(*Paper No. 4370.*)

"Design and Performance of a New Impulse Water-Turbine."

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FOR many years only two types of impulse water-turbines have been generally manufactured, namely, the Pelton wheel, and the Girard turbine. Of these two types the Girard turbine is practically obsolete for large power developments. Both types are so well known that it is not necessary to describe their design in detail. There are, however, certain advantages and disadvantages inherent in each type, which it is necessary to discuss at some length.

The Pelton Wheel.—The Pelton wheel is essentially a low-capacity turbine, that is to say, under the same conditions of fall a wheel of given diameter will develop relatively small power in comparison with other turbines of similar dimensions. The specific speed of a single-jet Pelton wheel of modern design does not exceed about 5 revolutions per minute, whereas a Francis turbine may be designed for any specific speed from about 11 revolutions per minute upwards. There is, therefore, a definite gap between the Pelton wheel and a low-capacity Francis turbine, and the Pelton wheel at its best compares unfavourably as regards efficiency with the latter. Efficiencies exceeding 82 per cent. are rarely claimed by first-class makers, and it is doubtful whether even in the largest sizes efficiencies exceeding this figure are obtained under working conditions. An important source of inefficiency in a Pelton wheel is the windage loss due to the fan action of the runner.

On the other hand the Pelton wheel has two great advantages over any other type of turbine in general use at present. It is very simple in construction, and its speed can be regulated with ease and efficiency. The advantage sometimes claimed for it, that a bucket, if it becomes damaged, can be replaced, is more apparent than real. The fact that its speed can be regulated with ease by deflecting the jet is, however, of the first importance.

The Girard Turbine.—The Girard turbine, if properly designed, is more efficient than the Pelton wheel, and has also a considerably higher specific speed. As generally constructed it has, however, the great disadvantage that its speed cannot be easily regulated. It cannot be governed by deflection of the operating jet or jets, and regulation must be effected by means of a sliding plate, which varies the size of the jet orifices. This cause more than any other has displaced the Girard turbine from a position of any importance in hydro-electric developments.

To summarize, the Pelton wheel is :—

(1) Low in capacity,
(2) Relatively low in efficiency,
(3) Weak in construction,

but it is :

(4) easily regulated.

The Girard turbine on the other hand has

(1) A relatively high capacity,
(2) A good efficiency,

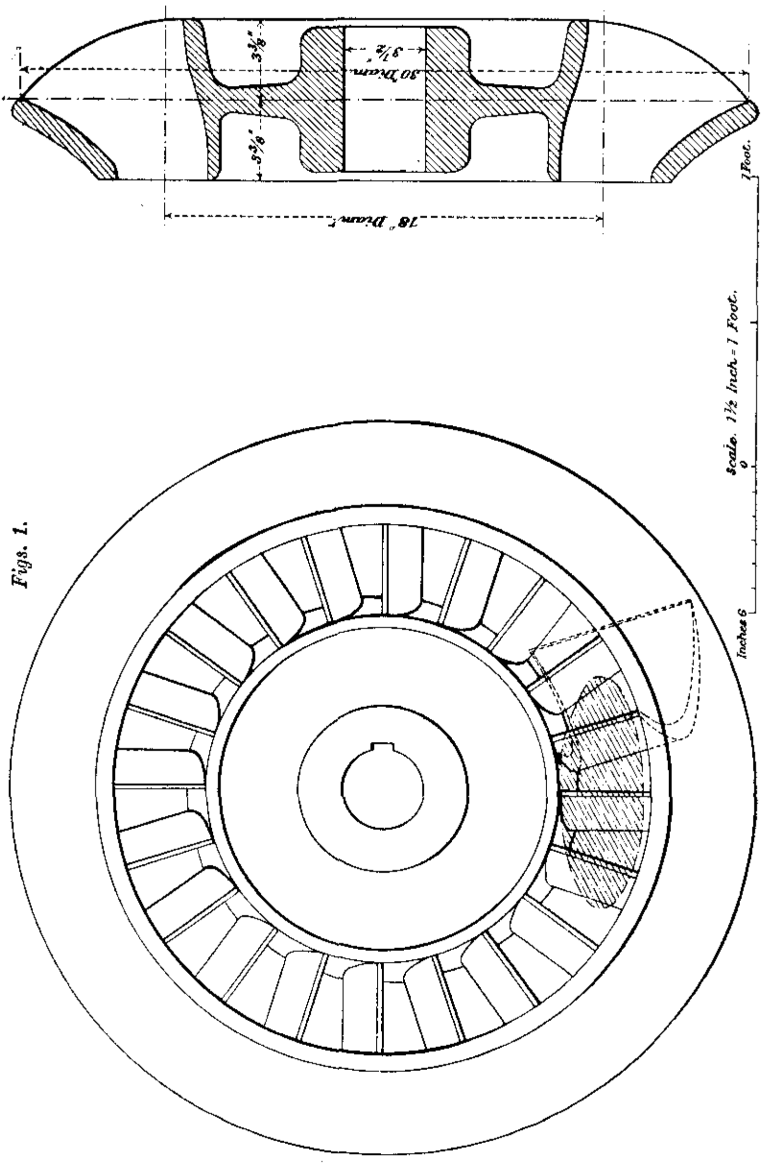
but it is :

