RECENT DEVELOPMENTS IN STEAM TURBINES.*

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WESTINGHOUSE TURBINES.

In the early experiments made at The Westinghouse Machine Company, under the direction of Mr. George Westinghouse, a system of turbine elements was developed by Mr. Westinghouse consisting of a combination impulse turbine for the high-pressure stages, and reaction or Parsons elements for the low-pressure portions of the turbine, the object of the combination being to secure, without sacrifice in economy, a more stable mechanical construction by shortening the machine and dispensing with the necessity for three balancing pistons, as obtained in the Parsons design originally exploited.

This combination was particularly desirable at the time of its introduction by Mr. Westinghouse on account of the relatively low powers then required in turbo-generator work, although very successful machines have been in operation here and abroad of the straight Parsons design, which lends itself admirably to certain operating conditions.

The ability of the Curtis element to extract at a given blade speed considerable energy in the high-pressure stage of the turbine with good efficiency resulted in its adoption by Mr. Westinghouse for the high-pressure portion of his machine, by which 20 per cent. to 50 per cent. of the energy of the total expansion is extracted, leaving to the Parsons blading in the low-pressure portions of the turbine the balance of the work to be done with the most efficient turbine elements.

The impulse wheel in this construction replaced a large number of rows of Parsons blading with equivalent economy, because in the smaller powers it was necessary in the straight Parsons machine, on account of total peripheral admission, to make the blade speeds relatively low in order that their height

* Continued from page 128.

might be as large as practicable, to reduce the proportion of leakage by the end of the blades due to the necessary blade clearances. Such a construction of spindles is shown in Fig. 31, where the comparison is obvious.

As this development proceeded, the demand for higher capacity generators increased, and to meet this demand it was necessary to materially enlarge the low-pressure portions of the turbine, which, in some instances, required a dividing of the Parsons blading into two sections, commonly known as the double-flow low-pressure expansion.

In 1903 Mr. Westinghouse filed his application for patent on the combination impulse reaction double-flow turbine illus-

FIG. 31.

Rotors of two 500-horsepower steam turbines; upper spindle with complete reaction blading, lower spindle with impulse and reaction blading.

trated in Fig. 32. With this construction the balancing pistons required by the straight Parsons machine were entirely dispensed with, and while for double-flow construction twice the number of blades were required as for similar conditions with single-flow, yet the application of the impulse wheel to the high-pressure stages made possible, in a great many instances, the proper mechanical design, on account of the considerable shortening of the turbine due to the introduction of the impulse wheel, thus allowing a length of spindle and cylinder of proper mechanical stability.

As this development progressed it was found advisable, on the score of efficiency, to resort to the Westinghouse single double-flow construction. This differs from the straight double-

flow principle in utilizing a single Curtis impulse wheel with two rows of revolving blades, followed by a single barrel of

FIG. 32.

G. Westinghouse fluid pressure turbine.

reaction blades, after which half of the steam is allowed to pass through the drum around a balancing piston (for the single

FIG. 33.

Turbine assembly, longitudinal section. Semi-double-flow turbine.

barrel of Parsons blades) to half of a double-flow expansion of reaction blading, and thence into a double exhaust opening at

either end of the turbine, as illustrated in Fig. 33. The intermediate reaction blading was thus allowed to be proportioned with blades of twice the height and half the number as would have obtained had the entire reaction blading been made double-flow.

Since the efficiency of reaction blading is a function of its length, assuming the blade clearances the same, it is apparent that this single double-flow construction is adaptable to turbine units of moderately high capacity, depending on the rotative speed. This construction is now used in turbo-generator units for alternating current from 3000 to 10,000 kilowatts at speeds of 1500 to 1800 revolutions per minute.

In the development of the higher-powered turbo-generator units (above, say, 10,000 kilowatts at 1500 or 1800 revolutions), owing to the large volume of steam handled, the best proportioning results by making the reaction blading double-flow with a Curtis impulse wheel carrying two rows of blades and utilized in the high-pressure portion of the turbine. Here the reaction blading may be operated at sufficiently high blade speeds to materially reduce the number of rows and make a comparatively short machine, notwithstanding the fact that twice the number of rows of blades are required on account of the double-flow principle than would be necessary if single-flow elements were resorted to. Fig. 34 shows such a construction for a machine of 15,000 kilowatts, maximum rating at 1800 revolutions per minute.

These various constructions have, as will be seen, their particular field of usefulness, depending on the operating conditions and the speeds for which the turbines are designed.

The great importance in the turbine art of this combination turbine has been realized principally in the past five years in the field of electric-power generation, because it gives great flexibility and permits the use of high pressures and high superheats without the deleterious effect these operating conditions might impose with the more difficult mechanical construction of long cylinders and spindles subjected to wide ranges of pressure and temperature. The effect of the combination machine may perhaps better be understood by taking the following practical example, which assumes operating conditions now common in Europe and being introduced in this country:

FIG. 34.

10,000-kilowatt turbine. Built for the City Electric Company, San Francisco.

Steam is to be delivered to the turbine at 200 pounds pressure gauge and 150 degrees superheat, and is to be expanded to $28\frac{1}{2}$ inches vacuum. Its temperature, therefore, entering

FIG. 35.

Turbine assembly, longitudinal section. Single-flow turbine with impulse element.

the turbine would be 538° F. With the Westinghouse turbine the steam would enter the nozzle chamber and be expanded through the nozzles to 78.8 pounds absolute with the impulse

FIG. 36.

Turbine assembly, longitudinal section. Single Parsons type turbine.

wheel operating at a blade speed of 500 feet per second. Its temperature and pressure would therefore be in the impulse chamber 328° F. and 78.8 pounds absolute respectively, which

are sufficiently low to give an entirely satisfactory mechanical construction of the low-pressure end. The construction of such a turbine is shown in Fig. 35. If the steam expansion were to take place in the equivalent Parsons turbine, the construction would be as illustrated in Fig. 36. By comparing the two it will be seen that a very material shortening of the spindle is secured by the replacement in the Parsons turbine of the high-pressure portions, corresponding to the equivalent expansion as would take place in the nozzles of the impulse wheel, and with the same efficiency, because the impulse wheel replaces the most inefficient portion of the Parsons turbine.

In the matter of economies at fractional loads the substitution of the impulse element for its equivalent Parsons blading in the high-pressure portion of the turbine allows greater efficiency, because, with the impulse construction, nozzle control can be resorted to, thus converting into useful work a portion of the energy of the steam which would be lost in a straight Parsons turbine due to throttling. In other words, with the nozzle control construction the impulse wheel becomes an efficient reducing valve when the turbine is operated at fractional loads.

While this advantage may not in a good many instances be very material, and may not warrant the ordinary complications incident to nozzle control, yet in principle it is proper to utilize it for the same flexibility with greater efficiency, and it is only necessary to design a mechanical nozzle control apparatus whose complication is less than the advantages gained by the economies in its use in order to make this arrangement a most satisfactory and desirable one.

