DIFFERENT APPLICATIONS OF STEAM-TURBINES.

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One of the most interesting developments in Mechanical Engineering during the years 1902 and 1903 was the greatly extended use of steam-turbines. This development is the result of efforts which have been made by various people to increase the efficiency of turbines to such a point that they may compete commercially with reciprocating engines. The author has been occupied for more than fifteen years in the study of rotary engines, and he has already published several papers upon the subject. He believes that, in presenting to the Institution of Mechanical Engineers an account of the practical result of his labours upon steam-turbines, he will contribute usefully to the study of this important question.

Although the principles which distinguish the different kinds of turbines are well known, it may be useful to recall briefly their distinctive characters in order that his own type of turbine may be more easily distinguished. In common with all other steam-engines, turbines transform into mechanical work the energy given out by steam during its expansion from the initial pressure of admission to the pressure at the exhaust. But whilst reciprocating engines effect this transformation of energy by means of variation in pressure of the steam, turbines can effect this transformation both by means

of the pressure and by means of the velocity of the steam while expanding. The employment of the velocity only in each moving wheel characterises the action or impulsion turbines, among which may be cited the De Laval and Curtis turbines, as well as that designed by the author; whilst the simultaneous employment of the velocity, and partial use of the pressure characterise the re-action turbine, of which the best known type is that of Parsons. Whatever may be the method in which the steam acts in the turbine, the chief problem consists in the employment, with good conditions of efficiency, of the very great velocities attained by the steam in expanding. When the expansion takes place in one stage, as in turbines with a single wheel, then the velocity of flow reaches, as is well known in a condensing engine, a value which is usually above 3,600 feet per second. But in order to obtain the maximum efficiency, the moving part of the machine should have a relative velocity which is approximately half that of the steam. As it is practically impossible to construct turbine wheels suitable for running with a peripheral velocity above 1,200 feet per second, the efficiency of turbines with a single wheel is necessarily low, this being due chiefly to the necessity for the employment of diverging inlet nozzles which give rise to great losses of energy by friction and eddying. On the other hand, angular velocities which correspond to these peripheral speeds prevent the direct driving of dynamos, and render necessary reduction gears of special and costly construction, which however cannot be protected from excessive wear, and are exposed to accidental breakage.

A consideration of these circumstances has induced inventors to divide the expansion of the steam into successive stages, and thus to produce turbines with multiple wheels, which are nothing but a series of simple turbines mounted upon the same shaft and driven successively by the same current of steam. This design of multiple turbines is by no means novel. It will be sufficient to mention the name of Tournaire, a French mining engineer, whose theoretical description to the Academy of Science in 1853 of a re-action turbine with multiple wheels is surprising when the description is compared with the Parsons turbine brought into use 30 years later.

Every simple turbine may be designed either as an impulse turbine or as a re-action turbine. In the former kind the fall of pressure under which the simple turbine works takes place solely in the distributor, whilst in the latter type the fall of pressure takes place, not only in the distributor, but also in the moving wheel, so that in the latter type there is a higher pressure in the guide-vanes than at the exit. Fig. 1 (page 740) represents the guide-vanes and the moving vanes of an impulse turbine, in which it is supposed that the fall of pressure is small enough at the vanes to render unnecessary the use of diverging nozzles as in the De Laval turbine. In the illustration the triangles of velocities are given at the entrance OAB, and at the exit OBC of the moving wheel. Similarly in Fig. 2 (page 740) are shown the guide-vanes and the moving vanes of a re-action turbine, and the triangles of velocities at the entrance OAB and at the exit OBC of the moving wheel are shown in the Fig. It is obvious from these illustrations that, by the mere inspection of the moving vanes, it is possible to distinguish between an action turbine and a re-action turbine. In the former type the vanes appear in transverse section in the form of an arc of a circle with entrance and outlet angles practically the same, whilst in the re-action turbine the cross section of the vanes is in the form of a parabolic arc, the entrance angle being more or less approximately 90° whilst the outlet angle is generally between 20° and 30° . Generally the vanes are increased in thickness in the middle, so that the steam spaces between them have approximately a constant transverse section for impulse turbines or decreasing section for re-action turbines. If however they are made of plate they may have a constant thickness as in 2, Fig. 1. In re-action turbines it is important that the increase in thickness should be suitably calculated, otherwise there will be an important loss of efficiency. It is otherwise with the impulse turbine, in which the efficiency is very little affected whether the vanes have a constant or variable thickness. In a given turbine with multiple wheels the different simple turbines may be all designed in the same way, or some may be designed in one way and some in another way to produce a hybrid system.

Drum Turbines and Multicellular Turbines.—From another point of view, drum turbines may be distinguished from multicellular turbines; in the former type, of which the Parsons turbine is the best known example, the moving vanes are fixed upon the periphery of a cylindrical drum; in the latter they are fixed to the periphery of wheels more or less flat and separated from each other by diaphragms which divide the interior of the turbine into cells. The author's turbine belongs to the latter type.

Fig. 3 (page 742) is a diagram of a drum turbine, and Fig. 4 a diagram of a multicellular turbine. These two types have different properties, so far as losses of steam are concerned. In the multicellular action-turbine leakage of steam can only take place through the play *c*, Fig. 4, which exists between the shaft and the ring of the fixed diaphragm which surrounds this shaft. In the re-action drum turbine, on the contrary, leakage may take place: 1st, around the moving wheels in *a*, Fig. 3 (by reason of the re-action); and 2nd, in *b* through the play between the guide-vanes and the moving drum. The first leakage can be suppressed, if desired, by making the wheels work by impulsion instead of by re-action, but then the second leakage between the fixed vanes and the moving wheel attains its maximum. It is because of this leakage at the periphery of the drum that turbines of the Parsons type require the most accurate workmanship. It is easy to understand that a play of some tenths of a millimetre is sufficient, at least upon the high pressure side, to produce a cross-sectional area of the leakage passages equal to that of the admission passages between the distributing vanes; if by specially exact workmanship the cross section of the leakage passages is made very small, it is to be feared that after some years of use the wear of the parts, either by friction of the moving parts against the fixed parts or by the action of the steam, will increase the cross section of the leakage passages and so cause a notable decrease in the efficiency of the machine.

In multicellular turbines the leakage, being confined to the periphery of the shaft, is reduced in proportion to the radii of the shaft and the drum; and moreover, friction being less to be feared at a place where the relative velocities are smaller, the play may be

Escape of Steam and Longitudinal Thrust in Turbines.

FIG. 4.—Multicellular Turbine (Rateau).

FIG. 3.

Re-action Drum Turbine (Parsons).

reduced to the absolute minimum. In practice, this play is not given any precise value. The machine is built with practically no play round the shaft, and when started the machine itself makes sufficient play to turn without touching the internal rings of the diaphragms.

Friction of the Moving Parts on the Steam.—It would appear probable that multicellular turbines produce relatively large losses, by friction of the moving parts upon the steam owing to the large surface of the moving wheels. This is however not the case. From tests which the author has made to determine the law of friction, and also from the results of experiments made upon complete machines, it has been found that frictional losses represent only from 2 to 4 per cent. of the normal power developed by the machine. These figures apply to turbines with an output exceeding 500 horse-power, and are comparable with those obtained in drum turbines, in which the friction of the moving vanes themselves must be added to that of the cylindrical surface of the drum and of the balancing pistons.

Turbines with Groups of Wheels.—Another type of turbine which may really be considered as belonging to the impulse class is being built upon a large scale, and is known under the name of Curtis. In this type, instead of using upon a single moving wheel the velocity of discharge of the steam leaving the distributor, it is used upon several wheels arranged in series in order to diminish the velocity gradually. The principle upon which the Curtis turbine is designed has been described in 1890 by Mr. Mortier in a discussion which took place after a Paper had been read by the author upon steam-turbines in general and upon the Parsons turbine in particular.*

The type which has been described as having a group of wheels possesses the notable advantage of allowing a great reduction in the speed of rotation for wheels of the same diameter and of the same

* See Comptes Rendus de la Société de l'Industrie Minérale à St. Etienne. Meeting of 12 April 1890.

number. It would therefore be preferred to all others if it had not, in the author's opinion, the grave defect of preventing a sufficiently high efficiency being attained. It is easy to see that, owing to the great velocity of flow of the steam in the first wheels, the losses of energy by friction and eddying are very great, reaching such a value that the second wheel develops less power than the first, and the third wheel and the fourth, when more than two are used, act more like brakes than prime movers. The Curtis turbine, as originally constructed at the Fisk Station in Chicago, had groups of four wheels, but the process of evolution reduced these to groups of three and then to groups of two wheels. Even with such a reduction in the number of wheels in a group the efficiency, according to calculation, is still at least 20 per cent. lower than that obtainable with the multicellular turbine which has only one wheel per cell. The author therefore ventures to think that the Curtis type of turbine will disappear, as in the process of evolution it will become the multicellular type pure and simple.