(3) difficult to regulate.

Description of the New Impulse Turbine.—It follows from the foregoing summary that, if a jet as used with the Pelton wheel be applied to a wheel of the Girard type, an impulse turbine with a relatively high capacity and efficiency should be obtained, which at the same time could be easily controlled. Such is, in fact, the turbine the design and performance of which is described in this Paper. The runner is a development of the axial-flow Girard wheel, with a jet of the Pelton type, acting on this runner. The nozzle is set in such a position relative to the wheel that the centre-line of the jet in plan makes an angle with the plane of the wheel. The most obvious criticism of such an arrangement is that the jet, striking the runner obliquely, will cause end thrust on the shaft, but this thrust, which in any given case can be determined experimentally, will be small in relation to the other forces involved. Thrust bearings are not now the bogey they used to be, provided they are designed with an ample margin of safety above the maximum load that can come upon them, and several satisfactory types are available.

Design of the Buckets.—The angle which the jet makes with the plane of the wheel is fixed at 20 degrees. The trace which the jet makes on the plane of the entrance of the wheel is shown in *Figs. 1.*

The velocity of the inlet edge of the bucket is made 0·45 to 0·47



Figs. 1.

TURBINE RUNNER.

of the velocity due to the head under which the wheel is working. The theoretical vane inlet-angle is then determined from the triangle of velocities, but this angle should be made larger than the theoretical value for the following reasons :—(1) Even if ground to a knife edge the vane must have some thickness, and the angle which the back (convex face) of the vane makes with the direction of the jet must certainly be not less than the theoretical angle, otherwise a portion of the jet will be deflected away from the runner. (2) The impact loss, due to the front (concave face) of the vane being set at a coarser angle with the direction of the jet than the theoretical angle, is negligible if, as in this turbine, the stream of water is properly controlled laterally. (3) The entrance angle, designed for the position when the entrance edge of the vane is on the vertical diameter of the jet, becomes actually less in all other positions of the vane relative to the jet.

