

THE DE LAVAL STEAM-TURBINE.*

BY MR. E. S. LEA, *Member, A.S.M.E.*, OF TRENTON, NEW JERSEY,
AND MR. E. MEDEN.

The fundamental principles of the De Laval Steam-Turbine are clearly shown in Fig. 1, Plate 81. It is a pure impact turbine, with a single turbine wheel, carrying one row of buckets, to which the steam is delivered in free jets at the highest possible velocity. These steam-jets come from stationary nozzles, tapered so as to increase their cross-sectional area toward the outlet end of the nozzle, and so calculated that the steam, before leaving the nozzle, has fully expanded down to the pressure prevailing in the exhaust chamber of the turbine, and has assumed a correspondingly high velocity, so that its whole available energy has been transformed into kinetic energy.

The velocity of the steam-jets varies considerably, owing to change in pressure of the steam before entering the nozzles, to varying exhaust pressure, and to a greater or less degree of moisture or superheat in the steam. The lower limit of this velocity found in general practice might be considered as about 2,000 feet per second, which is obtained at a steam pressure of about 45 lbs. per square inch, at an exhaust pressure equal to the atmospheric pressure,

* For further discussion of this topic, consult "Transactions, A.S.M.E.," as follows :—

"Note on the Steam Turbine," by J. B. Webb, vol. x, page 680.

"De Laval Steam Turbine," by W. F. M. Goss, vol. xvii, page 81.

"Steam Turbine," by R. H. Thurston, vol. xxii, page 170.

"Steam Turbine from Operating Standpoint," by F. A. Waldron, vol. xxiv, page 999.

and with steam containing 10 per cent. of moisture. The upper limit is found to be about 4,400 feet per second, at a steam pressure of 200 lbs. per square inch, at 27.5 inches vacuum, with the steam superheated 200° F. The velocity of the steam will determine the conditions under which a maximum of the transformation of the steam-jets' kinetic energy into useful mechanical work can be reached, these conditions being the same as for impact water turbines.

The nozzle angle, or the angle of the steam nozzle with relation to the plane of the wheel, should be as small as possible. A certain mathematical relation should exist between the nozzle angle, the velocity of the steam-jet, the peripheral velocity of the turbine wheel, and the inlet angle of the buckets. The outlet angle of the buckets should be the smallest possible. Practical considerations limit to a certain degree the attainment of proper angles for the very best efficiency. Thus, in the De Laval turbine, a nozzle angle of 20° has been established for all sizes of the turbine, the inlet and outlet angles of the buckets are made alike, and are 32° for smaller sizes, 36° for larger sizes. With these angles fixed, and taking into consideration the thickness of the buckets, it will be found that the best theoretical peripheral velocity of the turbine wheel will be about 950 feet per second for a steam-jet velocity of 2,000 feet per second, and about 2,100 feet per second for a jet velocity of 4,400 feet per second.

Contrary to popular belief, there are no reasons, either theoretical or practical, to prevent the building of a safe turbine wheel, with a peripheral velocity as high as 2,100 feet per second; only economical reasons have put a limit to it. In the turbines that have been built, the actual peripheral velocity varies between about 1,400 feet per second in the larger sizes, and about 500 feet per second in the smaller sizes. In comparison with existing machinery and other types of steam-turbines, these velocities are exceedingly high, and have necessitated the solution of some very interesting theoretical problems, such as the calculating of the strains in wheels revolving at high speeds, determination of flexible shafts suitable for carrying these wheels, etc. These theories would occupy too much space to

enumerate here, and as they have been published in the technical literature, they have been omitted. The authors would especially refer to a book on the steam-turbine, viz. "Die Dampfturbinen und die Aussichten der Wärmekraftmaschinen," by Dr. A. Stodola, of Switzerland, where, apart from slight inaccuracies, the said theories have been published in an exhaustive and able manner. This book is now being translated into English by Dr. Lewis C. Loewenstein, Instructor in Mechanical Engineering at Lehigh University, and will be published in about two months. This book also contains that part of thermodynamics dealing with the outflow of steam through nozzles, and the determination of expanding steam-nozzles, as well as the efficiencies obtainable from the work of the steam-turbines.

The diameters of the turbine wheels are such, in relation to the given peripheral velocities, that the speeds run from 10,600 revolutions per minute for the largest size, to 30,000 revolutions per minute for the smallest size. The speeds are reduced approximately 10 to 1, by helical gearing, giving driving shaft speeds of 900 to 3,000 revolutions per minute. A single gear-wheel is provided in the smaller types, Fig. 2 (page 700), and in the larger sizes they are double, Fig. 3 (page 701). If the larger types were single geared, the pressure in the pinion bearings, due to the pressure between the teeth of the gear and the pinion, would be too great at these speeds; therefore, the gears are made double so that half the load is taken by each wheel, the gear pressure on one side of the pinion balancing the pressure on the other side, thus eliminating the pressure in the pinion bearings.

The characteristic high velocities of the principal parts of the De Laval turbine also create some interesting practical problems.