The great flexibility of reaction blading over wide variations of pressure distribution, and its adaptability especially to handling large volumes of steam efficiently, make its application particularly desirable in the low-pressure portions of turbines. There are, however, operating conditions now arising in power-plant development which make more suitable the adoption of reaction blading for the high-pressure portion of turbines than has heretofore obtained. This condition has been particularly brought about by the development of the alternating-current generator to operate at high speeds, generating large power. This can probably best be illustrated by some recent

designs prepared by The Westinghouse Machine Company for generator units of 15,000, 20,000, 25,000, and 30,000 kilowatts capacity.

In such units it is of the utmost importance to secure the

FIG. 37.

25,000-kilowatt steam turbine, condensing, 1500 revolutions per minute

very maximum of economy in steam consumption, because the output of the units is so enormous that a small loss in efficiency means a very large increase in the cost of operation.

With the development of the electrical art, alternating-current generators of 5000 kilowatts can be built at 3600 revo-

FIG. 38.

25,000-kilowatt steam turbine, condensing, 750 revolutions per minute.

lutions per minute, from 5000 to 25,000 kilowatts at 1800 revolutions per minute, and from 5000 to 30,000 kilowatts at 1500 revolutions per minute, the different speeds corresponding to 60 and 25 cycles, as used in this country. Likewise, the tur-

bine is susceptible of generating in one unit the necessary power to drive the above generators at these particular speeds.

Some two years ago designs were proposed by The Westinghouse Machine Company for a 25,000-kilowatt unit to operate

FIG. 39.

20,000-kilowatt steam turbine, 200 pounds boiler pressure, 750 revolutions per minute, condensing.

a 25-cycle generator with operating conditions of 200 pounds steam pressure, 200 degrees superheat, and 29.2 inches vacuum.

Four solutions of this condition were worked out, as follows: (1) A Westinghouse double-flow turbine at 1500 revolutions per minute, direct connected to a 25-cycle generator of 25,000 kilowatts maximum capacity, as illustrated in Fig. 37.

FIG. 40.

20,000-kilowatt steam turbine, high-pressure element, 200 pounds boiler pressure, 1500 revolutions per minute.

(2) A single double-flow Westinghouse turbine running at 750 revolutions, direct connected to a 25,000-kilowatt generator, as illustrated in Fig. 38. (3) A tandem compound Westinghouse-Parsons turbine at 750 revolutions per minute, the high-pressure portion being single-flow and the low-pressure portion being double-flow, direct connected to a 25,000-kilowatt

generator, as illustrated in Fig. 39. It is evident that this machine could be arranged cross-compound with two generators of 12,500 kilowatts each. (4) A cross-compound Westinghouse-

FIG. 41.

25,000-kilowatt steam turbine, cross-compound condensing, high pressure 1500 revolutions per minute, 750 low pressure revolutions per minute. Section through low-pressure cylinder.

Parsons turbine with the high-pressure portion running at 1500 revolutions (Fig. 40), direct connected to a 12,500-kilowatt generator, and a double-flow, low-pressure, Westinghouse-Par-

FIG. 41A.

sons turbine running at 750 revolutions, direct connected to a 12,500-kilowatt generator, illustrated in Fig. 41. Fig. 41A shows the comparison to scale of the machines in 1, 2, 3, and 4 respectively.

All of these arrangements would give very excellent economy, and the choice of the unit would depend primarily on the two factors of first cost and economy, assuming that in each case the reliability for continuous operation is the same.

FIG. 41B.

A close study of the four arrangements indicates that the Westinghouse double-flow turbine at 1500 revolutions per minute, direct connected to a single generator (Fig. 37), is the cheapest construction. The large areas required in the low-pressure stages of this turbine make high velocity and long

FIG. 41C.

length of blade essential, with the necessity of careful designing to properly take care of the stresses due to centrifugal force in the low-pressure end.

The most economical combination of the four is the cross-compound Westinghouse reaction turbine with the high-pressure portion running at 1500 revolutions and the low-pressure por-

tion at 750 revolutions. With this arrangement the highest efficiency is obtained, because with the large volumes of steam required to develop the high power and the ability to combine the unit into high- and low-pressure cylinders, running at 1500 and 750 revolutions per minute, gives the condition for best blading proportions throughout the turbine without departing from standards of practice already established. This machine would be very remarkable for its high efficiency, which it is believed could not be reproduced with any other known form of turbine. The construction, however, is considerably heavier

FIG. 41D.

than the single unit commented upon above, and would be more costly to construct and install.

In such a case as the one cited there is little doubt that the cross-compound Parsons machine will give the maximum efficiency. At powers and speeds attainable with single alternating-current units of, say, below 15,000 kilowatts capacity the Westinghouse double-flow turbine between 10,000 and 15,000 kilowatts will be nearly, if not equal, to the cross-compound straight reaction turbine under like operating conditions, and any difference in efficiency would probably be offset by the difference in first cost in favor of the Westinghouse double-flow machine. Between 10,000 and 4000 kilowatts capacity the Westinghouse single double-flow machine would undoubtedly be

the proper construction, taking into consideration the first cost and efficiency.

The demand for turbo-generator machines is greatest in kilowatt output between the sizes of from 4000 to 15,000 kilowatts maximum rated capacity, within which range the Westinghouse single double-flow and Westinghouse double-flow machines most admirably meet the commercial conditions both with respect to cost and efficiency.

Below 4000 kilowatts capacity the Westinghouse turbine consisting of (1) an impulse wheel followed by single-flow reaction blading, illustrated in Fig. 35; (2) an impulse wheel followed by single double-flow reaction blading (Fig. 33); (3) an impulse wheel followed by double-flow reaction blading (Fig. 37), and (4) a single-flow Westinghouse reaction turbine (Fig. 36), represent the various choice of machines best suited for operating conditions (*a*) dependent on the initial pressure and superheat, (*b*) on the number of revolutions at which the generator or other driven machine rotates, and (*c*) on the capacity.

Thus with a speed of 3600 revolutions per minute, driving a 60-cycle generator at, say, 3000 kilowatts, the turbine design would follow the combination impulse double-flow reaction machine for best efficiency and lowest cost (Fig. 37). If, however, the generator to be driven was 25-cycle, with an allowable maximum speed of 1500 revolutions per minute and a capacity of 3000 kilowatts, the choice would lie between a design of a straight single-flow reaction turbine and a combination impulse single-flow reaction turbine, the former being used for moderate degrees of superheat and initial pressure, and the latter for high superheat and high pressure.

