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IMPULSE WATER WHEELS.

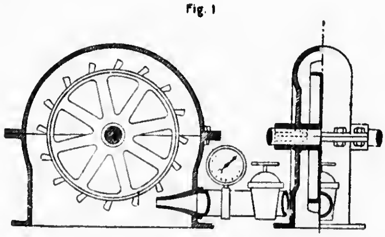
To be read Thursday, January 20th, 1898.

J. T. Farmer, Ma.E.

The devolopment of power by means of impulse water wheels has been receiving considerable attention during the past ten years. Water power . . be met with under varying conditions and in various surroundings; and the means best adapted for the utilization of the power vary with those conditions and surroundings.

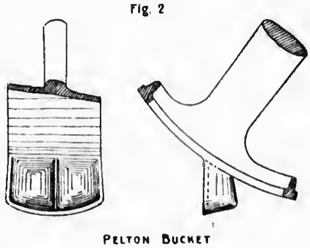
Among the means devised by man at different times before the advent of the impulse wheel for utilizing the water power that was going to wasto around him, one has easily taken the foremost place and, indeed, has, by a process of the survival of the fittest, practically ousted all other methods from a position of being worthy of serious consideration. The turbine has at the present day almost entirely taken the place of tho earlier devices in use, which have either been consigned to museums as curiosities or are regarded as picturesque additions to the landscape.

Fig. 1



The impulse water wheel probably differs as much from the various forms of turbine in construction and in action as the turbine does from an overshot or breast water wheel. The previous statement with regard to the turbine must therefore be further modified so far as it is found that the impulse motor is finding favour with those who utilise water power.

Fig. 2



PELTON BUCKET

It has been said that "countless wealth is being squandered in all the torrents and water courses of the world." But it might be added that unless the proper means are taken for its utilization that wealth of energy avails little more to man than that of the tides in Jupiter.

It seems at first sight a very simple matter to place a wheel in position to take up the energy of water; but in practice that arrangement is generally found to involve more or less costly construction in the way of dams, basins, canals, flumes and even tunnels. This is particularly the case where the use of turbines is contemplated, and this consideration is frequently sufficient to annihilate the expediency of thus attempting to utilize a known and otherwiso available source of power.

These adverse conditions are forcibly illustrated in the mountainous districts of the North American Continent. Water power is there in abundance, but it is that of mountain torrents; as a rule inconsiderable in volume of water, but, on account of the configuration of the country, affording large heads. The latter circumstance makes any constructive work very costly, and in most instances would put the use of an ordinary turbine out of the question.

It was from such causes that the Western States became the birth place of that system of water power of which the essential feature is an impulse water wheel. The simplification made possible in this system is that of the substitution of a pipe and nozzle of insignificant dimensions for the massive head race and wheel pit associated with the use of a turbine.

The first impulse wheels brought into use were of the very crudest description; with the increasing use of the system however came the development which attends every invention which has a large field of usefulness open to it. The impulse wheel of the present day ranks as fairly efficient among the various means of utilizing natural energy.

At this stage it becomes a question to what extent it may be desirable to employ the impulse wheel outside the conditions under which it first sprang into existence. This problem is specially interesting in a country where there is an abundance of water power, and at a time when the utilization of water power is assuming the place of one of the most important engineering questions of the day. The object of this paper is to record the results of some experimental research on this subject and also to discuss the question by the light of those results and from other considerations.

The history of the development of the impulse water wheel may advantageously be sketched briefly. The first wheels of this class were simply provided with flat projections on the rim of the wheel, and the jet was arranged to impinge normally on these flat surfaces. This was what was known as the hurdy-gurdy. It can easily be shown from theoretical considerations that the ideal efficiency of such a wheel is 50 p.c., but it is probable that most of those in use did not give a greater efficiency than from 20 to 30 per cent.

Fig. 3

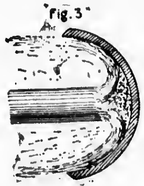
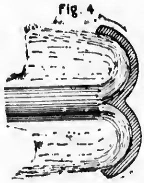


Fig. 4



The first notable improvement was that of substituting hollow cups for the flat vanes, so that the jet struck the interior part of the cup and was deflected back again until it left the vane, travelling, with respect to the vane, in almost the opposite direction to that in which it was travelling before impact, as shewn in fig. 3. This formation at once largely increased the efficiency, but in practice the efficiency was still far from what it theoretically might be.

The next modification was that of so curving the surface of the cup that the jet might follow the surface with very little deviation at the first point of contact. Thus some wheels are formed with a conical projection in the interior of the cup at the point where the jet strikes the surface, so that the water on striking may begin to pass along the generating lines of the cove, and may gradually be deflected further to follow the curved sides of the interior of the vane. This formation is illustrated in fig. 4. The more common construction is to place a wedge-shaped projection across the interior of the cup or vane. This modification was introduced about 1880. It may be seen in the bucket illustrated in fig. 2. The function of the wedge is two-fold.

