calculated consumption per I.H.P. per hour is 15.1 lbs., corresponding to 225 B.T.U. per I.H.P. per minute. It is to be noted that in these trials the turbine was driving its own air and circulating pumps. The trials were made after the turbine had been in operation for one year. In the maker's tests, when the turbine was not running the air and circulating pumps, the consumption was 24.1 lbs. per kilowatt per hour—i.e., practically the same as after one year's operation.

The guaranteed efficiency of the turbines for the Metropolitan Railway Co.'s plant, referred to above—17 lbs. of steam per kilowatt hour—is equivalent to 12.7 lbs. per E.H.P. This corresponds to a consumption of about 10.5 lbs. per I.H.P. per hour, or 213 B.T.U. per I.H.P. per minute.

There is very little data at hand concerning the economy of the Curtis turbine. A test made by the makers on a 600 kilowatt machine shows a consumption of 19 lbs. of steam per kilowatt hour. The initial steam pressure being 140 lbs. gauge, the vacuum 28.5 inches and no superheat. This is equivalent to 14.2 lbs. per E.H.P. per hour, or, expressed in B.T.U., 269 B.T.U. per E.H.P. per minute.

In trials on a 10 H.P. De Laval turbine at Purdue University by Professor Goss, the best consumption recorded is 47.8 lbs. of steam per B.H.P. per hour, corresponding to 805 B.T.U. per B.H.P. per minute. The initial pressure of the steam was 138 lbs. per square inch (gauge), and the brake horse power of the turbine, 10.33.

In a trial on a 50 H.P. De Laval turbine by Professor Cedarblom, of the Royal Polytechnic College at Stockholm, Mr. Andersson, assistant at the Royal Polytechnic College at Stockholm, and Mr. Uhr, Inspector of the Board of Trade, Stockholm, a consumption of 19.78 lbs. of steam per B.H.P. per hour was obtained. The initial pressure was 122.3 lbs. per square inch (gauge), and the vacuum 26.4 inches. The thermal consumption is 352 B.T.U. per B.H.P. per minute.

In trials on a 300 H.P. De Laval turbine by Dean and Main, an average consumption for six trials is recorded of 14 lbs. per B.H.P. per hour, corresponding to 272 B.T.U. per B.H.P. per minute. The initial steam pressure was 207 lbs. per square inch (gauge), the vacuum 27.2 inches and the superheat 84°F.

All things considered, it looks as if the steam turbine had made a permanent position for itself as a prime mover, and that it only needs time to extend its sphere of action. Probably the steam engine is the prime mover for nine-tenths of all the power that is developed, and any improvement producing a greater economy in its operation will have a powerful commercial influence. The reciprocating steam engine has apparently nearly reached its limits of economy. Although the turbine is not a perfect heat engine, it probably will, when those improvements are applied that experience alone can suggest, prove itself a more efficient machine than the reciprocating engine, and will mark one more step in the advancement of steam engineering.

There is a wide difference between the heat engines at present in commercial use and the perfect heat engine; and, although the thermal efficiency of the turbine is not as great as some internal combustion engines, the turbine, as it stands to-day, is a very simple and highly efficient steam engine. It is peculiarly adapted to the performance of certain kinds of work, and there is every reason to expect that those bright prospects for the future, which are indicated at present, will be more than realized.

The writer desires to thank the De Laval Steam Turbine Company and the General Electric Company for catalogues and information and especially wishes to express his indebtedness to the Westinghouse Machine Company, who through Mr. Duff, have placed photographs, results of tests and much useful information at his disposal.

TABLE 1.—Trials of a 24 K. W. Parsons Turbo Alternator.

Number of Test.	R. P. M.	Pressure at Stop Valve, lbs. per sq. in. Gauge.	Vacuum in Turbine Cylinder. Inches of Mercury.	Electric Output. K. W.	Consumption of Steam. lbs. per hour.	Consumption of Steam. lbs. per K. W. per hour.	Superheat °F.	REMARKS.
1	4990	80	28.8	24.7	712	28.8		Barometer = 30″.
2	4630	77	29 0	11.8	400	33.9		
3	4570	74	29.1	5.15	235	45.6		
4	4900	78	26.0	23.8	798	33.5		
5	4780	79	0	19.7	1350	68.5		

TABLE 2.—Trials of a 50 K. W. Parsons Turbo-Alternator for the Blackpool Corporation.

1	5044	126	28.0	52.7	1480	28.0		Barometer = 30″.
2	4880	132	28.5	—	320	—		

TABLE 3.—Trials of two 100 K. W. Parsons Turbo-Dynamos (d. c.) for West Bromwich.

1	3500	129	27.8	123	3144	25.5	54	Barometer = 30″.
2	3520	134	27.7	122	2913	23.8	64	

TABLE 4 —Trials of a 500 K.W. Parsons Turbo-Alternator at the Works of the ambridge Electric Supply Company in January, 1901. Turbine in operation one year. Trials conducted by Prof. Ewing.