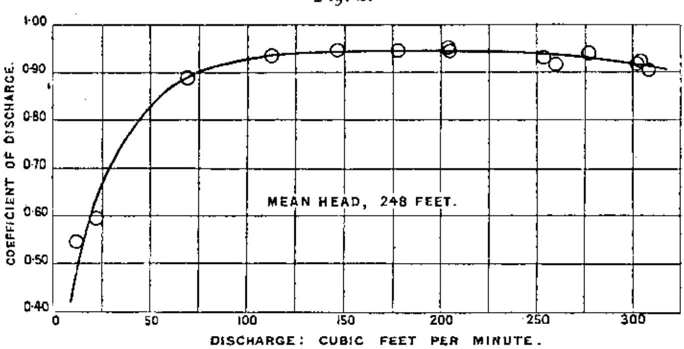
The design of the remainder of the vane depends very largely on the assumptions made as to the nature of the flow within the runner. It is assumed that :—(1) The best possible results will be obtained when the path of each water thread of the jet remains in a horizontal plane throughout its passage through the runner. (2) If each water thread behaves as assumed in (1), the actual relative velocity (neglecting friction and eddy losses) of the water and the surface against which it acts will vary in direct proportion to the distance from the axis of rotation of the point of contact of the water and the vane at any moment. (3) Fluid friction and eddy losses will have reduced this relative velocity by about 5 per cent. just before it leaves the runner.

The correctness of these assumptions is debatable, and the nature of the flow certainly differs from anything which may be deduced theoretically. The assumptions cannot be absolutely verified by experiment, but the actual efficiency and power approximate to the theoretical figures deduced from them.

The developed length of the buckets should evidently be made as short as possible, to reduce the loss of relative velocity due to the fluid friction. On the other hand, this length must be sufficient to prevent excessive eddy loss due to the bending of the water stream as it passes through the wheel.

By far the most important part of the runner design lies in the determination of the exit angle of the vane. Granted that the assumptions as to the flow in the runner are correct, it is easy to work out the angle of the vane at exit to give a minimum loss due to rejected energy in the water. In general it is found that this angle has to be made as small as possible—not more than 10 to 15 degrees to secure good results. On the other hand, in order to secure a reasonable capacity for the wheel, it is necessary that the angle should not be made too small. If it were so made, it is clear that the cross section of the passage for water at the exit section would be reduced too much; it is this cross section which largely governs the capacity of the wheel. Experiments go to show that the smaller the vane angle the greater will be the efficiency, and that the exit angles of the vanes can be made with advantage considerably smaller than would appear from theoretical consideration. The Author's view is that the actual relative discharge angle of the water is coarser than the exit angle of the vane, for at least two reasons: (1) There is a certain bending effect on the stream, due to surface tension between the fluid and the working face of the vane. (2) The stream of water flowing over the bucket will not be flowing at constant velocity, and the part close to the vane will be flowing at a lower relative velocity than the part more remote. This

Fig. 2.



entails a virtual velocity towards the edge of the vane as the water stream leaves it.