(1) To prevent the heaping of dead water upon the vane during its passage through the arc of action, or the part of its path in which the vane receives the jet of water.

(2) To give the diverted streams a direction of motion which will finally carry them clear of the wheel.

In a bucket unprovided with any conical or wedge-shaped projection, there is no sudden angular deflection of the water. Some of the water is heaped upon the flat surface upon which the jet is impinging, thereby forming a curved surface over which the following water is deflected, as shewn in fig. 3. With a stationary vane on which the stream is continuously playing, the loss of force due to this cause is very slight. When, however, the impact is taking place intermittently on a moving vane, the dead water is discharged after very inefficient action at the end of every short period of action, and the total loss in effective work may be considerable. This loss is reduced by placing a solid projection in the bucket, which takes the place of that formed by the water and leaves all the water free to be deflected in the most efficient manner. See fig. 4.

Fig. 5

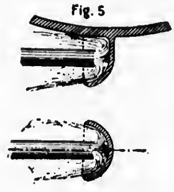
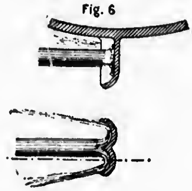


Fig. 6



As regards the second function of the wedge, it is well known that when a stream of water strikes normally upon a surface it is deflected equally in all directions. This is illustrated in the wheel bucket, of which two views are shown in fig. 5. The same action takes place when the stream strikes centrally upon the apex of a cone. This is undesirable in the case of the vane of a water wheel, as the water which is deflected towards the centre of the wheel gets into position to strike the back of the following vane, thus opposing the useful effect of the action. When the jet strikes a wedge, as in fig. 6, it is cut into two portions, which are deflected away from one another in a plane perpendicular to the cutting edge of the wedge. In a wheel this motion causes the water to be discharged at each side of the wheel where it is free from all liability to interfere with any following parts of the wheel.

Numerous modifications of the form of the curved surfaces of the buckets have been brought out at different times by inventors with a view of modifying the passage of the water over the vane in some particular, but it is not necessary to describe them more particularly.

Of the impulse wheels in use at the present day the best known is probably the Pelton water wheel. These wheels are made in sizes varying from 6 in. to 6 ft. in diameter, according to the head of water available and the velocity required. These wheels have been applied under heads ranging up to 1,700 feet and, as has been said, there is no doubt that under such conditions the highest efficiency is realized. On the other hand, there are said to be instances in which Pelton wheels are running with good results under heads of from 50 to 75 feet.

The writer recently made a series of tests on a small wheel of this class, catalogued as the Pelton Motor No. 3. This wheel is approximately 18 in. in diameter, and the weight of the whole machine is given as 320 pounds. Two elevations of this motor are given in fig. 1.

The tests were made in the Hydraulic Laboratory of McGill University, and a brief description of the methods employed will be given. It was impossible to make tests with heads as high as some of those under which these motors run. The maximum head employed was that afforded by the city supply from the high level reservoir, which gives a pressure of 125 lbs. per square in. in the laboratory, equivalent to a head of 290 feet. Lower pressures were also obtained by throttling the supply from the same source.

These trials will give an idea of what may be expected of this type of motor when used under ordinary heads of from 100 to 300 feet. In many districts these are as large heads as are commonly met with. Also, where it is proposed to take power from a water-works system, the pressure under which water is supplied would rarely exceed 125 lbs. per square inch.

In the present series of trials the wheel tested was small compared with many in use; the effective work done did not in any case exceed 7 horse-power. There is no doubt that with a machine designed on a larger scale, as with larger heads, the efficiency would show some increase over the values found in the present case.

The results obtained in these experiments are offered as bearing directly on the question of the utilization of this system for small amounts of power under the conditions usually met with in districts outside those referred to as abounding in very high falls of water. With reason and judgment, the general conclusions arrived at by the consideration of these results may be extended to cases where the machinery and the generation of power is on a larger scale.

For the purposes of trial the wheel was set up as received from the makers, and the auxiliary apparatus was fitted in accordance with their instructions. The water, after passing through the valve which was used to regulate the pressure, was led along a length of $2\frac{1}{2}$ inch pipe straight for 8 or 10 feet before reaching the nozzle. A Bourdon gauge was fitted on the supply-pipe less than one foot from the mouth of the nozzle-tip. This gauge was arranged on a pressure-chamber, enveloping the pipe and communicating with the interior through a series of small holes. Before being used the gauge was calibrated by means of a gauge tester. In the experiments the pressure in the pipe of course varied slightly. The pressure was read at intervals of one or two minutes, and the mean value during the whole trial was accepted as the pressure under which the flow took place. The extreme variation of the pressure was about one pound per square inch.

Three different sized nozzle tips were supplied with the wheel. These nozzles tapered gradually on the inside from the diameter of the supply pipe to that of the actual orifice. The outlet diameters were:

.5277″ .6307″ .7532″

Sets of trials were made using the largest and smallest of these nozzle tips, the largest giving the more satisfactory results.