In general, however, the application of the Westinghouse turbine, consisting of an impulse element for the high-pressure portion and reaction blading for the low-pressure portion, is admirably adapted for complete expansion turbines over wide ranges of power and speed, and since the introduction of this type in 1903, following the initial designs by Mr. Westinghouse, a large proportion of the firms building steam turbines have utilized this construction, either with Parsons or Rateau blading, following the Curtis impulse element or elements in the high-pressure end.

The advantages of the Westinghouse type are: (1) the

ability to shorten the machine by the introduction, in the high-pressure stages, of an impulse wheel of the Curtis type without material loss in efficiency where the powers and speeds are relatively low; (2) the reduction of pressure and temperature within the turbine cylinder by expanding the steam initially in fixed nozzles; (3) the elimination of two or more balancing pistons as obtain in the straight Parsons machine, consequently reducing the small, balancing piston clearance, and leakage of steam incidental to their introduction; (4) the retention of the most efficient turbine elements in that portion of the machine

FIG. 42.

Group of S. N. C. turbines.

where they can be most effectively employed for efficiency; (5) the ability to secure higher economies at fractional loads by the introduction of nozzle control on the initial impulse element; and (6) the retention of the drum construction for rigidity of spindle and low cost of manufacture, referring particularly to the reaction blading.

The disadvantages of the Westinghouse construction are: (1) for extremely high powers, say, above 15,000 kilowatts capacity, the impulse wheel is less efficient than its equivalent Parsons blading, depending, however, on blading proportions; (2) for capacities of 300 kilowatts at 3600 revolutions per minute, and 750 kilowatts at 1500 revolutions per minute, the construction, while applicable at these powers and speeds, becomes

as expensive with the same efficiency as turbines of other designs.

For the smaller powers, therefore, The Westinghouse Com-

FIG. 42A.

S. N. C. turbine. 05 S. N. C. nozzle blocks and reversing chamber.

pany developed some three years ago impulse turbines of the re-entry type, illustrated in Fig. 42, which are adaptable down to units of 1 kilowatt capacity, having the advantage of one revolv-

FIG. 43.

S. N. C. turbine. Arrangement of nozzles and reversing chambers, showing action of steam. Two stages, with two velocity drops to each stage.

ing row of blades, with which it is possible to obtain (α) a subdivided impulse turbine with one reversal (Fig. 42A), equivalent to a Curtis turbine with two revolving rows of blades, used for

medium-pressure drops; (*b*) a subdivided impulse turbine with two reversals (Fig. 43), equivalent to two Curtis elements with four revolving rows of blades, and (*c*) subdivided impulse elements with one or two reversals followed by one or more Rateau stages, or a series of Rateau stages without Curtis elements.

This construction has the advantage of the simplicity of the De Laval turbine operating efficiently at such rotative speeds as are commercially required for the purposes of direct connection to generators, pumps, blowers, and other machines which require relatively good efficiency with relatively low speed of rotation.

COMPARISON OF STEAM TURBINES.

Comparison of the relative merits of the turbines previously described cannot be fairly made in any general terms, as certain combinations of turbine elements must be varied to suit the particular operating conditions which may be imposed. If, however, the discussion be limited to the most common type, namely, the complete expansion turbine and its operating requirements, subdivided with definite conditions imposed, the general condition influencing the selection of a good type may be stated.

The three principal items in considering the complete expansion turbine for commercial purposes are (1) reliability, (2) capacity, (3) economy.

RELIABILITY.

In general, it may be stated that the reliability of turbines of different types depends more on the design of the details than on the inherent merits of the turbine elements themselves.

One of the principal items in detail design to which all turbine builders pay particular attention is the method of fastening the blades to the disks or drums, as the case may be. For many years after the introduction of the Parsons turbine the blading was secured to the drums by the well-known method of inserting, alternately, in a slightly dovetailed groove, the blades and packing pieces, and after a row had been so inserted in its groove, the packing pieces were caulked between the blades, to secure them by friction, as well as by a small nick in the base

of the blade. As the development in turbine design progressed it was found necessary with longer blades to lash the ends in some manner. Some ten years ago this was accomplished by silver-soldering a strip into the blades to secure the desired rigidity, but principally to prevent their vibration, due to their rotative speed and the velocity of the steam passing through them.

This general method of blading was devised by Mr. Parsons in 1896, and has been followed for a number of years by builders

FIG. 44.

Samples of the Westinghouse blade lashing.

of that type of turbine. To those skilled in turbine blading work the caulking process is known to be an excellent blade fastening. The first change which was made in this country in the Parsons form of blading was in the adoption of a different method of shrouding the blades, which consisted of punching a hole, in the shape of a comma, near the end of the blade and inserting therein a wire, drawn to the comma section. After the blades have been inserted in the spindle and cylinder and the comma wire put in place, a tool is used to shear the tail end of the comma at each blade and bend it around, as illustrated in Fig. 44. When this operation is performed, it is evident that there exists between

the blades a tight strut which effectively diminishes the tendency to vibration, is easily applied, and makes a much better and safer construction than the old method of silver-soldering. This construction substantially amounts to shrouding the blades without, however, creating any blank space between the real tip of the blade and the cylinder or spindle.

This lashing or shrouding of the blades prevents their individual vibration, but does not prevent their vibrating as a group.

It is a fair statement to make that scarcely any turbine of anybody's make ever gave trouble due to the blades breaking or coming out because of centrifugal force. It is not centrifugal force that breaks the blades, but vibration of the blades themselves, and until lashing of the longer blades was resorted to, breaking was occasioned by the individual vibration. The lashing or shrouding of the blades therefore reduced very materially this difficulty. Twelve years ago The Westinghouse Machine Company experienced some unaccountable breaking of blades which were made of bronze. This was attributed to fatigue of the metal of the blade, or repeated heating and cooling, and led, as a cure to the difficulty, to the adoption of steel as blade material. Later experience and analysis of these troubles reversed the impression first obtained and led to the conclusion that the difficulty was really due to blade vibration.

In the last few years the capacities of turbines have increased so enormously, resulting in long blade lengths, that it became necessary to devise better means of insuring blade stability, and with this in view the Westinghouse Company has developed in the past two years the blade construction illustrated in Fig. 45. The lashing or shrouding must not be continuous, as provision for expansion and contraction must be made, since the coefficient of expansion of the spindle or cylinder, as the case may be, is not the same as that of the blades, the material being different. Therefore, all lashed or shrouded blades must be arranged in segments not exceeding, say, two feet in length for large diameters. There is no reason, however, why the segments of blading will not vibrate as a whole, just as the individual blades do when not properly lashed, except that on account of lashing or shrouding the frequency of vibration will be very much higher, and the amplitude consequently less. Lashing or shrouding, therefore, is a palliative against

vibration, and not a cure. It is the practice of the firm with which I am connected to lash all reaction blades above one inch in length, and very long blades may have as many as three or four rows of lashing wire.