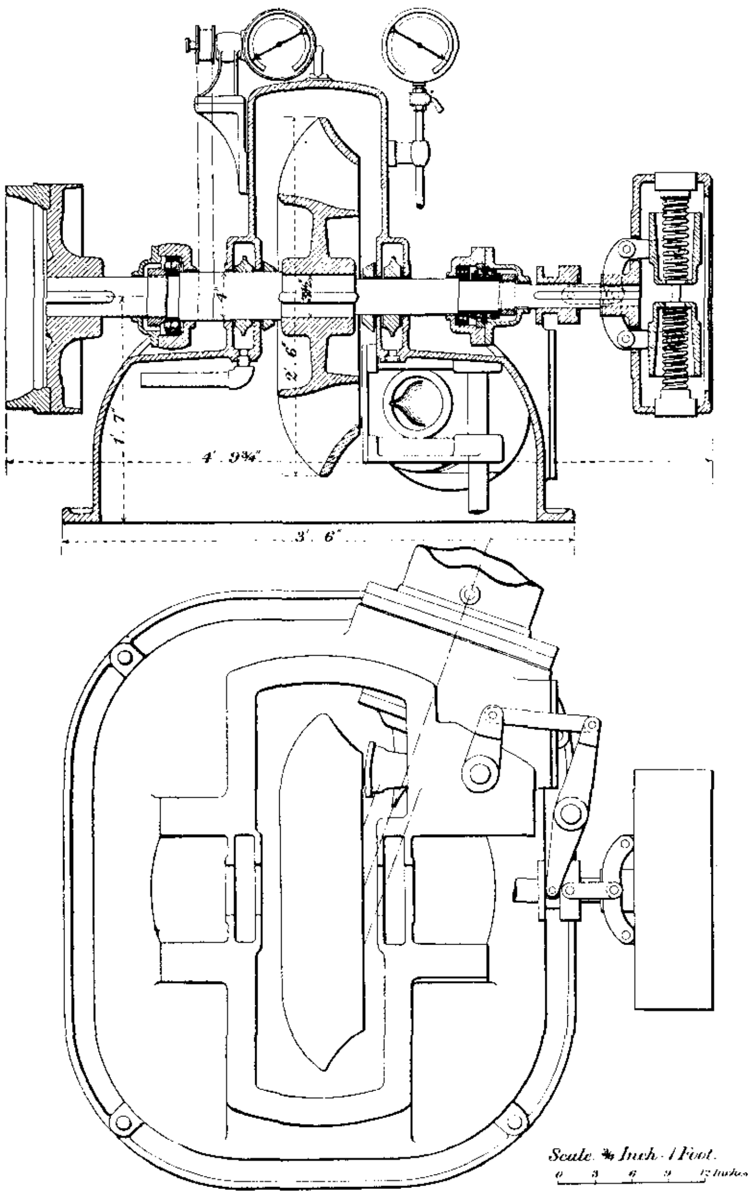
Nozzle.—The design of the nozzle is of the usual type adopted for Pelton wheels. For the operation of the economizer mechanism described hereafter, a passage for exhaust water is formed by making the spear rod hollow and drilling a small hole through to the spear tip. Though it was anticipated that this might have a bad effect on the shape of the jet, such has not been found to be the case, a particularly good jet being formed when the spear rod is nearly closed. This is confirmed by the very small quantity of water required to run the turbine at no load up to normal speed. The form of the supporting bracket which carries the weight of the spear rod and spear head within the nozzle pipe has been made with four carefully shaped arms These help to steady the flow in the nozzle pipe and, by reducing the eddies, render the jet more stable.

The fact that the spear rod is thus guided close to the spear head prevents any vibration in the latter which would destroy the form of the jet. *Fig. 2* shows the coefficient of discharge of the opening between the nozzle and the spear head. The area of opening for any particular position of the spear rod was calculated from workshop observations. The quantity of water actually passing for a corresponding position of the spear rod was observed. The ratio of the quantity actually passed to the quantity which it is calculated should be passed, assuming no contraction or loss of velocity in the orifice, has been taken as the coefficient of discharge. It does, of course, include a coefficient for loss of velocity due to friction, as well as a coefficient for contraction. The observations were not made, however, in a form to enable the relative effect of these two factors to be determined.

Bearings, Turbine-Case, and Other Details.—Ball bearings have been used throughout and have proved satisfactory. They are contained in cast-iron housings supported from the turbine casing. The latter is of cast iron, split on the horizontal centre-line, the lower portion being formed in such a manner as to allow ample space for the exhaust water to escape, and for the deflector mechanism of the governor to act. The manner in which water is prevented from escaping from the casing along the shaft is shown in *Figs. 3*. Any water which escapes past the first water-thrower collects in the annular space in which the second water-thrower runs, and is drained by a pipe into the lower part of the turbine casing.

Governor.—One of the most important features of any hydro-electric plant is the governor adopted to control the speed of the turbines under variations of load. Up to the present only one completely satisfactory type has been evolved, and this can only be applied to turbines of the impulse type, hence a governor of this type has been adopted for the control of the plant under description As it has only been introduced in recent years and may be unfamiliar to many engineers, it is proposed to describe it in some detail. The governor consists essentially of a powerful centrifugal pendulum, mounted directly on the turbine shaft, and operating directly a deflector interposed between the jet and the runner. No relay mechanism of any kind is provided. This direct action of the pendulum on the deflector reduces the so-called dead time of the governor to a negligible quantity, the action being so rapid as to be practically instantaneous. The result obtained is that, even with the whole load suddenly thrown off the generator, the momentary rise of speed above settled no-load speed, as indicated by an Elliott recording tachometer, is less than 1 per cent. It is doubtul whether