Description of the Rateau Turbine.—Since 1894 the firm of Messrs. Sautter-Harlé of Paris, with the assistance of the author, has been experimenting upon the construction of steam-turbines. The first turbine constructed had only a single moving wheel, like the De Laval turbines, and this wheel was formed with vanes in the form of a double arc similar to the buckets of a Pelton wheel, and these vanes were milled out of a solid block of steel. This type has since been copied in the Riedler-Stumpf type, built by the Allgemeine Elektrizitäts Gesellschaft, but which was rapidly abandoned by the author as it did not offer any chance of obtaining the maximum efficiency. The most recent Rateau turbine is of the action type, that is to say, expansion of the steam is fully carried out in the distributor for each group consisting of a distributor and one moving wheel. The steam therefore acts by its velocity and not by its pressure. These turbines are moreover multicellular, that is to say, they consist of a certain number of elements, each element comprising one distributor and one moving wheel. A very interesting characteristic of the type of action turbines is the possibility which it allows of leaving very

considerable play between the fixed parts and the moving parts, and this greatly facilitates construction and obviates the chances of dangerous friction if the bearings should become worn or the shaft somewhat bent. Besides this the wheels revolve in a chamber where the pressure is uniform. There is for that reason an absence of longitudinal thrust upon the moving parts and no necessity for the use of dash-pots for the purpose of overcoming the effect of this thrust, although such dash-pots are necessary in re-action or drum turbines. Finally, in the action type partial injection of steam is possible, that is to say, the steam may be directed upon a limited portion of the circumference. The moving wheels are formed of discs of sheet steel more or less thin, and upon the periphery of these discs are riveted vanes of cylindrical form, Figs. 5 and 6, Plate 83. A steel band riveted to the periphery maintains the correct spacing of these vanes and insures great rigidity to the construction. Wheels so constructed are extremely light, and remain in equilibrium at a velocity far higher than that to which they are subject in the turbine. These moving wheels turn between circular diaphragms provided with distributing vanes, which enter circumferentially into grooves formed in the interior of the turbine case. Between two adjoining diaphragms is therefore produced a cell or very flat chamber in which the moving wheel revolves. The shaft passes through the diaphragms in collars of anti-friction metal with very slight friction. In the first diaphragm which the steam passes through, the distributing vanes are placed only upon a part of the circumference. Partial injection of the steam is therefore obtained, and thus the velocity of the steam is better utilised. Moreover, to produce the same effect the useful part of each distributor is set with an angular advance on the preceding section; this angle of advance is calculated according to the speed of rotation, so that the steam leaving one moving wheel enters into the following distributor and never encounters a solid wall, which would produce a shock and therefore a loss of kinetic energy. For the last wheels it is necessary to employ total injection, that is to say, the distributing vanes must be set upon the whole circumference of the diaphragm, and moreover, owing to the expansion of the steam, the radius must

be increased. The bearings of these turbines are external, and by means of a special system of spring packing they are kept perfectly tight. No oil is carried by the exhaust steam to the condenser, and this has an important advantage as the water of condensation can be used for feeding steam boilers direct without any necessity for the use of an oil-separator. The speed of rotation is controlled by a centrifugal governor with a Denis compensator, which acts upon an obturator controlling the pressure of steam entering the turbine.

High and Low Pressure Turbines.—In the case of installations where exhaust steam must be used, a subject which will be considered in detail at a later stage, it is often desirable to supply the turbines temporarily with steam at high pressure when the primary machine is stopped or is furnishing less steam, and the work done by the turbine must remain the same. In order to obtain economical working, it is preferable not to expand this steam in order to lower it to the pressure for which the low-pressure turbine is built. Therefore the author has designed a turbine which may be described as of the mixed type, and this can be supplied either simultaneously or separately by steam at high pressure and by steam at low pressure, without any lowering of the efficiency of the mechanism. In order to attain this result the turbine is constructed in two parts, one designed for high-pressure steam and the other for low-pressure steam. The steam at high pressure having done work in the first portion will pass to the second portion, which may be fed either by steam coming from the accumulator or by the exhaust from the first portion. The admission of high-pressure steam into the first portion is automatically obtained by means of a special regulator which allows steam from the boilers to pass to the first portion, as soon as the pressure in the accumulator falls below a given value. This arrangement, which has been adopted in all the new applications of the system to the use of exhaust steam, works very economically, and is particularly suitable to cases in which the primary machine works irregularly, and where therefore the demand for live steam is frequent and somewhat prolonged. In the small machines the two portions are usually joined together.

Efficiency of Turbines.—It is essential that the word efficiency should be clearly defined in order to avoid any misconception as to the figures which are given below. The author uses the expression "Theoretical consumption of the perfect machine" to denote the maximum work which the steam is capable of supplying when starting from the saturated or superheated condition in which this steam is delivered to the engine, and expanding adiabatically with no loss of admission pressure P to exhaust pressure p . By comparing the actual consumption of steam, as measured during the tests of the machine, with this theoretical consumption of the perfect machine for identical conditions of pressure and similar states of the steam, the net efficiency of the machine is obtained. After special study of the question, the author has been able to draw a curve of theoretical consumption and to derive from it the following empirical formula for use when the steam is saturated and dry at admission :—

$$K = 0.85 + \frac{6.95 - 0.92 \log P}{\log P - \log p}.$$

This formula gives the consumption K in kilograms per horse-power-hour of 75 kilogrammètres as a function of the absolute pressures P and p expressed in kilograms per square centimètre. In British measures this formula becomes—

$$K = 2.13 + \frac{16.20 - 2.05 \log P}{\log P - \log p}.$$

in which K represents lbs. per horse-power-hour, and P and p lbs. per square inch. The curve which is given in Fig. 7 (page 748) will readily afford interesting information, particularly if it be necessary to estimate the consumption of steam with a given machine to obtain a certain useful effect. In that case the theoretical consumption per horse-power-hour given by the curve should be compared to the exact consumption of steam actually observed with the machine, and of course referring to the particular number of electrical horse-power developed by the dynamo driven, or to the power in water raised (expressed in horse-power-hours,) if pumps are driven by the machine. The relation between these two figures of consumption gives the combined efficiency of the group, and therefore takes account of all

the losses of the steam engine and of the driven mechanism. This combined efficiency is merely the product of the efficiency of the prime mover multiplied by the efficiency of the machine driven. If the steam admitted to the turbines is superheated, then of course in estimating the theoretical consumption the extra calorific energy corresponding to the superheat must be taken into account;* but

FIG. 7.

for the exact estimation of this extra energy adequate precise knowledge of the specific heat of water vapour is not yet known. At present we have only the figure obtained by Regnault after very cursory experiments (0.48), a figure which is merely a rough approximation.

* These results were published for the first time in the "Annales des Mines de Paris" in February 1897, and since that date have on many occasions been republished in various French and foreign publications.

The author has begun some experiments upon this subject, from which he believes he will be able to prove that the true specific heat of steam is not constant, as has been usually supposed up to the present. It varies inversely as the amount of superheat, and is probably approximately unity for very small values of superheat, and approaches 0.40 for large amounts of superheat.

Calculation of the Efficiency of Turbines.—The efficiency of a turbine may be calculated *à priori* when a preliminary study has been made of the practical coefficients which must be introduced into the theoretical formula. This efficiency can be calculated with quite a remarkable degree of accuracy, by dividing the losses of the machine under two headings, 1st, the internal losses which are produced by friction and eddying of the steam in the fixed and moving vanes; 2nd, the external losses which correspond to the leakages of steam in the play between the fixed and moving parts, and to the friction of the wheels upon the steam and to the friction of the bearings. The first kind of losses gives rise to the *internal efficiency*, which may therefore be called the “hydraulic efficiency,” by extending the use of the term employed for hydraulic turbines. This internal efficiency depends upon the more or less perfect form of the vanes, and also on the relation between the peripheral speed of the moving vanes and the speed of flow of the steam. It is possible to draw a curve showing the efficiency as a function of the peripheral velocity, and therefore it is easy to assign at once the degree of efficiency which a given turbine will realise.