The desirable thing, therefore, in turbine blading is to provide a blade which, since it is impossible to entirely eliminate vibration, will be of such form that the vibration, should it exist, will not concentrate the stresses at any one point. A turbine blade might be likened to a leaf spring secured at one end,

FIG. 45.

Turbine blading.

with the other end free to vibrate. If this leaf spring has a tapering section so that when loaded at its outer end the stresses throughout are uniform, it may be allowed to vibrate indefinitely without breaking. If the condition is reversed and the greatest section of the spring is at the outer edge, it will, on being deflected, have such deflection concentrated at the inner end close to the point of support. The concentration of these stresses at the supporting point would, in a short time, cause failure of the material and the breaking of the spring. If the spring were made of a uniform section throughout its length the

condition would, of course, be better than that as last stated, but it would be a question of sufficient time and sufficient amplitude of vibration for the fracture to take place at the point of support.

Some turbine builders diminish the section of the blade at the point of support; others maintain a straight section, while in the modern development of the Westinghouse construction (Fig. 45) the base of the blade where it is inserted in the spindle or cylinder is, in the process of manufacture, upset to a thicker section than at any other portion, such thickening extending beyond the point where the blade issues from the groove. It will also be noticed that there is provided an absolute interlocking system, the strength of the under-hanging foot of the blade being equal to that of the blade itself; also that this construction involves no caulking process which requires skill and is difficult to inspect. The foot at the ends of the blades is obtained by upsetting or forging, and the dies for carrying out this operation are so constructed that the blade material is increased at the root, tapering off to a standard blade section some distance above the surface of the blade-carrying member. This effectually eliminates concentration of stresses at the root of the blade and gives a construction approximating a beam of equal stress.

The advantages, therefore, of this new type of reaction blading may be summarized as follows:

(1) Absolute interlocking of the blades with the blade-carrying member, the strength of all fastening means being equal to the strength of the blades themselves.

(2) No caulking is employed. This permits ready inspection of the blades before insertion, and assurance that they are firmly secured when installed.

(3) The blades may be easily removed from their grooves and replaced an indefinite number of times without mutilation or injury to the grooves.

(4) The section of the blading is tapered, being heavier at the root, eliminating danger from vibration of the blades.

Development in blade-holding means for impulse turbines has been necessitated in the last two or three years, due to the increase in blade speeds. Prior to three years ago the blade speeds of the impulse wheel of the Westinghouse type did not exceed 400 feet per second. The blade fastening in the con-

struction used in these wheels is as shown in Fig. 46. The more recent developments, however, with the increase in impulse blade speeds, have resulted in the construction shown in Fig. 47, which materially reduces the weight of the impulse wheel at its head, and consequently the stresses in the disk, at the same time giving a blade fastening which is as strong as the blade itself. With this arrangement blade speeds of from 600 to 700 feet per second may be obtained without undue stresses in a through disk construction.

FIG. 46.

FIG. 47.

Blade fastenings.

Many details of design have been refined as the development of the turbine has progressed, such as better proportioning of cylinders and spindles to withstand distortions due to differences in temperatures, the supporting of the cylinders on the bed-plates, details of nozzle construction, balancing pistons, etc. Space will not permit a minute description of these recent developments, but it may be stated in general that in the early Parsons construction in 1900 it was thought impracticable to build units of larger capacity than 400 or 500 kilowatts without subdividing the cylinder construction into high- and low-pressure portions. The turbine represented in Fig. 30, constructed in 1901, marks a distinct advancement in the art in this respect, and was at the time of its introduction severely criticised for its large capacity,

and predictions were made for its failure in service. This machine might be said to have been the forerunner of the enormous development in power capacities in turbine work, which to-day reach an output of from 20,000 to 30,000 kilowatts in a single cylinder.

MAXIMUM OUTPUT OF TURBINES.

It is evident that the De Laval turbine cannot be utilized for large capacity, and the comparison will therefore have to be made between the Westinghouse or Parsons turbine, the Rateau turbine, and the Curtis turbine.

The maximum output of any turbine depends mainly on (1) the specific volume of steam at the exhaust of the turbine, which in turn depends on the vacuum; (2) the outlet velocity, which depends on the permissible leaving loss, and hence the efficiency; (3) the mean diameter of the blading of the last row of blades, which depends upon the maximum permissible peripheral speed; and (4) the maximum permissible blade height, which depends on the method of fixing the blade and the construction of that part of the spindle holding the last row of blades.

(1) Other things being equal in the Parsons, Rateau, and Curtis turbines, the specific volume of the steam passed by the last row of blades is the same, and with like blade speeds, blade heights, and blade angles no difference can exist between the three types as regards maximum capacity. Likewise, with equal blade speeds the leaving loss will be proportional to the outlet velocity. The maximum capacity of output, therefore, is dependent primarily on the length and mean speed of the last revolving row of blades and on the mean blade diameter, in any type of turbine. If a drum construction is used to carry the last row of blades, it can be shown that (other conditions remaining the same) a disk turbine can be built having twice the maximum output of a drum turbine. By those familiar with turbine construction, reaction blading is generally coupled with a drum construction. The recent designs of the Westinghouse high-power turbines, however, have the last rows of blading carried by a through disk and not by a drum, as may be seen by reference to Fig. 37.

The construction of the blade fastening in the Westinghouse turbine can be shown to be as strong as the blade itself; conse-

quently the limit of output, considering the last row of blades in this type of turbine, would be proportional to the permissible stresses in a through disk, which allows, therefore, the maximum capacity that any turbine is capable of, and particularly so as the construction of the Curtis and Rateau machines involves a disk with a hole in the centre, which, as shown in Appendix III, cannot be as strong and reliable as a disk without a hole in the centre.

The specific volume of one pound of steam expanding adia-

FIG. 48.

batically from 26 inches to 29 inches vacuum is plotted on the curve, Fig. 48. With a disk speed of 500 feet per second in the low-pressure end of the turbine there will be required, with reaction blading, two moving rows to complete the expansion from 26 inches to 29 inches vacuum. The height of the revolving blades to accommodate the increase in specific volume between 26 inches and 29 inches vacuum will be relatively as 1 to 3.4, depending somewhat on the capacity of the machine.

If the leaving losses in a Rateau or Curtis turbine are to be the same as in the reaction turbine, the last blades must be of equal length and equal speed for the same leaving loss and capacity, since the last moving row of the reaction turbine for

like efficiency may be made an impulse blade, which would result in a reduction of the blade height compared to what it would be if made for pure reaction.