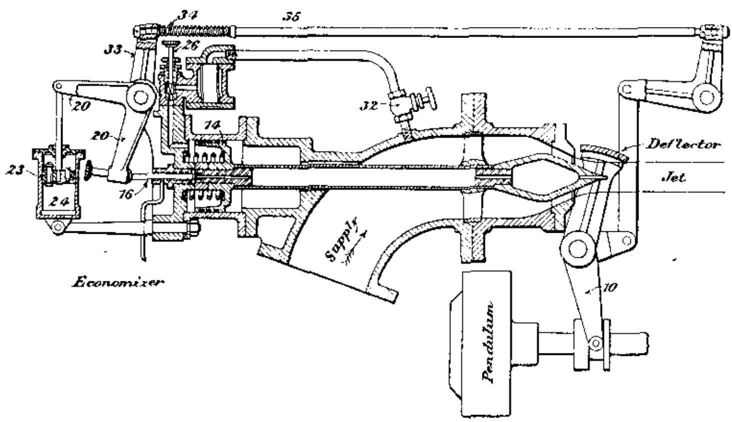
Figs. 3.



ELEVATION AND PLAN OF TURBINE.

better governing can be obtained even with a heat-engine. It should be borne in mind that there is very little energy stored in the rotating parts, which consist merely of the turbine runner and generator armature with their respective shafts, and the flexible coupling and governor head. It is interesting to compare the results with those obtained on the Pelton wheels at Kinlochleven. The rise of speed, when the whole load was suddenly thrown off, amounted to upwards of 23 per cent. for the 2,200-kilowatt sets and 10 per cent. for the 600-kilowatt sets, after heavy flywheels had been fitted on the smaller sets; before flywheels were fitted, the rise of speed with only 50 per cent. of the load thrown off was about 29 per cent.[1] As the method of governing by deflection is

Fig. 4.



DIAGRAMMATIC ARRANGEMENT OF GOVERNOR MECHANISM.

wasteful of water at fractional loads, an economizer mechanism for the operation of the spear rod has been introduced, which is controlled by the governor proper.

Economizer.—The governor and economizer are shown diagrammatically in *Fig. 4*. For the sake of clearness the pendulum is shown mounted on a separate shaft, but it is actually mounted on the turbine shaft, and is connected directly by means of levers 10 to a shaft, on which is mounted the deflector. As the speed rises or falls the deflector is forced into or out of the jet. The spear rod is hollow, and has a hole running through the full length of it, the hole at the cylinder end making a seating for poppet valve 16, so that, when the valve is pressed against the hole, communication

[1] Minutes of Proceedings Inst. C.E., vol. clxxxvii, p. 80.

through the rod is closed. Pressure water is brought through valve 32 to the back end of the cylinder. When valve 16 is pressed against the seating, the pressure on both sides of the piston 14 is equal, but the pressure on the spear head is constantly endeavouring to pull the spear into the nozzle. The deflector is connected by means of rods and levers to the bell-crank lever 20, one end of which presses against the poppet valve 16. If the load be thrown off, the deflector is forced into the jet, and the poppet valve is pushed hard up against the spear rod. Pressure accumulates behind the piston, and the spear is slowly forced into the nozzle, reducing the area of the jet. As the area of the jet is reduced, the deflector is lifted out of the jet, until finally the deflector and spear come to rest in a position where the deflector is just clearing the jet. If the load is thrown on, the deflector is raised, the bell-crank lever 20 is drawn away from the poppet valve, and the pressure in the cylinder forces the valve outwards away from its seating. The pressure behind the cylinder is now relieved, and a superior pressure in front of the piston forces the spear back, thus forming a larger jet.