The water was discharged from the motor into a flume beneath, whence it ran into measuring tanks, and all the water used was thus actually measured. For the purposes of these trials two tanks were used, each of the capacity of 1,000 gallons; these had both been previously calibrated.

The power given by the wheel was estimated by means of an absorption brake and a revolution counter.

The shaft was provided with an 18″ diameter brake wheel of special design, and the power was taken off this. In the earlier trials the brake consisted of one or more cords embracing a suitable arc of the periphery of the brake wheel, and having spring balances attached to the tight and slack ends to indicate the corresponding tensions in the cord. As the power varied slightly all the time, both readings were taken at intervals of one or two minutes, and the means used in calculating the final result of the trial. Later a direct-reading, self-adjusting brake, designed by Mr. Withycombe, was substituted for the cords and spring balances with very satisfactory results.

An ordinary revolution counter was used, but arranged to be thrown in and out of engagement with the shaft at the beginning and end of each trial. The necessary readings could thus be made at leisure, ensuring greater accuracy.

In addition to the revolution counter a tachometer was connected to the shaft. This served as a guide when adjusting the load on the brake wheel previous to a trial to give a desired speed of running. It also served to indicate any considerable departure from the intended speed which might take place during a trial, and which would vitiate the accuracy of the calculated results.

Before passing to the examination of the experimental results of the trials, it may be well to make a brief theoretical analysis of the subject.

The elementary theory of an impulse wheel is very simple—so simple indeed that no attempt seems to have been made to consider to what extent known and observable phenomena may modify theoretical calculations; but rather the elementary theoretical result is generally taken as the last word which can be said on the subject from a theoretical point of view.

In the following investigation the efficiency is deduced from a consideration of the circumstances, as far as they can be mathematically expressed, under which the mechanical action takes place.

In the elementary theory of the impulse water wheel the assumptions generally made are substantially as follows:—

1. That the jet has the theoretical velocity due to the available head of water.

2. That the jet strikes the vane centrally and tangentially to the wheel.

3. That the jet passes over the surface of the vane without any loss of relative velocity.

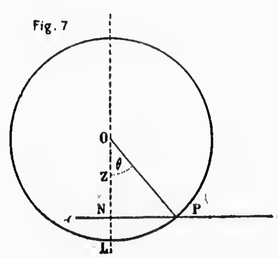
4. That the vane is so formed as to turn the stream through an angle of 180° completely back on itself.

In all these particulars there are some modifications which can be more or less exactly stated:—

(*a*) The velocity of the impinging jet is reduced in the ratio of a coefficient of velocity depending on the pipe line and nozzle.

(*b*) Instead of striking the vane tangentially, the jet generally as in the case particularly alluded to strikes at a point nearer to the nozzle. Thus suppose a horizontal jet is applied underneath a wheel. If O in fig. 7 be the centre of the wheel and L its lowest point, the jet strikes at P, where $L\ O\ P$ makes an angle θ say. Of course the water begins to play on the vane before it reaches P, and continues for a short space afterwards until the stream is cut off by the next approaching vane; but P may be taken as a mean position.

Fig. 7



The wedges of the vanes then are formed normal to the jet at this point instead of being radial to the wheel. This involves their being inclined at an angle θ to the radius of the wheel.

(c) The force of impact is reduced owing to the velocity lost by the water in passing over the surface of the vane. Some previous experiments on this subject afford data which will be used in approximating to the loss due to this cause.

(d) It is impossible, practically, to turn the water completely back on itself on account of the reaction which would take effect on the back of the succeeding vane.

Let u be the resolved velocity of the vane at P in the direction of motion of the jet.

v that of the jet.

The water strikes the vane with relative velocity $(v - u)$ and leaves it with relative velocity $c_w\ (v - u)$, where c_w is the ratio of final to initial relative velocities.

The force exerted on the vane in the direction of motion of the water is equivalent to the momentum of water destroyed per unit time, which is :—

$$\frac{m}{g} \left\{ (v - u) - c_w (v - u) \cos \delta \right\}$$

where δ is the angle of deflection of the water

$$\therefore F = \frac{m}{g} (v - u)(1 - c_w . \cos \delta)$$

Now $u = OP . \omega . \cos \theta$.

ω being the angular velocity of the wheel.

If the line of the jet cuts OL in N, and ON be called z,

$$\frac{ON}{QP} = \cos \theta$$

$$\therefore OP = \frac{z}{\cos \theta}.$$

$$\therefore u = \frac{z}{\cos \theta} \omega . \cos \theta$$

$$= z\,\omega$$

$$\therefore F = \frac{m}{g} (v - z\omega)(1 - c_w \cos \delta)$$

The moment of this force about the centre of the wheel is

$$Fz = \frac{mz}{g} (v - z\omega)(1 - c_w \cos \delta)$$

which is constant.