The turbine wheel itself will be first considered which is shown in section Fig. 4 (page 702). The wheel is designed with a factor of safety at normal speed of about 8, and the radial and tangential stresses due to the centrifugal force constant throughout the wheel. The profile of the wheel is a logarithmic curve asymptotic to the radial axis of symmetry of the wheel section. The buckets, which are inserted into milled slots in the rim of the wheel, when actuated

FIG. 2.—30-H.P. Turbine Dynamo.

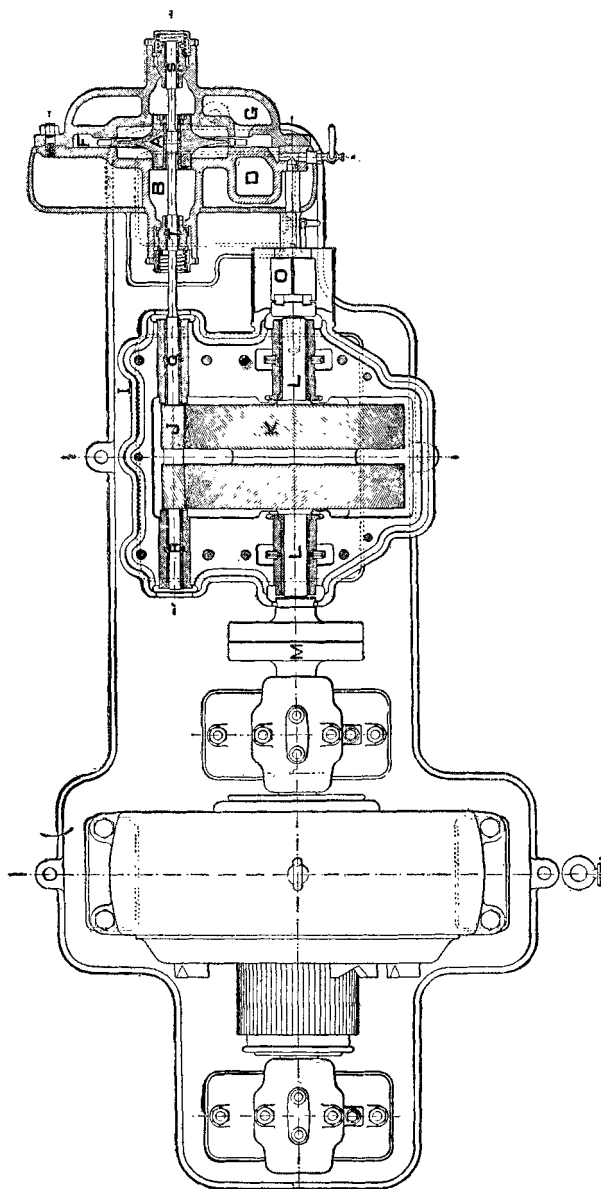
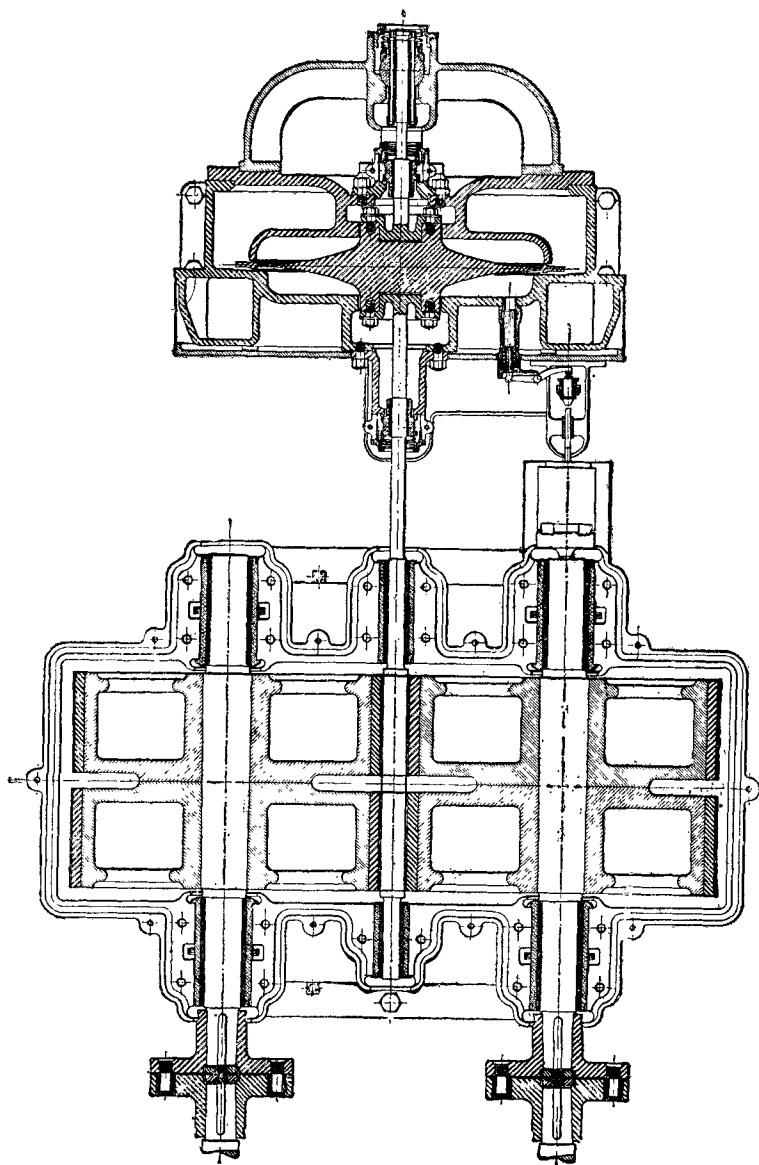
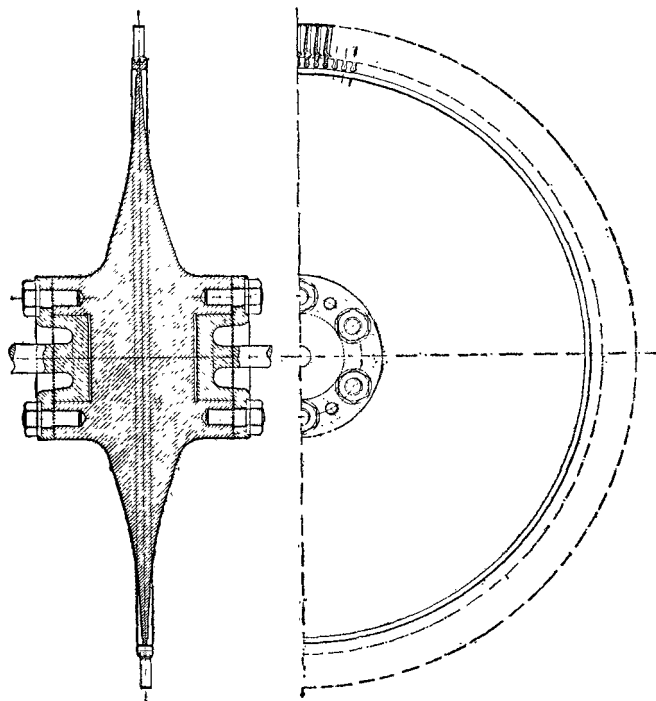