In order to prevent undue rise of pressure in the pipe-line the inward movement of the spear must be slow, but in order to prevent rise in speed the cutting off of the jet must be extremely rapid. This is allowed for by the connection of the rod 35 to the lever 33 being through the spring 34; the rapid movement of the deflector compresses the spring, which keeps the poppet valve up against the end of the spear rod. Pressure accumulating behind the piston forces the piston, and with it the spear, inwards, the rate of closing being regulated by means of a by-pass valve in the piston 23 moving in the oil dashpot 24. A non-return valve is arranged in the piston 23, allowing the oil to pass rapidly from the upper to the lower side of the piston, and thus permitting the lever 20, and with it the poppet valve, to be moved rapidly. The spear rod, following the movement of the poppet valve, is drawn back rapidly. The regulating valve 26 is provided; this is not intended to regulate the movement of the spear, but only to ensure that there is not more water passing into the cylinder than can be carried through the hole in the spear.

For convenience in carrying out the efficiency tests, described hereafter, the economizer mechanism was disconnected from the governor proper, and the bell-crank lever was locked so that the position of the spear rod could be set as described by means of the small knurled-headed screw. This screw carried twelve threads per inch; thus it was a simple matter to adjust the spear to any desired position with an error of less than $\frac{1}{100}$ inch.

Test of the Turbine.

Installation.—The water supply for the turbine is obtained from the stream Afon Calettwr, a tributary of the River Dee. A low weir built across the stream diverts the flow into a concrete settling-tank. From this tank a line of 15-inch concrete tubes laid at a slope of 1 : 150 leads the water along the side of the valley for a distance of about 2,800 feet to a concrete straining-chamber, from which the water flows in a 15-inch cast-iron pipe-line, 1,572 feet long, to the power-house and turbine. The catchment-area of the stream at the point where the weir is situated is about 5 square miles. It is anticipated that in extremely dry weather the flow in the stream may fall as low as 60 cubic feet per minute. Arrangements are being made to store the continuous flow of the stream in dry weather for use during the time when power is required. The static head from the overflow of the chamber at the pipe intake to the centre-line of the turbine nozzle is 253·3 feet.

The turbine is coupled directly to a 65-kilowatt continuous-current compound-wound interpole generator, wound for 220 volts at 725–750 revolutions per minute. In order to allow an ample margin of capacity in the turbine, it was designed to be capable of developing 150 HP. at full load.

It may be of interest here to compare the capacity of this turbine with that of a normal single-jet Pelton wheel. The full-load diameter of the jet is 3·375 inches. To obtain the best possible efficiency the mean diameter of a normal Pelton-wheel runner would have been not less than about eleven times the diameter of the jet, namely, 37 inches. Taking the mean bucket velocity of the Pelton wheel at $0{\cdot}45 \times \sqrt{2gH}$, the speed of rotation would have been 350 revolutions per minute. To attain the speed of 725 revolutions per minute a Pelton wheel consisting of two runners, each fitted with two jets, would have been employed. Though this arrangement is fequently adopted, it is at best a makeshift, and such Pelton wheels are never so efficient as single-jet single-runner machines.

Method of Carrying Out the Test.—The test was carried out by measurement of the water head and quantity, and by simultaneous observations of the output of the dynamo. The head was measured by means of a 6-inch Budenberg pressure-gauge, reading in feet of water from 0 to 500. The gauge was connected to the turbine inlet pipe. The bore of the pipe at the point of attachment was 10 inches, and the centre of the gauge was 2·5 feet above the centre of the nozzle. To calibrate the gauge, levels were taken from the centre of the gauge to the overflow of the tank at the intake to the cast-iron pipe-line. The reading of the gauge under static conditions, and the figure obtained by direct levelling, were compared, and the error so obtained was assumed to be constant over the range of the test observations. The reading was finally corrected for the head due to the velocity and the height of the gauge above the nozzle.