The energy in the adiabatic expansion of one pound of steam from 26 inches vacuum to 29 inches vacuum is plotted in the curve in Fig. 49, from which it will be seen that between 28 inches and 29 inches vacuum the energy is practically the same as between 26 inches and 28 inches vacuum. With turbines of high capacity, therefore, it is essential to obtain very high vacuum in the exhaust on account of the enormous energy contained in

FIG. 49.

the expansion at low pressures. If we assume, therefore, that turbines of high capacity are built for operating conditions with high vacuum, say 29 inches, the last row of blades must be considerably higher than the preceding row to handle the large increase in specific volume of the steam passing to the exhaust. Again, as the stress in a disk due to the blades is proportional to the weight of the blade, the longer the blade and the higher the blade speeds, the greater the necessity for a construction to properly care for the stresses due to centrifugal force; and as the last one or two rows in the reaction turbine can be secured to a through disk, this design lends itself to a turbine of

maximum output, because (1) the rows immediately preceding the last rows of blades handle steam of considerably less specific volume, and (2) because in the reaction turbine a design with through disks supporting the last row or two of revolving blades can be made with such through disk construction, which, as shown in Appendix II, gives the maximum strength.

If the low-pressure portions are made double-flow, as is common practice with the reaction turbine, the maximum capacity obtainable is, of course, double the capacity of a single-flow machine. The facility with which reaction turbines are made double-flow, therefore, allows a maximum output of at least double the capacity obtainable in a design of Rateau or Curtis machines with equal blade speeds and equivalent operating conditions, and with equal reliability in service.

ECONOMY.

In order to compare the efficiencies which can be obtained with the Curtis, Rateau, and Parsons machines, reference is made to Figs. 10 to 16 inclusive, and to the consideration that the maximum efficiency for certain blade speeds in any type of turbine is dependent on (a) design of the blade elements, (b) the relative ratio of the velocity of the blades to the velocity of the steam, (c) the capacity of the turbine, and (d) the operating conditions.

Let S_b = mean blade speed,

Let S_s = mean steam speed,

Let E = blade efficiency.

If steam is admitted to a wheel with one row of impulse blades (Fig. 10), at a certain speed, S_s , and the blades are revolved at different speeds, S_b , theoretically, the efficiency is at a maximum for a certain blade speed given by the ratio,

$$\frac{S_b}{S_s} = \frac{\text{mean blade speed}}{\text{mean steam speed}} = \frac{1}{2}$$

For any other ratios of $\frac{S_b}{S_s}$ the efficiency will be smaller, becoming zero when S_b = zero and when S_b = S_s .

The actual maximum efficiency and the ratio $\frac{S_b}{S_s}$ at which it is obtained depend mainly on the angles of the blades and

the frictional losses in the machine. Curve *c*, Fig. 50, represents a fair average of test results actually obtained in well-proportioned Rateau blading, the maximum efficiency being $82\frac{1}{2}$ per cent. for $\frac{S_b}{S_s} = .47$ approximately.

For impulse wheels with two rows of moving blades the maximum efficiency is obtained for $\frac{S_b}{S_s} = .28$ approximately,

FIG. 50.

Efficiency curve. Parsons, Rateau, and Curtis wheels.

as shown on curve *b*, Fig. 50, such maximum efficiency being $72\frac{1}{2}$ per cent.

For impulse wheels with three, four, or five rows of blades the maximum efficiency is obtained for $\frac{S_b}{S_s}$ less than $\frac{1}{6}$, $\frac{1}{8}$, and $\frac{1}{10}$ respectively. Of these, however, only the impulse wheel with three rows of blades is of practical importance, the actual efficiencies obtainable being given by curve *a*, Fig. 50, showing a maximum efficiency of 60 per cent. for $\frac{S_b}{S_s} = .17$ approximately. The efficiency of impulse wheels with more than three rows of moving blades is less, and they are therefore not

used except in cases where efficiency becomes a secondary matter.

For reaction turbines with multiple stages properly proportioned, and with relatively large capacities, the maximum efficiency is obtained for $-\frac{S_h}{S_s} = .9$ approximately, at which the efficiency obtained on test has reached 90 per cent., with corresponding reductions when greater or less values of $-\frac{S_h}{S_s}$ are used, as given by curve *d*, Fig. 50. The total pressure drop with the best efficiency obtainable must have: one wheel with three rows of blades, maximum efficiency 60 per cent.; or three wheels with two rows of blades, maximum efficiency $72\frac{1}{2}$ per cent.; or twelve wheels with one row of blades, maximum efficiency $82\frac{1}{2}$ per cent.; or forty-eight rows of reaction blades with a maximum efficiency 90 per cent.

From a comparison of these efficiency curves it is evident that the reaction blading is most desirable for the low-pressure portions of turbines where the blade heights become suitable for maximum efficiency, due to the reduction in the percentage of leakage loss, which vitiates the efficiency of the blade in proportion to such loss, and consequently becomes greater as the height of the blade becomes less. With, therefore, a blade clearance of ten-thousandths of an inch in the low-pressure portion of a reaction turbine with blades ten inches in length, the loss would be approximately four-tenths of one per cent. If, however, the blades were one inch in length, the loss would be approximately four per cent.; and as the capacity of the turbine diminished, requiring smaller blade heights, the leakage per cent. would have sufficient vitiating effect on the economy to warrant the introduction (in place of the reaction blading in the high-pressure stages) of a Rateau or Curtis element to get equal efficiency and the large initial drop in pressure which is desirable for mechanical reasons, given elsewhere in this paper. On the other hand, where the volume of steam, and consequently the capacity of the turbine, becomes great, on the score of efficiency, the impulse elements in the high-pressure stages can be dispensed with to advantage, because the reaction blading gives, under such conditions, reaction blade proportions for maximum efficiency. Such a condition has already been discussed in comparing the relative merits of designs for a 25,000-kilowatt machine.

The frictional resistance due to the revolving parts would undoubtedly favor the drum construction or reaction turbine, because the surface exposed in an equivalent design becomes less than in equivalent Curtis and Rateau machines.

The losses are, in any event, a small proportion of the total output of the machine, especially in high capacities, and the comparison of the advantages of the different types in this respect may be eliminated.

The improvement in economy due to the development of turbo-generator units up to 1906, of turbines built in this country, is indicated in Table I.

The most important tests obtained in the last five years on the complete expansion turbine, together with the authority for such tests, are given in Table II.

The efficiency of the turbines as shown in the tables is the ratio of the

$$\frac{\text{Actual useful work done by the turbine.}}{\text{The mechanical equivalent of the heat drop under the operating conditions, due to adiabatic expansion.}}$$

This ratio is commonly known as the Rankine-cycle efficiency. With turbo-generators this efficiency is generally stated by taking the actual energy output at the switchboard in kilowatts, as the numerator of the fraction expressed above, and is consequently less in value than the actual efficiency of the turbine itself, because the efficiency of the generator is less than unity.

When comparing the performance of turbines, the Rankine-cycle efficiency is a more correct basis than the water rates obtained on tests, because if the operating conditions are not similar the comparison of water rates is an unfair one, unless proper correction factors are applied.