FIG. 3.—300-H.P. Turbine.



by centrifugal force, load the solid wheel body at its outer periphery to an amount equal to the centrifugal stresses in the body. The stresses vary with the square of the speed, and with increasing speed they will gradually increase to a point where the wheel will burst.

FIG. 4.
De Laval Turbine Wheel.



In spite of speed-regulating mechanism and safety stops, a motor of any kind might race, as all regulating devices are liable to derangement, and safety stops, which as a rule are seldom used, sometimes fail to operate. This, of course, applies also to steam-turbines. It is therefore necessary to provide means for the prevention of serious damage. In the De Laval turbine this protection is obtained by reducing the thickness of the wheel

close to the periphery, which naturally decreases the strength of the wheel at this point, the stresses here being about 50 per cent. higher than in the rest of the wheel. At normal speed the factor of safety at this point of the wheel is about 5; consequently the wheel will burst here at about double its normal speed, and in such a manner that the rim holding the buckets is broken up into pieces, which, on account of their small size, are unable to do any damage to the wheel case. At the moment the rim leaves the wheel, the stresses in the solid wheel body are considerably reduced, at the same time the wheel becomes unbalanced; and as the clearance between the heavy hub of the wheel and the safety bearings in the surrounding wheel casting is very small, the hub of the wheel will come in contact with the latter, which efficiently act as a brake on the wheel, and bring it to a stop in a short time, as with the buckets gone, the steam has no effect whatever on the wheel. Exhaustive experiments have verified these statements, it having been found that turbine wheels without this decrease in section at the outer periphery, having purposely been speeded up, would burst through the centre in two or three heavy pieces, which, at the high velocity, a wheel case of ordinary proportions would not resist. Such pieces have been driven through an experimental wheel-case of steel castings, having walls 2 inches thick. With the wheels as made, however, they are perfectly safe, and in the event of the rim being stripped, no damage will result except to the wheel itself.

As it is possible to design a turbine wheel for any radial and tangential stresses, it might be asked why the wheels are not made so strong that it will be impossible, with the available steam velocities, to run them up to the bursting point. The reply to this is, that it would be too expensive and not practicable to design the rest of the turbine and connected machinery to run safely at a corresponding speed.