The quantity of water was measured by observing the flow over a rectangular notch 3 feet in length. The tail water from the turbine was passed into a rectangular wooden box, 5 feet 6 inches wide by 6 feet long, sunk into the ground, the down-stream face being fitted with a board in which the notch was cut. The edges of the notch were bevelled to $\frac{1}{8}$ inch in thickness. The up-stream face of the notch was flush with the side of the box to which it was attached. The edges of the notch were 1 foot 3 inches distant from the sides of the box. The sill was 2 feet 1 inch above the bottom of the box. A hook gauge reading by vernier to $\frac{1}{1000}$ foot was placed 5 feet back from the sill. The hook of the gauge was surrounded by a wooden casing to enable accurate observations of the surface of the water to be made. To ascertain the amount of leakage, the box was filled with water to the sill level, and observations of the rate at which the water-level fell were made. The leakage so obtained was less than 1 cubic foot per minute, and corrections have been made to allow for this in the results.

The quantity was worked out from Francis's formula $Q_1 = 3 \cdot 33 (L - 0 \cdot 2\ H_1)\ H_1^{\frac{3}{2}}$. No correction was made for the velocity of approach, as this was less than 0·3 feet per second even at the largest quantities observed. The end contractions of the notch were taken to be complete.

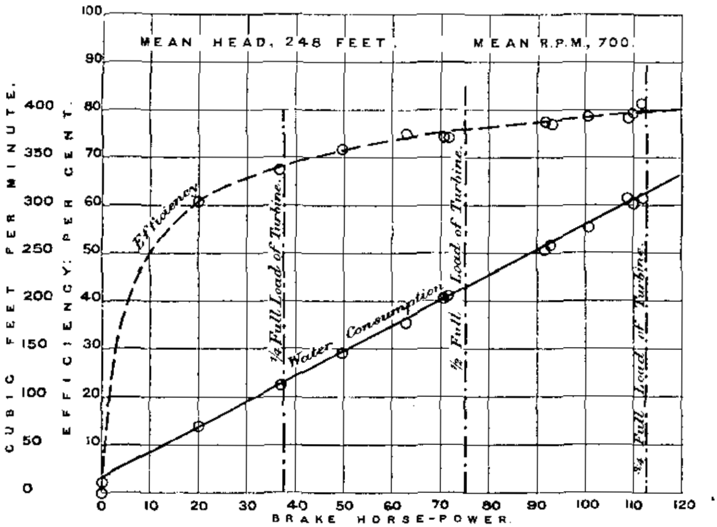
A Bailey gyrometer reading from 350 to 1,050 was used to measure the revolutions per minute. This was calibrated at two or three speeds by means of a direct-reading revolution counter.

To measure the generator output a portable testing ammeter and voltmeter was employed. Immediately after the test the instrument was sent to the National Physical Laboratory for calibration. A test of sub-standard accuracy was there carried out on it, and the corrections so obtained have been allowed for. To obtain the brake-horse-power output of the turbine, the efficiency of the dynamo, as given by the makers' test curve, has been taken.

Results.—The results obtained are shown in *Fig. 5*. It was unfortunate that, owing to the limitations of the concrete supply-pipe and the dynamo, the full power of the turbine could not be developed. From the remarkably straight line of the water quantity results it would appear that the hydraulic efficiency and the mechanical losses remain practically constant over the range of the tests.[1]

The Author is indebted to Messrs. Gilbert Gilkes & Co., Ltd., the makers of the turbine and governor, for permission to publish the results and drawings. He is also indebted to Sir Henry Robertson, the owner of the plant, for the facilities given him for carrying out the tests. The whole of the design and manufacture of the plant

Fig. 5.



described, with the exception of the ball-bearings, is entirely British. The Author has to acknowledge the help given him by Mr. G. E. Thorpe in the description of the governor and economizer.

The Paper is accompanied by three drawings and two diagrams, from some of which the Figures in the text have been prepared.

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[1] Since the Paper was written, tests of a similar turbine of the same capacity have been carried out by Dr. A. H. Gibson, M. Inst. C.E., Professor of Engineering, Manchester University. The results show a maximum efficiency of 83·5 per cent. under a head of 200 feet, delivering 106 HP. at 640 revolutions per minute.