Number of Test.	R. P. M.	Pressure at Stop Valve, lbs. per sq. in. Gauge.	Vacuum in Turbine Cylinder. Inches of Mercury.	Electric Output. K.W.	Consumption of Steam.		Superheat. °F.	REMARKS.
					Lbs. per Hour.	Lbs. per K.W. per Hour.		
1	2670	148	25.7	518.0	12,970	25.0		Trials 1 to 5 Barometer 29.93".
2	2741	145	25.4	586.0	14,320	24.4		" A and B " 29.99".
3	2630	151	27.2	273.5	7,730	28.3		Turbine driving its own air and circulating pump.
4	2590	151	27.8	160.5	5,320	33.1		In test by makers, when new, machine showed load
5	2580	121	28.1	0	1,850	—		526.4 K.W. Consumption, 24 1 lbs. steam per K.W. hour.
A	2880	145	25.1	535.0	13,350	25.0		Conditions of steam pressure and vacuum about same as
B	2800	150	26.2	300.0	8,270	27.6		in tests at Cambridge.

TABLE 5.—Trials of a 1000 K.W. Parsons Turbo-Alternator, for Elberfeld Corporation, at the makers' works, in January 1900. Trials conducted by W. H. Lindley, Prof. Schröter, and Prof. Weber.

II.		129	28.2	1190.1	23,067	19.43	18.4	For full particulars of these exhaustive trials see Revue
I.		134	28.5	994.8	20,024	20.15	20.0	de Mécanique, November, 1900.
III.		139	28.4	745.3	165,924	22.31	14.4	
IV.		132	28.5	498.7	125,180	25.20	52.5	
V.		129	28.5	246.5	83,028	33 76	30.6	
VI.		132	28.7	0	40,568	—	24.0	No load alternator excited.
VII.		134	29.0	0	26,026	—	24.3	No load without excitation.

Table 6.—Trials of a 1,000 K. W. Parsons Turbine driving dynamos for Newcastle & District Electric Lighting Company's Power Station. Trials made by Mr. Hunter, Engineer to Newcastle & District Lighting Company.

No. of Test.	R. P. M.	Press at Stop Valve lbs. per sq. in. Gauge	Vacuum in Turbine Cylinder. Inches of mercury.	Electric Output, K. W.	Consumption of Steam.		Superheat °F.
					lbs., per hr.	lbs. per K. W. per hr.	
1	1690	138	26	1011.6	21,734	21.48	71
2	1680	140	26	909.0	18,610	20.47	86
3	1700	142	26.2	894.6	17,760	19.85	128
4	1660	144	26.3	890.7	17,020	19.1	132
5		144	26.3	882.9	17,171	19.43	135
6	1700	145	26.3	874.0	16,983	19.42	137
7		145	26.3	901.1	17,500	19.29	137
8	1680	146	26.3	896.7	17,302	19.11	136
9	1640	142	26.4	862.2	16,479	19.15	136
10		142	26.6	787.2	16,800	18.96	133
11	1640	146	26.3	944.2	17, 03	18.78	131
12	1660	140	26.3	986.6	18,434	18.52	142
13	1710	135	26.3	942.5	17,559	18.52	146
14	1710	135	26.4	942.6	17,460	18.52	182
15		140	26.5	878.1	15,857	18.06	195
16	1710	146	26.6	863.3	15,493	17.94	221
17		145	26.5	897.8	15,922	17.73	237

Barometer = 30″

TABLE 7.—Trials of a 10 H. P. De Laval Turbine. Trials conducted by Prof. Goss.

Number of Test.	R. P. M.	Press. at Stop Valve, lbs per sq. in. Gauge.	Vacuum in Condenser. Inches of mercury.	B. H. P.	Consumption of Steam. lbs. per hour.	Consumption of Steam. lbs. per B. H. P. per hour.	Superheat. F.	REMARKS.
1	2138	130	Exhaust at Atm. Press.	0.00	120.8	....	None	Trials 1 to 10 were made with four nozzles in operation.
2	2545	130		1.63	210.3	128.6		
3	2038	130		2.36	230.8	99.8		
4	2118	130		2.97	254.6	85.7		
5	1917	130		3.46	275.5	79.6		
6	2072	130		4.38	313.0	71.5		
7	2128	130		5.10	328.5	64.4		
8	2576	130		7.52	403.0	53.6		
9	2453	130		8.24	422.8	51.3		
10	2411	130		10.33	491.8	47.8		
11	2584	130		0.00	121.4	....		Trials 11 to 14 were made with three nozzles in operation.
12	2112	130		3.95	267.8	67.8		
13	2125	130		4.77	286.0	60.0		
14	2490	130		6.50	346.3	53.3		
15	2546	130		0.00	99.3	....		Trials 15 to 18 were made with two nozzles in operation.
16	2049	130		1.95	162.6	83.4		
17	1909	130		3.43	222.9	65.0		
18	2412	130		3.87	229.6	59.3		

TABLE 8.—Tests on a 300 H. P. De Laval Turbine. Conducted by Wilh. Jacobson:

No. of Test.	R. P. M.	Press at Stop Valve lbs. per sq. in Gauge.	Vacuum in Turbine Cylinder. Inches of mercury.	B. H. P.	Consumption of Steam. lbs. per hr.	Consumption of Steam. lbs. per B. H. P. per hr.	Superheat °F.	REMARKS.
1	754.6	151		342.1	5 270	15.4	20	Installed in a Paper and Cellulose Factory at Pötschmühle, Bohemia.
2	750.0	147		297.8	4,590	15.5	17	
3	760.0	143		252.6	3,960	15.7	20	
4	753.0	147		214.3	3 820	15.5	17	
5	750.0	154		165.0	2,640	16.0	10	
6	762.0	162		120.5	1,988	16.5	4	
7	762.0	162		74.5	1,476	19.8	4	
8	762.0	165		30.8	663	21.5	5	Exhaust at atmospheric pressure.