Once having fixed the internal efficiency, then, in order to obtain the net efficiency, the losses by leakage must be deducted as well as the losses by friction of the wheels upon the steam, and also those in the bearings. Let us take, for example, a multicellular turbine of 1,500 brake horse-power upon the shaft with a speed of 1,500 revolutions per minute. By means of practical coefficients found by experience it can be calculated that the internal efficiency of such a machine may easily rise to 69 per cent.; on the other hand, the losses by leakage and by friction in the bearings absorb 1.5 per cent. of the normal power, and the losses due to friction of the wheels

upon the steam amount to 2·5 per cent., making a total for the external losses of 4 per cent. The net efficiency upon the shaft at the speed stated will then be 0·69 multiplied by $0·96 = 0·66$. From this value of efficiency it is easy to calculate the consumption of steam per horse-power-hour which would be required by this turbine under conditions of pressure and of superheat of the steam already determined. All calculations for designs of turbines that have been made by this means have always proved correct within 1 to 2 per cent. of actual practice. Such accurate calculations are not possible in the case of reciprocating engines owing to the action of the cylinder walls, which makes calculations uncertain and often inaccurate, whilst in the turbine the continuity of flow of the steam allows practical calculations of a high degree of accuracy to be based upon theory, once having determined the fundamental coefficients which are employed in the formulæ. For a more detailed consideration of the use of these coefficients, the author refers to the work which he published in 1903 in the 'Revue de Mécanique,' entitled "Théorie élémentaire des Turbines à Vapeur."

The Results of actual Practice.—The author began the construction of his first multicellular turbines in the year 1898 in collaboration with the firm of Messrs. Sautter-Harlé of Paris, and there are now already at work or in process of manufacture turbines developing more than 25,000 horse-power in units varying from 10 to 2,000 horse-power, irrespective of the designs now in preparation. Some of these turbines are used for driving dynamos, others for pumps, for fans, and for the propulsion of vessels. The author has appended a brief abstract of the conditions and economical results of the most interesting of these installations.

Turbo-Dynamos for Direct Current.—The company working the Peñarroya Mines in Spain installed in their central electric lighting station a little more than a year ago three groups of turbo-dynamos, developing 500 electric horse-power with direct current at 240 volts. Each of these three generating sets comprises two turbines for high and low pressure, and two dynamos for direct current, the latter

directly driven from the turbines upon the same shaft and upon the same bedplate, and the two dynamos supply a three-wire network with a potential of 480 volts between the outer wires. The speed of rotation is about 2,200 revolutions per minute, and the floor space occupied by the turbine with its two portions is only 12 feet by 5 feet 6 inches, with a height of 5 feet inclusive of a bedplate 1 foot high. The condensation of the steam is carried out by means of ejector-condensers of a type designed by the author.

Figs. 8 and 9 (pages 752 and 753) show a longitudinal section and a plan of the turbine which has twenty-four moving wheels. The works tests of the first group were made in September, 1902, with the greatest care, and the results are given in Table 1. The condensation was made for these tests by means of a surface condenser belonging to the works, so that the water of condensation might be collected and measured. It will be seen that the vacuum at the condenser declined as the power increased. This result arises from the fact that the condenser was designed for machines of 250 horse-power only, and therefore was of insufficient size for the larger volumes of steam.

TABLE 1.

Data.	$\frac{1}{4}$ Load.	$\frac{1}{2}$ Load.	Full Load.	Over-load.	Over-load at 2400 rev.
Electrical H.P. at brushes .	135	259	525	627	641
Admission pressure, absolute, } lbs. per sq. in. }	46.21	76.6	136	156	156
Exhaust pressure, absolute, } lbs. per sq. in. }	1.24	1.33	1.63	1.82	1.82
Theoretical steam consumption } of perfect engine per H.P. } hour lbs. }	10.93	9.8	8.89	8.73	8.73
Actual steam consumption per } electrical H.P. hour at } brushes lbs. }	21.3	18	15.8	15.39	14.90
Combined efficiency of the } electrical generating set . }	0.513	0.540	0.560	0.569	0.580

500 Electric H.P. Turbo-Dynamo. 2,400 Revs. (*Peñarroya.*)

FIG. 8.
Longitudinal Section.

500 Electric H.P. Turbo-Dynamo. 2,400 Revs. (Peñarroya.)

FIG. 9.—Plan.

A. Steam Pipe.

B. Stop Valve.

C. Throttle Valve.

D. Vanes of Moving Disc.

E. Passages in fixed diaphragm.

F. Steam Pipe.

G. Exhaust.

H. Bearing.

I. Speed Indicator.

J. Throttle Valve.

L. Auxiliary Valve.

N. Ejector Condenser.

It follows from the preceding figures that a turbine developing 644 electric horse-power, and working without appreciable superheat between an admission-pressure of 156 lbs. absolute and an exhaust-pressure of 1.8 lbs. (notably higher than the results given by good condensers in practice), has given the reduced consumption of 14.9 lbs. per electrical horse-power at the terminals; the combined efficiency for the set being then 59 per cent. when compared with the energy contained in the steam for the same fall of pressure. With steam at 180 lbs. superheated 50° and a vacuum of 29 inches of mercury, the consumption at 2,400 revolutions per minute would decrease to 11.5 lbs. per electrical horse-power-hour.

Fig. 10 gives the curves corresponding to the figures taken at a speed of 2,400 revolutions per minute. The abscissæ denote the power of the machine in electric horse-power at the terminals of the dynamos. The ordinates vary with the curves. For curve A the ordinates denote the absolute pressure of the steam on reaching

FIG. 10.

500-H.P. Turbine Dynamo (Peñarroya).

Curves of Pressure, Consumption and Efficiency, with varying power at constant speed of 2,400 revs. per minute.

the turbine in lbs. per square inch. For curve B they show the absolute back-pressure at the exhaust in lbs. per square inch. For curve C they show the total consumption of steam in lbs. per hour, and for curve D they show the consumption per electrical horse-power in lbs. per hour, and lastly for curve E they show the combined efficiency. These graphic curves show clearly a remarkable characteristic of turbines which is the very low relative consumption of steam at small loads. It will be seen that, in order to drive the machine at its normal speed with no load but with the dynamo excited, the total consumption of steam is only 10 per cent. of that at full load. With reciprocating engines the result would be very different. Under similar conditions the consumption of steam at no load with the dynamos excited usually reaches from 20 to 25 per cent. of the consumption of steam at full load. It therefore follows that turbine sets are more advantageous than reciprocating engine sets when running at light loads, even supposing it be admitted that there is equality at heavy loads. Another valuable property of steam-turbines is that they may be considerably overloaded without difficulty. Thus, in the case of the machines at Peñarroya, which were designed for 500 horse-power, they have been able to develop 650 electrical horse-power, and the output could have been raised even still higher had it not been for the overheating of the commutators of the dynamos. In the author's machines he is enabled to take an overload by supplying steam to a distributor more or less distant from the inlet in the series of successive distributors. The stop-valves which regulate this supplementary admission can be moved by hand or controlled mechanically by the governor. The second group was submitted to a competent committee (consisting of Professors Stodola, Wyssling, and Farny of the Zürich Polytechnicum), and this commission found the figures for consumption of steam a little higher (from 2 to 4 per cent.), the ampère meter brought from Zurich having given somewhat higher readings than the instruments standardized by the central laboratory of the *École Supérieure d'Electricité* in Paris.