The work done per second is

$$F z w$$

$$= \frac{m\,z\,w}{g} (v - z\,w)(1 - c_w \cos \delta)$$

The energy available per second is mh,

h being the available head of water. The efficiency therefore is

$$\eta = \frac{z\omega}{gh} (v - z\omega)(1 - c_w \cos \delta).$$

If N be the number of revolutions per minute

$$\frac{N}{60} = \frac{\omega}{2\pi}$$

$$\therefore \eta = \frac{2\,\pi\,N\,z}{60\,gh} \left(v - \frac{2\,\pi\,N\,z}{60}\right)(1 - c_w \cos \delta) \text{ (I).}$$

In calculating the value of v to be inserted in this expression it must be remembered that the velocity of the issuing jet is less than that theoretically due to the head.

In practice there will be a reduction of velocity due to two causes :

(1) Resistance of pipe line.

(2) Loss in discharge from the nozzle.

If l be the length of pipe,

d the diameter,

The loss h_1 due to pipe friction is calculable by the well-known formula

$$h_1 = 4f \frac{l}{d} \frac{V^2}{2g}$$

V being the velocity of the water in the pipe and f the coefficient of frictional resistance of pipes.

If A_1 a be the areas of the pipe and nozzle respectively,

$$A_1 V = av$$

$$\therefore h_1 = 4f \frac{l}{d} \frac{a^2}{A_1{}^2} \frac{v^2}{2g}.$$

Other losses due to bends, etc., can be considered included in a coefficient K, which would have to be derived from consideration of the special circumstances in every case. This would make the total loss of head in the pipe line

$$h_1 + h_2 = \left(4f \frac{l}{d} \frac{a^2}{A_1{}^2} + K\right) \frac{v^2}{2g}$$

The remainder of the available head is spent in producing the velocity v, and part is absorbed in the resistance of the nozzle. If the nozzle offered no resistance the velocity would be increased in the ratio of $1 : c_v$, and therefore the velocity equivalent to the remaining head is $\frac{v}{c_v}$.

The energy remaining is therefore

$$\frac{v^2}{c_v{}^2\, 2g}$$

$$\therefore h = h_1 + h_2 + \frac{v^2}{c_v{}^2\, 2g}$$

$$= \left(\frac{1}{c_v{}^2} + 4f \frac{l}{d} \frac{a^2}{A_1{}^2} + K\right) \frac{v^2}{2g}$$

Numerical values inserted in this formula will give

$$h = \left(1.06 + .026 \frac{l}{d} \frac{a^2}{A_1{}^2} + K\right) \frac{v^2}{2g}$$

The results given by this formula would only approximate more or less closely to the actual state of affairs, which would best be determined by actual measurement of the pressure close to the nozzle by means of a gauge, when the conditions of flow are those actually occurring in practice.

In the trials under discussion the heads given are those measured close to the point of discharge, so that no loss due to the pipe line need be considered. The only loss of velocity is that which occurs in the discharge from the nozzle, and the value of v therefore is calculated from the formula

$$v = c_v \sqrt{2gh}$$

where c_v is the coefficient of velocity for the nozzle used.

In the nozzles used in the experiments the stream issued from a parallel throat, and consequently there would be no appreciable contraction of the jet. On this consideration it is reasonable to attribute all the deficit in discharge to the loss in velocity or

$$c_v = c_d.$$

The co-efficients of discharge were determined for these nozzles for heads up to 20 feet, above which point the variation becomes very slight. The results obtained therefore give an approximation to the true velocity of the jet.

In addition to these determinations the co-efficient of discharge was calculated from the data afforded by each of the trials. These coefficients agreed very closely with those obtained directly in the case of the $\frac{3}{4}$ in. nozzle. The mean values were .972 and .980 respectively.

The discrepancy is not surprising when it is considered that in the former case the outflow was from the end of a long pipe, while in the latter it was from a large body of water at rest.

The discrepancy is more marked in the case of the nozzle ½″ diameter, where the two values are .909 and .976. It is suggested that this difference is due to the fact that the interior of this nozzle was covered with rust at the time of its being used in the water wheel, as it had been in place for some time. The coating of oxide on the interior would diminish the actual area of outlet, so that the co-efficient would appear to be smaller than it actually was. In addition to this there can be no doubt that the rough surface of the oxide would diminish the velocity of the outflowing water; this may be partly the reason why the trials with the ½″ nozzle show a smaller efficiency than those made with the ¾″ nozzle.

If this explanation is correct, it would point to the desirability of having the interior surfaces of the nozzle-tips clean and free from rust. To accomplish this, it would probably be worth while to have detachable nozzle-tips made of brass or some other metal not so liable to be acted upon as iron in the presence of moisture. It would also be advisable for the user to periodically take out and clean the nozzle tip, especially if made of cast or wrought iron.

With a dirty nozzle there is a direct loss of efficiency corresponding to whatever loss of velocity is caused by the rough surface of the nozzle. More than this, there is a diminution in the area of the outlet, and therefore in the discharge, with the result that the power developed by the motor falls off. This may become a serious consideration if the motor is not much more than equal to the demands usually made upon it.

The particular values will now be inserted in the expression for the efficiency.