In this connection we will consider the speed regulation mechanism of the De Laval turbines. This consists of a common centrifugal governor, actuating a throttle-valve in the steam-supply line of the turbine. With this the pressure can be closely

controlled, but not entirely shut off; in most cases, though, sufficient to prevent the turbine going above its normal speed when running light. This is especially true of turbines running non-condensing. In condensing turbines operating with very high vacuum, the passive resistances are sometimes extremely small, and even if the governor-valve throttles the steam considerably below the atmospheric pressure, the remaining pressure may be sufficient, at no load, to increase the speed above the normal. To prevent this speed increase, a second regulating mechanism is provided, the purpose of which is to decrease the vacuum in the wheel case. This apparatus consists of a small valve which is directly actuated by the governor, but only after the governor-valve in the steam line has been shut off. This valve either lets air into the wheel case, decreasing the vacuum, or in such cases where the vacuum in the condenser must be maintained for other machines, it admits air into a regulating valve mechanism placed in the exhaust line of the turbine. When air is let into this valve, it more or less shuts off the communication between the wheel case and the condenser, thereby raising the pressure in the wheel case, which then increases the passive resistances of the wheel, and checks the expansion of the steam in the nozzles, and, together with the steam throttle-valve holds the speed within the normal limits. In case of accident to the governor-valve mechanism, this air-valve will also effectually prevent destructive racing.

The peripheral velocity of the gear wheels is about 100 feet per second. The pinion is made of high-grade high-carbon crucible or nickel steel. The gear wheels are made of soft steel of low carbon. The teeth are carefully generated at an angle with the shaft centre and the pitch is very small, insuring a smooth contact with a minimum amount of noise. The noise cannot be entirely eliminated, but with great care in cutting the teeth, and giving close attention to alignment and centre distances, it has been possible to reduce it to a minimum and to a point where it is in most cases of no consequence. The gears are continually lubricated, but with a very small amount of oil. If they get the proper amount of lubrication and care is taken that no sharp grit, such as cement dust, coal dust,

or the like, is allowed to enter them, they will operate for many years without visible wear. The gears are encased as much as possible, to prevent the entrance of dust or foreign matter. The gear wheels were originally made of bronze, but it soon developed that this material, as a rule, became crystallized after about two years of continuous operation, when pieces of the teeth were broken off and destroyed the gears. Steel gears have now been in operation for about nine years, without showing any of the disadvantages of bronze.

Little is to be said about the bearings. They are all lined with white metal. The low-speed bearings for the gear shafts are similar to bearings for electrical machinery of same speed, and are provided with ring oilers. Ring oiling, on the other hand, has not proved to be satisfactory for the high-speed bearings. The turbine wheel-shaft usually vibrates slightly, which is communicated to the oil rings; they then refuse to follow the shaft, and consequently do not furnish proper lubrication. It is also found that the temperature of the oil in this case will increase too much, and drip lubrication has been found more satisfactory, only a small quantity of oil being required. With the high speed it is very important that the lubrication should not be interrupted, as it takes but a short time for the bearing to run hot. Wick lubrication has so far proved the most reliable. It must, however, be arranged so that the oil leaves the wick tube in drops, and with a sight glass below the tube through which the amount of feed can be ascertained. The oil is filtered by the wick, which ensures clean oil in the bearing, and the oil will flow as long as any oil remains in the tank. With oil tanks of ample size there will not be much attendance required. It seems, though, in the present advanced stage, that opposition is sometimes met with in having this method of lubrication used. The common sight-feed lubricator, with such a small number of drops as are required, has the disadvantage of a very small opening for the oil, so that a small amount of dirt will suddenly interrupt the lubrication. The bearing will then immediately heat. Any mechanical arrangement for forced lubrication is in itself more or less apt to get out of order.

It is all right for slow-speed machinery, which, in case of interruption of the oiling, can run a considerable time on the oil already supplied, and until the trouble can be discovered and remedied; but it is more or less uncertain for high-speed apparatus.

It might be interesting to touch on the practical difficulties which the De Laval steam-turbine, like any other radically new machine, was compelled to meet, after it had been put to work. The turbine naturally had its troubles from defects due to faulty material and workmanship, but these have been remedied. There have been troubles with bearings becoming overheated. This was partly due to faulty workmanship, but in many cases it can be ascribed to the lubrication, either to failure in keeping the oil reservoir filled, or else to the sight-feed lubricators, which in themselves might have caused trouble. As more machines have been put on the market, they have become more fully understood, and are therefore receiving better attention; consequently these troubles have been gradually reduced. Furthermore, there has been trouble with the buckets. It has sometimes happened that one or more buckets have broken, and come out of the turbine wheel, but without doing any further damage. Generally the turbine, after losing a bucket, can be continued in operation, as the turbine shaft is sufficiently flexible to take care of the unbalancing, though it is best to take out the turbine wheel and replace the buckets. The only explanation of these troubles is that the buckets are subjected to vibratory strains of more or less unknown origin, as their ability to withstand centrifugal force and the action of the steam-jet is amply sufficient. In the smaller sizes, below 100 horse-power, broken buckets have been very rare. In the larger sizes, it has been somewhat more frequent. Although the causes of bucket breakage are not yet accurately determined, it has been possible to remedy the trouble where it has occurred. One cause of the undue vibrations of the buckets may have its source in the turbine wheel itself, which, if not homogeneous, will, under action of the centrifugal force, expand unevenly in different directions, thereby unbalancing and causing vibration of the wheel at full speed. This trouble has