These sets of turbo-dynamos for direct current have proved somewhat unsatisfactory in practice, owing to the sparking of the

brushes occasioned by the vibration of the armatures. The commutators of direct-current dynamos are, as is well known, the delicate parts of these machines, particularly when they have to run at such a great angular velocity as is necessary if direct coupled to steam-turbines. In order to obtain sparkless running, special arrangements must be made for absorbing the vibration, but it is very difficult to maintain in good equilibrium such heterogeneous bodies as the armatures of dynamos, consisting of bars of copper which become slightly displaced under the action of centrifugal force, owing to the compression of insulating material. Special arrangements for static equilibrium, and also for dynamic equilibrium have been designed, but the construction of the armatures is still always somewhat delicate, and it is only after several improvements that it has been possible to make them work well. At Peñarroya the vibrations were caused principally by want of level between the bearings, and this was brought about by settlement of the masonry supporting the machine. In order that bodies in rapid rotation upon very long shafts supported by several bearings may turn easily without vibration, it is indispensable that the principal axis of inertia of each part should coincide with the axis of rotation, and the bearings must be in correct alignment. In spite of these difficulties, it has been possible to build good direct-current dynamos for direct coupling to turbines, and when the voltage is high the problem is simpler. Owing to the high peripheral velocity of the commutator, it is usually necessary to employ metallic brushes, but in certain cases, notably with turbines supplied by steam at low pressure when the angular velocity is lower, it has been possible to use carbon brushes. The machine at Bruay, which will be referred to later on, is an example. Among other sets for direct current built by Messrs. Sautter-Harlé may be mentioned one of 500 horsepower for Huta-Bankowa (Russian Poland), which is interesting from the fact that the two portions of the turbine and the two dynamos were built on two parallel shafts. This arrangement offers the advantage that one of the halves of the group can be used if the other is accidentally damaged.

Turbines with Alternators.—The construction of generators for two-phase and three-phase current for direct driving by turbines offers much less difficulty than the design of direct-current dynamos for the same purpose. The absence of the commutator renders possible higher angular velocities for alternators than for direct-current dynamos. Three types of alternators have been tried successively. The machine with the solid rotor, that with the rotating field magnet and that with the rotating armature. The first type would be, from a mechanical point of view, the ideal generator for coupling to a steam-turbine. Unfortunately from its design it is impossible to use speeds higher than 1,500 revolutions per minute for a frequency of 50 periods per second. On the other hand, this type of machine permits much greater magnetic leakage than the others, and this necessitates disproportionately large dimensions and a comparatively low efficiency. These circumstances are unfortunate, for the alternator with the solid rotor would permit of making the movable part of very solid construction, which is easy to balance once for all and thus avoid any chance of mishap. It is on these grounds that types of alternators with revolving field-magnets or revolving armature are in actual use, and Messrs. Sautter-Harlé have already built a great number of them. The view in Fig. 11, Plate 84, represents a turbo-alternator with revolving field-magnets to develop 400 electric horse-power at 5,500 volts, and this machine is now working in the generating station of the Loire Electricity Co. The moving part of this machine makes 3,000 revolutions per minute. The results of tests have been as follows :—

Pressure of steam	170 lbs. absolute.
Back pressure of the exhaust	2·85 lbs.
Output of the terminals	388 electric horse-power.
Consumption of steam per electrical horse- power, including excitation	19·2 lbs.
Combined efficiency	48·7 per cent.

The efficiency obtained could have been much improved by increasing to a slight extent the dimensions of the turbine, which had only twelve moving wheels, but even now it is comparable with turbine sets of the same power of other types. Three similar sets

working with superheated steam are now being constructed for the factories of Pavin de Lafarge at Teil (Ardèche). The results guaranteed by the makers are as follows:—

Pressure of steam on admission to the turbine .	156 lbs.
Vacuum in condenser	26 ins.
Temperature of steam	270° C.
Consumption of steam per electrical horse- power at the terminals, including excitation }	15 lbs.

All these different machines are fitted with ejector condensers.

A machine of 2,000 kilowatts at 1,500 revolutions per minute would have a turbine efficiency of 68 per cent., which would enable the following guarantee to be given, assuming the use of superheated steam and the condenser giving a very good vacuum, which it is easy to obtain with steam-turbines in which there is no entrance for the air:—

Pressure of steam on admission to the turbine .	200 lbs.
Vacuum in condenser	29 ins.
Temperature of steam	660° F.
Consumption of steam per horse-power-hour upon the shaft of the turbine }	8·5 lbs.
Consumption of steam per kw.-hour at the terminals }	12·2 lbs.

This is a far better result, the author believes, than can be given by reciprocating engines, and this can be obtained with steam-turbines under the conditions specified.

Turbines for Vessels.—The author read a Paper on the 25th March 1904 in London, before the meeting of the Institution of Naval Architects, upon the Application of Steam Turbines to the Propulsion of Vessels. He then described the difficulties occurring in the application of turbines to the propulsion of vessels, which are as follows:—

- (1) The difficulty of adapting screw propellers to the high speeds of rotation of the turbines.
- (2) The poor efficiency of turbines at low velocity.
- (3) The inconvenience of the combination in stopping and approaching quays.

The author believes that the best solution will be reached in the employment of a reciprocating engine of small power and of steam-

turbines coupled to independent shafts, so that the reciprocating engine would always be working at any speed of the vessel. Then each type of machine would be perfectly adapted to the part which it has to play, and excellent results in consumption of fuel would be obtained at all speeds. The author gives below a short account of the actual results obtained by turbines in the propulsion of vessels.

Torpedo Boat 243.—The equipment of the French torpedo-boat No. 243 with steam-turbines was decided upon in the year 1898. The French Admiralty wished at that time to make experiments upon turbines and upon the working of multiple propellers. In order to reduce the costs of this experiment as far as possible, it was thought desirable to make use of the hull of an ordinary torpedo boat, but great difficulty was found in making room for the turbines, and moreover the propeller shafts were placed very high at the turbine end and were necessarily considerably inclined; in this case to 11 per cent. from the horizontal. The back supports had to be considerably increased, and therefore the total resistance of the boat was much greater. All these conditions were extremely unfavourable, and the screws fixed upon the inclined shafts gave a very bad efficiency. The two turbines, each of a nominal output of

TABLE 2.

Torpedo Boat No. 243. Trials of 22 January 1903.

Six Propellers: Diameter, 23·6 inches; Pitch, 19·7 inches.

Number of Trial.	I.	II.	III.	IV.
Speed of vessel (mean of three runs) knots)	17·07	19·59	20·94	21·26
Rotation of turbines . revs. per min.	1,348	1,572	1,748	1,774
Effective pressure of steam on admission to turbines . . lbs. per sq. in.)	68·26	100·98	129·42	132·26
Condenser vacuum . . . ins.	28	28	27	27·5
Mean slip of propellers . . .	0·217	0·230	0·260	0·260

900 horse-power were completely independent of each other. They were provided on the exhaust side with a single moving wheel for movement in a reverse direction. As the experiment was solely made to see if the installation of turbines with propellers of small diameter was suitable for producing in a vessel a speed comparable to that given by a reciprocating engine, the backward movement of the turbine was willingly sacrificed. Very careful experiments had been made with this machinery, and it will be seen from Table 2 (page 759) that a speed of 21 knots was attained. It is quite certain that if the shafts had not been set at such a great inclination a speed of 24 knots would easily have been exceeded, for the turbines gave an output somewhat higher than was expected.

The turbines worked in a most satisfactory manner, as is proved from the numerous reports of the trials made under the control of the Engineers of the French Navy.

A Vessel built by Messrs. Yarrow and Co.—This vessel is similar to the first class torpedo boats usually constructed by this firm, and similar, with the exception of the turbines and propellers, to the "Tarantula" upon which Parsons turbines are used. The boat is of 140 tons burden, and is provided with three propeller shafts driven simultaneously and separately by a turbine divided into two portions, and a reciprocating engine. This latter engine develops 250 brake horse-power, and drives a central shaft which is completely independent of the turbines. The side shafts are driven by a turbine in two parts arranged in series and rotating in opposite directions. The reciprocating engine was used in order to obtain a velocity of 10 to 14 knots in the best conditions of coal consumption, and so that movement astern might be easily effected. The above combination adopted by Messrs. Yarrow and Co. gave the expected results; but it is not the only possible design, and the author thinks that it would have been even more advantageous to arrange the driving mechanism so that the reciprocating engine, instead of allowing its exhaust to escape directly to the condenser, should pass it in to the turbine at low pressure, and perhaps even into the turbine at high pressure during

FIG. 12.
Turbine for Messrs. Yarrow and Co.'s Boat
Longitudinal Section of H.P. portion.

a slow speed run. The turbine of the Yarrow boat can develop more than 2,000 horse-power. The high-pressure portion is represented in longitudinal section in Fig. 12 (page 761). At the points where the shaft passes through the frame tightness is obtained by a system of special packing, the same as that employed upon land turbines. A special governor assures constant pressure upon the four packings so as to prevent all entrance of air, and owing to this arrangement it is quite easy to obtain an excellent vacuum in the condenser. Several tests were made upon this vessel between the 13th October 1903 when it first went out and the present date. The Table of experiments made on the 19th January 1904, after the propellers had been improved, is given on the next page. The speed of 25 knots which had been attained in the first trial of the boat has been increased to 26.39 knots, by giving to the turbines a little more than the quantity of steam for which they were designed.