Turbines are generally designed for given operating conditions, involving (*a*) quality of steam (superheat), (*b*) pressure, and (*c*) vacuum, and the correction for the change in these operating conditions as applied to a given machine must be taken into account when a comparison of the steam consumption of different machines is made.

Such corrections for steam consumptions should also be divided into two classes: (1) a comparison of turbines designed for different operating conditions, and (2) a comparison of turbines operating under conditions other than those for which they were designed.

For such corrections, standard steam conditions should be assumed to which the performance of all turbines may be reduced by applying proper correction factors.

Such standard conditions as now obtain in this country may be taken as follows:

For high-pressure turbines, 175 pounds pressure per square inch (gauge), 100° F., 28 inches vacuum (referred to 30-inch barometer);

For low-pressure turbines, 16 pounds pressure per square inch (absolute), 0° F. superheat, 28 inches vacuum, (referred to 30-inch barometer).

In Appendix III a list of correction factors is given for application to the general cases and operating conditions met in this country.

These correction factors are of great advantage to prospective purchasers of turbines, in order that they may more intelligently specify those operating conditions which are best suited to the economy of their plant.

FIELD OF APPLICATION OF THE TURBINE.

In the past five years the field of application of the turbine has been wonderfully increased, and the flexibility inherent in its construction has been developed by its adoption to: (1) non-condensing operations, (2) utilization of low-pressure steam, (3) reducing or bleeder turbines, and (4) mixed-pressure turbines.

In general, it may be stated that the distinguishing feature of these applications resolves itself into expansion of steam to and from atmospheric pressure.

Space will not permit a lengthy discussion of the various types of these special cases of the complete expansion turbine, except to say that conditions are arising to continually broaden the field of application of these special machines. Some general considerations of the designs of such machines obtain, however, which may be briefly stated as follows:

NON-CONDENSING TURBINES.

The non-condensing turbine may be made for any given condition, utilizing either the reaction or impulse machine with approximately the same efficiency, except where relatively large

TABLE I.

No.	Maker of turbine	Type	Date of test	Load K.W.	R.P.M.	Steam pressure Lbs. Abs.	Super-heat	Vacuum referred to 29.92" barom-barometer	Pounds of steam per KW.Hr.	Efficiency ratio	Reference
8	Westinghouse Machine Co.	Parsons	1903	1098	1200	170.1	11.56	26.91	19.10	58.9	Purchaser's Test, Hartford Elec. Lt. Co.
84	Westinghouse Machine Co.	Parsons	1905	5601	750	191.2	131.30	27.41	16.20	60.7	Purchaser's Test, Penn. T. & T. Co.
253	Westinghouse Machine Co.	Parsons	1907	9865	750	191.7	96.09	27.31	14.98	67.2	Electrical World, Jan. 29, 1908.
	General Electric Co.	Curtis	1907	16186	720	190.7	147.00	29.47	12.99	62.1	Electrical World, Jan. 29, 1908.

TABLE II.

No.	Maker of turbine	Type	Date of test	Load K.W.	R.P.M.	Steam pressure Lbs. Abs.	Super-heat	Vacuum referred to 29.92" barom-barometer	Pounds of steam per KW.Hr.	Efficiency ratio	Reference
1	Esie Brunner M.F.G.	Curtis-Parsons	1910	2128	1500	156.2	120.31	27.89	13.82	71.8	Periodische Mitteilungen, Zeit. d. V. D. Ing., 12/10/10.
2	Esie Brunner M.F.G.	Curtis-Parsons	1910	6000	960	184.9	197.64	28.18	12.56	71.3	Periodische Mitteilungen, Zeit. d. V. D. Ing., 12/10/10.
3	Esie Brunner M.F.G.	Curtis-Parsons	1910	7442	960	192.0	205.51	28.18	12.625	68.3	Periodische Mitteilungen, Zeit. d. V. D. Ing., 12/10/10.
4	Westinghouse Machine Co.	Curtis-Parsons	1910	9173	1800	181.7	59.97	27.81	14.57	68.9	Trans. A. S. M. E., Vol. 32.
5	Esie Brunner M.F.G.	Curtis-Parsons	1910	1416	1260	128.2	135.69	27.60	15.18	68.8	Periodische Mitteilungen, Zeit. d. V. D. Ing., 12/10/10.
6	Allis-Chalmers	Parsons	1908	4300	1800	186.4	197.97	27.06	14.02	68.4	Shiley Jour. of Eng. 1/17/11
7	Richardson & Westgarth	Parsons	1910	6237	1200	203.7	176.00	29.10	11.90	68.4	Electrician, Vol. 66, p. 862, 1911.
8	General Electric Co.	Curtis	1912	12460	720	207.5	191.00	27.52	13.63	68.2	Jour. A. S. M. E., Aug. '12, p. 1109.
9	Brown-Boveri & Cie.	Curtis-Parsons	1911	3053	1360	150.2	146.40	29.00	13.01	68.0	Dinglers P. J., 6/17/11.
10	Brown-Boveri & Cie.	Curtis-Parsons	1911	1730	1500	176.4	214.50	27.08	14.23	67.5	Zeit. f. d. G. Turb., 5/30/11.