been overcome by replacing the wheel. The buckets are also subject to more or less wear due to the action of the steam. The cause of this is also very difficult to determine. It may be that the buckets are chemically affected and that thin films of oxide are blown away by the steam, or it may be caused by mechanical wear due to small solid particles coming with the steam, such as rust, or scale from the pipes. It may also be due to some electrical phenomena. However this may be, it is a fact that wear takes place, and it is very doubtful that it can be entirely prevented. It has been found in a few cases that buckets have been worn out in a year, necessitating replacement. In other cases the wear has been very slight, even after a run of four to five years. The wear affects only the steam inlet side of the buckets, and will only increase the steam consumption to a slight degree. In tests made on a turbine of 100 horse-power, where the edge of the buckets had been worn away about one-sixteenth inch, the steam consumption was about 5 per cent. higher than with new buckets. The wheel and buckets are, however, so designed that an insertion of a new set of buckets can be easily made at a small cost.

In looking about for proper fields of usefulness, the De Laval steam-turbine, in common with other steam-turbines, first developed the direct connected electrical unit, no difficulty being met in adapting both direct and alternating current generators for direct connection to the gear shafts at their moderate speeds. In many cases De Laval turbines can also be used, with advantage, for belt transmission.

However, the field where the De Laval turbine is particularly suitable is in connection with centrifugal pumps; these pumps require certain determined velocities to enable them, at a given lift and water quantity, to give the best efficiency. With the De Laval turbine it is easy to produce the most suitable velocities; with the small turbines, having one gear shaft, for all lifts from 15 feet to 150 feet, and with the large turbines, with two gear shafts, for lifts from 40 feet to 300 feet. These velocities are often difficult to obtain with other steam motors. Fig. 5, Plate 81, shows a 55 horse-power single-stage turbine pump.

For a greater lift the centrifugal pump has been directly connected to the high-speed turbine shaft. The pump wheel will then revolve with a velocity of 10,000 to 30,000 revolutions per minute, depending on the different sizes. The pump wheel will naturally be very small, and will not produce any suction, but must be fed with another pump, which is connected to the gear shaft, running at a considerably reduced velocity. This latter pump sucks the water and presses it into the high-speed pump wheel, which then gives the high pressure required. Pumps of this type have been made for lifts up to a normal head of 850 feet on a single wheel, which, at a decreased water quantity, can go up to 1,000 feet, the small pump-wheel giving an efficiency of about 64 per cent. They have in some cases been made, and are in operation, as boiler feed-pumps. A pump of this type is shown in Plate 82.

Another field where the De Laval turbine is well adapted is for direct connection to blowers for all pressures above 4 inches of water, for which a blower can be practically built, the high velocity of the turbine being particularly suitable for this purpose.

About the steam consumption it is difficult to make any general statements. It varies for the same size turbine with the steam conditions, in about the same manner as for other steam motors, but the degree of variation can be considerably different for the various sizes of turbines, dependent upon the diameter and speed of the turbine wheel. It may be sufficient to give here a few diagrams showing the steam consumption of different sized turbines under more or less favourable conditions.

Fig. 7 shows the result of a test of a 10-kilowatt non-condensing turbine dynamo. The boiler steam-pressure is 140 lbs. per square inch; steam dry saturated. The curve I, II, III, IV gives the steam consumption per kilowatt-hour with all steam nozzles open during the varying conditions of the load, the governor valve alone having to take care of this variation by throttling the steam pressure. The curve I-V shows the steam consumption with the nozzles shut off in proportion to the varying load. In this manner the nozzles will be supplied with steam of the full steam pressure at all times.