The propeller shafts were for this test each fitted with two propellers. The author's opinion on this subject is that the use of a single propeller upon the same shaft is certainly likely to give a better efficiency. The arrangement of propellers, one in front of the other, is defective. The aft propeller works in the moving tail water from the former. It is certain that the speed of 26.4 knots can be still further improved, and will become equal or even higher than the maximum speed given by reciprocating engines. The use of a single propeller for each shaft necessitates a peripheral diameter greater than the pitch, in order to obtain sufficient propulsive surface. Under such conditions the efficiency can never be good, unless the inclination of the shafts to the horizontal is very slight. In spite of this, it is probable that the efficiency of these propellers with a relatively small pitch is distinctly less than that of ordinary propellers. Fortunately, this inferiority can be compensated by the fact that the efficiency of well-designed turbines is a little higher than that of the reciprocating engines used on board ship.

Turbine-Driven Pumps.—The application of steam turbines for the purpose of direct driving of centrifugal pumps enables most remarkable results to be obtained, and particularly renders possible

TABLE 3.
Messrs. Yarrow and Co.'s Torpedo Boat. Trials of 19 January 1904.

Number of Trial.		I.	II.	III.	IV.
Effective pressure of steam on admission to H.P. turbine	. lbs. per sq. in.	50	100	150	170
Condenser vacuum	 inches	28	27·5	27	27
Speed of vessel (two runs)	 knots	15·58 20·00	19·25 23·53	23·22 26·67	25·714 27·067
Mean speed of vessel	 knots	17·79	21·39	24·94	26·39
Rotation of reciprocating engine	 revs. per min.	458	508	555	576
Rotation of H.P. turbine	 revs. per min.	836	1,052	1,207	1,258
Rotation of L.P. turbine	 revs. per min.	836	1,065	1,232	1,307
Slip of propellers	Reciprocating engine per cent.	28·7	22·4	17	15·3
	H.P. turbine per cent.	13·6	17·4	16·4	14·8
	L.P. turbine per cent.	24·0	28·2	27·8	27·8

height of lifts much greater than those which are at present obtained. It is well known that the pressure produced by the wheel of a centrifugal pump increases directly as the square of the peripheral speed, and therefore with the velocity produced by steam-turbines it becomes possible with a single wheel to raise large volumes of water to heights of more than 900 feet in a single lift. The author, in the "Rapport au Congrès de Mécanique" in 1900, published the drawing of a turbo-pump of only one wheel, enabling the attainment of a head of 700 feet and developing a useful power of more than 40 H.P. The test of this pump and its driving turbine, made in 1900, was published in "Ventilateurs et Pompes Centrifuges à Haute Pression" 1902. If it is desired to obtain either greater pressures or to obtain the same results with relatively lower speeds, it is sufficient to couple steam-turbines with pumps consisting of several similar wheels arranged in series, so that each one increases by an equal quantity the pressure already given by the preceding wheels. The author has himself designed a particular type of multicellular centrifugal pumps. These machines are designed upon the general ideas set forth in a Paper which he published in 1892 in the "Bulletin de la Société de l'Industrie Minérale." The principal results of the general theories established at that time were further developed in the "Treatise on Turbo Machines" which appeared in 1897-1900, in which the author showed their application to hydraulic turbines, pumps, and centrifugal fans. From a practical point of view, by combining the advantage of high velocities of rotation with the advantage obtained by putting the wheels in series, it would be quite easy to deliver to heights greater than 1,500 feet. Some notes upon this subject were given in a communication made in 1902 to the "Société d'Encouragement pour l'Industrie Nationale." At that time a considerable number of turbine-driven pumps had been manufactured by Messrs. Sautter-Harlé. Multicellular turbine pumps form very light sets which are compact and extremely simple. Their maintenance is very low, for the wear is almost nil and the cost of lubricating oil extremely small. In the examples referred to later, it will be seen that the consumptions of steam per brake horse-power in water raised are comparable with those of good piston-pumps.

Turbine-Driven Pump at Falkenau (Bohemia).—The first machine of this kind which was made was delivered to the mines of Falkenau in Bohemia. This apparatus consisted of a single bedplate upon which were mounted a steam-turbine with multiple wheels enclosed in the same casing, and a pump with four wheels also mounted in a single case. The regulation of speed was obtained by a centrifugal governor arranged in the base of the principal bearing, Fig. 13, Plate 84. The lubrication of the end bearing of the shaft on the pump side was effected by means of a little hydraulic Servo motor. A compensating piston enabled the longitudinal thrust received by the moving part of the pump to be exactly counter-balanced. This thrust is moreover inconsiderable, owing to the special arrangement adopted for the wheels of the pump. Condensation of the steam is effected by means of an ejector-condenser on the system designed by the author. The results obtained in the different tests of this set are set forth in Table 4 (page 766).

It appears from these figures that it is possible with such a turbine-driven pump to obtain combined efficiencies of 36·5 per cent., which corresponds to an efficiency for the pump of 67 per cent., and an efficiency for the turbine of 55 per cent.

Turbo-Driven Pumps at Bruay.—A more powerful turbo-pump has been made and delivered recently to the Mining Co., of Bruay, in the Pas-de-Calais. This set can raise 950 gallons a minute to a height of 1,200 feet. The complete set is mounted above a small tank into which discharges the ejector-condenser, which is used for condensing steam, and from which is taken the suction of the principal pump. At the end of the shaft is a small pump which is directly coupled to the turbine and furnishes water to the ejector-condenser at a pressure equivalent to a head of 18 to 20 feet. The results obtained with this pump at full load were as follows:—

Discharge in gallons per minute	980 gallons.
Total lift	1,195 feet.
Revolutions per minute	2,200.
Absolute pressure of the steam on admission to the turbine	101 lbs.

TABLE 4.

Admission pressure of steam P . . . lbs. per sq. in.	82.5	91.9	94.6	97.5	104
Exhaust pressure of steam p (vacuum 24.8) ins. of mercury) . . . } lbs. per sq. in.	2.47	2.47	2.47	2.47	2.47
Revolutions per minute	3175	3200	3200	3200	3280
Total height raised feet	721	721	695	669	688
Discharge of water gals. per min.	405	492	551	608	665
Useful work in water raised H.P.	89	107.8	116.7	123.6	139.4
Theoretical consumption of steam per H.P.-hour . . . lbs.	11	10.7	10.62	10.5	10
Actual consumption of steam per useful H.P. of water raised } . . . lbs.	34.61	32.15	30.3	29.5	27.8
Total combined efficiency of the turbine pump . . .	0.315	0.335	0.350	0.355	0.360

Absolute back-pressure at the exhaust	1.61 lbs.
Power in water raised	359 H.P.
Consumption of steam per horse-power hour in water raised	22.7 lbs.
Total net efficiency	42.5 per cent.

It will be observed that the total efficiency of this set is very high, attaining 42.5 per cent. It will be pointed out hereafter what a slightly improved efficiency denotes in consumption of steam when high pressures of steam can be used. One of the chief advantages of turbo-pumps, besides their excellent efficiency, is the extremely small floor space which they occupy when compared with piston-pumps. Their dimensions are very small, and they require very little height, so that they can be placed in a room which has no greater dimensions than that of an ordinary working gallery in a pit. In Fig. 14 are shown the comparative dimensions of the

FIG. 14.

Pump Chambers at Bruay.

- A. Reciprocating Steam-pump.
B. Steam-Turbine (Rateau).

chamber in the mine necessary to accommodate the turbo-pump which has just been described, and of a chamber necessary to accommodate a piston-pump of the same power. The cubic volume of the masonry work is approximately in the proportion of 1 to 10.

Turbo-Pumps for Boiler-Feed Purposes.—This class of pump can be used very readily for feeding steam-boilers, and several such sets have already been constructed. When there is a group of generators of the same size, a turbo-pump can easily carry out the

whole feed service with a consumption of steam much less than that of the small feed-pumps usually employed. Whilst ordinary reciprocating pumps consume from 150 to 250 lbs. of steam per brake horse-power in water moved, the turbo-pump for boiler feed service will not usually require more than 40 to 60 lbs. per brake horse-power. An automatic system of control has been designed by the author which enables the apparatus to work automatically in a continuous manner, even when the demand falls to zero. The work of the stoker, so far as the feed service is concerned, consists simply in opening the feed valve to the necessary amount.