11	James Howden & Son.....	1909	6383	1000	202.7	136.08	27.33	14.395	67.5	Engineer, London, 10/29/09.
12	M. A. N.....	1910	1200	3000	180.7	180.53	27.40	14.21	67.4	Zett. d. V. D. Ing., 12/10/10.
13	R. Crutcheon & Westgarth.....	1910	5060	1200	203.7	166.00	29.13	12.10	67.3	Electrician, Vol. 66, p. 862, 1911.
14	Westinghouse Machine Co.....	1906	9830	750	192.2	96.43	27.32	15.15	67.0	Trans. A. S. M. E., Vol. 32.
15	British Westinghouse.....	1911	5066	1500	190.2	174.31	28.68	13.00	67.0	Electrical Review, 6/23/11.
16	Brown-Boveri & Cie.....	1910	3764	1500	161.2	106.78	28.77	13.04	66.8	Zett. f. d. G. Turb., 5/30/11.
17	Escher Wyss & Co.....	1910	2552	3000	103.9	203.70	28.50	13.01	66.6	Zett. f. d. G. Turb., 2/20/11.
18	General Electric Co.....	1907	7326	720	184.4	134.07	28.25	13.73	66.5	Jour. A. S. M. E., Aug. 12, p. 1169.
19	British-Thomson-Houston	1911	2987	1347	184.7	144.07	26.73	15.00	66.5	Engineering, 10/30/11.
20	Bergmann.....	1909	1545	1500	188.5	204.04	28.59	12.97	66.4	Zett. d. V. D. Ing., 12/10/10.
21	Westinghouse Machine Co.	1907	9870	750	191.7	95.00	27.35	15.00	66.3	Zett. d. V. D. Ing., 1907, p. 1782.
22	General Electric Co.....	1907	12000	750	200.0	125.00	28.00	14.22	66.2	Jour. A. S. M. E., Aug. 12, p. 1169.
23	Orclikon.....	1911	3166	1500	213.9	275.12	20.25	11.44	66.1	Engineering, 10/20/10.
24	General Electric Co.....	1911	6500	1800	180.0	100.00	28.00	14.78	66.1	Jour. A. S. M. E., Aug. 12, p. 1169.
25	Escher Wyss & Co.....	1911	5118	1000	133.7	199.48	27.55	15.18	65.7	Dinglers P. J., 7/15/11.
26	Bergmann.....	1910	2477	1500	140.0	168.91	28.31	13.93	65.6	Elect. Zett., 4/20/11.
27	Brown-Boveri & Cie.....	1903	3500	1360	156.4	137.40	28.84	13.71	65.6	Zett. d. V. D. Ing., 12/10/10.
28	Westinghouse Machine Co.	1909	11466	750	191.7	105.65	28.07	14.45	65.5	Trans. A. S. M. E., Vol. 32.
29	Escher Wyss & Co.....	1910	4189	1000	179.7	183.08	28.66	13.39	65.5	Zett. f. d. G. Turb., 2/20/11.
30	Richardson & Westgarth.....	1910	4256	1200	206.7	162.00	29.19	12.40	65.1	Electrician, Vol. 66, p. 862, 19 1.
31	British Westinghouse.....	1911	2930	1500	210.2	187.90	28.18	13.72	64.9	Electrical Review, 4/28/11.
32	A. E. G.....	1908	4249	1500	188.3	285.50	28.40	13.02	64.9	Stodola 4th ed., p. 494.
33	Escher Wyss & Co.....	1911	3540	1500	155.1	107.50	28.21	15.07	64.8	Stodola 4th ed., p. 494.
34	F. Ringhoffer.....	1908	3060	1000	170.7	101.18	27.60	13.52	64.8	Dinglers P. J., 7/15/11.
35	Brown-Boveri & Cie.....	1907	3522	1500	156.9	136.00	28.945	13.70	64.8	Zett. d. V. D. Ing., 12/10/10.
36	Es. Masch. Fabr.....	1910	3500	1500	177.7	142.00	28.435	14.07	64.8	Zett. d. V. D. Ing., 1910, p. 1500.
37	M. A. N.....	1910	1250	3000	183.1	207.80	28.82	13.09	64.4	Zett. d. V. D. Ing., 12/10/10.
38	Brown-Boveri & Cie.....	1910	3000	1500	165.0	258.91	27.68	14.75	64.3	Die Turbine, 6/20/11.
39	C. A. Parsons & Co.....	1907	5164	1200	214.3	121.26	28.95	13.18	64.3	Stodola 4th ed., p. 439.
40	British Westinghouse.....	1911	3645	1500	208.3	180.00	27.78	14.03	64.3	Electrical Eng'g, Vol. 47, p. 461, 11.
41	A. E. G.....	1906	3766	1500	185.2	247.00	28.78	12.80	64.2	Zett. d. V. D. Ing., 1907, p. 386.
42	Escher Wyss & Co.....	1910	1641	5000	221.0	281.63	27.91	13.08	64.1	Zett. f. d. G. Turb., 2/20/11.

capacities are demanded, in which case the reaction turbine would have the advantage in efficiency. Since a non-condensing operation would require approximately twice the amount of steam for a given output as would be necessary with a complete expansion turbine, opportunity would be afforded for better proportioning of the reaction blading to give maximum efficiency, and, due to the incomplete expansion, a relatively short spindle would result on account of a smaller number of rows of

FIG. 51.

Longitudinal section, 15 N.C., 3600 revolutions per minute, single-flow high-pressure turbine with Kingsbury thrust-bearing.

blades being required to absorb less than half of the total energy drop occurring in a complete expansion machine. A design of such a turbine is illustrated in Fig. 51.

LOW-PRESSURE TURBINES.

In a low-pressure turbine design the reaction elements again lend themselves admirably to the efficient extraction of the energy of low-pressure steam, because, for a given output, the blade proportions of the low-pressure turbine would be practically twice the capacity of those required in a single expansion turbine, resulting, as in non-condensing operation, in blade proportions of maximum efficiency, a short, simple construction of the spindle and cylinder, and (with moderate capacity) the

elimination of balancing pistons by resorting to the double-flow construction. The same arguments for simplicity would, of course, apply to impulse turbines, but the fact that reaction blading is, in general, best adaptable to the low-pressure portions of any turbine should be a distinct advantage in favor of such a machine in low-pressure work. Figs. 52 and 53 indicate single- and double-flow low-pressure machines as built by The Westinghouse Machine Company.

FIG. 52.

Low-pressure single-flow turbine.

REDUCING OR BLEEDER TURBINES.

Where it is desirable to utilize steam at about atmospheric pressure for heating or industrial purposes, the turbine lends itself admirably to fulfil such a condition. It is extremely simple to bleed steam out of the turbine, at any particular pressure, in quantity proportional to the amount of flow through the turbine. Suitable automatic valves may be applied to the bleeder exhaust of the turbine to maintain (irrespective of the load the turbine may carry) that quantity of steam required to maintain a constant pressure in the bleeder line.

The conditions having been determined as to the amount of steam to be supplied from the turbine, the proportions of the elements beyond the bleeder valve are made to utilize the steam

remaining under the operating conditions imposed, thus extracting the maximum amount of energy from the complete expansion of the steam required for generating power, and giving at the same time a constant steam supply at a given pressure (usually about 15 to 20 pounds absolute) for heating or industrial purposes. Likewise, with suitable valve arrangements, steam at relatively low pressures, say 15 pounds absolute, can be bled into the low-pressure portion of the turbine, and the energy from the expansion of such steam to a high vacuum may be utilized in the low-pressure portion of a bleeder turbine for the

FIG. 53.

Low-pressure double-flow turbine.

development of power. The bleeder valves and the initial steam valve, in the high-pressure portion of the turbine, may be governor-controlled and make, therefore, a purely automatic machine, which will take care of variable operating conditions by the governor control. An illustration of such a machine is shown in Fig. 54, where the high-pressure portion is designed with an impulse wheel and the low-pressure portion with reaction blading.

MIXED-PRESSURE TURBINES.

The mixed-pressure turbine is really a modification of conditions requiring a low-pressure turbine. Suppose a source of

low-pressure steam being available from which it is desirable to extract the energy in the development of power, and where, at certain times in the operation, sufficient low-pressure steam is not available to generate the amount of power required: To

FIG. 54.

Bleeder turbine assembly.

FIG. 55.