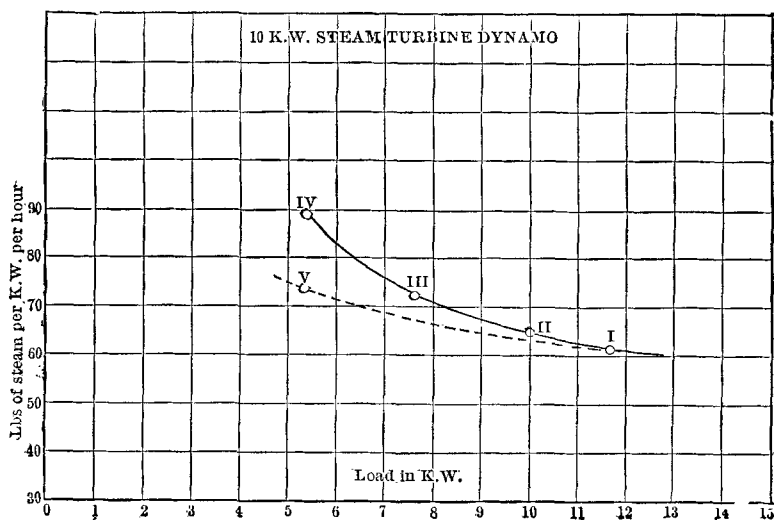
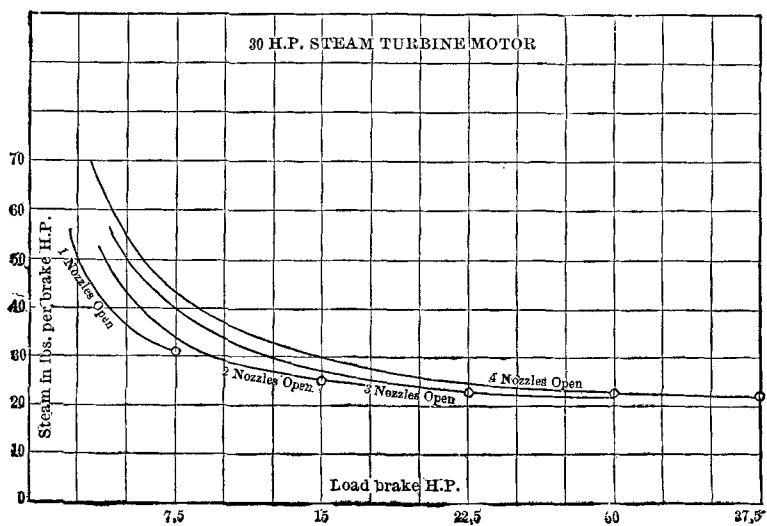
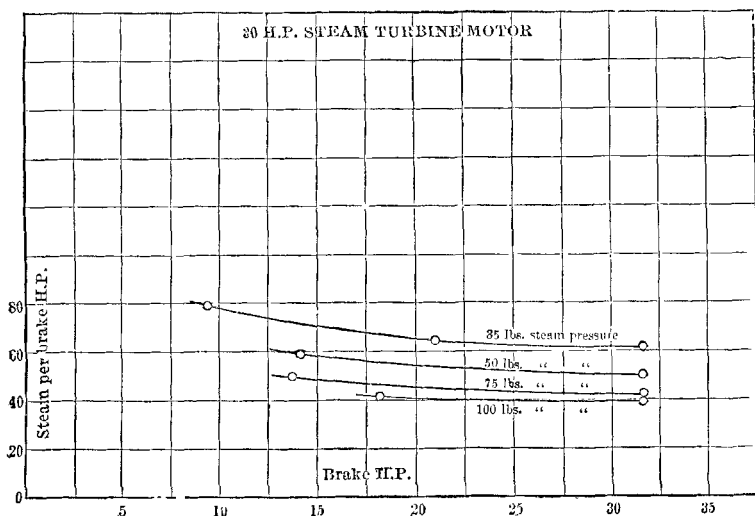
FIG. 7.—*Steam Consumption.*FIG. 8.—*Steam Consumption : number of Nozzles constant.*

Fig. 8 (page 709) shows a test made on a 30 horse-power steam-turbine motor, condensing, with $25\frac{1}{2}$ inches vacuum, steam pressure being $125\frac{1}{2}$ lbs. per square inch above the governor valve. Steam ; dry saturated. The four different curves show how the steam consumption per brake horse-power varies with the varying load, with and without regulating the number of nozzles opened.

Fig. 9 shows a test made on a 30 horse-power steam-turbine motor, non-condensing, with different steam pressures above the

FIG. 9.

Number of Nozzles varied to suit Steam Pressure.



governor valve, with the nozzles suitable for the different steam pressure. The curves represent respectively 35, 50, 75, and 100 lbs. boiler steam-pressure per square inch. The number of nozzles opened have been varied according to the varying load.

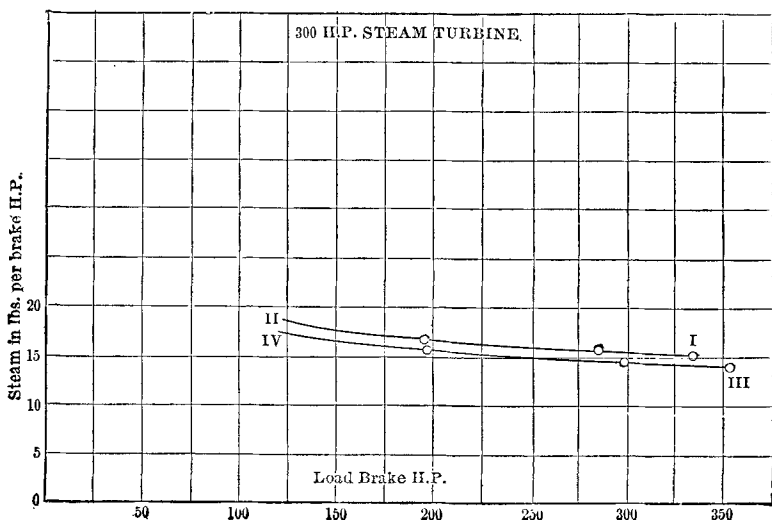
Fig. 10 shows the result of a test made on a 300 horse-power steam-turbine, steam pressure about 200 lbs. per square inch, vacuum about 27 inches. The curve I, II gives the steam consumption for dry saturated steam. The curve III, IV gives

the consumption for superheated steam. The superheat varied from 90° F. at maximum load to about 20° F. at the smaller loads.