Pressures = 75 and 100 lbs. per sq. in.

Corresponding heads = 175 and 235 ft.

$$v = 103.1 \text{ and } 119.6 \text{ ft. per sec.}$$

$$z = .666 \text{ ft.}$$

$$\delta = 170^\circ$$

$$\cos\delta = -.9848$$

The value of c_w can be deduced, as mentioned previously, from an expression derived from a series of experiments on vanes of this description,

$$r = \frac{w}{v_o} = .0266 \ \frac{A}{a} \ \frac{1}{v_o{}^{.3}}$$

where w is loss of velocity, v_o the mean velocity of the water, a the sectional area of the jet and A the wetted area of the vane.

$$c_w = 1 - r \text{ approximately.}$$

The ratio $\frac{A}{a}$ is about 6.6

$$\text{whence } r = .176 \ \frac{1}{v_o{}^{.3}}$$

The quantity expressed by v_o here is the mean velocity with which the water passes over the surface of a bucket and may be taken as

$$\left(v - \frac{2\ \pi\ N z}{60} \right)$$

in expression (I).

Substituting values for the different conditions under which the wheel is run, the following table of values of c_w can be deduced :—

TABLE I.

N	h = 175 ft.	h = 235 ft.
300	.953	.955
400	.952	.954
500	.950	.953
600	.948	.952
700	.946	.951
800	.944	.949
900	.941	.947
1000	.937	.945

These values lead to values of the factor $(1-c_w \cos \delta)$ as given in the following table :—

TABLE II.

N	h = 175 ft.	h = 235 ft.
300	1.939	1.940
400	1.938	1.940
500	1.936	1.939
600	1.934	1.938
700	1.932	1.937
800	1.930	1.935
900	1.927	1.933
1000	1.923	1.931

The following values are thus obtained for the theoretical efficiency of the wheel with a $\frac{3}{4}$ in. diam. nozzle :—

TABLE III.

N	h = 175 ft.	h = 235 ft.
300	59.3	53.0
400	72.3	6[illegible] 7
500	81.8	75.8
600	88.0	83 [illegible]
700	90.9	88 5
800	90.3	91.0
900	86.6	91.2
1000	79.4	88.8
	max. 91.1 @ 738	max. 91.5 @ 857

The differences between these calculated values and the actual values obtained are exhibited in Table IV. These results are illustrated graphically in figures 16 and 17.

TABLE IV.

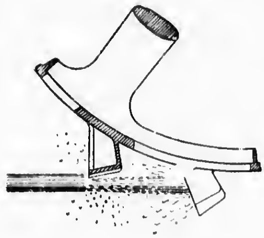
N	h = 175 ft.			h = 235 ft.		
	MATHEM	ACTUAL	DIFF.	MATHEM	ACTUAL	DIFF.
300	59 [illegible]	......	......	63.0	......	......
400	72.[illegible]	58.4	13.9	65.7	52.5	13.2
500	81.8	64.8	17.0	75.8	59.3	16.5
600	88.0	68.2	19.8	83.4	65.0	18.4
700	90.9	69.2	21.7	88.5	69.5	19.0
800	90.3	66.2	24.1	91.0	70.0	21.0
900	86.6	......	......	91.2	66.2	25.0
1000	79 4	......	......	88.8	......	......

From this last table it is apparent that there is still a waste of from 15 to 25 per cent. of the original energy of the water which has not been accounted for. The loss due to friction of bearings would be small in a simple machine of this sort, and the greater part of the 15 to 25 per cent. loss must be due to some departure in practice of the phenomena of action from those assumed.

It is suggested that the loss arises wholly or in part from the imperfect action of the vanes or buckets in turning back the water.

It will be remembered that one of the functions of the wedge was described to be to cause the water to be discharged to the side of the wheel. A little consideration, however, will show that during a part of the period of action the wedge does not perform this function.

Fig. 8

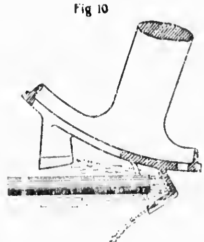


When the vane begins to intercept the jet, as in fig. 8, it is the outer lip or scoop which first comes in contact with the jet. The small amount of water which strikes the blunt edge of this outer lip is scattered, and thus only gives up a proportion of its energy to the wheel. More than this, it probably causes considerable disturbance and consequent loss of energy in the rest of the stream.

Fig. 9

As the vane passes further into the path of the jet, as represented in fig. 9, the water strikes on the interior curved surface of the outside scoop portion of the bucket on each side of the outer end of the wedge. The curve of the bucket at this point is such that the water is mainly deflected in an inward and backward curve in the plane of the wheel, so that it emerges from the vane surface in a plane tangential to the wheel rim; it proceeds in the same direction until it strikes the back of the following vane, producing upon it a force of impact opposite to the direction of motion of the wheel.