Turbo-Pumps of large output for raising Water in Towns.—Turbo-pumps, both from the point of view of efficiency and also of ease in working, are particularly suitable for raising water into reservoirs for the service of towns. The following figures indicate results which can be obtained with this class of machine in similar conditions to those that exist in large towns.

Delivery in gallons per minute	5,200.
Height to which the water is raised	460 feet.
Actual horse-power in water raised	730 H.P.
Combined efficiency including condensation	46 per cent.
Pressure of steam in lbs. per square inch	210.
Vacuum at the exhaust	28 inches.
Superheat in the steam	100°.
Consumption of steam per horse-power hour in water raised	15 lbs.

It will be observed that the system described is quite as efficient as that of the best piston-pumps now employed.

Turbo-Fans.—The great angular velocity of steam-turbines makes them particularly suitable for the direct driving of fans for high-pressure used for blowing engines, and even for air compressors. The author has already described them in his work entitled “Fans and Centrifugal Pumps for High-Pressures” inserted in the “Bulletin de la Société de l’Industrie Minérale,” of 1902, and has given particulars of actual results obtained from a turbo-fan with

a revolving wheel of 10 inches in diameter. With an angular velocity of 20,200 revolutions per minute, this fan discharged 22.5 cubic feet of air per second under a pressure of 19 feet of water. The useful work contained in this compressed air was about 41 horse-power, and the combined efficiency of the group 30.7 per cent. The pressure of the steam on admission to the turbine was 145 lbs. per square inch. Since that time the system described has been improved, and several sets of this apparatus have been built for different purposes. Some are used in sugar refineries for the compression of air charged with carbonic acid extracted from the lime furnaces, and requiring to be sent into the de-carbonising vats. A special system of regulation enables the pressure of discharge to be kept constant whatever may be the volume of discharge, and on the contrary to maintain a constant discharge whatever may be the pressure. The set which is shown in perspective in Fig. 15, Plate 85, is actually at work in the forges of Chatillon and Commentry, and is used for blowing a blast-furnace. The set consists of two fans arranged in parallel and each driven by one turbine (the two turbines are placed in series), and the apparatus will discharge 80 cubic feet of air per second under a pressure of 10 inches of mercury. The energy in the compressed air rises as high as 100 horse-power. The moving wheels of the fans and of the turbines are only 11.2 inches in diameter; Table 5 (page 770) gives an abstract of the results of trials to which the apparatus was submitted.

It will be seen from this Table that the combined efficiency of the apparatus reaches the remarkable figure of 38 per cent. Similar apparatus of larger dimensions can be built, and these might replace advantageously the large blowing engines used for blast-furnaces. One important characteristic of these sets is their remarkably small dimensions, and they can be put down at a cost which is very small in comparison to large piston blowing-engines. By coupling several wheels of centrifugal fans, as the wheels of multicellular pumps are coupled, much higher pressures of air can be obtained. In a few months the construction of a centrifugal turbine-compressor of 350 horse-power will be finished, capable of delivering compressed air

TABLE 5.

Turbo Fan at Commentry. 6th-8th August 1903.

Orifice.	Revs. per minute.	Pressure of Steam Absolute.		Air-pressure at the Outlet of Fan.	Total Consumption of Steam.	Air Discharge.	Horse-Power Theoretical.	Horse-Power Useful.	Total Efficiency.
		Admission.	Exhaust.						
Sq. ins.		Lbs. per sq. in.	Lbs. per sq. in.	Feet of water.	Lbs. per hour.	Cubic feet per sec.			
19·1	15,200	113·8	4·53	10·60	3037	91·4	260·4	99	0·380
	15,700	128·0	4·80	11·61	3403	93·5	299·5	112·6	0·377
22·3	14,600	113·8	4·92	9·0	3037	99	256·9	93·2	0·363
	15,200	128·0	5·32	9·90	3403	103	289·6	106·52	0·368
	15,800	142·2	5·68	11·0	3760	107·5	327·2	121·91	0·372
27·8	14,300	113·8	4·44	7·55	3037	114	261·8	91·08	0·348
	14,900	128·0	4·80	8·25	3403	117·5	299·5	102·46	0·342

at a pressure of 90 lbs. per square inch. This apparatus will be driven by a low-pressure turbine using the exhaust steam from a winding engine used in a pit. The turbine has a high-pressure turbine as a stand-by, so as to ensure economical working during the time when the winding engine is out of use. This special branch of the application of steam-turbines to air-compression has, in the author's opinion, a great future, not only for work in mines but also in metallurgy, where blowing engines of all kinds for blast-furnaces and for the tuyeres of steel works may be replaced by centrifugal machines of a much simpler and smaller character, costing far less and requiring fewer repairs than the present piston-engines.

*Turbines for Low Pressure with Steam Accumulators.**—One of the most interesting and promising applications of steam-turbines is the employment of steam at low pressure, and particularly exhaust steam coming from engines working intermittently. This last class of engines, which are, of course, usually employed in installations for the working of minerals or for rolling mills and pile drivers, has not until recently been able to benefit by the same advantages as in the case with engines working continuously, and this is owing to the special difficulties in the application of condensing plant, multiple expansion, and super-heating to such engines. The majority of such engines discharge the exhaust freely into the air, and the quantity of steam which is thus lost is considerable. The author has given special attention to the interesting problem of the employment of such waste steam, and he has obtained satisfactory results by means of his regenerative accumulator of steam, combined with turbines at low pressure, which are themselves coupled directly to dynamos, centrifugal pumps, or fans. The regenerative accumulator is intended to regulate the intermittent flow of steam before it passes to the turbine, and it consists essentially of a vessel containing solid and liquid materials which play the part of a fly-wheel for heat. The steam is collected and condensed as it arrives in large quantities in the apparatus

* Proceedings, 1901, Part 4, page 945.

FIG. 16.

Regenerative Accumulator (Rateau).

FIG. 17.

Water Heat-Accumulator (Rateau).

and is again vaporised during the time when the exhaust of the principal engine diminishes or ceases. The necessary variations for condensation and regeneration of the steam correspond to fluctuations in pressure in the accumulator, this pressure rising when the apparatus is being filled and descending when it is discharging into the turbine. Water which has a very high calorific capacity has been used as a heat fly-wheel, but, in order to communicate rapidly to a liquid mass a considerable quantity of heat corresponding to the latent heat of steam to be condensed, it becomes necessary, owing to the poor conductivity of water, either to arrange it in thin layers or to cause a rapid circulation, in order to increase the surface of contact between the steam and the water itself. The first solution of the problem gave rise to the accumulator with flat cast-iron trays, in which water is contained in shallow vessels arranged one above the other, Fig. 16 (page 772). The second solution of the problem gave rise to the accumulator with water only in which a rapid circulation was produced by the injection of steam into the body of the liquid itself, Fig. 17 (page 773). The low-pressure turbine, fed by the regular flow which comes from the accumulator, and working, for example, between an admission pressure of 15 lbs. per square inch and a vacuum at the condenser of 27 inches of mercury (back-pressure of 1.6 lbs.), can furnish an electric horse-power for about 31 lbs. of steam per hour. In a moderate sized pit, consuming 13,000 lbs. of steam per hour, it is possible to economise under these conditions about 350 electric horse-power. In steel works, where reversible steam-rolls are employed consuming about 45,000 lbs. of steam per hour, it will be easy to develop, by means of accumulators and turbines, an extra output of over 1,100 electric horse-power. It is desirable to mention that the prime movers are in no way injuriously affected in their working by the presence of the accumulator and turbines. This method was applied for the first time at Bruay, and the installation has worked most satisfactorily since August 1902, when it was first put into use. The exhaust steam from a winding engine is first of all treated by an accumulator with cast-iron trays, and then passes to a low-pressure turbine of

300 horse-power, which itself drives two dynamos keyed upon the same shaft; this group is shown in Fig. 18, Plate 85. A particular feature of this type of installation is the "Automatic Expander," which comes into use when the winding engine is not giving sufficient steam, and also when this engine is not working. This apparatus permits the admission to the turbine of live steam which is passed from the boilers through reducing valves; this expander can be adjusted by means of a spring, and comes into use as soon as the pressure falls below a pre-determined point.

Table 6 (page 776) gives the results of tests made upon this set in

FIG. 19.

Comparison of Efficiencies (1902 and 1903).