Figs. 11 (page 712) and 12 (page 713) are curves from very exhaustive tests made recently by Professors Kent and Denton, on turbine pumps. These curves in themselves need no further

FIG. 10.

Tests with Saturated and with Superheated Steam.



explanation. The curve in Fig. 11 is obtained from a pump shown in Plate 82; the curve in Fig. 12 from a pump of the type shown in Fig. 5, Plate 81.

In the foregoing, some frank statements have been made to illustrate the difficult theoretical as well as practical problems encountered, the solution of which has produced the present successful De Laval steam-turbine.

FIG. 11.—Tests made on High-Pressure Centrifugal Pump (Plate 82).{

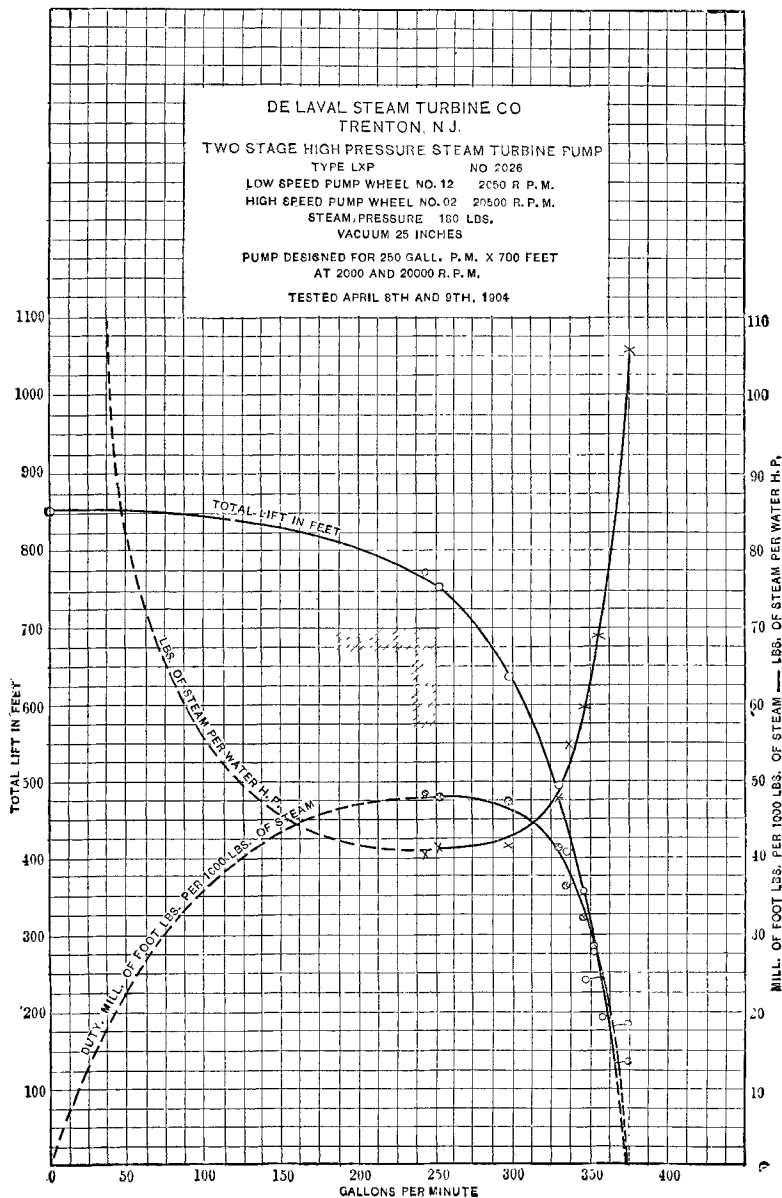
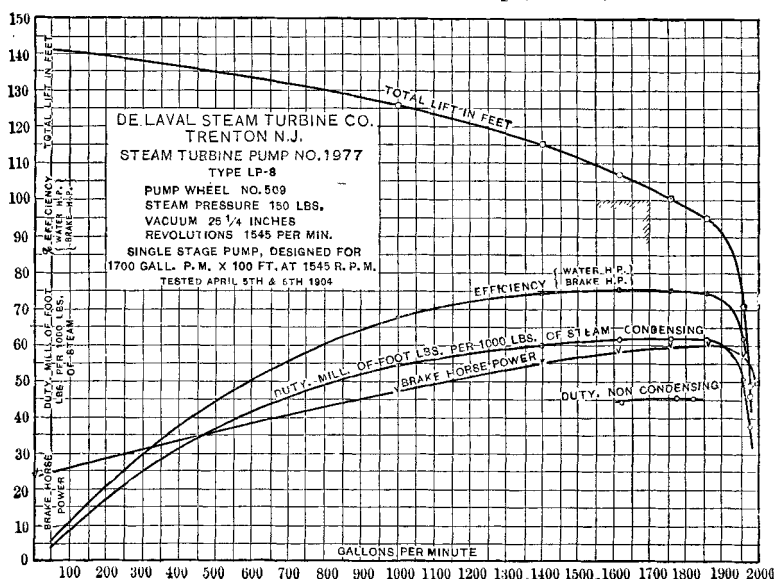


FIG. 12.—Tests made on Turbine Pump (Plate 81).