Fig 10



As the wheel moves into such position that the jet plays upon the central portion of the wedge, the stream is deflected to each side in a plane parallel to the axis of the wheel; and it is then and only then that the conditions of action assumed are approximately fulfilled. This position is shewn in fig. 10.

It may be estimated that the action of the water is not what it is assumed to be while the vane moves over from 1-5 to 1-3 of the total arc of action. During this interval the action of the water is more or less inefficient.

Fig. 11

It will be noticed that the deficit of the actual from the calculated efficiency increases steadily as the speed is increased. It is suggested that this may be attributed to two causes.

(1) The best effect of the impact occurs when the sharp edge of the wedge is perpendicular to the line of the impinging jet. This condition only occurs at one point in the arc of action. At all other points the position of the edge of the wedge departs more or less from the perpendicular position, as in fig. 11, and the deflection does not take place in the manner assumed with the consequence that the efficiency of the impact is more or less impaired. The higher the speed of the wheel the greater is the arc of action, and consequently the greater will be the departure of the cutting edge of the wedge from perpendicularity to the line of the jet. This would mean that the loss of efficiency of the impact is less when the arc of action is smaller, or the speed small, and that the loss of efficiency increases as the arc of action increases or as the speed is increased.

(2) It was pointed out how the action of the outer lip or scoop at the beginning of the arc of action tended to impair the efficiency of the wheel. It will be seen that if the arc of action is large enough the same effect will take place at the end of the arc of action, as well as at the beginning. If, therefore, the speed is increased to such an extent as to allow this to occur, there will be a further cause of loss of efficiency at high speeds.

It is estimated that the efficiency would not suffer diminution from this latter cause until the velocity reaches a value of 800 revolutions per minute with the 175 foot head or a value of 900 revolutions per minute with the 235 ft. head. It will be noticed on reference to table IV that the discrepancy between the theoretical and actual efficiencies shows a marked increase for those respective speeds.

In addition to the trials quoted and compared with the theoretical results, trials were also made with the small $\frac{1}{2}$ inch nozzle. Complete tables of all the results obtained are given.

I.—NOZZLE .5277″ DIAMETER.

(*a*) Pressure 50 lbs. per sq. inch.
Equivalent head = 115 feet.
Discharge = 45 galls. per second.

Speed.	Horse Power.	Efficiency.
252	.79	50.7
322	.76	48.5
398	.93	59.1
400	.86	57.0
407	.84	53.5
438	.87	56.7
450	.94	59.9
497	.96	62.8
506	.94	60.4
545	.95	60.6
551	.95	61.0
565	.84	55.7
585	.76	47.3
588	.83	52.6
625	.91	55.7
638	.91	58.3
665	.83	52.2

(*b*) Pressure 75 lbs. per sq inch.
Equivalent Head = 175 feet.
Discharge = 53 galls. per min.

Speed.	Horse Power.	Efficiency.
345	1.38	49.8
409	1.52	54.9
477	1.68	60.7
523	1.68	61.4
582	1.80	64.7
594	1.79	65.8
632	1.75	64 4
632	1.76	63.1
672	1.78	64.1
677	1.56	55.3
725	1.63	58.3
726	1.65	59.6
737	1.61	59.7
768	1.55	55.1
779	1.57	57.0
847	1.40	51.2
879	1.31	47.7

(*c*) Pressure 100 lbs. per sq. inch.
Equivalent head = 235 ft.
Discharge = 63 galls. per minute.

Speed.	Horse Power.	Efficiency.
276	1.70	37.9
306	1.85	41.2
345	1.98	44.4
387	1.90	44.5
459	2.22	49.7
470	2.36	52.8
541	2.52	56.0
605	2.67	60.0
644	2.80	63.2
698	2.76	64.0
760	2.96	65.0
834	2.65	59.5
858	2.78	61.8
914	2.76	58.6
939	2.76	57.9

(*d*) Pressure = 125 lbs. per sq. inch.
Equivalent head = 290 ft.
Discharge = 70 galls. per minute.

Speed.	Horse Power.	Efficiency.
494	3.25	52.0
536	3.32	53.6
592	3.50	57.5
664	3.79	62.1
702	3.83	63.3
765	4.00	65.2
813	3.89	62.8
867	3.93	64.8
918	3.95	64.0

II.—NOZZLE ·7532″ DIAMETER.

(*a*) Pressure 75 lbs. per sq. inch.
Equivalent Head = 175 feet.
Discharge = 120 galls. per minute.

Speed.	Horse Power.	Efficiency.
402	3.68	58.5
501	4.10	65.0
618	4.34	68.8
675	4.34	68.9
750	4.33	68.7
770	4.30	67.7

(*b*) Pressure 100 lbs. per sq. inch.
Equivalent Head = 235 feet.
Discharge = 138 galls. per minute.

Speed.	Horse Power.	Efficiency.
370	4.82	49 4
371	4.86	50.0
475	5.67	58.4
515	5.95	60 7
588	6.20	63.9
654	6.60	67.8
698	6.66	68.6
756	6.88	70.8
815	6.72	69.3
911	6.35	65.6

The results given in the foregoing tables are represented graphically in figures 12–17.