Low-pressure turbine with live steam valve.

suit this condition, a low-pressure turbine is supplemented by a high-pressure element which can utilize steam at high pressure through complete expansion. When sufficient low-pressure steam is available to furnish the required power, the high-pressure steam is cut off by the governor or by other means, and when the low-pressure steam is not sufficient for the power required,

the governor of the turbine admits steam to the high-pressure element to supplement the low-pressure supply for the development of the power required. This same arrangement can be accomplished with the low-pressure turbine by applying a high-pressure valve admitting steam into the low-pressure line direct. Figs. 55 and 56 illustrate machines of this construction, and the desirability of either type depends entirely on the operating conditions imposed.

FIG. 56.

Low-pressure turbine with live steam valve, installed at Peace Dale Manufacturing Company, Peace Dale, R. I.

RECENT MISCELLANEOUS TURBINE APPLICATIONS.

Due to the recent development of small turbines of relatively low rotative speeds that have recently come into quite extended use for driving auxiliaries in power plants and for other applications, apparatus which formerly was operated by reciprocating means has been developed for rotary drive with marked reduction in the cost, weight, space occupied, and efficiency obtained.

Perhaps the best example of this development has been the

commercial introduction of the Westinghouse-Leblanc condenser, which, by the introduction of a rotary vacuum pump, permits the driving of the condenser circulating and air pumps by direct connection to a small turbine unit, as illustrated in Fig. 57.

FIG. 57.

Turbine-driven type E high-speed condenser.

The application of the small turbine to the field of centrifugal pumps is rapidly replacing the reciprocating engine, the advantages of the turbine being due to the fact that for low lift pumps and high-pressure pumps the speed for the best design is beyond that suitable for direct connection to a reciprocating engine.

Fig. 58 is an application of the small turbine to a centrifugal circulating and Leblanc air pump for surface condensers. Fig. 59 illustrates a boiler feed centrifugal pump direct connected to a small Westinghouse turbine. Similarly, the small turbine is admirably adapted to the driving of centrifugal blowers, an example of which is illustrated in Fig. 60.

As auxiliary apparatus for large power stations, or in places where it is important to reduce attendance to a minimum and where reliability is essential, the small direct-current turbo-generator unit is now generally accepted in place of the reciprocating engine, direct-connected or belt-driven apparatus, principally on account of its reduced size and cost, and the many advantages incident to rotary apparatus. Fig. 61 represents several sizes of these machines, from 1 kilowatt up, as manufactured by The Westinghouse Machine Company, both in direct- and alternating-current sets.

FIG. 58.

Air and circulating pump unit for 2000 square feet surface condenser, built for Louisville & Nashville Railway Company.

cating engine, direct-connected or belt-driven apparatus, principally on account of its reduced size and cost, and the many advantages incident to rotary apparatus. Fig. 61 represents several sizes of these machines, from 1 kilowatt up, as manufactured by The Westinghouse Machine Company, both in direct- and alternating-current sets.

APPLICATION TO MARINE SERVICE.

The application of the turbine to marine propulsion is perhaps the broadest single field for the utilization of maximum horsepower output. Until within the last few years this field was limited to approximately 5 per cent. of the total merchant

marine vessels, on account of the difficulty of correlating the high speed of the turbine and the relatively low speed of the ship's propeller.

It has been, of course, possible to apply turbines to the slower speed ships, but it has not been practicable to do so, because there would have been no saving in weight, space, or economy over the reciprocating engine, since the turbine is essentially a machine of relatively high rotative speeds for efficient

FIG. 59.

Small turbine driving boiler-feed pump.

operation, while the screw propeller requires relatively low speed of rotation.

Recognizing the possibilities of the steam turbine for marine propulsion, Mr. George Westinghouse, in 1903, commissioned the late Rear Admiral George W. Melville, Ex-Engineer-in-Chief of the United States Navy, and Mr. John H. Macalpinc, Consulting Engineer, to make an impartial investigation and report on the status of the marine steam turbine.

This report, a number of copies of which were printed for private circulation in May, 1904, did not indicate that the reciprocating engine would be immediately superseded by the

steam turbine, as it was then being applied to marine propulsion, but concluded with the statement that: "If one could devise a means for reconciling in a practical manner the necessarily high speed of the turbine with the comparatively low rate of revolution required by an efficient propeller, the problem would be solved and the turbine would practically wipe out the recip-

FIG. 60.

06 V.N.C. turbine with 30-inch 25,000 cubic foot Navy type blower.

cating engine for the propulsion of ships. The solution of this problem would be a stroke of great genius."

Acting on this suggestion, Mr. Westinghouse caused designs to be made for a reduction gear for large power transmission, which was constructed and tested in the works of The Westinghouse Machine Company at East Pittsburgh, after designs

of Messrs. Melville and Macalpine embodying a floating frame carrying the pinion, which had been shown mathematically to be necessary to the successful operation of high-speed gearing transmitting large powers.

This experimental gear transmitted under test 6800 horsepower with a pinion speed of 1500 revolutions per minute and a gear speed of 300 revolutions per minute. These tests demonstrated conclusively that the floating frame would insure correct alignment of the gear teeth, which is absolutely essential to the transmission of large powers. It was believed that this par-

FIG. 61.

1-kilowatt to 10,000-kilowatt turbines.

ticular design was more rigid than desirable, because the connection between the floating frame and the casing was a metallic one, and in consequence noise and vibration were not eliminated to the degree that it was believed to be possible.

As a substitute, therefore, for the Melville-Macalpine design, Mr. Westinghouse devised a hydraulically supported pinion frame having a degree of flexibility in itself, and which was supported on the main frame by oil in the hydraulic cylinders in the floating frame. This construction also allowed vertical movements of the pinion and floating frame, which were impossible in the Melville-Macalpine design.

A test on a large scale of the Westinghouse floating frame demonstrated that this accomplished all the objects of the original inventors, and, by reason of the absence of metallic connection between the pinion frame and the casing, the noise and vibration incident to the operation of the gear were almost wholly eliminated.

The tests on this large experimental gear aroused great interest in naval and merchant marine circles, and resulted in the

FIG. 63.

25,000 G.P.M. circulating pump, reduction gear and turbine drive. Built for Edison Electric Illuminating Company, Brooklyn, N. Y.

application of turbine gearing sets of 3600 horsepower each to the twin screw U. S. S. *Neptune*, and the general application of turbine gearing to the larger sizes of direct-current generators, centrifugal pumps, rolling-mill machinery, etc. A complete technical description of the experimental reduction gear and machinery in the U. S. S. *Neptune* may be found in *London Engineering* of September 17, 1909, and November 17, 1911, respectively.

Twenty-one sets of the Westinghouse turbine reduction gearing have been constructed and installed by The Westinghouse Machine Company with eminently satisfactory results in opera-

FIG 62.