The Paper is illustrated by Plates 81 and 82, and by 9 Figs. in the letterpress.

Discussion.

MR. CHARLES B. REARICK said that he did not represent the authors, but he wanted to make a remark or two that bore on the De Laval machine, and he wished also to say something about the question of condensers. On the question of condensers, he thought the point of 26 inches vacuum being sufficient was very well taken, but it would seem that the amount of vacuum was for the engineer to determine. If he had a condition where fuel was very low in cost, it followed that he could afford to put in a plant at low cost and sacrifice economy. On the other hand, if his fuel were comparatively high, he could easily afford to spend a little more money for his condenser. What was good in one case might not be good in

(Mr. Charles B. Rearick.)

another, and each case should be dealt with in reference to the conditions under which the machine would operate, as to the cost of fuel, cost of getting water for the condenser, and other points that entered into the proposition. There was one point where the De Laval turbine had made some progress, in which other turbines in the field, in the United States, had not done anything. He regretted that he had not any actual figures as to the number of De Laval machines driving pumps in actual service today, but there were a large number varying in size from 20 horse-power up to 225 horse-power, and he believed they were building several at the present time of 300 horse-power. They were particularly satisfactory for centrifugal pump work, which should interest engineers in general, since those pumps generally speaking were low in efficiency. The speeds of De Laval turbine driving-shafts enabled higher efficiencies for certain lifts than could be got at the lower speeds obtained from engines. Some of these efficiencies were showing up as high as 75 and 77, and some of the larger units that they were building were expected to give as high as 80 per cent. He had seen actual tests of 75 and 76 per cent. made by prominent engineers, Professors Denton and Kent conducting them.

*(For further Discussion on this subject, see pages 678-696 ;
734-735 ; and 780-785.)*

Fig. 1. *Wheel and Nozzle.*

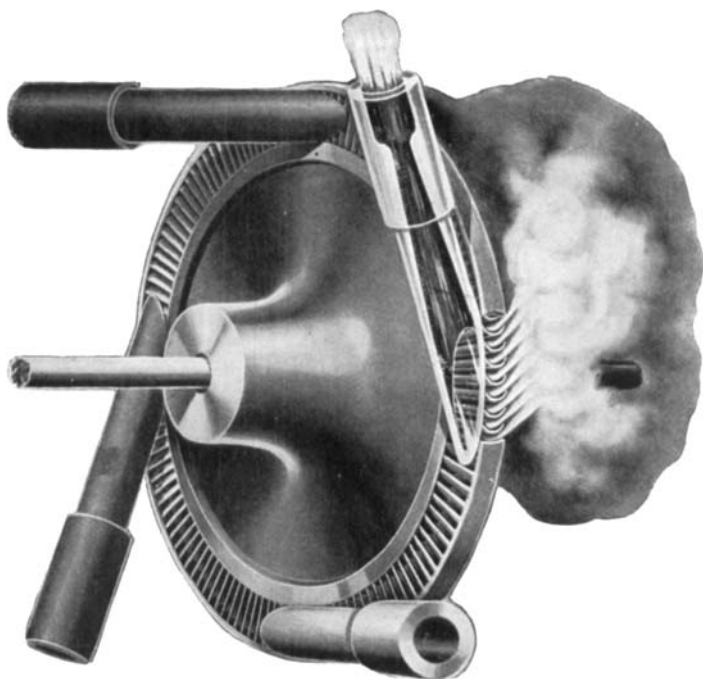


Fig. 5.

55-H.P. Turbine Pump.

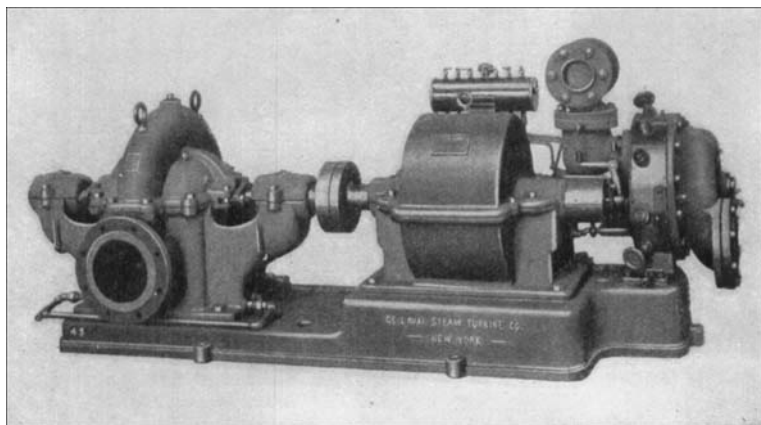


Fig. 6. *High-Pressure Centrifugal Pump.*