In connection with the above results it is interesting and important to notice that the highest actual efficiency appears at a speed which is about ·9 of that which theoretically should give the maximum efficiency.

Fig 12

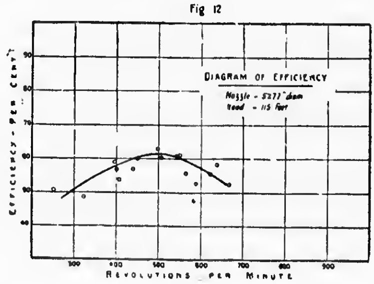


Fig 13

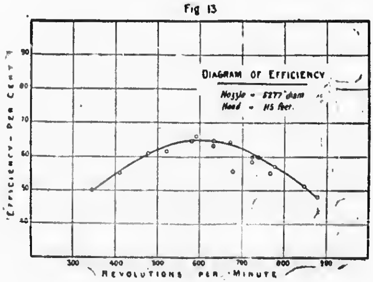


Fig. 14

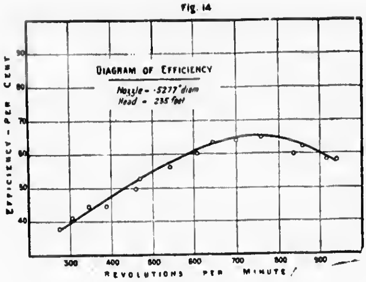


Fig 15

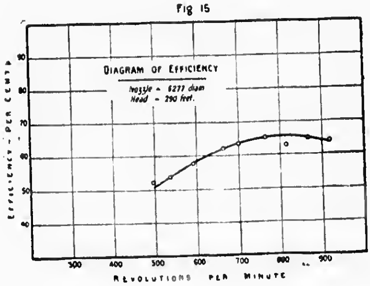


Fig. 16

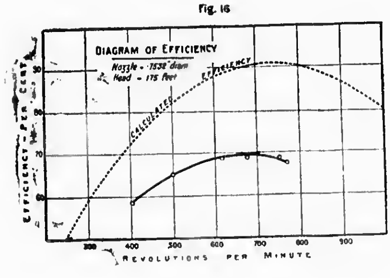
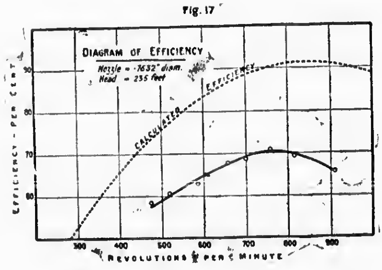


Fig. 17



A most important difference between an impulse water wheel and a turbine of either the impulse or pressure type is that the construction of the latter allows a larger area of water to be applied to the wheel for the same dimensions of wheel. In the turbine the wetted surfaces bear a much larger proportion to the size of the wheel than in an impulse wheel, and those surfaces in the turbine are constantly in action, while in the impulse wheel their action is intermittent. When the head of water is small a correspondingly large quantity has to be used to give a required horse-power, and in this case the turbine has the advantage of passing a much larger quantity than the impulse wheel. When the head is very large this feature of the turbine becomes a disadvantage, as it becomes a difficult problem to curtail the total discharge of water so that the total power developed may be handled without mechanical inconvenience by the working parts of the motor.

For a given area of outlet the horse-power of the issuing stream varies as $h^{\frac{3}{2}}$. To illustrate this the horse-power of a stream one inch in diameter is given in the following table for a series of heads and also the number of such jets which would have to be applied to aggregate 1000 horse-power.

TABLE V.

Head.	Horse power.	Jets to give 1000 H.-P.
100	4.96	202
200	14.02	72
300	25.75	39
400	39.65	26
500	55.41	18
600	72.84	14
700	91.79	11
800	112.15	9
900	133.82	8
1000	156.73	7
1100	180.82	6
1200	206.03	5
1300	232.31	5
1400	259.63	4
1500	287.94	4
1600	317.21	3½
1700	347.40	3
1800	378.50	3
1900	410.48	2½
2000	443.31	2½

In order to develop considerable power, with a comparatively small head using an impulse wheel, one of two things must be done; either the area of the nozzle and consequently that of the vanes must be made very large, which is only practicable to a limited extent, or else the number of nozzles and wheels must be multiplied. Thus the use of impulse wheels under small heads involves a large amount of machinery for the power obtained.

On the other hand, the impulse wheel has many points in its favour, chief among which is its simplicity of construction, which leads directly to the absence of mishaps and to ease of maintenance. The bearings are simple, being merely those on the horizontal shaft, in such a position as to be easily got at when necessary to make any repairs or adjustments. There are no bearings running under water; and the bearings are not subject to any other reaction than that due to the useful effort of the water on the wheel; no difficulty is met with corresponding to that of balancing the static pressure of the water on a turbine, which becomes such an important problem when large heads are being used. The impulse wheel has no water-tight joints, as there is no water pressure to be maintained among the working parts. The mechanism also does not contain any parts which are likely to work loose or otherwise become deranged and so lead to trouble.