April 1902 and January 1903. Numerous similar tests have been made at different times since the apparatus was first started, and Fig. 19 enables a comparison to be made between the efficiencies which were calculated from the results obtained in April 1902, January 1903, and November 1903.

It will be noted from this Table that, although the tests of November 1903 were made with an output much lower than the normal output of the turbine and for a part of the curve in which the efficiency varies rapidly, the mean point which these tests graphically represent is very distinctly higher than the

TABLE 6.

Experiments with the Low-pressure Turbine and Dynamo of 300 Horse-power for the Bruay Collieries.

Revolutions per Minute. <i>n</i>	Kilowatts at the Terminals.	Absolute Pressures.		Temperature of Steam.		Flow of Steam measured.	Consumption of Steam per horse-power-hour.		Efficiency. ρ^*	
		Steam to Turbine. <i>P</i>	Condenser. <i>p</i>	<i>t</i> °		<i>I</i>	Measured. <i>K</i>	Theoretical. <i>K</i>		
		Lbs. per sq. in.	Lbs. per sq. in.	Cent.	Fahr.	Cubic feet per hour.	Lbs.	Lbs.		
3rd April 1902.										
1,337	169.5	12.02	2.13	132	269.6	165	{	45.2	21.9	0.485
1,500	176.5	"	2.22	"	"			42.8	22.2	0.518
1,610	186.0	"	"	"	"			40.6	"	0.547
1,690	190.5	"	"	"	"			39.7	"	0.559
1,840	198.0	"	"	133	271.4			38.1	"	0.581
1,510	225.0	14.37	2.49	135	275.0	201	{	41.0	21.2	0.516
1,605	232.5	"	2.52	"	"			39.8	21.3	0.534
1,700	240.5	"	2.57	"	"			38.5	21.5	0.559
1,800	247.0	"	2.62	"	"			37.4	21.7	0.580
5th April 1902: Experiments made by Messrs. Sauvage and Picou.										
1,589	70.3	5.42	1.25	111.0	231.8	79	51.1	25.1	0.492	
1,600	140.9	9.37	1.82	135.0	275.0	130	42.1	22.3	0.530	
1,591	202.0	12.83	2.32	137.0	278.6	177	39.7	21.1	0.531	
1,598	232.5	14.70	2.79	147.0	296.6	203	39.5	21.9	0.556	

* The efficiency is the relation between the electric power measured at the terminals of the dynamo, and the theoretical energy contained in the current of steam utilized for such conditions of pressure: this yield is therefore stated after the deduction of all losses in the turbine and in the dynamo.

curve of 1902. The combined efficiency of the turbo-electric group has therefore been maintained at an excellent value since the set was first put into use. When, as at Bruay, the low-pressure turbine is fed with expanded live steam during the time when the prime mover is at rest, the consumption of steam is obviously higher than that of an engine working at the full pressure of the boilers. If the demand for live steam is only made during a relatively short period, the slightly higher consumption during these periods will have scarcely any influence upon the total economy. If this is not the case, and, on the contrary, such periods are of considerable length as compared to that during which exhaust steam is used, as, for example, when the prime mover is only working during the day, and it is desired to have a regular service day and night from the secondary set, then it would be best to add to the low-pressure turbine a high-pressure turbine which, during the time that the prime mover is not working, would receive live steam at boiler pressure. The steam passing out of this high-pressure turbine would pass to the low-pressure turbine, which would use equally well either the steam coming from the accumulator or the exhaust from the first portion of the turbine. The admission of live steam at high pressure is, moreover, automatically obtained by a suitable system of obturators controlled by a speed regulator on the one hand and by the pressure in the accumulator on the other hand. Such a group works under all conditions with the maximum possible economy with an ordinary vacuum, and working at full load it would consume only 18 lbs. of steam per electric horse-power-hour if supplied with steam at 110 lbs. pressure, and it would consume 26 to 36 lbs. of steam when it was fed by exhaust steam at atmospheric pressure. This system of regulation by means of steam accumulators and turbines, which is particularly applicable to mines and steel works, enables very considerable economy to be effected. Installations of this nature are now in course of erection at the following places:—

350 Horse-Power at the Béthune Mines (turbine-driven air-compressor).

600 Horse-Power at the Mines of Reunion in Spain (electric generating sets).

2,000 Horse-Power at the Donetz Steel Works (electric generating plant).

700 Horse-Power at the Aciéries de Poensgen à Dusseldorf (electric generating plant, etc.).

As the author showed in a note published in January 1904 in the "Revue Universelle des Mines," this system offers great advantages when there already exists a central condensing plant, and the Poensgen installation is being carried out in that manner.

Conclusion.—There is no need to insist upon the well-known advantage in turbines resulting from their absolutely steady rotation, but, on the contrary, the enormous speed at which they must work in order to suit the velocity of flow of the steam is, in many cases, a serious inconvenience, particularly in the application of the turbine to the propulsion of vessels. It was a similar disadvantage for a considerable time in the driving of dynamos, but now it is possible to construct dynamos for very high speeds, and these can be coupled to steam-turbines forming light and inexpensive sets which take up little room, although they are very powerful. Such sets are very simple and are very easy of maintenance. Besides the constant value of the turning effort, the chief advantages which can be claimed for steam-turbines may be thus summarised: very low consumption of steam, particularly at small loads. From this point of view they are notably superior to piston-engines of even the best-known types. Further advantages are the small floor space occupied and the absence of expensive foundation work; the cost of oil is very low, particularly for the author's turbines, which have oil reservoirs and ring lubrication to all the bearings. The oil in the reservoirs need be changed only once in two months, and it is not thrown out by the rotating parts, so that the machinery always remains clean. With steam-turbines, momentary or even permanent overloads are very easy to deal with. It is even quite possible to have overloads of more than 50 per cent. higher than the normal

load. The regulation of speed in steam-turbines is carried out in a more perfect manner than is possible in piston-engines, and this is due, on the one hand, to the absolute constancy of the turning effort, and, on the other part, to the great kinetic energy accumulated in the revolving parts. This last characteristic makes the machines very insensible to variations in load. The full load may be taken off and put on again without causing a variation in speed of more than 2 to 3 per cent. When coupled to dynamos, fans, or centrifugal pumps, steam-turbines render it possible to obtain surprising results, owing to their capability of giving great power at very high speeds. The author has constructed turbine-driven fans, giving a pressure of 7.5 lbs. per square inch, and turbine-driven pumps raising water to several hundred yards. Such sets can now be built to develop several thousand horse-power, compressing air to more than 90 lbs. per square inch, or raising water to more than 2,000 feet high. Turbines are able to use very advantageously steam at low pressure, for they have an efficiency rising in value as the pressure of the steam becomes lower. In combination with steam accumulators of the type designed by the author of this Paper, turbines permit of the employment of exhaust steam in large quantities produced by engines working intermittently, such as winding engines and engines used in metallurgical works. In this Paper, the author believes that he has given typical examples of the chief applications in which steam-turbines are available. It will be seen that their field of application is very wide, and it is certain that a great future is in store for them.

The Paper is illustrated by Plates 83 to 85, and 13 Figs. in the letterpress.

Discussion.

Mr. FRANCIS HODGKINSON said that in Fig. 10 (page 754) and Table 1 (page 751) Professor Rateau had given a few details of some tests. He expressed the efficiency of his turbine by dividing the steam consumption of an ideal engine by the consumption actually observed. This efficiency was undoubtedly the basis upon which engine performances might best be compared, but he himself rather took issue with Professor Rateau in the manner he had expressed this efficiency at fractional loads, inasmuch that he appeared to charge his machine with the heat value of the steam at the internal pressure in the cylinder, after it had been reduced by the governor-valve instead of the pressure in the steam-pipe adjacent to the throttle. The speaker might be in error in this assumption, since no readings of boiler pressure were given. The steam on being wire-drawn through the governor-valve from presumably, in this particular test, a pressure in the steam pipe of about 156 lbs. to the figure he had given of 46.21 lbs. at one-quarter load, would do much work in the way of superheating, such that the efficiency figures quoted would be naturally unfairly high. He would calculate the efficiency at one-quarter load to be about 39.2 per cent. instead of 51 per cent., assuming a pressure of 155 lbs. absolute at the throttle.

The engineering world, he believed, would look forward with the greatest possible interest to the results of Professor Rateau's experiments to find the value of the specific heat of steam at high temperatures and at other pressures than atmospheric, at which pressure Regnault made his experiments. America, however, had not been idle in the matter, as Professor Carpenter had lately concluded a large series of experiments with steam of high pressures, and he believed Mr. Emmet had some information on the subject.