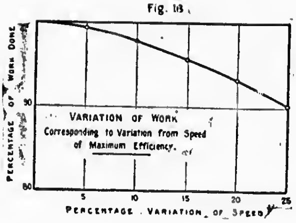
Another good feature is the absence of passages in the working parts of the motor, which would be liable to become choked by debris carried through by the water. Once through the orifice the water has a perfectly clear and open course until it falls into the tail-race.

It is an easy matter in designing an impulse motor to arrange the diameter of the wheel so as to give a desired speed of rotation under an available head with the most efficient results. Small departures from this speed do not affect the efficiency to any great extent, as may be judged from an examination of the tabulated results. The following table is compiled to show the percentage loss of output of work due to a subsequent departure from the pre-arranged speed. This is illustrated graphically in fig. 18.

TABLE VI.

Percentage. Increase or Decrease of speed.	Percentage. Decrease of output of work.
5	$\frac{3}{4}$
10	$2\frac{1}{4}$
15	$4\frac{1}{2}$
20	$6\frac{3}{4}$
25	$9\frac{3}{4}$

Fig. 18.



An important point in determining the practical usefulness of water motors is their adaptability to be run with a fair degree of efficiency under a fraction of the full load. This state of things is generally liable to occur either intermittently as where a number of loads are being continually put on and off the mechanism driven by the motor; or periodically as where for portions of a day or week or year the work required from the motor is heavier than at other times.

Three methods will be mentioned which are employed to vary the output of work from the wheel.

It was mentioned that three nozzle tips of different sizes were supplied with the wheel with which the tests were made. By changing these the quantity of water discharged under a given pressure can be varied as the area of the orifice. The power of the jet will consequently vary in the same ratio; and so any change of load which can be anticipated and will last for a considerable period can be provided for.

The changing of the nozzle tips need not be a very difficult operation. It it, however, a very inconvenient plan to have to resort to regulate the output of power from the wheel.

These wheels are sometimes built with several nozzles placed at intervals round the periphery of the wheel. When this is the case the power can be reduced by shutting off the stream from one or more of the nozzles.

The third method is to employ a value or gate in the supply pipe which can be shut off to any desired extent by hand or by some automatic regulating machinery. This method is almost always necessarily employed in addition to those aforementioned. It will be noticed that the effect of the value to reduce the power is reached by throttling the water as it passes the gate, thus reducing the pressure of the water as it reaches the orifice and consequently reducing also the discharge. It need hardly be pointed out that there is a great loss of efficiency when the motor is running under a light load, as the pressure energy which is not required to drive the machine is all absorbed without useful effect in the resistance of the partially closed valve

An idea of the actual efficiency reached can be gained from a consideration of the foregoing results, obtained for the small nozzle, for the range of heads from 120 to 300 feet. In calculating the efficiencies previously given, the available work was calculated on the assumption that the pressure under which the test was made was the total pressure available. But if that pressure is not the total available pressure as when the pressure is reduced by throttling from 125 to 100 or 75 lbs. per square in., then the total available work must be considered to be the product of the weight of water used and the head equivalent to the total available pressure before any throttling took place.

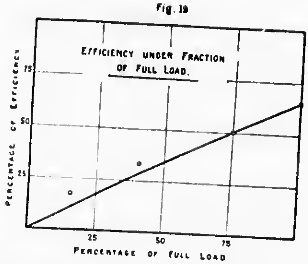
Proceeding in this way, and using the amounts of work done obtained in the trials for the various heads, but considering the energy available in every case to be that due to a pressure of 125 lbs. per square in., gives result as follows :

TABLE VII.

Pressure.	Load.	Efficiency.
125	Full	65%
100	.75	50%
75	.40	33%
50	.17	about 18%

This is illustrated in fig. 19.

Fig. 19



Besides the fact that a large amount of energy is wasted in the throttling in the pipe, a further cause of loss of efficiency under a light load exists in the fact that the motor is probably working under unfavorable conditions of pressure and velocity. In general it is desirable to keep the velocity constant, although the work done may vary. This means that if the motor is designed to run at the most efficient speed under full pressure, it will exceed the best speed for lower heads.

It will thus be seen that the efficiency obtained on the total expenditure of water will be considerably less than that indicated by the previous tables if there is any considerable variation in the load put on the motor.

In the preceding remarks an attempt has been made to describe and discuss the action of impulse water wheels, and more particularly of the wheel on which the experiments described were carried out ; the question of efficiency has been illustrated and examined, and the advantages and disadvantages connected with the use of such a system have been pointed out. It is hoped that these notes may throw some light on this interesting and important subject.

The writer wishes to repress his indebtedness to Prof. Bovey for his kind co-operation in allowing the use of apparatus and every facility for carrying out the experiments at McGill University, and also to Mr. Withycombe for useful advice and assistance with regard to many practical details connected with the trials.