Mr. E. MEDEN said that Professor Rateau stated (page 738) that it was practically impossible to construct a turbine wheel to run at a peripheral velocity above 1,200 feet. If Professor Rateau had been familiar with the De Laval turbine, he would have known that turbine wheels, running at a velocity of over 1,200 feet per second, had been in successful operation at least since 1897.

The author further in the same paragraph stated that the efficiency of a single wheel was necessarily low, chiefly due to the necessity of employing diverging nozzles, giving rise to great loss of energy due to friction, etc. These losses of energy undoubtedly occurred, but the comparatively efficient results obtained from this type of turbine did not warrant Professor Rateau's assumption.

On page 738 the author spoke of the necessity of employing gears of special construction which were subject to excessive wear and accidental breakage. He himself did not understand what was meant by "gears of special construction," as the De Laval type of gearing was very old, and with the method employed for the manufacture of these gears, the cost was not at all excessive. As regards wear, many of these gears had been in operation for upwards of nine years without showing any appreciable wear. Regarding accidental breakage, they had yet to learn of a piece of machinery which was not subject to this misfortune.

Professor RATEAU regretted that his lack of familiarity with the English language had prevented his active participation in the discussion in a public way, and for this reason he regretted that he was unable, as would have been appropriate, to express his thanks for the cordial reception which had been accorded to his Paper. It was particularly agreeable to an investigator to bring his work before a meeting of engineers and scientists in America, who could appreciate the services that science was rendering to industry. To prevent his Paper from becoming too technical, he had avoided any theoretical development of the question. Those who desired to go further than this would find certain articles of the author's presented in the "*Bulletin de la Société de l'Industrie Minérale à St. Etienne*," and in the "*Revue de Mécanique*" (Paris).

He would reply first to the criticisms made by Mr. Meden, which would serve to bring out certain points only touched on in the Paper. Mr. Meden stated that if the author had been familiar with the De Laval form of turbine, he would know that wheels turning at a velocity in excess of 1,200 feet per second had been in operation

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since 1897. He did not ignore this fact, particularly since this result could only be attained by the use of that form of movable wheel which he had been producing in Paris since 1895. He had revolved wheels at higher speeds than this, but they were made from one piece of metal and the vanes were very short and milled out from a solid block of metal. In his opinion, although fractures of these wheels had been little to be feared, he held that such high velocities were not to be recommended in practice. He considered these speeds to be dangerous when the vanes were attached to the rim, and when they were relatively long, as was the case in the De Laval design. He had not seen the De Laval turbine at the time he was preparing his Paper, and only gave the result of his personal experience.

Since Mr. Meden spoke for the De Laval turbine, he himself might, therefore, be allowed to say that he knew of a considerable number of breakages of this type of machine. He would only recall the recent accident where the explosion of such a turbine had caused the death of several persons at Seraing in Belgium. This form of accident was apparently caused by the fracture of one or of several vanes after they had been worn by the steam. There then followed an eccentricity of the revolving disc, which by reason of the very high velocity of rotation brought about the breaking of the shaft. Sometimes the disc remained in the chamber, which surrounded it and turned there for a time. But it also happened that this chamber was broken and a species of explosion ensued. He knew a case where the disc kept on revolving at some distance from the chamber, turning on one of the ends of the shaft like a Dutch top. It was without doubt to mitigate the intensity of these tendencies that certain makers of the De Laval turbine had determined to make their chambers of cast-steel rather than of simple cast-iron. It was for these reasons that he regarded his statement justified that it was not possible "in a practical way" to exceed to any great extent the peripheral speed of 1,200 feet per second. Furthermore Mr. Meden said in his Paper on the De Laval turbine that the breaking of vanes occurred more frequently in large turbines than in small ones; this must surely result from the higher peripheral speed.

The author was versed in the question of the resistance of a revolving body. There would be found under his name a Paper on this subject in 1890 in the "Bulletin de la Société de l'Industrie Minérale à St. Etienne." In this work he showed that discs of uniform thickness should begin to fracture at the centre, and Mr. Meden had pointed out exactly certain experiments which showed that this condition would establish itself. He had further shown that greater velocities at the circumference could be attained by utilizing the radial strength of the material, that is to say by using discs enlarged at their centre. The illustration in Mr. Meden's Paper of the De Laval design subsequent to 1897 showed exactly the form of flexible wheel made under the author's specifications since 1895 by the shops of Messrs. Sautter-Harlé in Paris, except that their blades were milled in the rim while the De Laval design showed blades attached by a tenon. In 1895 the form and construction of the De Laval wheel were quite different.

The wear of the blades in the current of steam, which was insignificant in multiple turbines, was on the contrary sufficiently rapid in the turbine of a single wheel, or of two or three wheels only; this followed from the greater velocity of the current of steam. The wear appeared to be due principally to the shock of small particles of water which the steam contained after its expansion in the nozzles. It was diminished by utilising superheated steam. This liquid friction was a cause of important lessening of the output, besides being the occasion which led to the fracture of the vanes after a certain time, as stated above.

Mr. Meden criticised him for having designated the gearing of high-speed turbines by the word "special." These high-speed gearings had this in particular that the teeth being helical were of very fine pitch; they required, therefore, very careful construction. They were cut by machines specially made for this purpose, and not by general or ordinary types. Furthermore, after cutting they were rectified by grinding them with emery powder. This would appear to him to constitute a "special" construction. It was not a matter of much consequence provided the cost of this construction was not excessive, and Mr. Meden said it was not. Some reference had also been made

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in a summary fashion to the question of losses in divergent nozzles. He had already given certain explanations on the subject in the "Revue de Mécanique," September 1903. In his communication Mr. Meden described the De Laval turbo-pumps and presented certain results of experiments. He himself recalled the fact that he had been, he thought, the first to show that very high delivery might be secured with centrifugal turbo-pumps.

His first experiments went back to the year 1900, and he published them in the "Bulletin de la Société d'Encouragement" of 1902. Furthermore, the graphic curves given by Mr. Meden showed that the economy as a function of the discharge of the pump was manifestly inspired by the more general curves, which he himself had pointed out under the name of "Characteristic curves of Centrifugal Pumps."

His first turbo-pump was little else than an experimental apparatus. Its high speed of rotation made it a machine of little advantage to industrial uses according to his opinion, by reason of its delicacy; and he had produced a more satisfactory type by using turbines and multi-cellular pumps. The economy was thus much better, and the use of gearing, which was the delicate part, was completely avoided. His communication referred to turbo-pumps of this latter type. A combined economy of pump and turbine was obtained of more than 42 per cent., if judgment could be based on the figures which he had given for a pump of 350 effective horse-power.

The curves given by Mr. Meden for a pump of 55 horse-power exhibited a maximum economy of 21 per cent. only. This was one-half of the economy of his (Professor Rateau's) design, although allowance should be made for the smaller size. The better result was in fact a consumption of 41 lbs. of steam per horse-power hour in water lifted, with a pressure of the steam of 180 lbs. per square inch, and a vacuum of 25 inches. Under these conditions the theoretical consumption should be 8.72 lbs. of steam per horse-power hour, whence a total economy would result $\frac{8.72}{42} = 0.213$. For a similar capacity of 55 horse-power the

turbines with multiple wheels should easily attain a combined economy of 36 per cent., instead of 21 per cent.

He found himself in complete accord with Mr. Hodgkinson in his statement that, to estimate the net economy of the combined machine, it was wise to start from the pressure of steam in the pipe, in advance of the throttling valve, rather than from the pressure after the throttle. That second pressure could only be used as he had done, if there was only in view the economy of the turbine itself independent of its method of regulation. It was easy to pass from one system to the other, if the two pressures were known before and behind the throttle. He regretted that he had not given in the Tables the pressure in front of the valve. In the case examined by Mr. Hodgkinson this pressure was very nearly equal to that which he stated.

(See also discussion on pages 678-696 ; 713-714 ; and 734-735.)

Fig. 5.

Fig. 6.

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Fig. 11. 400 Electric H.P. Turbo-alternator, 3,000 revs.,
with revolving field-magnets.

Fig. 13. Turbine-driven Pump at Falkenau
(Bohemia).

Fig. 15. *Turbo Fan for Blast Furnace
at Commentry.*

Fig. 18. *Low-pressure Turbine driving two dynamos.
Bruay Collieries.*